PUMP VIBRATIONS

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ABSTRACT

The reliability and failure mechanisms of high energy pumps used in the utilities industry and the high suction specific speed pumps used in the hydrocarbon processing industry show surprising parallels as to root causes. Both groups of pump operators have realized that many types of repeated failures are related to flowrates and other factors of pump hydraulic design. Despite the high number of pump failures reported by both groups, a relatively limited amount of work has been conducted into recognizing and correcting damaging hydraulically induced vibrations. A better understanding of the symptoms of hydraulic instabilities as demonstrated in vibration patterns is needed to detect impending failures. The authors discuss, in a simple manner, some of the basic questions of where to start and how to conduct the problem analysis and then apply corrective measures to centrifugal pumps.

INTRODUCTION

Predictive maintenance (also known as condition monitoring) is determining when to do preventive or service type maintenance. It pinpoints deviations from accepted operating parameters of a piece of machinery. Predictive maintenance can provide a more reliable means of recognizing impending equipment failures. The concept offers opportunities for maintenance cost reductions. Its major limitation is that good engineering guidelines for determining the root causes of many observed pump problems are not available. Vibration monitoring is one method of predictive maintenance. The parameters of hydraulically induced vibration are just now being adequately described and evaluated to permit their recognition in vibration spectrum.

COMPONENTS OF A PUMPING SYSTEM

A typical pumping system can be divided into eight component areas for purposes of operating problem solving: 1) the foundation. Poor foundations, grouting, and flexible baseplate designs can cause many problems; 2) the driver. Excitations from the vibrations of the driver (motor, steam turbine, gearing) can be transmitted to other components; 3) mechanical power transmission. Excitations from the coupling area, especially due to misalignment of the driver or eccentrically bored coupling hubs. Incorrect positioning of driver and pump such that distance between shaft ends (DBSE) exceeds the axial flexing limits of the coupling; 4) the driven pump. Design of the pump can greatly influence the hydraulic interaction between the rotor and the casing and thus the problems encountered. Pump thermal growth misconceptions can alter the alignment targets; 5) the suction piping and valves. Unfavorable incoming flow conditions like cavitation, intake vortex, or suction recirculation due to poor design and layout of suction piping and valves can cause flow disturbances; 6) the discharge piping and valves. Unfavorable dynamic behavior of piping because of loads from dynamic, static or thermal causes including resonance excitation; 7) the instrumentation for control of pump flow. Control system/pump interaction during startups or other periods of low flow can produce pressure pulsations. High pressure pulsations can occur due to hydraulic instability of the entire pumping system; and, 8) the alignment anchoring devices. Once it is established, dowels or other devices on the baseplate must hold the pump alignment.

As is true in many vibration guides, most of these areas are mechanical aspects of the pump. Only items 4) and 5) consider the hydraulic nature of the pump, yet hydraulic problems are as common as the mechanical ones that are covered. A typical vibration guide that is published as an aid to vibration analysis by many manufacturers of monitoring equipment is shown in Table 1. It also addresses only mechanical aspects of pump vibration. Hydraulically induced vibrations are not referenced as possible problems.

Table 1. Traditional Vibration Guidelines.

Detected Symptoms (Vibration)	*Frequency Characteristic*	*Suspected Problem*
High and steady radial vibration of shafts exhibiting amplitudes about 2 × higher than the axial vibration	1 × cpm	Unbalance
High axial and radial vibration of bearing and shaft	1 or 2 × cpm	Bent shaft/misalignment
Beat rate	1 × cpm	Two pumps running close CPM
Detected signal disappears when electric power is shut off	1 × cpm	Eccentric armatures of electric motors
Radial and axial vibration disappears instantly when electrical power is turned off	1 or 2 × line frequency	Defective electric motors (electrically induced vibration)
High axial vibration	2 × cpm	Mechanical looseness
High axial vibration (as high as twice)	1,2,3 × cpm	Misaligned coupling
Periodic flow pulsations	vane passing frequency	Increased turbulence of defective pump
High radial vibration which disappears suddenly with lowering of speed	Less than 0.5 × cpm	Oil whirl in sleeve bearings
Unsteady radial and axial low amplitude vibration	10 to 100 × cpm	Defective rolling element bearings
Subharmonic vibration or shaft rotational frequency	1/2 to 1/3 × cpm	Journal bearings loose in housing
Radial & axial vibration with sidebands around tooth meshing	Number of teeth × cpm PINION frequency	Worn gears

SIMPLIFIED PUMP HANDBOOK

Traditional pump handbooks fail to tie the whole pump together. Most consist of many pages of technical data, tables of pressure drops, conversion charts, etc., relating to hydraulic design. However, the handbooks do little to explain the practical considerations of centrifugal pump selection, operation, and maintenance. The following three brief paragraphs help to show the relationship between hydraulic conditions, possible pump vibration and resulting mechanical problems [1].

• Centrifugal pumps should be selected and normally operated at or near the manufacturer's design rated conditions of head and flow. This is usually at the point of best efficiency (BEP). Pump impeller vane angles and the size and shape of the internal liquid flow passages can only be designed for one point of optimum operation. For any other flow conditions, these angles and liquid channels are either too large or too small.

• Excess capacity. Any pump operated at excess capacity; i.e., at a flow significantly greater than BEP and at a lower head, will surge and VIBRATE, resulting in potential bearing and shaft problems along with requiring excessive power.

• Reduced capacity. When operating at reduced capacity; i.e., at a flow significantly less than BEP and at a higher head, the now incorrect vane angles will cause eddy flows within the impeller, casing, and between the wear rings. The radial thrust on the rotor will increase, causing higher shaft stresses, increased shaft deflection, and potential bearing and mechanical seal problems. Coupled with increased radial VIBRATION and shaft AXIAL movement, continued operation in this mode will result in the accelerated deterioration of the mechanical and hydraulic performance of the pump.

Flows that are substantially different from design can be vibration problems. Most handbook authors do not recognize this effect of flow.

VIBRATION MONITORING NEEDS OF CENTRIFUGAL PUMPS

The hydraulic and mechanical design and construction of a centrifugal pump are interrelated and the two cannot be separated without disastrous results. After hydraulic and mechanical factors are evaluated, corrective measures must developed for the root causes of the pump failure mechanisms. Most vibration monitoring and analysis programs address only the mechanical causes.

Predictive Maintenance should consist of three phases: a) surveillance or monitoring; b) analysis or diagnosis; and, c) correction or remedy.

Many monitoring programs get bogged down in step "b" because of the problems in interpreting data. Step "c" frequently consists of a recommendation to replace the journal bearings, which treats the symptoms but not the root cause of the vibration trouble. Vibration data must lead to a correction. "Trending" alone is not economical. Less data taking and more decisions are needed to provide the economics of condition monitoring for pumps.

Knowledge of the pump's hydraulic design features, along with its present mechanical condition, is necessary to accurately evaluate any vibration data. Predictive maintenance in the form of vibration monitoring and analysis is widely used in the hydrocarbons processing industry for compressors and steam turbines. Vibration monitoring is not used as often for determining potential pump problems. While there are many technical papers published on vibration monitoring to evaluate performance problems on steam turbines and turbocompressors, there are few such guidelines for pumps. The smaller size and less critical nature of pumps are factors that cause this disparity. Another factor is that turbines and turbocompressors are pneumatic in nature. The fluid flowing through the machine has relatively little mass compared to the rotor weight. The compressible fluid normally has only a small impact on the vibration patterns of the rotor. Most of the vibration problems in pneumatic machines are mechanical in nature. A pump, on the other hand, is a hydraulic machine. The handled fluid has considerable mass when compared to the rotor weight and transmits any pressure pulsations throughout the system undiminished. The interaction between the rotor and casing at off design flows becomes of prime importance and a point of confusion about the vibration patterns of the rotor. The restoring force acting in opposition to the pump rotor deflection has the effect of altering the natural frequencies. Interpretation of vibration data is very difficult.

In every pump, there are many potential causes of vibration. In a multistage pump, the forces occurring in the wear rings and the thrust balancing devices have a considerable influence on the vibrational behavior of the rotor. A balancing drum has a great influence because of its large area and high pressure differential. The forces acting on the rotor in the tangential direction can lead to instabilities in the form of self-excited vibrations. Some of the vibration causes, both mechanical and hydraulic, are shown on the spectrum on Figure 1. The hydraulic forces are as numerous and as great in magnitude as the mechanical forces causing vibration.

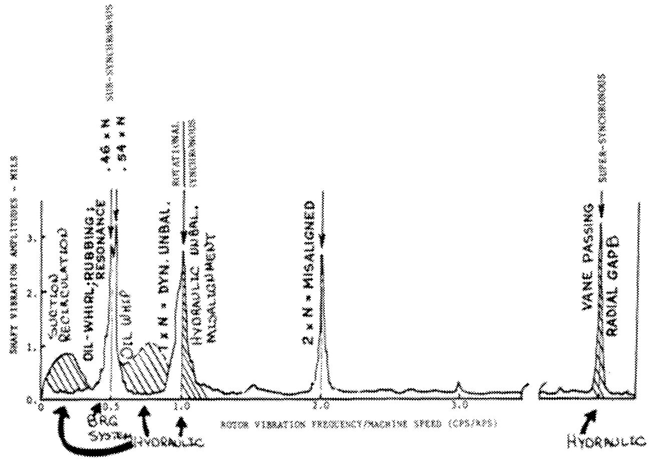


Figure 1. Typical Vibration Spectrum for a Centrifugal Pump.

RECIRCULATION—A NEGLECTED FACTOR

Liquid flow in the impeller internal channels and the casing is a very complex phenomenon, especially at off-design conditions. Most pump handbooks assume uniform flow, although the actual flow does not even approximate these conditions. The internal flow is unstable and unsteady, with violent changes sometimes occurring from impeller channel to channel. Stall, back-flow, eddy-type circulation, turbulence, and cavitation can and do take place, as shown in Figure 2. The locations of these "eddy flows and hydraulic losses" of the simplified three-paragraph pump handbook are:

• Flow "recirculation" at the impeller eye. Generally found during off-design flows, damage caused by this flow instability is always found at the impeller eye or inlet areas of the casing.

• Flow "recirculation" at the impeller vane tips. Damage is at the impeller outside diameter. This eddy flow is caused by reduced capacity flows and improper impeller tip clearance or alignment.

• Flow "recirculation" around impeller shrouds. Damage is seldom seen on the impeller; it is seen as thrust bearing damage, particularly on double-suction impellers.

Severe hydraulic instabilities result from the recirculation effects. The liquid recirculates within the impeller and casing hydraulic channels so that the fluid has an opportunity to strike the vane surfaces a number of times before finally being discharged. This turbulent flow results in a situation that is often confused with "bubble" cavitation because it generates noise, and sometimes by raising the available suction head, some improvement can result.

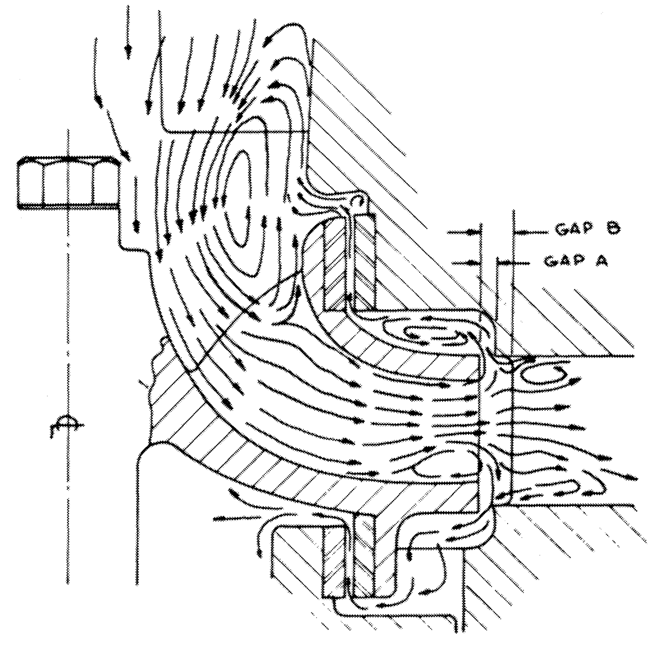


Figure 2. Eddy Flow Pattern at Suction Eye and Impeller Vane Tip at Off Design. Flow Operations [6].

The minimum flow the pump is capable of operating is strongly influenced by the recirculation tendencies. Recirculation is always present in a pump. The turbulent flow, or recirculation, must be recognized for what it is, because it can be much more destructive than cavitation. Traditional pump curves, especially the acceptable flow ranges, are misleading or even invalid.

Recirculation can cause impeller erosion and failure along with high failure rates of mechanical seals and bearings. The sand cast impeller of a centrifugal pump will have some degree of internal dimensional inaccuracies that cause both mechanical and/or hydraulic unbalance. Hydraulic unbalance is the result of uneven flow patterns between vanes due to uneven spacing of the vanes or shrouds. This can be caused during the casting process by core shifts, material flow, and/or uneven shrinkage. Good quality control at the foundry is the only way to minimize this problem. Because it affects the liquid flow characteristics, it is impossible to eliminate or even to reduce hydraulic unbalance by mechanical balancing of the impeller.

Recognizing Recirculation

There are a number of practical considerations relating to the pump design that must be understood to recognizing recirculation troubles.

• A low NPSHR or high suction specific speed impeller is much more susceptible to recirculation than a normal design. Because of highly undesirable low flow characteristics, a calculation of suction specific speed is about the only predictive measure of the trouble.

• Double suction pumps are more vulnerable than single suction pumps, because casting problems create more flow disturbances.

• Energy levels of 650 ft of head and 250 hp per stage are usually the lower limits of really serious recirculation problems, but smaller pumps can be affected.

• To suppress the destructive characteristics of recirculation, a very high NPSHA may be necessary or operations limited to only at or near the BEP flowrate with little if any turndown allowed.

• Recirculation is much more damaging with some liquids than with others. Pure liquids such as water are homogeneous, and vaporization can occur instantaneously. In addition, water has a high vapor-to-liquid volume ratio. A mixed chemical or petroleum liquid, composed of fractions that vaporize at different temperatures and pressures will have a much less violent cavitation reaction. Narrow boiling range liquids tend to react like water.

• In multistage pumps, suction recirculation occurs only in the suction or first stage. Recirculation and the associated unstable hydraulic characteristics can result in mechanical failures of the pump.

Hydraulic Stability Guidelines

One measure of the hydraulic stability of a pump, the minimum flow required and its ability to operate away from the BEP flow conditions is called suction specific speed. Unlike discharge specific speed, suction specific speed is not a "type number," but a criterion of a pump's performance with regard to stable hydraulic operation and limited recirculation. Suction specific speed is defined by the following formula:

$$N_{ss} = \frac{(\text{rpm}) \times (\text{gpm/eye})^{1/2}}{(\text{NPSHr})^{3/4}}$$

Values of capacity and NPSH required are at the pump's best efficiency point or design point. In double suction impellers, which actually have two inlets, or eyes, suction specific speed is based on the performance per inlet, or per eye such that the total capacity would be divided by two.

Although opinions vary on the N_{ss} value for a conservative *vs* a marginal design, many engineers in industry have found the following to be desirable maximums:

• For cold water and general service applications, suction specific speeds are selected in the 8500 range and lower.

• For boiler feed and condensate applications, and for general hydrocarbon service, values of suction specific speeds typically range between 8500 and 11,000.

• Pumps designed for suction specific speeds in excess of 12,000 are generally for special applications only.

One well known pump engineer's opinions on stability ranges of various pump impeller patterns are presented in Figure 3 [2]. Failure to understand the problems of recirculation, minimum flow and hydraulic stability can lead to severe maintenance and operating troubles. Several disastrous fires have resulted from abrupt mechanical seal failures, shaft breakage, impeller fracturing, etc., of pumps that can be directly traced to high suction specific speed problems.

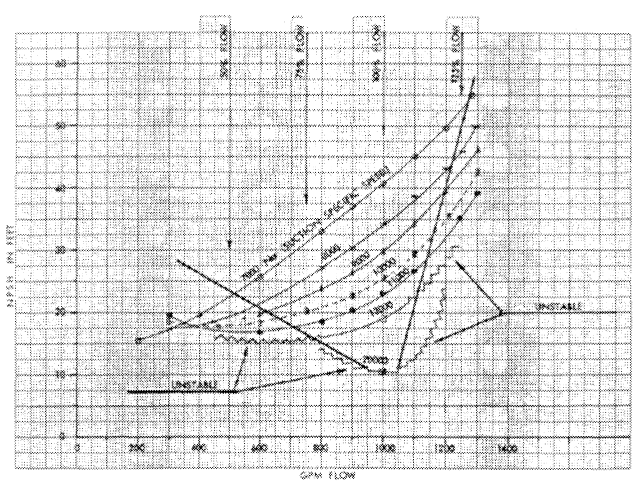


Figure 3. Stable Operating Ranges with Impeller Patterns of Different Suction Specific Speeds [2].

The results depicted in Figure 4 are of a study [3] of several hundred pump failures over a five-year period. The study indicates that pumps with high suction specific speeds (above 11,000) had a failure rate of approximately double that of the ones with lower values.

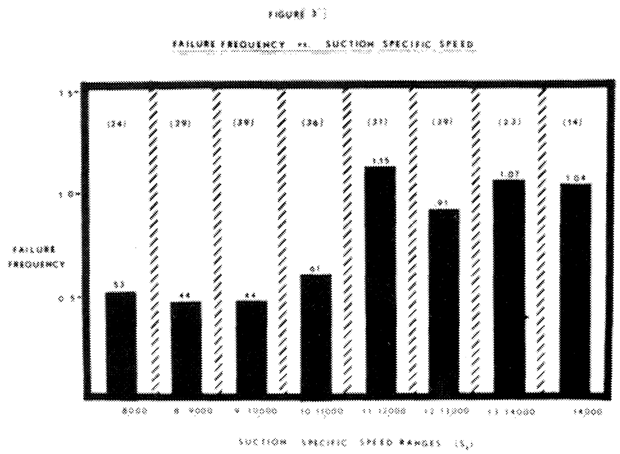


Figure 4. Failure Frequency Vs Suction Specific Speed [3].

MINIMUM FLOW VIBRATIONS

What should the minimum flow limits described in paragraph two of the simplified *Pump Handbook* be, in order to prevent serious mechanical vibration and damage from the recirculation effects discussed earlier? There are no fixed rules for establishing these limits. There is disagreement as to why the minimum flow is required. API 610, "Centrifugal Pumps For General Refinery Services," starting in 1981, has used two definitions of minimum flow.

• Minimum continuous thermal flow, "the lowest flow at which the pump can operate and still maintain the pumped liquid temperature below that at which net positive suction head available equals net positive suction head required."

• Minimum continuous stable flow, "the lowest flow at which the pump can operate without exceeding the noise and vibration limits imposed by this standard."

Thermal Minimum Flow

Thousands of curves found in vendors' catalogues show head *vs* flow as continuous from zero or shutoff flow to "end-of-curve" flow often exceeding 150 to 175 percent of the BEP design flowrate. If asked if the pump had any minimum flow restrictions, the vendor will say no or maybe set it at 2.5 to 5.0 percent of BEP flow because of temperature rise. In a panel session at the Fifth International Pump Users Symposium, Gopalakrishnan presented Figure 5, which graphically states the minimum flow factors [4]. Note that suction and discharge recirculation, impeller erosion, seal and bearing problems, and cavitation occur long before the effects of temperature are encountered. The minimum thermal flow is so far to the left on the curve and so many major problems are encountered before that point that a thermally based flow is almost meaningless. Temperature rise of the liquid in the pump is basically a function of the pump inefficiency and can be calculated easily. The critical pump area where vaporization or flashing could occur is usually at the pump inlet on single stage pumps. For multistage pumps, this may be across the axial thrust balancing device. For some chemical process pumps, the maximum temperature rise is limited not by vaporization, but by that temperature necessary in order for some chemical reaction to take place; e.g., polymerization.

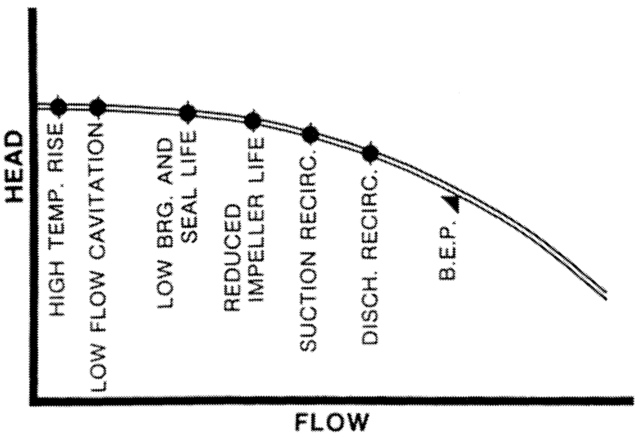


Figure 5. Head-Flow Curve Illustrating Points of Onset of Events That Adversely Affect Pump Operation [4].

Minimum Continuous Stable Flow

In addition to the many items discussed above, there are a number of practical considerations that must be understood when setting minimum flows.

• The selection of a suitable minimum-flow level for a pump depends heavily on the time factor. Is the low-flow condition almost continuous, a momentary emergency situation, or something in between?

• Recirculation cavitation in the impeller inlet is most likely to occur in pumps designed for lowest NPSH (high suction specific speed) requirements, therefore these impeller designs require higher minimum flows.

• Recirculation is much more damaging with some liquids than with others. Water vaporization can occur instantaneously, so the minimum flow must be higher.

• Double suction pumps are more vulnerable to recirculation than single suction ones, and may require minimum flows in the 60 to 70 percent of BEP range.

• Energy levels of 600 to 650 ft of head and 250 to 300 hp per stage are usually the lower limits of really serious instability.

• NPSH available is a major factor in determining minimum flows.

• To protect the pumps from instability during low flow conditions, bypass systems are necessary in many larger pumps, assuming speed reduction is not available.

• Axial positioning of the pump rotor in the casing will greatly influence the stable flow range. The several positioning factors and overlap of liquid channels of a multistage pump are shown in Figure 6.

Minimum Flow Guidelines

In the last ten years, engineers have come to realize that establishing the proper minimum flow requirement of a centrifugal

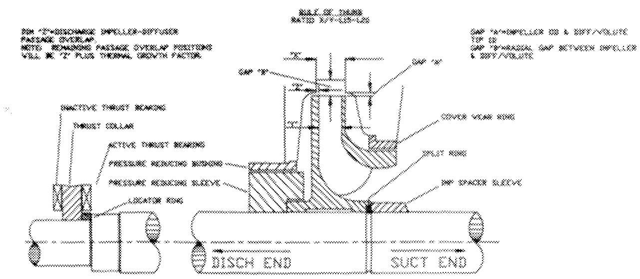


Figure 6. Impeller-diffuser Passage Overlaps—Multistage Type Pump.

pump is more complicated and very critical to the reliable operation of that pump. Rather than temperature rise, the proper minimum flow requirement for most centrifugal pumps should be based on hydraulic design and is referred to as the minimum continuous stable flow in API 610. This flowrate is much higher than that based only on temperature rise. This minimum flow is a function of the energy level of the pump, the NPSH margin available, the hydraulic design of the impeller/casing (suction specific speed or N_{ss}), the characteristics of the fluid being pumped and wear of areas such as the wearing rings. Minimum continuous stable flow, as a percentage of best efficiency capacity, increases as suction specific speed increases as shown in Figure 3.

Avoiding low-flow conditions is very important. Experience with a particular pump design is a major factor in establishing minimum flows.

Warren Frazier has presented some guidelines to estimate the point of inception of recirculation flows by use of certain measured dimensions of an impeller pattern into equations [5]. This will give some quantified factors, but it requires information that is not normally available on the hydraulic data sheet or pump curve. Some simple guidelines that can be used without any dimensional data for estimating minimum flows are:

• Assume that no pump is designed to operate for longer than about fifteen minutes below 50 percent of BEP flow and make this the baseline.

• Provide a minimum-flow bypass piping for high-energy pumps. The bypass line should be piped to permit heat dissipation.

• Impeller patterns with suction specific speeds greater than about 11,000 require a minimum flow in the 60 to 70 percent of BEP range.

• A double suction impeller also will require minimum flows in the 60 to 70 percent of BEP range.

• A pump with a suction specific speed of 20,000 may require a minimum flow of 100 percent since the pump is stable only near BEP.

• Correction factors that reduce the minimum flows when handling hydrocarbons should be used carefully. It is true a mixed chemical or petroleum liquid is composed of fractions that vaporize at different temperatures. The cavitation that occurs during recirculation is of a mechanical origin and is not totally temperature related. Also, some narrow boiling range hydrocarbons tend to react like water.

• Pump liquid channels must have proper overlap and positioning as shown in Figure 6.

Any single percentage number for minimum flow can only be arbitrary. If the pump is expected to run at other than design flows, applying some of the guidelines discussed above, a smoother operating pumping system can be achieved.

OTHER HYDRAULIC VIBRATIONS

A random 60 to 80 percent frequency vibration is sometimes found in multi-stage pumps. Its causes are hydraulic unbalance, the effects of leakage points such as wear rings, balancing drums or center stage bushings casing interaction when operating away from the design point. These dynamic forces may be very strong with both diffuser and volute types of pumps. Remedies such as antiswirl devices (Figure 7) at leakage points such as at the balancing drum bushing have been successful on multistage straight through flow pumps [9]. Total correction of the dynamic forces is difficult, since very little is known about proper volute or diffuser design at present.

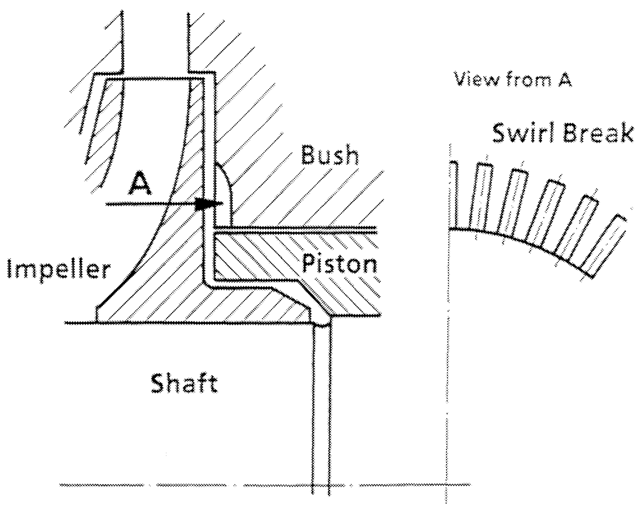


Figure 7. One Design of a Swirl Break at the Balancing Drum.

Vane Passing Frequency

Vane passing is a hydraulically induced vibration at a frequency determined by the number of impeller vanes, the number of stationary vanes and the pump rpm. The vibration is created by the momentary disturbance of the wake of the liquid exiting the impeller liquid channels by the stationary diffuser or volute vane tips. The larger the gap "B," the more the unguided flow path can smooth out before it contacts a diffuser vane or volute tongue. This increases the possibility of the flow reaching a more favorable inlet path into the diffuser or volute, especially at of design capacities. Guidelines for recommended impeller vane and diffuser or volute tip clearances are given in Table 2 [6].

Determination of the vane passing frequency sounds easy but can be confusing. The most common pump design has an impeller with an odd number of impeller vanes and a double volute in the casing. The vane passing frequency in this pump is the rotational speed times the number of vanes. Even combinations of impeller vane and volute or diffuse vanes can induce very high forces that can fracture vane tips, impeller shrouds and volute tongues. Many manufacturers use a four vane impeller in a double volute casing. The supposed objective is a very low NPSH required. The fewer the vanes the lower the NPSH required. Three vaned impellers used in most condenser hot well pumps are very inefficient, so four vanes are chosen by some manufacturers as a compromise.

For diffuser type pumps the larger number of diffuser vanes coupled with the closer "Gap B" than that of a volute pump causes a different interaction of the rotor and casing. The match up of stationary and rotating vanes has a different frequency. This will give a vane passing frequency that does not correspond to either the number of vanes in the impeller or the casing. Common vane interactions are given in Table 3 [7].

Table 2. Recommended Radial Gaps—Impeller to Casing [1].

Recommended Radial Gas for Pumps				
Type Pump Design	Gap "A"	Gap "B"*- Percentage of Impeller Radius		
		Minimum	Preferred	Maximum
Diffuser	50 mils	4%	6%	12%
Volute	50 mils	6%	10%	12%

$*B = 100 \frac{(R^3 - R^2)}{R^2}$

R^3 = Radius of diffuser or volute inlet

R^2 = 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%.

Source: Dr. Elemer Makay

Too small a Gap "B" can have a detrimental effect on various structural parts of a pump along with generating a high noise level. The axial positioning is also very important, as discussed earlier.

Random Positioning of Impellers

In multistage volute pumps, the degree of positioning of the volutes is severely limited by case design. It is necessary to randomly cut the keyways in the impellers to assure that vanes on adjacent impellers are not aligned and do not pass volute tongues simultaneously. Frequently, this random 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, and reassembly on a new pump, or when replacing the impellers during maintenance.

Correction of Vane Tip Shape

Impellers manufactured with blunt vane tips can cause trouble by generating hydraulic disturbances in the impeller exit wake area, even when the impeller is the correct distance, "Gap B," from the cut water. This disturbance may be greatly reduced by sharpening the impeller vanes on the underside or TRAILING edge of the vane as shown in Figure 8.

Table 3. Vane Passing Frequencies for Impeller-vane Combinations [7].

NO OF DIFFUSER VANES	Number of Impeller Vanes				
	4	5	6	7	
	8	*	15	9	7
	9	8	10	4	28
	10	6	*	6	21
	11	12	10	12	21
	12	*	25	*	35
	13	12	25	12	14
	14	6	15	6	3
	15	16	*	6	14

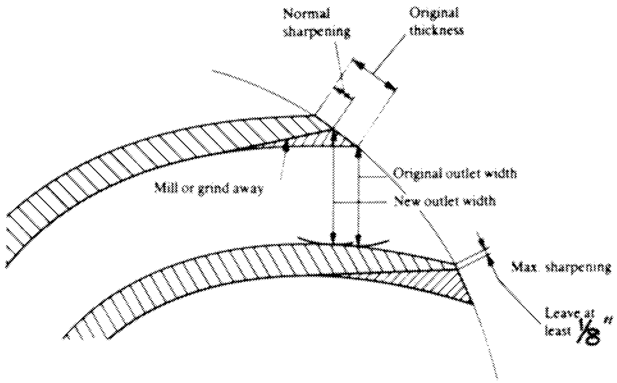


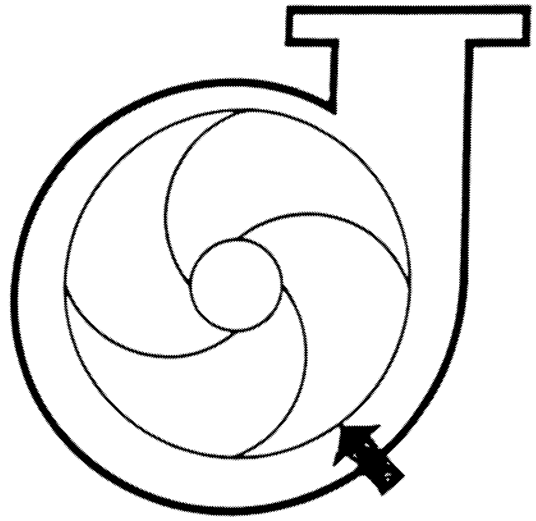
Figure 8. Sharpening of Impeller Vanes by Underfiling.

Vortexing

Vortexing can produce similar problems as encountered when the NPSHa is less than that required. Vortexing is a swirling and funneling action in a liquid. When this occurs between the liquid's surface and the draw off nozzle in a vessel or basin, air or vapor can be drawn into a pump. The formation of a vortex can be very damaging because vortexing effectively reduces the NPSH available. With the suction head of a pump close to a minimum value, air entrainment is likely to occur. As little as two percent, by volume, of air or gas entrainment can result in a 10 percent loss in pump capacity.

Radial Hydraulic Loads

The designing and casting of an impeller and casing is much of the "black art" of pump design. The form of the single and double volute case may cause an uneven distribution of pressure around the periphery of the impeller, producing a radial load perpendicular to the shaft axis. The magnitude of the load varies directly with impeller width, diameter, and the head developed. The greatest radial load occurs at low flows and is minimal at the BEP flowrate (Figure 9). This radial load unbalance increases bearing loads and shaft deflections that could lead to premature seal, shaft, or bearing failure. Shaft deflection due to radial reaction can be predicted quite accurately and therefore, should be taken into consideration when setting the minimum flow requirements. In a well designed pump, radial load sets the minimum flow at about five to ten percent of BEP.



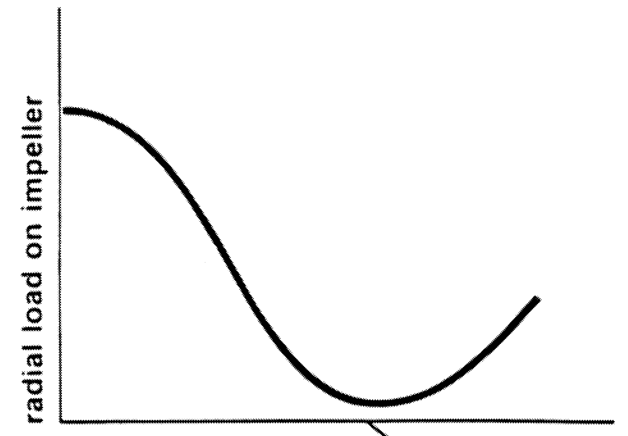


Figure 9. Radial Loads on Pump Impeller.

MECHANICAL VIBRATIONS

Power Transmission Vibrations

A major source of externally induced vibration is misalignment between the pump and its driver. This misalignment can result in an axial vibration reading as much as 1.5 times the vertical or horizontal readings. Vibration generally occurs at the running speed of the pump, although it may occur also 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. Machines that normally operate at elevated temperatures must tolerate vibration during a temporary cold misalignment until normal operating temperature is reached.

Flexible couplings will accommodate sizable amounts of 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. A gear type coupling will transmit axial thrust imposed upon it, and normally give a two times running speed. A membrane or disc coupling gives little likelihood of the vibration caused by misalignment being two times running frequency, due to the axial "softness" of the coupling.

Oil Whirl

A condition known as "oil whirl" may occur in lightly loaded sleeve bearings (under 90-100 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 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. The shaft vibration resulting from oil whirl or whip can quickly reach 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. The only real fix is a simple one: eliminate any trace of that vibration component. A rule of thumb used by many vibration experts says that oil whirl movement should not exceed 50 percent of the bearing clearance. Since the vibration level can go much higher without warning, this is a risky guideline. At higher vibration levels, the machine must be shutdown immediately to avoid serious damage.

Correction of Oil Whirl

The only real fix for oil whirl is complete elimination by preloading the bearing with a pressure dam or reducing the net area of the bearing, raising the unit loading above 150 psi, or installing tilting pad journal bearings that cannot whirl.

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.

Ball Bearing Fits

Unfortunately, many pump manufacturers 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. Do not install bearings with ODs outside of the given tolerance band since this might result in either excessive or inadequate outer race looseness. The rule of thumb shown in Table 4 is a good guideline for looseness.

Table 4. Ball Bearing Housing Fits Guide.

Rules of Thumb:	
Housing Fits	
1.	Bearing OD to housing clearance – About 0.00075 inch loose with 0.0015 inch maximum.
2.	Bearing housing out of round tolerance is 0.001 inch maximum.
3.	Bearing housing shoulder tolerance for a thrust bearing is 0 to 0.0005 inch per inch of diameter off square up to a maximum of 0.002 inch.

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 slight interference of the antifriction bearing inner ring 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. The rules of thumb shown in Table 5 give some good guidelines.

Table 5. Ball Bearing Housing Fits Guide.

Rules of Thumb:	
Shaft Fits	
1.	Fit of bearing inner race bore to shaft is 0.0004 inch tight for small sizes: 0.00075 inch tight for large sizes.
2.	Shaft shoulder tolerance for a thrust bearing is 0 to 0.0005 inch per inch of diameter off square up to a maximum of 0.001 inch.

Detection Of 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. Detection of defects is best done using accelerometers or shock pulse meters.

There are some guidelines that can help 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. 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 (retainer 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.

Piping Vibration Limits

How much vibration can be permitted in the piping system of a pump? One rule of thumb states that permissible unfiltered velocity readings taken on the piping at the midspan of its supports can be three times the permissible readings taken on the pump bearing caps. Bearing cap readings near 0.5 to 0.6 in/sec 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 per second as the limit for piping.

VERTICAL PUMP VIBRATION PROBLEMS

Despite the widespread use of the vertical pump, many old engineering problems are frequently encountered. Vertical pumps present maintenance problems that are distinctively different from those of horizontally mounted pumps. Many more parts are required to rebuild a vertical turbine pump, since typically, four stages of this mixed flow design are required to produce the head of one stage of the radial flow horizontal pump. More frequent repairs and failures are encountered than with horizontal pumps.

The vertical pump's rotor is not gravity stabilized. There is more cantilever action of the rotor than in a horizontal pump. The gyroscopic effect of rotation can cause lots of damage to the rotor and the casing when problems arise. Vibration analysis is very difficult in vertical pumps. As shown in Figure 10, the line shafting and the impeller section of the pump have vastly different rotordynamics that must be accounted for in the analysis. Vertical pumps also have a high cost of pull and rebuild, along with extended out-of-service time required to do the work. There is great potential for cost savings if excessive wear and impending failure could be detected.

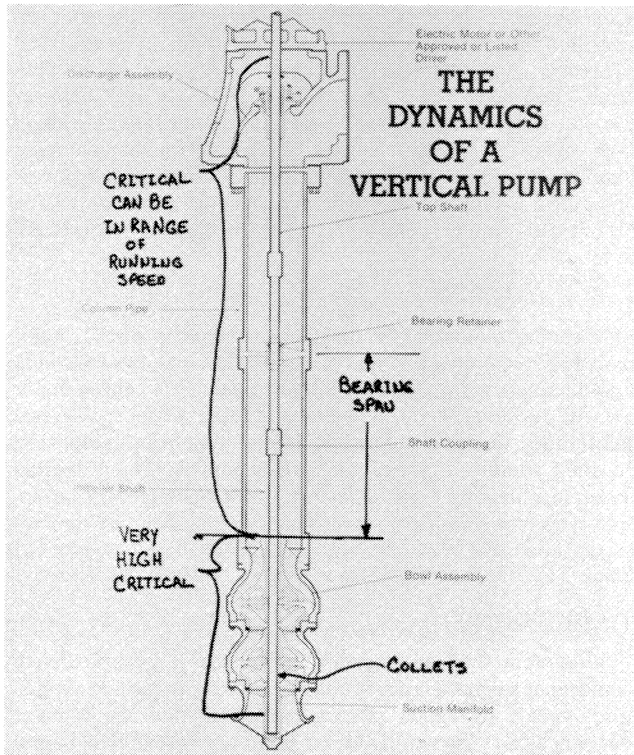


Figure 10. The Dynamics of a Vertical Pump.

The quality of vertical pump designs varies widely. Major differences exist in shaft diameter and construction, spacing and length of shaft guide bushings, impeller mounting, casing construction, degree of thrust load (down and/or up), speed selection, and quality of construction.

The most desirable location for a vibration sensor is at the very bottom bushing of the pump, a very hostile environmental for any electronic device.

Maximum Spacing Between Shaft Guide Bushings

Through the years, some vertical pumps have been manufactured with extremely large spacing between the shaft guide bushings. Problem pumps may be helped by repairs that involve shortening the bearing span. Column spacing between shaft guide bushings shall be 4.0 ft, or as indicated in API-610 7Th Edition, Paragraph 2.9.2.1, whichever is less. The reasons for this requirement are to reduce the "whip action" of the line shafting, and to increase the wet criticals of the shafting above operating speeds of the vertical pump. This "whip action" drastically increases the chance of seal failures and other mechanical problems, due to the runout on the shaft.

Shown in Figure 11 are line shafting spacing requirements from API 610, Seventh Edition, "Centrifugal Pumps For General Refinery Service."

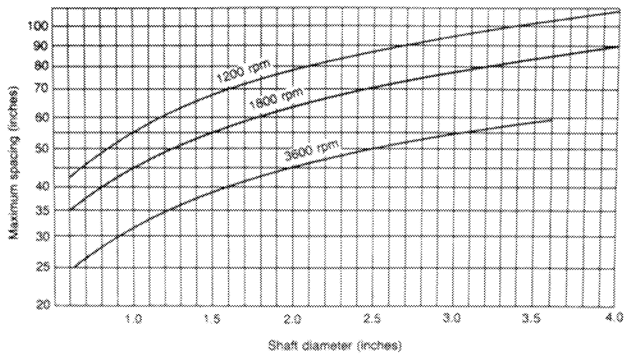


Figure 11. Maximum Spacing Between Shaft Guide Bushings (Vertical Pumps)—API 610-7th Edition.

VIBRATION ANALYSIS OF SEALLESS PUMPS

Many changes are taking place in the pumping industry because of environmental and economic necessities. There are a number of sealless pump designs. All of them have their strong points and some severe limitations as summarized in Table 6 [10].

Table 6. Comparisons of Special Sealless Pumps (Table 4, Modified).

Rule of Thumb — Pump Type Comparisons		
Pump Type	Advantages	Disadvantages
Shaft Sealed	Low First Cost, Small Sized, Highly Efficient Motor	Short MTBF, Seal Leakage
Wet Winding	Zero Leakage Good Reliability Good Motor Efficiency	High Maintenance Costs Insulation Failures, Bearing Failure, Elec Lead Seal Leakage, Large Size/ Weight
Canned Motor	Zero Leakage	High Costs, Lower Motor Efficiency, Damage to Can — Major Leak
Magnetic Drive	Zero Leakage High Motor Efficiency	100-150 HP Limit Currently, Damage to Can — Major Leak

Wet Winding Pumps

Wet winding pumps allow the pumped liquid or a separate dielectric fluid to flow into the stator and the rotor cavities of the motor. The main advantage of this design is zero leakage to the atmosphere. A greater potential for electrical, mechanical and thermal breakdown of the insulation material problems are major disadvantages. Corrosion of the stator and rotor iron and particulate matter in the circulation dielectric fluid are major problems. The motors require a thicker, less space-efficient winding insulation that keeps motor efficiency in the lower eighty percent range. This increases motor size and weight.

A bearing failure is generally evident by an electrical failure, which is a very costly indicator.

Canned Motor Pumps

The major canned motor advantage is that the motor windings are not in contact with the high pressure fluid. This permits motor construction of more conventional materials. Because of increased gap between the stator and the rotor, the efficiency of the motor is reduced by about 10 points from the conventional and about five points from the wetted winding designs.

Part of the pumpage at discharge pressure passes around the rear casing, cooling and lubricating the bearings, sleeves, and the small thrust bearings, and returns to the suction side of the impeller. Canned motor pumps are inherently very quiet during operation.

Shaft deflection is high because the product lubricated sleeve bearings are widely spaced. The rotor diameter is reduced to make the can walls thinner and its length increased to produce the necessary torque. On the good side, there is no need for overhang to accommodate a mechanical seal. The product lubricated bearings are usually made of chemically inert sintered silicon carbide. One of the strong objections to canned pumps is that when run dry, major repairs are necessary. Since the bearings are pumped product lubricated, any dry running can cause bearing failures. If the rotor drops, it can cut through the inner can, which is very thin to reduce the air gap. A major hydrocarbon release will result. To offset this problem, canned motor pumps MUST BE equipped with a leakproof junction box that prevents the introduction of process fluid into the electrical conduit in the event of a stator liner rupture.

Canned motor pumps can be furnished with a bearing wear detector. This device warns that the bearings are close to the maximum allowable wear tolerance and it is time to change bearings. However, this device must penetrate the liquid chamber and increases the likelihood of a product leak.

Some canned motor pump users utilize a programmable microprocessor/based device, designed for protecting the three phase motors. This protection is accomplished by sensing the motor current with three isolated current transformers. Trip level can be set by the user to establish the desired degree of protection for the following hazardous conditions:

- Over current (running to the right of the curve)
- Under current (dry running)
- Rapid cycling (process problem)

This device also can be used to prevent dead heading the pump and for dry run protection.

Canned motor pumps are advertised as requiring less space and having lower initial installation costs than conventional pumps. It is true, canned motor pumps do not require a special base and concrete foundation and/or grouting as normally used on conventional pumps. The installation costs are greater by the time the monitoring devices needed are provided.

Magnetic Drive Pumps

Magnetic drive centrifugal pumps are being installed to fit the increasing need for a leak-free installation demanded by various governmental regulations. The basic design transmits torque from a motor shaft to a pump shaft by a permanent magnetic coupling. As the outer magnet assembly is turned by the motor, magnetic flux passes through the rear casing causing an inner magnet assembly to rotate. The inner magnet assembly is attached to a pump shaft that turns at the same sped as the outer magnet. The design is very close to the canned motor style,except the rotor tends to be shorter between bearings. The magnetic rotor is exposed to the pumped liquid. Mechanical seals are not required because pumpage is contained with in a heavy wall pumping chamber. The magnetic couplings are slip free and generate very little heat. The magnetic drive pump offers the ability to use a conventional motor with its lower costs and higher efficiencies. Shaft deflection is low because the product lubricated sleeve bearing is immediately behind the impeller. There is no need for overhang to accommodate a mechanical seal. The product lubricated bearings are usually made of sintered silicon carbide, a chemically inert material. Few manufacturers offer the bearing wear detector device feature as standard on a magnetic drive pump. About two percent of the pumpage at discharge pressure passes around the rear casing, cooling and lubricating the bearings, sleeves and the small thrust collars and returns to the suction side of the impeller.

A major disadvantage of the design is that the thrust action developed by the impeller must be balanced by the ratio of diameters of the front and back wear rings. There is NO thrust bearing capacity in the product lubricated bearings.

In the submerged motor, the close clearances surround the long thin rotor. The type of hydraulic destabilization or oil whirl encountered in cylindrical journal bearings can be present. The fluid filled clearance space around the pump rotor, either electrical or magnetic, acts like a simple journal bearing with larger than normal clearances. The bearing has a large length over diameter ratio. Because the fluid surrounds the entire rotor,it has the effect of increasing the effective mass of the rotor and making hydraulic vibrations more likely. This will be a major problem as hp levels are raised. Some severe rotordynamics and other problems will be encountered as motor hp are increased above the present levels and the rotors get longer [8].

Need For Monitoring Of Sealless Pumps

The canned motor and the magnetic drive pump share a major problem. If the product lubricated bearing wear allows the rotors to drop and come in contact with the containment cans, a major leak will develop. The utilities industry has experienced several shaft failures, detected and undetected by monitoring equipment, in wetted winding type reactor coolant pumps [11]. Vibration monitoring, bearing wear detectors, rotor positioning sensors, electric current sensors, or other methods of predicting the mechanical condition of a pump becomes vital to all of these types of pumps.

CONCLUSIONS

Vibration monitoring of pumps as a part of a good predictive maintenance program is becoming increasingly important as pressures increase to reduce environmental pollution and damage. Economics rules the world and a vibration program must have a pay out. Better, more reliable analyses of data are needed to evaluate the condition of pumps to achieve these goals. The hydraulic factors must not be ignored because of difficulties in evaluating their causes.

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