REDUCING RELIABILITY INCIDENTS AND IMPROVING MEANTIME BETWEEN REPAIR

by

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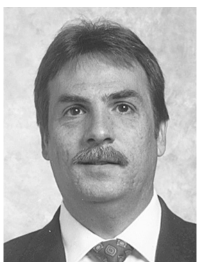

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ABSTRACT

Recently much attention is being given to the total cost of pump ownership. In many cases, for process plants, the cost of unscheduled maintenance can become critical since these costs can impact production runs and result in significant environmental cleanup. The cost of unexpected downtime and lost production can rival energy costs and the cost of replacement parts in its impact (Hydraulic Institute, 2001). The loss may be thousands of dollars per hour and can dwarf all other elements of life cycle cost.

A study was proposed to help a major user achieve its goal of reducing vibration levels for rotating pump equipment in order to reduce reliability incidents and improve mean time between repair. A test program was subsequently conducted to investigate the influence of certain pump hydraulic factors and to determine the effects of speed, impeller diameter, and operating point on vibration reduction and reliability improvements.

This paper quantifies the benefits of lower speed operation on vibration reduction as compared to traditional operation at synchronous speed. It also builds on and expands the investigation into the effects of speed, operating point and impeller diameter on centrifugal pump reliability previously undertaken by Bloch and Geitner (1994), Erickson, et al. (2000), and Budris, et al. (2002).

INTRODUCTION

Looking at the total cost of pump ownership is increasingly being recognized as an important way to reduce plant costs. In 2001 the Hydraulic Institute and Europump published a complete guide to life cycle cost (LCC) analysis for pumping systems. The major components of life cycle cost are initial cost, installation cost, operating cost, and maintenance cost. For practically all applications the initial cost has been determined to be a small percentage of the total life cycle cost.

Typically the cost of power is the largest segment of the cost of ownership. More recently this is especially true as energy costs continue to spiral upwards. This is at a time when process plants are under constant pressure to reduce costs in order to compete in today's global economy. The opportunities for improvements in operating cost by machinery selection are limited since pump and motor efficiencies are close to theoretical maximums. Any significant improvements in operating cost will need to come from process operation. Variable speed operation may present significant opportunity to reduce operating costs. Opportunities abound where pumps are sized for multiple setpoints or where the pump has been oversized to cover limited operation during peak demand. Operation at just a single setpoint may also result in significant opportunity for processes that have variable system conditions.

In many applications, the cost of unscheduled maintenance may be the largest contributor to life cycle cost in process plants. These costs are not only associated with the repair of the equipment but also for interruptions in production that may cost thousands of dollars per hour and can dwarf all other elements of life cycle cost. One Gulf Coast refinery projected savings of $2 million per year by reducing the number of unscheduled equipment shutdowns (White, 2004).

Plants have made significant improvements in mean time between repair (MTBR) through better installation practices, alignment, and operating procedures. However, further improvements can be realized if certain hydraulic factors could be quantified to guide the application engineer during the pump selection process by a more objective approach than just relying on general guidelines.

Bloch and Geitner (1994) have published a method that quantifies the effects of three application factors: speed, impeller diameter (tip clearance), and operating point. These define an overall reliability index that gives a quantitative approach to comparing pumps of similar design or evaluating alternate operating conditions. This method can better guide the application engineer to the best fit solution when confronted with several choices. Bloch and Geitner's method has been the subject of recent papers by

Erickson, et al. (2000), and Budris, et al. (2002), in an attempt to validate the process through experimental testing and field data.

One major end user has put a great deal of emphasis on reducing vibration levels for rotating equipment (pumps) through their precision maintenance program. This end user has made great strides to reduce vibration levels to .075 inch/second peak velocity (unfiltered) through mechanical improvements of pumping components. A focus on tighter impeller balance criteria (ISO G1.0), balanced couplings, specified keyway tolerances, standardized specifications and installation improvements have contributed to lower vibration operating levels.

In an effort to help this end user achieve and accelerate its goal of improving pump reliability, a comprehensive test program was conducted to determine the influence of pump hydraulic factors other than impeller balance on specific reliability indicators.

The test program objective was to quantify the benefits of the following pump hydraulic factors:

- Optimal pump speeds (variable, nonsynchronous motor speeds)
- Impeller diameters (percentage trim ratios)
- Operating flow point (percentage of best efficiency point [BEP] flow)

Benefits or reliability indicators were quantified by the following output measurements:

- Bearing housing temperatures
- Bearing housing vibrations
- Discharge pressure pulsations
- High-frequency detection (HFD) energies

A second objective of the test program was to further validate the Bloch and Geitner and Erickson, et al. (2000), reliability factors for capacity and impeller diameter.

EXPERIMENTAL PROGRAM

Test Models

Two test pump models were selected for this study. They were a 1.5 × 3-13 ANSI B73.1 chemical process pump and a 4 × 6-18 paper stock/process pump. Tables 1 and 2 give the test conditions and design details for the models selected. Both models are existing single-stage, overhung process pumps with semi-open type impellers. The pumps have rolling-element type bearings with oil lubrication. The pumps were fitted with single mechanical seals and Plan 11 seal flush. General sectional assembly drawings are shown for each test model in Figures 1 and 2. These pumps were chosen since they are the two most commonly used sizes within the end user's plant.

Table 1. Test Models and Test Conditions.

Pump Size	Ns	Nss	Suction Energy $X10^6$	Design Capacity Gpm	Design Head Ft	Max Power Bhp	Test Speeds Rpm	Test Impeller Diameters Inches
1.5x3-13 ANSI B73.1 Chemical Process	769	8333	74	550	518	141	3560, 3150, 2700, 2250, 1800	13, 12, 11, 10, 9
4x6-18 Pulp & Paper Process	994	6692	72	1500	286	190	1785, 1575, 1350, 1125, 900	17.5, 16.75, 16, 15.25, 14.5

Table 2. Test Model Design Details.

Pump Size	Casing Type	No. Imp Vanes	Cutwater Gap
1.5x3-13 ANSI B73.1 Chemical Process	Single Spiral Volute	5	0.56" (6.4%)
4x6-18 Pulp & Paper Process	Single Spiral Volute	4 (2V inlet, 4V outlet)	0.30" (4.6%)

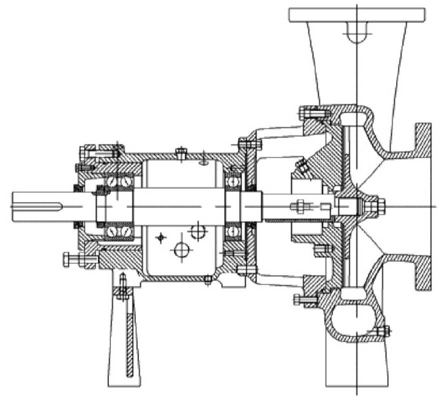

Figure 1. Test Model ANSI B73 Chemical Pump—Typical Construction.

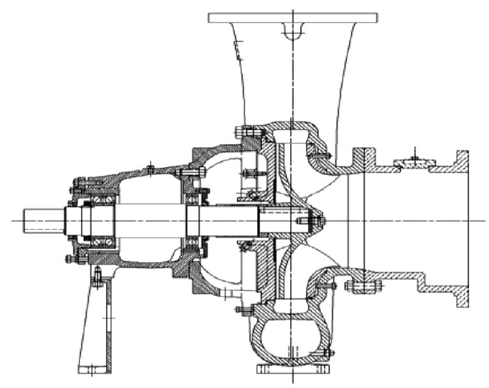

Figure 2. Test Model Pulp and Paper Process Pump—Typical Construction.

Instrumentation

Each test pump was instrumented with a dynamic pressure transmitter to obtain pressure pulsation data. The pressure transmitter was located two pipe diameters downstream of the pump discharge flange and oriented parallel to the pump shaft. Discharge side pressure pulsations were taken at vane pass and all pass frequencies. Bearing housing vibration was measured in both vertical and horizontal planes at the line and thrust bearings with an accelerometer. Axial vibration was also taken at the thrust bearing. Peak vibration velocity was gathered at vane pass and all pass frequencies. A high frequency piezoelectric accelerometer that measured high-frequency detection (HFD) energy was mounted in a plane horizontal to the thrust bearing. Thermocouples were used to measure stabilized temperature for the thrust bearing, oil sump, ambient, and pumpage. The test media was clean water at ambient temperature. The test setup is shown in Figure 3.

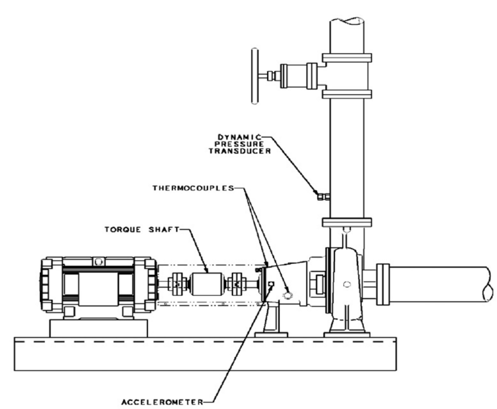

Figure 3. Test Setup.

Test Procedure and Setup

The test setup was in accordance with Hydraulic Institute Standards. Discharge pressure pulsation, HFD energy, and vibration data were all collected on a vibration data processor. The data processor is a vibration signal analyzer that can perform overall (RMS) and spectral analysis for one signal input at a time. Pressure pulsation (psi peak-to-peak) data were recorded at vane pass and for an overall frequency range of 0 to 1000 Hz. Bearing housing peak vibration (in/sec) data were also recorded at vane pass and for an all pass frequency range of 0 to 1000 Hz. HFD data (g's peak) was recorded over a frequency range of 5 kHz to 60 kHz.

All data were recorded at ten equally spaced flow points between closed valve and runout flow. For the 1.5 × 3-13 size data were not taken at closed valve due to excessive heating. Each pump was tested at the speeds and impeller diameters shown in Table 1. Stabilized bearing temperature data were recorded for each speed and impeller diameter.

For the 4 × 6-18 size the throttle valve was located at least 10 pipe diameters from the discharge flange. The throttle valve for the 1.5 × 3-13 pump was located well downstream of the pump in the discharge header. Throttle valve position was recorded for each constant speed test. For the 4 × 6-18 pump backpressure was applied downstream of the throttle valve to minimize noise and vibration due to possible cavitation across the throttle valve.

Pump performance (head, flow, and power), discharge pressure pulsation, HFD, and vibration data were recorded for each impeller diameter and operating speed combination at 10 equally spaced flow points. A torque bar was used to determine power. For each test model and impeller diameter a variable speed test with a fixed system curve was conducted with the throttle valve full open and the backpressure valve adjusted to achieve the maximum test flow at maximum test speed. All baseline testing was conducted with net positive suction head available (NPSHA) values 10 percent greater than the required pump net positive suction head required (NPSHR) at each flow point. For comparison purposes, two additional tests were conducted with NPSHA values equivalent to two times the NPSHR. NPSHR testing was conducted for five flows at maximum speed for each impeller diameter.

Prior to testing, each impeller was balanced to ISO G1.0. For each test model the same instrumentation, piping, piping supports, coupling, and bedplate setup were used throughout testing. The pump and motor were mounted on rigid blocks bolted to a laboratory bedplate as it was not practical to use a grouted-in baseplate. Torque bars were aligned to within 0.002 to .003 inch radial and axial runout. Each test setup was checked for soft foot and pipe strain.

NORMALIZATION OF DATA AND INDICATORS

The discharge pressure pulsation, HFD, and vibration data were normalized based on the equation:

$$R = \left(1 - \left(V / V\,\max\right)\right) + C \qquad \textbf{(1)}$$

where:

R = Relative indicator of reliability
V = Data point being analyzed
Vmax = Maximum data value
C = A constant added to set the peak value of R equal to 1

The R value in Equation (1) is a relative reliability number between 0 and 1. A value of zero does not necessarily indicate zero reliability but rather is intended to discourage application of the pump at these conditions. Similarly a value of 1 does not indicate infinite reliability but is intended to be a relative indicator of the best operating conditions for a given pump. Since the mechanical design of a pump can also affect reliability these values should not be used to compare pumps of different design or manufacturer. It is intended to compare alternate selections of a given design or the effect of alternate operating conditions. These relative reliability factors can offer guidance to the user in answering questions such as: How will the reliability of my pump be affected by operating it at a higher or lower speed or using a smaller or larger impeller diameter? Or how will a different operating point affect the life of my pump?

Impeller Diameter Indicators

The reliability indicator chosen for impeller diameter was vane pass pressure pulsation and high frequency vibration detection. Dynamic deflections due to impeller vane-cutwater pressure pulsations can affect mechanical seal life. These pressure pulsations cause a radial shaft deflection each time a blade passes the volute cutwater. This repetitive motion results in a translating motion at the seal face rather than a steady pure rotational motion for which the seal faces were optimally designed. This motion can tend to pull in abrasives across the seal faces, which can increase wear. The smaller the gap between the impeller vane and casing tongue, the less chance there is for the fluid angles, which become mismatched at off design flow, to adjust to the cutwater angle. This can result in significant pressure pulsations.

HFD is another good indicator of the effect of impeller diameter. HFD is a measure of the intensity of the energy generated by impacts in a pump. These impacts can be caused by mechanical and/or hydraulic phenomena and can affect pump reliability. Some examples of these impacts are metal-to-metal contact, cavitation, suction recirculation, discharge recirculation, etc. For impeller diameter reliability the same Vmax was used for all impeller diameters for a given test model. This will determine the most reliable impeller diameter/speed combination.

Operating Point (Capacity) Indicator

As the pump is operated further to the left or right of the BEP flow turbulence is created in the pump casing and impeller, which increases hydraulic loads on the pump shaft and bearings. These increasing and unsteady loads can have the same deleterious effect on mechanical seal life as discussed previously. Vibration at the pump bearings was the reliability indicator selected for operating point since vibration level will increase as operation moves away from the best efficiency flow. It was found that both pressure pulsation and HFD generally tracked the shape of the vibration data at each capacity. For capacity reliability Vmax was determined for each impeller diameter. This will indicate the most reliable speed/operating point for a given impeller diameter.

RESULTS AND DISCUSSION

Operating Speed

The effect of operating speed directly affects pump reliability through:

- Increased wear due to excessive heat generation between seal faces through rubbing contact.
- Excessive heating, which reduces viscosity and lubricity of the bearing lubricant and subsequent life of the bearing.
- Excessive temperature, which increases the oxidation rate of the lubricant resulting in decreased lubricant life and increased corrosion potential in bearings.
- Increased speed can also accelerate the wear of hydraulic components in a pump if abrasive particles are present.

Figure 4 shows the effect of reduced speed operation on lowering thrust bearing housing skin temperature for the 4 × 6-18 size. Note this comparison is based on a maximum impeller diameter. Comparison tests were also made with varying impeller diameters operating at maximum speed for the 4 × 6-18 size as shown in Figure 5. Note when operating at higher speeds with large impeller trim diameters rolling-element bearings can become too lightly loaded if no preload exists. Skidding of the rolling elements can cause unexpected increases in bearing temperature and accelerated wear. This is clearly shown in Figure 6, which is a replot of the same data shown in Figure 5. The curve for impeller diameter ratio shows the effect of trimmed impeller diameters versus temperature when operating at maximum speed. In this plot a value of one represents maximum diameter impeller and zero represents a minimum diameter impeller. Note the temperature increase for the minimum diameter impeller is nearly the same as that for the maximum impeller diameter. The intermediate diameters show the lowest temperature rise, which would indicate that the bearing loads have been lowered but are still sufficiently high to prevent skidding. The plot of speed ratio (actual speed/maximum speed) is based on a maximum impeller diameter operating at varying speeds. In this plot since the speed is decreasing with load, there is no skidding and bearing temperatures decrease steadily with decreasing speed.

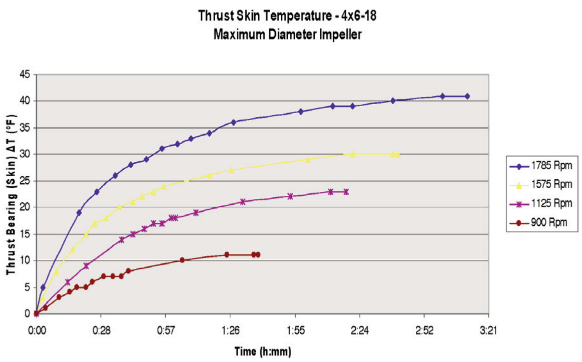

Figure 4. Thrust Bearing Skin Temperature Versus Speed.

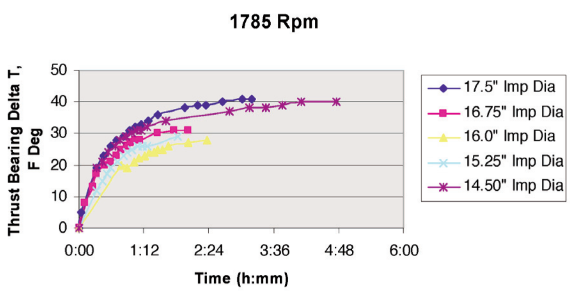

Figure 5. Thrust Bearing Housing Temperature Versus Impeller Diameter.

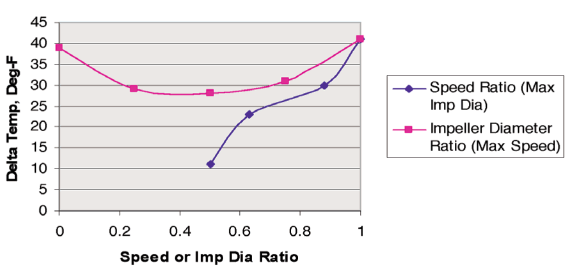

Figure 6. 4 × 6-18 Stabilized Thrust Bearing Housing Temperature Versus Speed and Impeller Diameter Ratio.

Figure 7 was presented by Erickson, et al. (2000), and shows the contribution of the aforementioned speed related reliability criteria on a nondimensional relative life scale between zero and unity where zero represents least expected life and unity represents the best possible life for a given application. The combined curve shown in Figure 8 is a best fit curve for all data from Figure 7. It is near linear and agrees closely with the original work done by Bloch and Geitner (1994). The erosion, seal wear, and oil degradation curves shown in Figure 7 are based on theoretical data. The bearing temperature curve represents adjusted bearing life based on a bearing lubricant life adjustment factor (a3), which is well documented in ANSI/AFBMA Standard 9 and bearing manufacturer's literature. Therefore no changes are recommended to the original work for the operating speed reliability factor as presented by Bloch and Geitner (1994) in Figure 8.

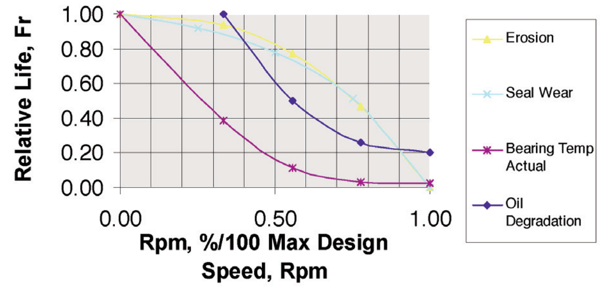

Figure 7. Operating Speed Characteristics.

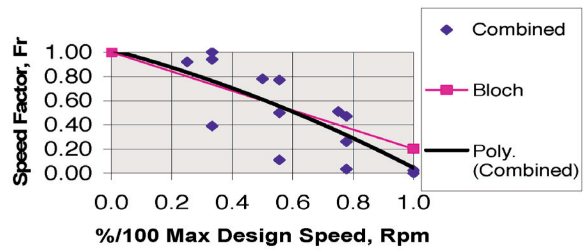

Figure 8. Speed Reliability Factor Comparison.

Operating Point (Capacity) Reliability

As mentioned previously, pump operating point can affect reliability through the turbulence created in the pump casing and impeller as the operating point moves farther away from the best efficiency flow. As discussed by Budris, et al. (2002), operation at reduced flow rates may cause the pump to operate in the suction recirculation zone. This can lead to excessive noise, vibration, and possible cavitation damage, particularly in high suction energy pumps.

Suction energy (Equation 2) is a measure of the liquid momentum in the suction eye of the impeller. It is a function of the mass and velocity of the liquid in the inlet. It can be approximated as follows:

$$Suction\ Energy(S.E.) = De \times N \times S \times s.g. \tag{2}$$

where:
De = Impeller eye diameter, inch
N = Pump speed, rpm
S = Suction specific speed, defined as: $((\text{rpm}) \times (\text{gpm})^{0.5})/(\text{NPSHR})^{0.75}$
s.g. = Specific gravity of the liquid being pumped

Typically for end suction pumps the high suction energy threshold occurs at a value of 160×10^6 and very high suction energy begins at 240×10^6. By inspection of Equation (2) lower operating speeds will lower the suction energy value. This also improves the NPSH performance. More on this subject can be found in Budris and Mayleben (1998). Both test models in this study have values in the range of 74×10^6 and clearly fall into the low suction energy category. Although the two test models are classed as low suction energy pumps, comparison testing was carried out at maximum impeller diameter and maximum speed using an NPSHA/NPSHR ratio of 110 percent and 200 percent for both test models. Unless specified otherwise all other testing in this study was conducted with an NPSHA/NPSHR ratio of 110 percent. By Hydraulic Institute definition the NPSHR of a pump is the NPSH value that will result in a 3 percent drop in the total head of a pump due to the blockage of flow caused by cavitation bubbles between the impeller vanes. It should be noted that the incipient point of cavitation, i.e., where bubble formation first begins, can require an NPSHA/NPSHR ratio of from 2 to 20 times to fully suppress it. This depends on pump design and operating point. The higher values are normally associated with high suction energy, high specific speed pumps with large impeller inlet areas. Since most applications are operating with a substantially lesser NPSHA/NPSHR ratio than mentioned above, it can be concluded that most pumps are operating in some degree of cavitation. However, it is the amount of energy associated with the collapse of the cavitation bubbles that causes noise, vibration, and damage from cavitation due to an insufficient margin.

Figure 9 shows the results for the 4 × 6-18 pump. In this test model suction recirculation is shown by a peak at 28 percent of the best efficiency flow. Although severity levels are low and cavitation damage is unlikely, Figure 9 does show there is some benefit to increasing the NPSHA/NPSHR ratio to reduce vibration when operating in the recirculation zone for low suction energy pumps.

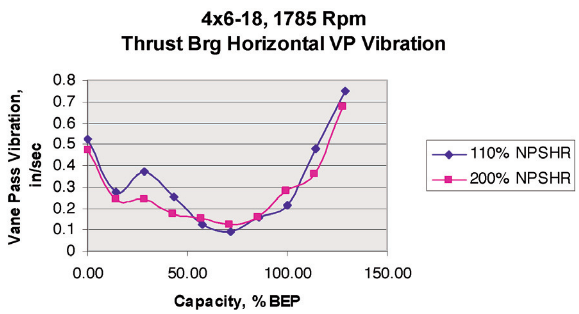

Figure 9. Effect of NPSHA on a Low Suction Energy Pump.

As previously discussed vibration was selected as the indicator for operating point (capacity) reliability. For each test model both overall and vane pass vibration were measured for each impeller diameter and speed. Overall vibration is affected by impeller balance, coupling alignment, and test setup. All of these criteria were tightly controlled as discussed previously under test setup; therefore, data are presented in both formats throughout this paper. In nearly all cases the point of highest measured vibration occurred at the thrust bearing horizontal plane as this is the least stiff plane. The forces working in the vertical plane are in tension or compression and as such have a higher stiffness than in the horizontal plane.

The 1.5 × 3-13 (Figure 10) and 4 × 6-18 (Figure 11) test models indicate that the lowest point of vibration occurs in the capacity range of 75 to 85 percent of the best efficiency flow. Also note that once the speed drops to 75 percent of nominal motor speed the vibration level is reduced substantially and operability increases. Figures 12 and 13 show the normalized HQ curves for each test model at maximum impeller diameter for reference.

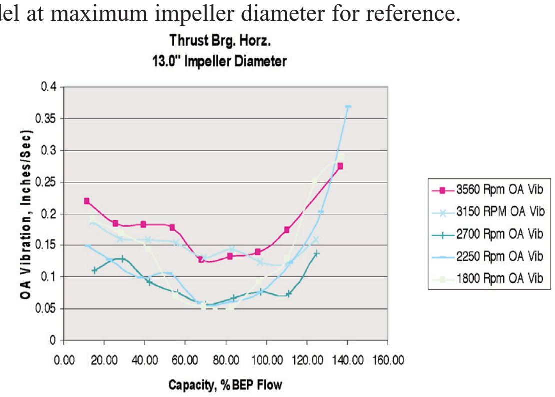

Figure 10. Overall Vibration Test for 1.5 × 3-13 Pump for Maximum Impeller Diameter.

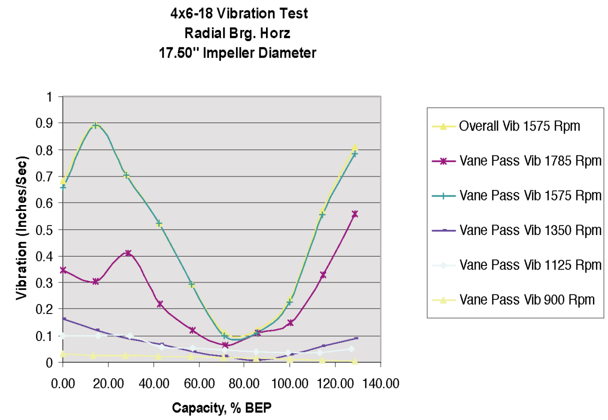

Figure 11. Vane Pass Vibration Test for 4 × 6-18 Pump for Maximum Impeller Diameter.

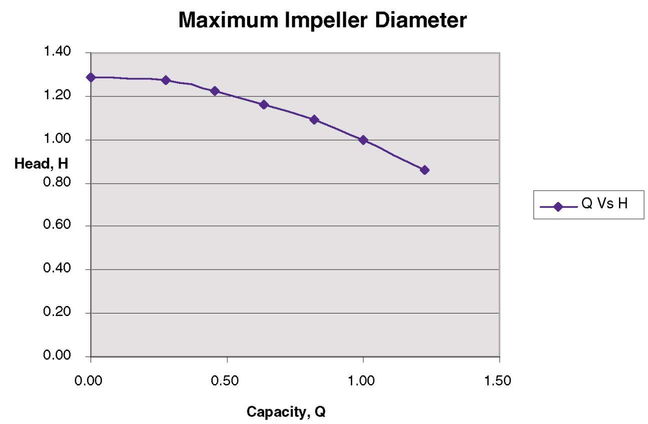

Figure 12. 1.5 × 3-13 Normalized HQ Curve.

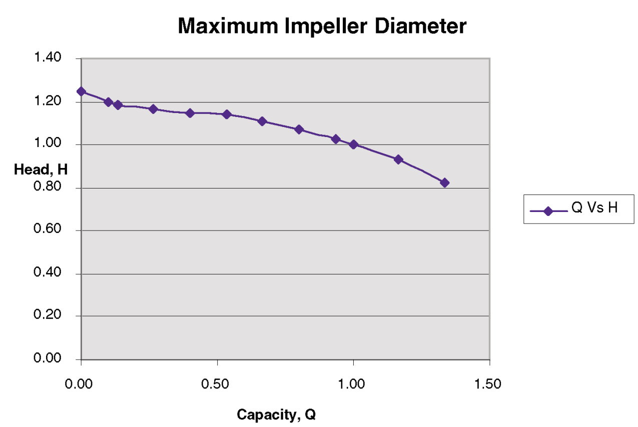

Figure 13. 4 × 6-18 Normalized HQ Curve.

It was found for the 4 × 6-18 size an impeller structural resonance occurred at 105 Hz, which corresponds to a shaft speed of 1575 rpm as shown in Figure 11. This resonance has a narrowly defined bandwidth of under ± 5 Hz about the resonant frequency and is not observed at motor synchronous speed. A "ring" (or modal-bump) test was conducted on the bearing frame/rotor assembly (less casing) out of the system by mounting an accelerometer on the impeller vane (horizontal and vertical planes were tested) and "ringing" the impeller. The vibration data processor was used to analyze the frequency response, which indicated that the resonance related to the stiffness of the impeller vane in the vertical and horizontal direction. This is an area of variable speed operation that the user must be aware of; but note variable frequency drive (VFD) firmware can lock out these speed bands during operation. It should be noted that data obtained for the 4 × 6-18 1575 rpm test was not used in the vibration reliability analyses that follow.

Figure 14 is an overall vibration comparison for various speeds and impeller diameters for the 4 × 6-18 size. It shows that vibration levels drop and operability increases for all impeller diameters when speed is lowered to 75 percent of maximum rated speed. A further decrease in speed reduces vibration level below 0.1 in/sec.

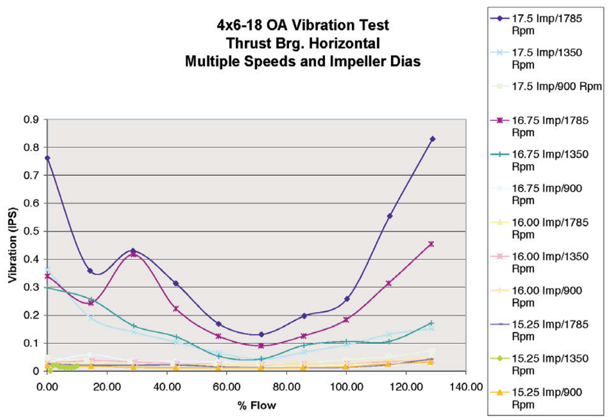

Figure 14. Overall Vibration 4 × 6-18: Multiple Speeds and Impeller Diameters.

Figure 15 shows a plot of vibration versus speed ratio and impeller diameter ratio at best efficiency flow. For the impeller diameter ratio a value of one represents maximum impeller diameter and zero represents minimum impeller diameter. The speed ratio is the test speed divided by the maximum rated speed. The impeller trim ratio comparison shows how vibration varies with impeller trim when the speed is held constant at 1785 rpm. The speed ratio comparison shows how vibration varies with speed for a maximum impeller diameter. Data for both comparisons were taken at best efficiency flow. The plot clearly shows that varying speed is a more effective method for vibration reduction than trimming an impeller.

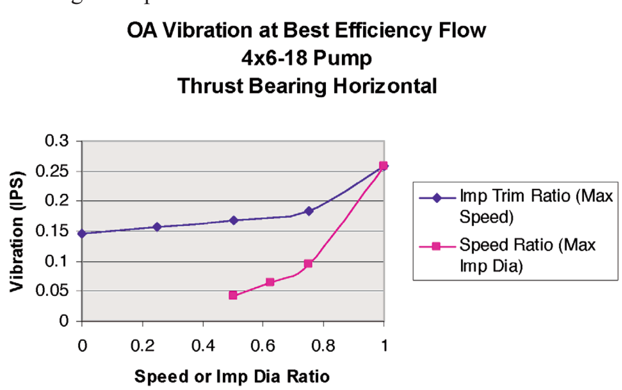

Figure 15. 4 × 6-18 Overall Vibration Versus Speed Ratio and Impeller Diameter Ratio.

Vibration data were normalized according to the method previously described and compared to the Bloch and Geitner (1994) reliability factors. Figure 16 shows close agreement between Bloch and Geitner (1994) and the results for the 4 × 6-18 test model for maximum impeller diameter. Note an offset is shown at peak reliability between the test data in this study and the Bloch and Geitner (1994) method. This corresponds to the lowest point of vibration, which occurred at 75 percent of best efficiency flow. Similar results were obtained for the 1.5 × 3-13 test model.

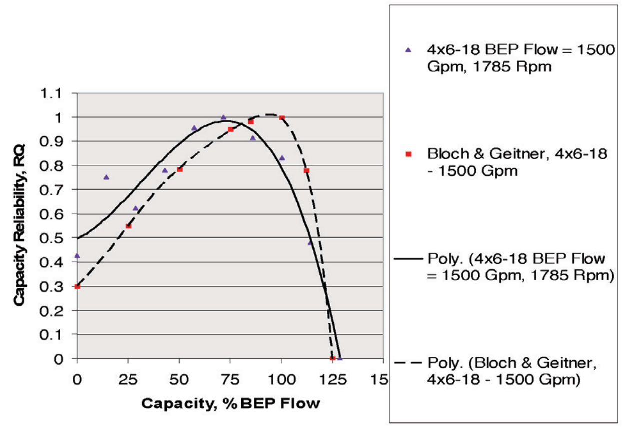

Figure 16. Comparison of Capacity Reliability for 4 × 6-18 at Maximum Impeller Diameter.

Note the Bloch and Geitner (1994) method sets the peak value of reliability equal to one for each impeller diameter without considering the effect of lower speed operation. To consider the effect of reliability on lower speed operation the pump operating range was evaluated for each impeller diameter over the entire speed range. This approach is validated by the increased operability and reduction in vibration attributed to lower speed operation shown in Figures 10, 11, and 14.

Figure 17 shows the effect of speed on the capacity reliability factor (all impeller diameters) for the 4 × 6-18 pump. It is fairly intuitive for this pump that reliability should increase as speed is reduced since the useable operating range has increased and vibration has decreased as shown in Figures 11 and 14. To put it differently, the energy density of the machine decreases as speed decreases. Power decreases by the third power of the speed reduction but the machine structure remains unchanged. Although there is a reduction in vibration attributed to impeller diameter alone it is accounted for in the impeller diameter reliability chart.

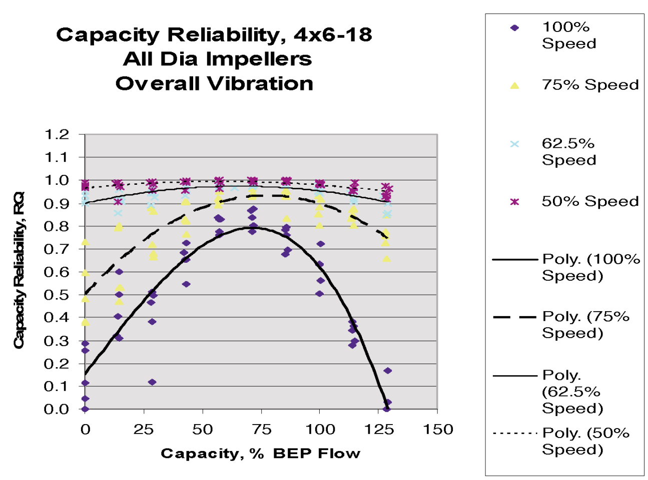

Figure 17. 4 × 6-18 Capacity Reliability Versus Speed.

Plots similar to Figure 17 showed similar trending for the 1.5 × 3-13 test model but with a larger scatter in test data often attributable to low specific speed designs, which are sensitive to changes in geometry and clearance. Capacity reliability factors for both test models were then plotted together in a similar format. Adjusted trend lines for these data are compared to Bloch and Geitner's (1994) original capacity reliability (dotted lines) chart in Figure 18. As can be seen from the result separate speed lines (solid lines) have been added to reflect reliability increases attributable to lower speed operation found in this study.

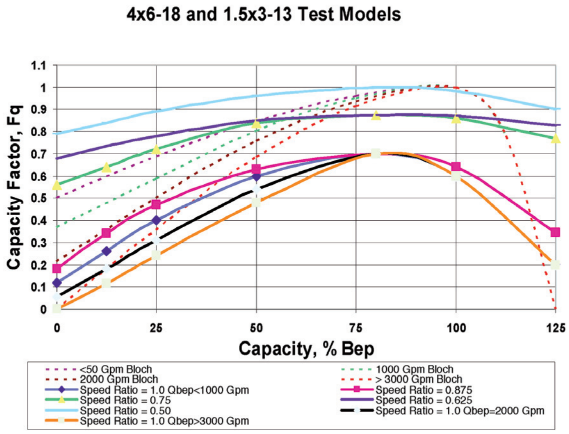

Figure 18. Capacity Reliability Comparison for Test Models Versus Bloch and Geitner (1994).

Impeller Diameter Reliability

As discussed previously, dynamic deflections caused by vane pass pressure pulsations can affect bearing and mechanical seal life. High frequency detection is a good indicator of mechanical and hydraulic conditions in a pump. For this reason discharge pressure pulsation and HFD were selected as the indicators to evaluate the benefits of various impeller diameters.

It was found for both test models that vane pass pressure pulsations decreased with impeller trim, i.e., the larger the impeller to cutwater clearance, the lower the pressure pulsation values. Figure 19 (4 × 6-18 vane pass pressure pulsation versus impeller diameter) and Figure 20 (1.5 × 3-13 vane pass pressure pulsation versus impeller diameter) show that for impeller diameters less than maximum the pressure pulsations can be substantially reduced.

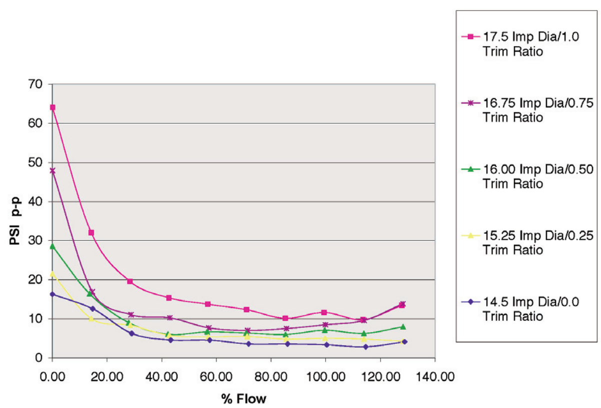

Figure 19. 4 × 6-18 Vane Pass Pressure Pulsation Versus Impeller Diameter at 1785 RPM.

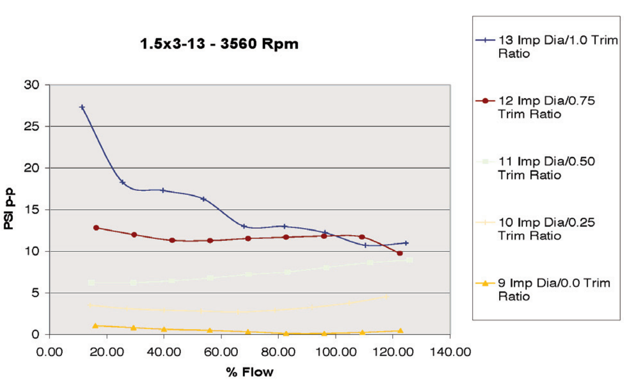

Figure 20. 1.5 × 3-13 Vane Pass Pressure Pulsation Versus Impeller Diameter.

The vane pass pressure pulsation data were normalized based on the best impeller diameter. As explained in the "NORMALIZATION OF DATA AND INDICATORS" section for impeller diameter reliability a single Vmax was determined for all impeller diameters, capacities, and speeds for a particular test pump. Relative reliability was then calculated using this Vmax value. This would determine the most reliable (best) impeller diameter and speed combination for a given test pump. Figure 21 shows relative reliability based on vane pass pressure pulsation for both test models. These data show that the most reliable (best) impeller speed combination occurs for minimum impeller diameter and minimum speed. Certainly from an economic standpoint this is not very practical; however, the chart does suggest that an impeller trim ratio (Equation 3) between 0.5 to 0.75 is a reasonable range for reducing the effects of pressure pulsation.

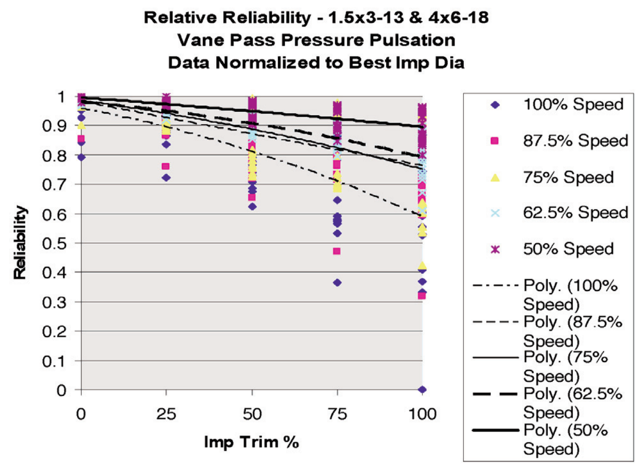

Figure 21. Vane Pass Pressure Pulsation Relative Reliability.

Impeller trim ratio is defined as:

$$\left(Actual\ Dia - Min\ Dia\right) / \left(Max\ Dia - Min\ Dia\right) \qquad \textbf{(3)}$$

Figure 21 also shows that the greatest improvement in reliability occurs for larger impeller diameters operating at lower speeds.

Figure 22 (HFD based on speed for the 4 × 6-18 test model) shows that HFD decreases with speed for the maximum impeller diameter. Similar results were found for all other impeller diameters and also for the 1.5 × 3-13 test model. Note the HFD test using 110 percent NPSHA (1785 rpm) shows a strong peak below 20 percent BEP flow. This is where the relative severity is highest in the suction recirculation zone. Also note the curve for 200 percent NPSHA (1785 rpm) indicates that this peak has been suppressed by the increased suction pressure.

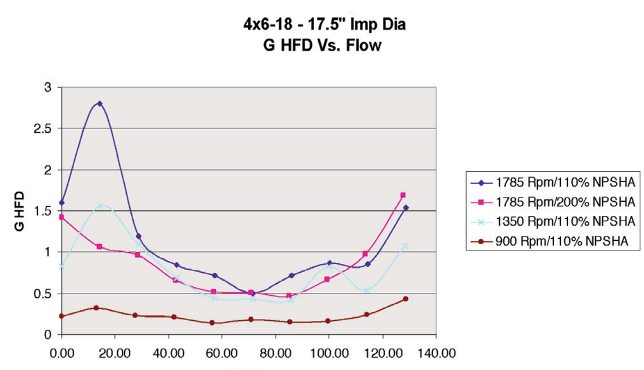

Figure 22. HFD (Thrust Bearing Horizontal) Versus Flow for 4 × 6-18 Test Model Based on Speed.

Figure 23 (HFD based on impeller diameter for the 4 × 6-18 pump) does not indicate any clear trend with impeller diameter variation. The data for the 1.5 × 3-13 test model showed quite a bit more scatter but did not show a clear HFD trend either. Note the work done under Erickson, et al. (2000), did show a clear trend; in all cases HFD values increased as the impeller diameter was reduced. In the Erickson, et al. (2000), study two of three test models were of high suction energy design. One explanation why Erickson, et al. (2000), saw increased HFD values with decreased impeller diameter is that the severity levels in high suction energy pumps result in greater impacts and therefore higher HFD values. In pumps with larger impeller-cutwater gaps the damaging effect of pressure pulsations is minimized but the effects of suction and discharge recirculation can become more of an issue (Erickson, et al., 2000), especially if vane overlap is lost for smaller impeller trim ratios. Figure 24 shows a trimmed impeller where vane overlap is lost. In this case discharge recirculation can extend directly into the suction eye since the adjacent vane is not there to block it.

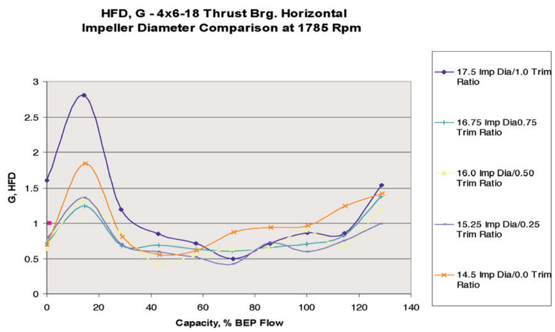

Figure 23. HFD Versus Flow for 4 × 6-18 Test Model Based on Impeller Diameter.

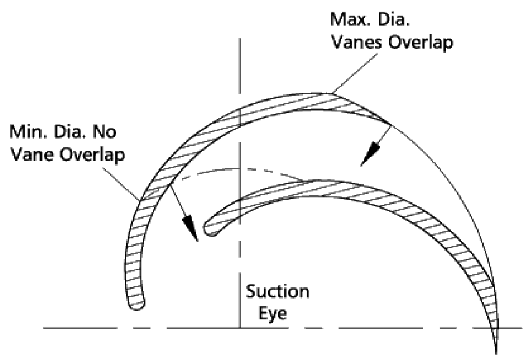

Figure 24. Impeller Showing No Vane Overlap at Minimum Diameter.

Impeller diameter reliability curves based on percentage speed for all pressure pulsation and HFD data for the 1.5 × 3-13 and 4 × 6-18 test models were plotted. HFD data were normalized by the same method as the pressure pulsation data. The data are shown plotted in Figure 25 (solid lines) along with the original impeller diameter reliability chart (dotted lines) by Bloch and Geitner (1994). The work done by Erickson, et al. (2000), and Bloch and Geitner (1994) indicated that an optimum impeller diameter trim should exist since pressure pulsations decrease with impeller diameter and HFD increases with impeller diameter. It was reasoned that the maximum reliability occurs somewhere within the trim range. This optimum impeller trim is shown at 75 percent in the original Bloch and Geitner (1994) chart. The results of this study showed that vibration levels continued to drop for smaller diameter impellers operating at reduced speed. Therefore no optimum trim diameter (other than minimum diameter) was found. This is likely attributed to the test models being of low suction energy design in this study.

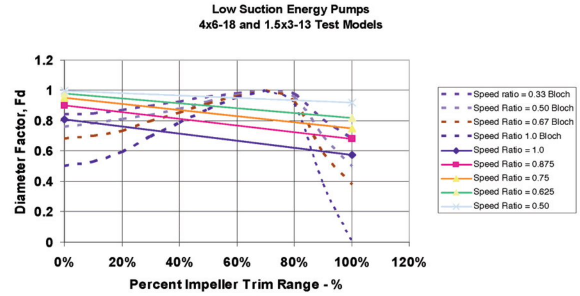

Figure 25. Impeller Diameter Reliability Comparison for Test Models Versus Bloch and Geitner (1994).

Budris, et al. (2002), proposed a method for predicting reliability in high suction energy pumps by the addition of an NPSH margin reliability factor but did not have a method for predicting reliability for low suction energy pumps for failures caused by factors other than cavitation. The test data in this study suggest that the original Bloch and Geitner (1994) curve is valid for high suction energy pumps based on the Erickson, et al. (2000), test data but that a separate chart is required for low suction energy pumps (Figure 25).

Variable Speed Operation

It has been common practice in industry to oversize pumps to satisfy peak demand although this condition may be required only a small percentage of time. Other reasons may be to add safety factors to cover system losses that may be difficult to define or for possible future expansion. In order to meet demand in constant speed systems control valves are used that use some of the available pumping energy to control the process (Stavale, 2000). The amount of consumed energy will vary depending on the method of control, valve sizing, and operating point. In addition operation at off-peak design and oversizing negatively affects the reliability of the pumping equipment due to excessive hydraulic forces (both static and dynamic) as shown in this study. Cavitation and wear can also impact pump reliability.

Since changes in impeller diameter and changes in speed generally follow the same affinity laws for flow and head, a pump's operating point can be met by one of several pump configurations:

- A constant speed system with maximum diameter impeller (oversized) with significant control valve throttling
- A constant speed system with trimmed impeller to suit the operating point
- A maximum impeller operating at reduced speed
- A trimmed impeller operating at reduced speed

Figure 26 shows how overall vibration varies for each of these configurations for the 1.5 × 3-13 test pump using the closest approximations for the speeds and impeller diameters tested.

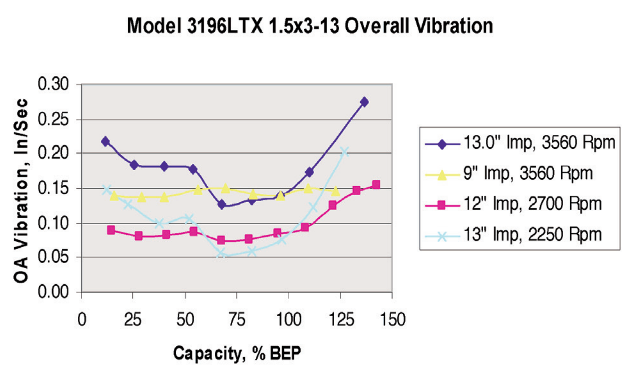

Figure 26. Overall Vibration for Different Pump Configurations.

For the fixed speed arrangement there are two configurations shown:

- A maximum impeller diameter (13 inches) operating at 3560 rpm. Note at capacities to the left and right of BEP vibration levels begin to pickup. This results in a fairly narrow sweet spot (70 to 100 percent BEP) to obtain low vibration levels.
- A minimum impeller diameter (9 inches) operating at 3560 rpm. This configuration is effective at reducing vibration levels to the left and right of BEP, although vibration levels have not been optimized.

Note the operating point can be met in a variable speed system for the following two configurations:

- A trimmed impeller diameter (12 inches) operating at 2700 rpm. The 12 inch impeller diameter operating at reduced speed achieves the lowest vibration levels over the widest operating range.
- A maximum impeller diameter (13 inches) operating at 2250 rpm. This selection also gives good results over a wide operating range.

Table 3 below shows the reliability factors for speed, impeller diameter and flow for each of these conditions as well as the reliability index. The reliability index (RI) shown in Equation (4) (Erickson, et al., 2000) is the product of the speed, impeller diameter, and capacity reliability factors. It is calculated as follows:

$$RI = Fr \times Fd \times Fq \qquad \textbf{(4)}$$

where:
Fr = Speed reliability factor
Fd = Impeller diameter reliability factor
Fq = Capacity reliability factor

Table 3. Reliability Factors for Pump Configurations as Shown by Figure 26.

Impeller Dia	Imp Trim Ratio	Speed rpm/%Max	Fr Speed	Fd Imp Dia	Fq 25% Bep Flow	Fq 100% Bep Flow	Fq 125% Bep Flow	RI 25% Bep	RI 100% Bep	RI 125% Bep
13.0”	1.0	3560/100	0.2	0.58	0.4	0.6	0.2	0.05	0.07	0.02
9.0”	0.0	3560/100	0.2	0.80	0.4	0.6	0.2	0.06	0.10	0.03
12.0”	0.75	2700/75	0.42	0.80	0.72	0.86	0.78	0.24	0.29	0.26
13.0”	1.0	2250/62.5	0.5	0.81	0.78	0.87	0.82	0.32	0.35	0.33

Similar to the reliability factors, values of RI range from zero to one with higher values indicating greater reliability. The index does not take into account design considerations; therefore it cannot be used to compare pumps of different design. It is intended to compare alternate selections of a given design or alternate operating conditions. Inspection of Table 3 shows that the 12 inch diameter/2700 rpm and 13 inch diameter/2250 rpm selections are best choices from a reliability standpoint.

Overall vibration was measured for a variable speed test and compared to a constant speed system with throttle valve for the 4 × 6-18 test model as shown in Figure 27. During the variable speed test the system was fixed, i.e., there was no throttling of the control valve to obtain changes in flow. Prior to the variable speed test, the control valve was opened wide and the backpressure valve was adjusted to obtain maximum flow at maximum speed. Subsequent changes in flow were achieved by varying speed only. It was found that vibration was reduced substantially when compared to the constant speed test for speeds below 75 percent of nominal motor speed. Note the results of the variable speed system shown in Figure 27 are valid for flow control systems having friction head only. In systems with relatively flat head-capacity curves and high static head a change in flow no longer becomes proportional to speed as it does in all friction systems. Caution must be exercised in these types of systems as a small turndown in speed could give a large change in flow and may result in operation below minimum flow or deadheading. These systems may be better controlled by proportional-integral-derivative (PID) torque rather than the more traditional PID speed. PID torque results in more stable control with less oscillation when adjusting to system changes where a small change in speed can be accompanied by a relatively large change in flow as shown in Figure 28.

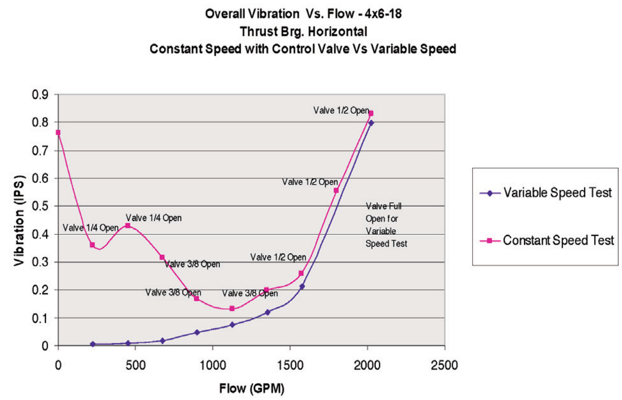

Figure 27. Overall Vibration for Constant Speed and Variable Speed Condition.

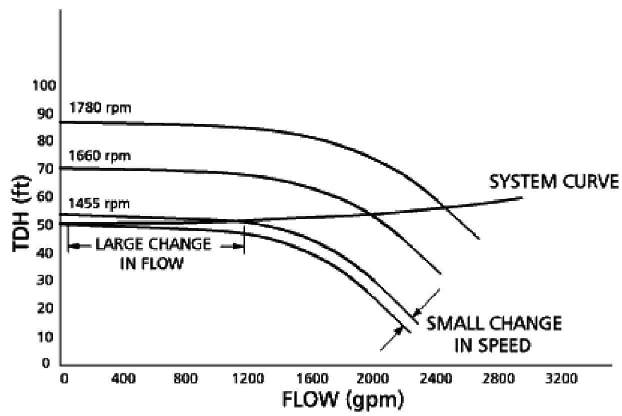

Figure 28. Pump System Showing Small Change in Speed with Large Change in Flow.

CONCLUSIONS

The study was prompted by a need to provide critical data to a major end user for use in reducing reliability incidents and improving mean time between repair. The conclusions of the experimental test program are as follows:

- It was found that for a given speed the point of minimum vibration occurred at 75 to 85 percent of the best efficiency point flow rather than 100 percent BEP flow. Pressure pulsation data followed a similar trend.
- It was found that the operating point (capacity) reliability factors varied significantly with operating speed. Results showed that vibration levels dropped significantly with speed over the entire operating range of the pump.
- The results of this study show that reliability continuously improved for smaller impeller diameters operating at lower speeds. This is likely attributed to the test models being of low suction energy design in this study.
- The reliability increase attributable to reduced speed is greatest at maximum diameter impeller and least at minimum diameter impeller.
- The reliability increase attributable to reduced impeller diameter is greatest at maximum speed and least at minimum speed.

- The increase in reliability achieved by reducing speed alone (maximum to minimum speed) is greater than by reducing impeller diameter alone (maximum to minimum impeller).

- For the models tested a 50 percent to 75 percent impeller diameter trim ratio operating at 62.5 percent to 75 percent of nominal speed optimizes both reliability and application range.

- In addition to the benefits of variable speed operation shown in this study, it is clear that systems that do not have variability may also benefit from an optimum speed (nonsynchronous)/impeller diameter combination to lower vibration and improve reliability.

- An impeller structural resonance was found at 105 Hz for the 4 × 6-18 test model. This corresponded to a shaft speed of 1575 rpm. The 1575 rpm data were not used in the 4 × 6-18 analysis.

- There was no trend found for HFD versus impeller diameter for either the 4 × 6-18 or 1.5 × 3-13 test models. This is likely attributed to the test models being of low suction energy design.

- For the models tested the study showed that overall vibration was lowest for a combination of trimmed impeller diameter operating at reduced speed.

- A comparison of the capacity reliability curves between Bloch and Geitner (1994) and the two test models in this study at maximum impeller diameter and maximum speed showed good agreement except for the capacity offset mentioned above (75 to 85 percent versus 100 percent BEP flow).

- There was no optimum impeller trim found in this test study. Bloch and Geitner (1994) indicated peak reliability occurs at 75 percent of the impeller trim range. The results of this study show that reliability continuously improved for smaller impeller diameters operating at lower speeds. This is likely attributed to the test models in this study being of low suction energy.

RECOMMENDATIONS

The author recommends that this work continue. In particular, further investigation is needed in the following areas:

- Since it was found that operating point (capacity) reliability factors varied significantly with operating speed, suggested changes have been made to the capacity reliability chart to account for lower speed operation achievable in variable or nonsynchronous speed applications.

- The study done by Erickson, et al. (2000), did suggest there was an optimum impeller diameter between 60 to 80 percent of the trim range. However, the test models in the Erickson, et al. (2000), study were of mostly high suction energy design. It is suggested that Bloch and Geitner's (1994) impeller diameter chart be used for pumps of high suction energy design. A separate chart for low suction energy pumps has been proposed.

- Additional experimental data are required to further validate and refine the proposals made for the capacity and impeller diameter reliability charts in this study as well as Bloch and Geitner's (1994) charts. Data are required over a wide range of pump types, specific speeds, and suction and discharge energy designs. Discharge energy is defined by Budris, et al. (2002), as a function of total head, specific gravity and specific speed.

- Testing of higher specific speed low suction energy pumps is required. The scatter in the 1.5 × 3-13 data may be attributed to the fact that it is a low specific speed design. Additionally, more testing is required to determine HFD trending versus impeller diameter.

- Although lab data are important, the reliability factors presented in this study can only be truly validated by user experiences and data from the field. However, it is mandatory that any field data be cleansed by factors other than hydraulic selection, such as operator error, hard-to-handle liquids, duty cycles, system problems, and the mechanical design of the pump. It is highly recommended that further studies be undertaken to correlate these experimental findings with field data.

REFERENCES

Bloch, H. P. and Geitner, F. K., 1994, *An Introduction to Machinery Reliability Assessment,* Houston, Texas: Gulf Publishing Co.

Budris, A. R. and Mayleben, P. A., 1998, "Effects of Entrained Air, NPSH Margin, and Suction Piping on Cavitation in Centrifugal Pumps," *Proceedings of the Fifteenth International Pump Users Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 99-108.

Budris, A. R., Erickson, R. B., Kludt, F. H., and Small, C., January 2002, "Consider Hydraulic Factors to Reduce Downtime," *Chemical Engineering*.

Erickson, R. B., Sabini, E. P. and Stavale, A. E., 2000, "Hydraulic Selection to Minimize the Unscheduled Maintenance Portion of Life Cycle Cost," Pump Users International Forum 2000, Karlsruhe, Germany.

Hydraulic Institute, 2001, "Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems," Parsippany, New Jersey.

Stavale, A. E., 2000, "Smart Pumping Systems: The Time is Now," Proceedings of PACWEST 2000 Jasper Conference, Pulp and Paper Technical Association of Canada.

White, D. C., October 2004, "Improve Your Projects Prospects," *Chemical Processing*.