EFFECTS OF ENTRAINED AIR, NPSH MARGIN, AND SUCTION PIPING ON CAVITATION IN CENTRIFUGAL PUMPS

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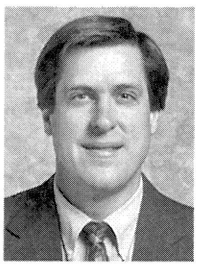

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Mr. Mayleben received a B.S. degree (Mechanical Engineering) from the University of Cincinnati (1979). He has published articles on pump hydraulics and applications.

ABSTRACT

Explanations and guidelines for the phenomena of cavitation noise above the $NPSH_R$ of a pump are presented, backed by laboratory and field tests.

The Hydraulic Institute concepts of low, high, and very high suction energy, and NPSH margin are explained and related to cavitation noise and suction pressure pulsations. Cavitation is shown to exist above the $NPSH_R$ of a centrifugal pump, and be related to the suction energy level. High suction energy pumps above a certain threshold, with low NPSH margins, can produce severe cavitation noise, especially at reduced flowrates.

Test results on an end suction pump also demonstrate the interaction between the air content in the water and cavitation noise within the pump, over varying NPSH margin ratios, flowrate, and speed (energy level). A small amount of air is shown to dramatically reduce the suction pressure pulsation levels. Further, suction pressure pulsation levels are seen to increase as the NPSH margin is reduced, until dissolved air begins to come out of solution and reverse the trend.

Finally, test results show the negative effects of piping elbows close to the pump inlet on cavitation noise, especially with high suction energy levels.

INTRODUCTION

There are many misunderstandings about cavitation, cavitation noise, and the potential of cavitation damage in centrifugal pumps. They include such misconceptions as:

- There is no cavitation in a pump if the NPSH available to the pump exceeds the NPSH required by the pump (even if the margin is only one foot).
- Entrained air will always make a pump noisy.
- The inlet piping configuration has little effect on standard (price book type) pumps.
- Cavitation will always cause damage and shorten pump life.

This study was conducted by ITT A-C Pump to explore these issues. The cornerstone of the project was a laboratory test program that monitored inlet pressure pulsations for varying suction pressure, pump speed, and flow. The main test pump was a six inch inlet, high suction energy, end suction pump, with different inlet piping configurations. The air content in the water was varied and measured. The test results are augmented by laboratory and field tests of other end suction and axial split case centrifugal pumps, along with input from a recent publication by the Hydraulic Institute (1997) on the subject of NPSH margin.

A prime objective of the research was to demonstrate the importance of suction energy levels on the effects of cavitation, and the intensity levels that can be expected with high suction energy pumps. Low suction energy pumps are normally not bothered by the cavitation that exists above the $NPSH_R$ of a pump.

CAVITATION ABOVE THE $NPSH_R$ OF A PUMP

Cavitation can exist above the $NPSH_R$ (net positive suction head required) of a centrifugal pump. The $NPSH_R$ of a pump is not the point where cavitaiton starts. According to the Hydraulic Institute (1997), it is the $NPSH_A$ (net positive suction head available) that will cause the total discharge head of the pump to be reduced by three percent. This head drop is caused by flow blockage from cavitation vapor in the impeller eye, which means that there was cavitation in the impeller to cause the two percent and one percent head drops as well. The three percent head drop criteria was based on the ease of determining the exact head drop-off point, and because most standard low suction energy pumps can operate with little margin above the $NPSH_R$ point, without seriously affecting the operational integrity of the pump. Hydraulic Institute (1997) and Budris (1993) state that it normally takes an $NPSH_A$ of four to five times the $NPSH_R$ of a pump to fully suppress cavitation within a pump.

That is why some pumps can generate cavitation noise even when the $NPSH_A$ of the system exceeds the $NPSH_R$ of the pump. Whether or not operation in this post $NPSH_R$ cavitating region will cause noise, and/or be detrimental to the life of the pump, is a function of suction energy, NPSH margin, flowrate, and the air content of the liquid, as discussed in this paper.

Cavitation is difficult to measure in a pump, unless the cavitation can be visually photographed and the bubbles measured. The human ear is quite effective in picking up cavitation noise, but it is not a well calibrated instrument. Although not perfect, the level of pressure pulsations in the suction of a centrifugal pump has been found to be one of the better ways to obtain quantitative measurements of the amount and intensity of cavitation in a pump, and it is the method used in this study. The pressure pulsation method was chosen over other methods, such as vibration, since the measurement is closer to the action and more sensitive to the collapse of the cavitation bubbles. It is also less influenced by structural variables.

LABORATORY TEST SET UP/PROCEDURE

The four and six inch suction size ANSI pumps were set up on a closed loop test facility. The test liquid was water (90°F to 115°F). Static pressure measurements were obtained using differential pressure transducers for head, and an absolute pressure transducer for suction pressure.

The dynamic suction pressure pulsation levels were obtained with a PCB piezoelectric transducer mounted in a .125 inch pipe tap, directly in the suction nozzle, approximately 0.75 inches from the impeller leading edge. The transducers sensing face was not flush with the inside wall of the passage. The transducer output was routed to a Hewlett Packard FFT. Suction pressure pulsation measurements presented in this report are overall levels, spanning a frequency range of zero to 500 Hz for 3600 rpm machines, and zero to 250 Hz for 1800 rpm.

Flowrate was monitored with a magnetic flowmeter. All pump performance data were recorded with a data acquisition computer.

The single tube mass flowmeter was a key component in the test set up for the six inch pump. As shown in Figure 1, the meter was used to obtain a sample of the two phase fluid, and provide a real time estimate of the ratio of air to liquid water.

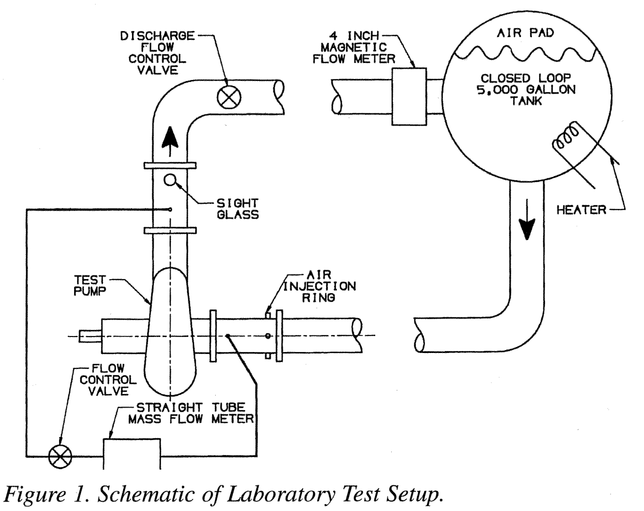

Figure 1. Schematic of Laboratory Test Setup.

A sample line from the pump discharge was directed through a flow control valve, into the meter, and then back into the pump suction. The gas portion of the two phase flow expanded across the flow control valve to a volume equivalent to the mixture in the pump suction line. By keeping the sample flowrate low (0.15 ft^3/min), the line losses were considered to be negligible. Therefore static pressure in the meter was assumed to be equal to the static pressure in the pump suction. For accuracy, the meter was placed near the centerline of the pump suction.

The mass flowmeter was set up and calibrated by the manufacturer specifically to compute the percent air. The mass flow sensor works by measuring the total mass of the air/water sample, which it divides by the known density of water. The meter automatically compensates for changes in water density due to temperature.

The scheme eliminates the need to continuously inject and monitor the air flow into the test loop. The meter measures the total mass of the combined two phase flow. Obviously, the meter cannot discern the composition of the mixture. The term *percent air* is simply used to define the ratio of the volume of all gas present in the sample, to the volume of liquid pumped, The term is not meant to imply that the mixture necessarily contains all air. The mixture should be mostly air, due to the fact that the static pressure in the meter remained above the water vapor pressure.

It should be noted that although the total amount of air in the test loops' water supply can be held constant, the volumetric *ratio* of liquid to the air can change significantly by adjusting the suction pressure, or NPSH available. It is the *volume* of the air relative to the liquid (or percent air) that can affect the pumps suction performance.

NPSH margin ratios were controlled by pressurizing or evacuating the air pad above the water in the closed loop test tank. Entrained air was added to the water through an air injection ring located just upstream of the pump inlet (Figure 1). The test pump helped to distribute air throughout the test loop. For most of the tests, the external air supply was shut off when running tests. Air content was reduced after the conclusion of the test by drawing a vacuum above the water surface in the tanks, while elevating the water temperature.

Performance data for the four and six inch end suction pumps are shown in Figures 2 and 3.

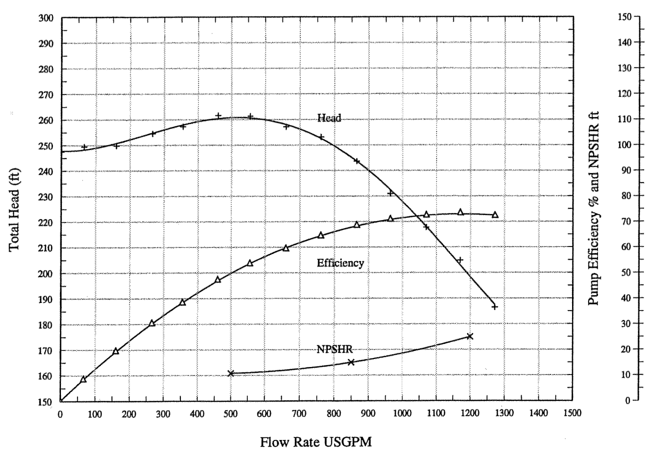

Figure 2. Six Inch End Suction @ 3550 RPM Performance.

NPSH MARGIN

Definition of NPSH Margin

The amount of system NPSH over the $NPSH_R$ is referred to as the NPSH margin and is defined as the NPSH available at the pump inlet, minus the NPSH required by the pump. The NPSH margin ratio is the $NPSH_A$ divided by the $NPSH_R$. According to

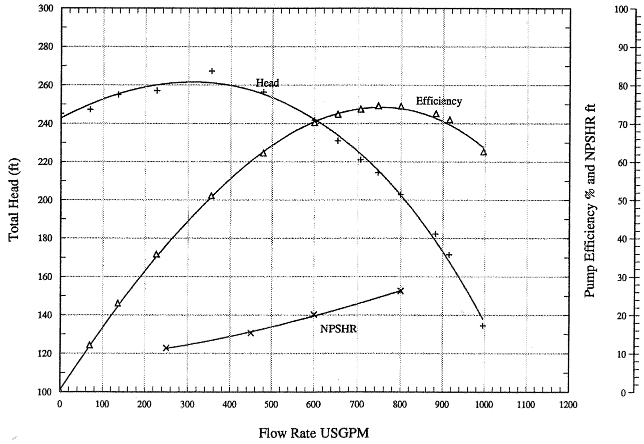

Figure 3. Four Inch End Suction @ 5000 RPM Performance.

Hydraulic Institute (1997), it can take a NPSH margin ratio of from 1.05 to 2.5 just to achieve the 100 percent discharge head value, and from two to 20 to fully suppress all cavitation. A listing of recommended minimum NPSH margin values for various industrial markets and pump suction energy levels can be found in a pending Hydraulic Institute technical specification on NPSH margin. The HI recommended that margin ratios range from 1.1 to 2.5, which means that cavitation does frequently exist in centrifugal pumps.

Tests with Varying NPSH Margin Ratios

End suction and a radial (double) suction, axial split case pumps were tested at constant flowrates, with varying NPSH margins. Pressure pulsations were monitored in the suction passages of the pumps, as shown in Figures 4, 5, 6, and 7. The percent entrained air was also measured as stated in Figures 4, 5, and 6. Additional air was injected near the pump inlet in Figure 6.

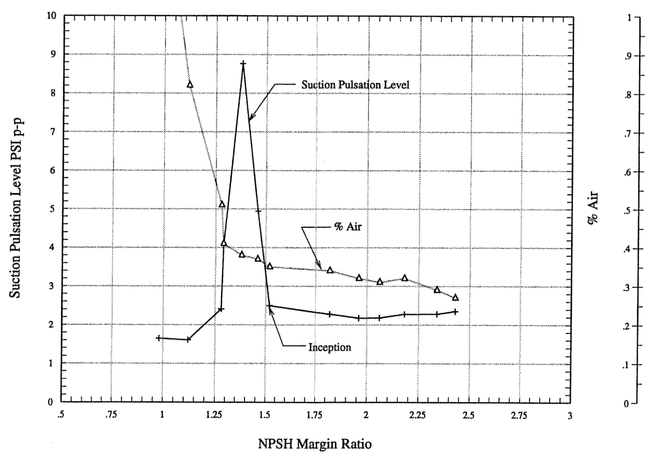

Figure 4. Suction Pulsation Level and Percent Air Versus NPSH Margin Ratio, Six Inch End Suction @ 3550 RPM, 100 Percent BEP (Low Air Content).

All of the results, with the exception of the test with additional air added (Figure 6), follow the same trend. They show relatively constant, low levels of suction pressure pulsations at higher NPSH margin ratios, a slight increase in pulsations (cavitation) as the margin is reduced from high levels, and a marked increase in pressure pulsations as the margin is reduced further and the impeller eye becomes engulfed in cavitation vapor. As the $NPSH_A$ level is reduced beyond this peak, the pressure pulsation level drops rapidly. This drop can be explained by the increase in vapor that accompanies the cavitation buildup. It appears that dissolved

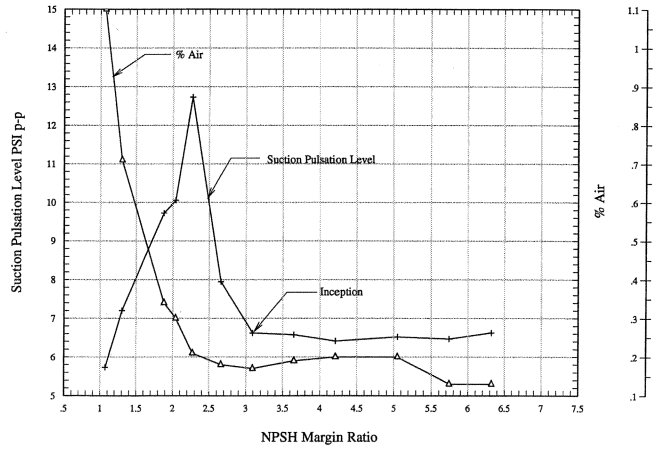

Figure 5. Suction Pulsation Level and Percent Air Versus NPSH Margin Ratio, Six Inch End Suction @ 3550 RPM, 42 Percent BEP (Low Air Content).

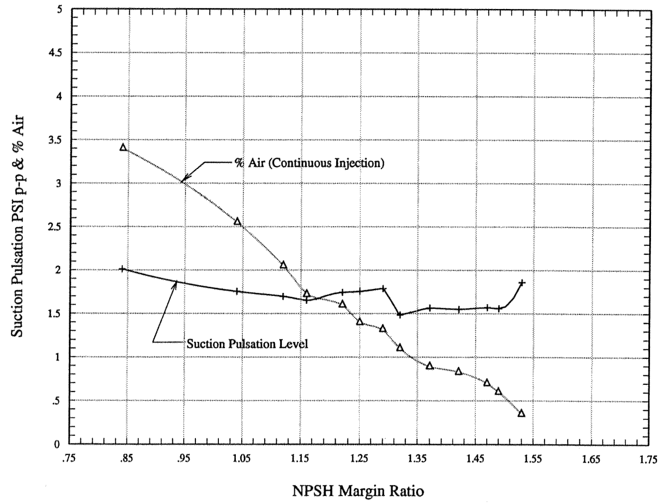

Figure 6. Suction Pulsation Level and Percent Air Versus NPSH Margin Ratio, Six Inch End Suction @ 3550 RPM, 100 Percent BEP (High Air Content).

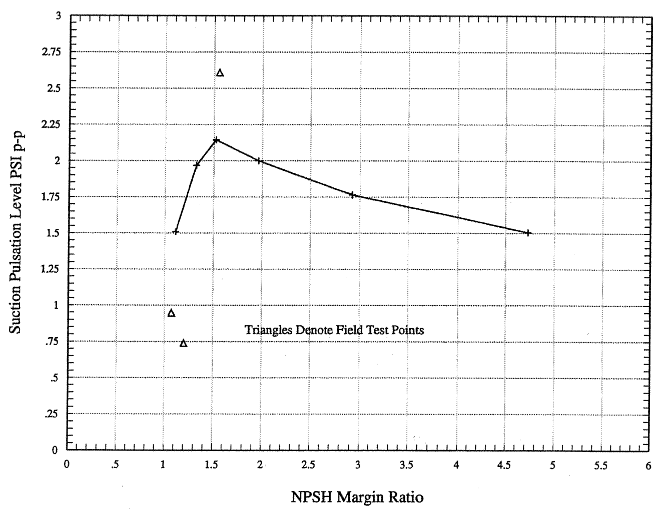

Figure 7. Suction Pulsation Level Versus NPSH Margin Ratio, 12 Inch Split Case Suction Pulsation Levels, 100 Percent BEP @ 1780 RPM.

air comes out of solution as the water flashes into cavitation vapor, and the additional entrained air cushions the pressure pulsations, to the point that the suction pressure pulsation levels near the $NPSH_R$ of the pump (NPSH margin ration = 1) are actually lower than at the highest margin values.

Field tests on three identical 12 inch suction split case pumps, in two cooling tower applications (Figure 7), demonstrate the same trend as the laboratory test results, which show the highest pulsation values around a NPSH margin ratio of 1.5, with much lower values at reduced margin ratios. The deviations from the laboratory test can be attributed to different amounts of entrained air and suction piping in the field.

Some pumps will actually get noisier with slight increases in the $NPSH_A$ (increases in NPSH margin), if the NPSH available is close to the $NPSH_R$ of the pump and air is present in the pumped liquid. This can lead to the incorrect conclusion that the problem is not caused by cavitation or an insufficient NPSH margin.

Also noteworthy is the fact that an NPSH margin ratio greater than 1.5 was required to avoid significant cavitation in the six inch end suction pump at the BEP flowrate (Figure 4). A margin ratio of over 3.0 was required at 42 percent of BEP (Figure 5), which is in the suction recirculation region of this pump, and below the recommended operating region.

These tests demonstrate why it is critical to provide an adequate NPSH margin for high suction energy pumps, especially at reduced flowrates, and if the liquid pumped contains little or no entrained or dissolved air.

SUCTION ENERGY

Definition of Suction Energy

The amount of energy in a pumped fluid that flashes into vapor and then collapses back to a liquid in the higher pressure area of the impeller inlet determines the extent of the noise and/or damage from cavitation. For simplicity, the Hydraulic Institute defines the suction energy level of a pump as shown in Equation (1).

$$\text{Suction Energy} = D * n * S \tag{1}$$

Where:
D = Casing suction nozzle size in inches
n = Pump shaft speed in rpm
S = Suction specific speed in rpm $\times$ $gpm^{.5}/NPSH_R(ft)^{.75}$

The suction nozzle size is used because it approximates the impeller eye diameter and is a measure of the flowrate of the pump. The speed (along with the impeller eye diameter) relates directly to the inlet tip speed of the impeller, and relative inlet velocities. The suction specific speed is appropriate in that it includes the flowrate and speed, plus larger impeller eye diameters are normally required for lower $NPSH_R$ values. References including Hallam (1982), base the likelihood of cavitation damage solely on the suction specific speed, which is an over simplification. Further, this definition of suction energy is very similar to the definition of the "suction recirculation factor" (SRF) found in Budris (1993), where high SRF values are shown to be indicative of suction recirculation damage.

The specific gravity term was excluded from the definition of suction energy for simplicity, since a high percentage of pumps handle water, but can be added for fluids with a specific gravity other than one.

The following suction energy milestone points were used to calibrate the suction energy levels of the pumps used in the study. They approximate the values of high and very high suction energy from Hydraulic Institute (1997), and field experience gained by the authors. High suction energy is generally where cavitation noise begins, and very high suction energy is the level at which severe cavitation damage starts (approximately 1.5 times the start of high suction energy). To improve accuracy, these energy milestones are based on the impeller eye diameters (D_e), instead of the inlet nozzle size (D), as shown in Equation (2).

$$\text{Suction Energy} = D_e * n * S \tag{2}$$

- Start of high suction energy:
  - End suction pumps - 160×10^6
  - Horizontal split case pumps (radial inlet) - 120×10^6
- Start of very high suction energy:
  - End suction pumps - 240×10^6
  - Horizontal split case pumps (radial inlet) - 180×10^6

The graph shown in Figure 8 illustrates the correlation between the suction energy term shown in Equation (2) and the inlet relative velocity W_{1t} at the vane tip leading edge. The figure is based on 130 common end suction, and horizontal split case industrial pumps. The method used for calculating W_{1t} in Figure 8 assumes a uniform inlet velocity with no preswirl. An explanation of the calculations can be found in Japikse and Brennen (1989). The same reference reveals that W_{1t} is directly related to the inlet relative kinetic energy.

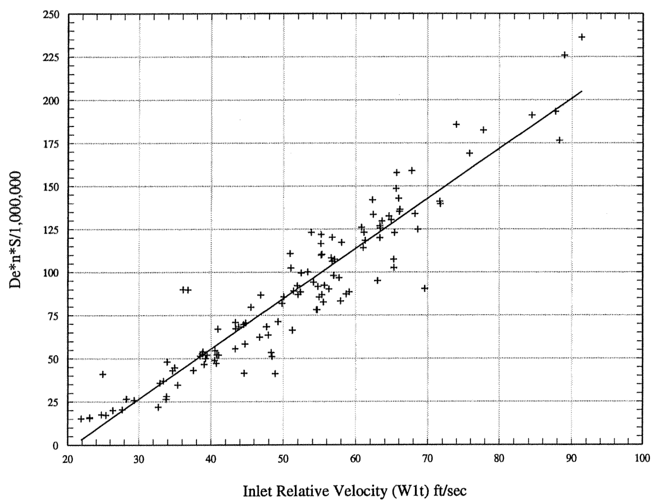

Figure 8. Suction Energy Correlation.

If the equations for W_{1t} are simplified to assume no preswirl, or shaft blockage, it can be shown that W_{1t} is directly proportional to the impeller eye diameter, inlet flowrate, and shaft speed according to Equation (3).

$$W_{1t} \propto [(Q/D_e^2)^2 + (D_e \times n)^2]^{1/2} \tag{3}$$

Where:
Q = Volumetric flowrate
D_e = Impeller eye diameter
n = Pump shaft speed

Since there is a reasonably linear relationship between W_{1t} and the suction energy term, Equation (2) or Equation (3) can be used to estimate relative suction energy levels. Equation (2) may be easier for the end user to apply, and that is why it was selected by the Hydraulic Institute.

Tests at Varying Suction Energy

Suction energy values for the pumps included in this study are shown in Table 1, along with other key metrics. The study includes laboratory and field tests on both end suction and radial suction pumps of varying high suction energy levels.

Table 1. Test Pump Metrics.

Type Suction	Suction Size	HP	Speed (rpm)	Suction Energy ($D_e \times n \times S$)	Suction Specific Speed
End Suction	6	85	3550	208	11,000
"	6	120	4000	234	11,000
"	4	25	3550	164	11,600
"	4	70	5000	230	11,600
Radial Suction	12	175	1780	118	8,727
"	18	650	1185	136	9,207
"	16	650	1780	177	8,920
"	14	900	1780	183	10,516
"	8	270	3550	193	9,804

Suction pressure pulsation levels are shown for a six inch end suction pump at two speeds (two suction energy levels), and against varying NPSH margin, at the BEP flow in Figure 9. Similarly, speed changes for a four inch end suction pump are plotted against varying flowrate in Figure 10. Pulsation levels are also plotted against suction energy for four radial suction axial split case pumps, at two flowrates (Figure 11). All but one of these pumps (the 12 inch split case pump) are considered high suction energy pumps, with the 12 inch split having borderline high suction energy.

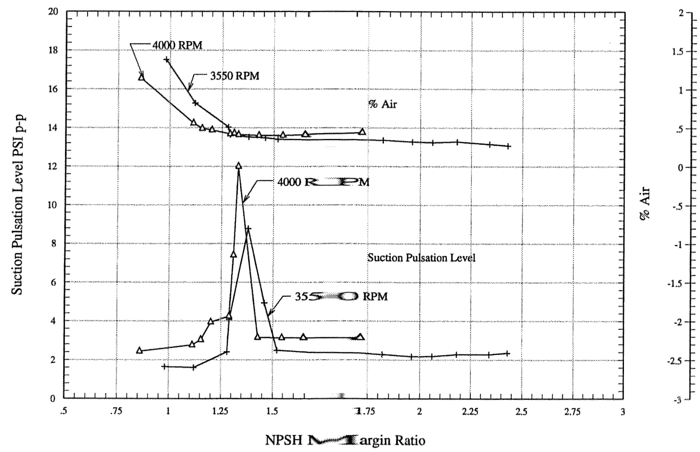

Figure 9. Suction Pulsation Level and Percent Air Versus NPSH Margin Ratio, Six Inch End Suction @ 3550 and 4000 RPM, 100 Percent BEP (Low Air Content).

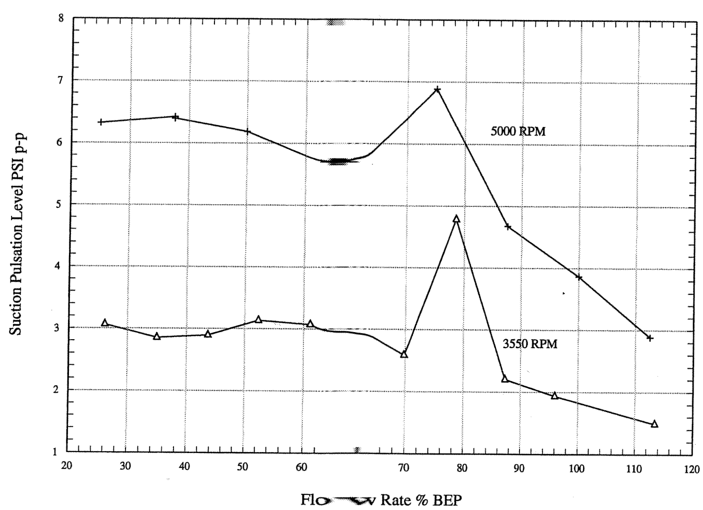

Figure 10. Suction Pulsation Level Versus Flowrate Percent BEP, Four Inch End Suction @ 3550 and 5000 RPM, NPSH Margin Ratio Greater Than 3.0.

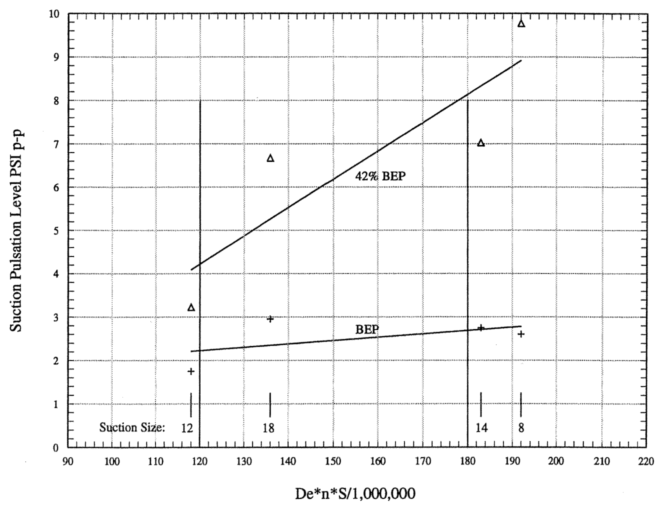

Figure 11. Suction Pressure Pulsation Level Versus Suction Energy, Radial Suction Split Case.

In all cases, a definite trend of increasing suction pressure pulsations with increasing suction energy can be seen, especially at reduced flowrates and lower NPSH margin ratios. By contrast, there were no similar trends for suction specific speed or horsepower (compare suction sizes in Figure 11 with Table 1). This helps to validate the Hydraulic Institute recommendation for higher NPSH margins with high and very high suction energy pumps.

Field Example of High Suction Energy

The damage potential in very high suction energy pumps was recently demonstrated with five 16 inch suction, 650 horsepower, split case pumps, on cooling tower service. These pumps, with a suction energy of $D_e \times n \times S = 177\ 10^6$, were operated beyond the rated (best efficiency) flowrate, and as a result had a very low NPSH margin (NPSH margin ratio = 1.05). The pumps experienced high vibration and severe cavitation damage after only several months of operation. Attempts to modify the pumps, including changing the impeller material from bronze to stainless steel, were unsuccessful. The NPSH available to the pump could not be raised more than a few feet, which did not help the situation. It was not until the units were replaced by lower suction energy pumps ($D_e \times n \times S = 136 \times 10^6$) that the vibration and erosion problems were solved. The replacement pumps, which still had high, but not very high, suction energy, have now operated for over a year without experiencing any erosion damage from cavitation.

ENTRAINED AIR

There are many differing opinions on the effects of entrained air in a centrifugal pump. Air is often considered detrimental to performance. Large amounts of air (over about five percent) can collect in the eye of the impeller and cause air binding. Air can increase the noise level in high (discharge) energy pumps, which have high impeller outlet tip speeds and small volute tongue clearances. However, small amounts of entrained air have been seen to cushion the implosive effects of cavitation, thus reducing their noise and erosive results. Tests were conducted to quantify this quieting effect, and show how it relates to NPSH margin, flowrate, inlet piping configuration, and the suction energy of a pump.

Test results of the six inch inlet end suction pump, at the maximum pulsation margin ratio, with varying amounts of entrained air, are shown in Figure 12. The addition of only 1/2 percent of additional air (.38 percent to .89 percent) is seen to reduce the suction pressure pulsation levels by 82 percent. However, the pulsation level showed little additional improvement as the air content was increased further (to three percent). This is why applications that already have small amounts of air will not always see further improvement when additional air is injected.

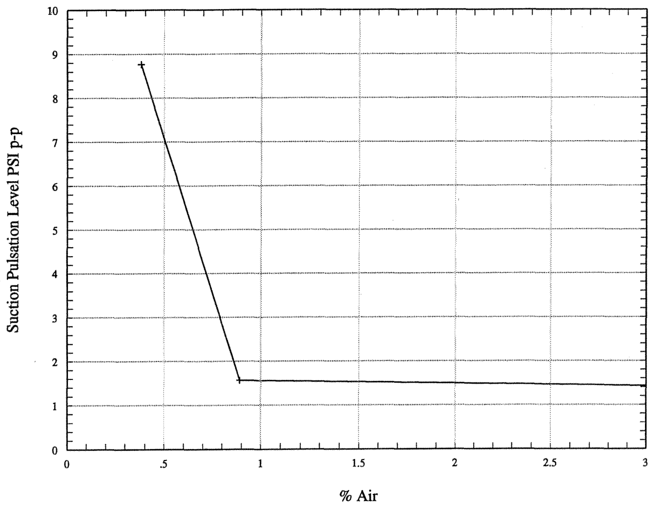

Figure 12. Suction Pulsation Level Versus Percent Air, Six Inch End Suction @ 3550 RPM, 100 Percent BEP, NPSH Margin Ratio = 1.38.

This dramatic cushioning from small amounts of entrained air explains why the suction pressure pulsation levels drop dramatically when air is released from solution, as shown in Figures 4 and 5.

This phenomenon has been frequently demonstrated on high suction energy pumps in the field as well. One such case, a 16 inch suction (176×10^6 suction energy) split case pump is presented in Figures 13 and 14. In this case, the cavitation was indirectly measured by monitoring the axial vibration of the bearing housing. As can be seen by comparing Figures 13 and 14, the pump experienced a 69 percent reduction in vibration velocity (cavitation intensity) with air injection.

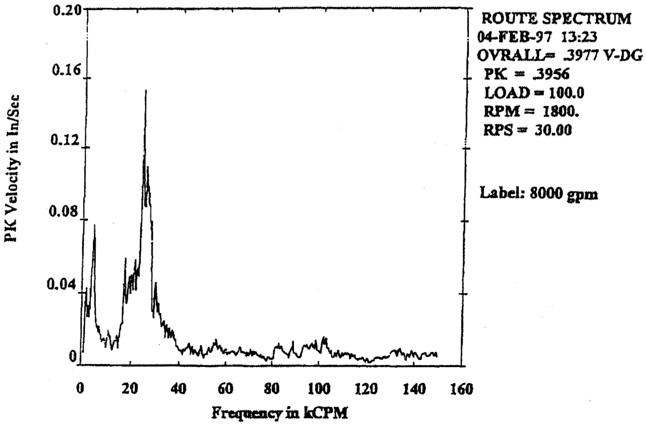

Figure 13. Vibration Versus Frequency, 16 inch Radial Suction @ 1780 RPM and Low Air.

Not all effects of air are positive, however, which can be seen in Figure 15. The results of two NPSH tests on a six inch suction pump are shown at BEP, with and without continuous air injection. The added air (from .5 percent to 1.0 or 2.0 percent) increased the apparent $NPSH_R$ of the pump from 28 ft to 30 ft.

The above results lead to several conclusions about air in a centrifugal pump. First, that small amounts (.5 to 1.0 percent) can have very positive effects on reducing cavitation noise and damage, if permitted in the system. Second, that too much air

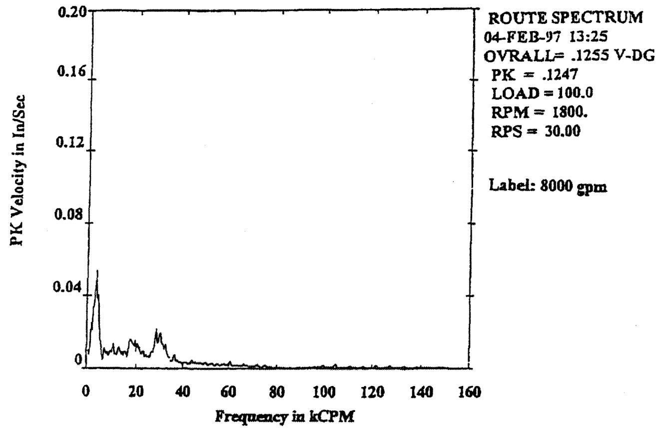

Figure 14. Vibration Versus Frequency, 16 inch Radial Suction @ 1780 RPM and High Air.

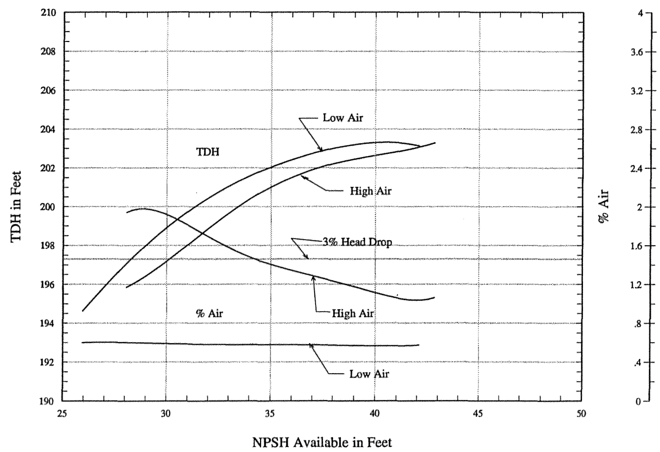

Figure 15. TDH and Percent Air Versus NPSH Available, Six Inch End Suction @ 3550 RPM, 100 Percent BEP, NPSH Test.

(much over 1.0 percent) can have negative effects on the performance of the pump. Further, since the percentage of air is often not known, it can complicate the analysis of cavitation noise problems. The situation is further compounded by the percentage of air, which varies inversely to the absolute pressure in the suction of a pump, for a given mass of air. Finally, the interaction between the dissolved and entrained air, and the amount of air coming out of solution during the formation of cavitation bubbles, makes the prediction of air effects even more difficult.

VARYING FLOWRATES

Much has been written on the results of operating centrifugal pumps at reduced flowrates in the suction recirculation region. Two such papers are listed in Budris (1993) and Hallam (1982). This study is not intended to delve deeply into the general subject of suction recirculation, but to simply study it in terms of high suction energy, effects of entrained air, and inlet piping.

The tests depicted in Figures 16, 17, 18, and 19 are intended to demonstrate the effects of the flowrate on the suction pressure pulsation/cavitation intensity of high suction energy pumps. Although the test results are generally similar, they do show slightly different trends for the two end suction pumps (Figures 16 and 17), compared with the radial suction split case pumps (Figures 18 and 19). The pressure pulsations increase for all pumps as the flowrate is throttled below the BEP, and then take a quick jump as the pumps enter the recirculation mode. However, the pulsation level then drops for the end suction pump as the flowrates are reduced further, while it continues to climb for the split case pumps. This difference may be explained by the fact that suction recirculation starts at higher relative flowrates for the end suction pumps, coupled with a higher air content at reduced flowrates, as shown for at least one of the end suction pumps (Figure 16).

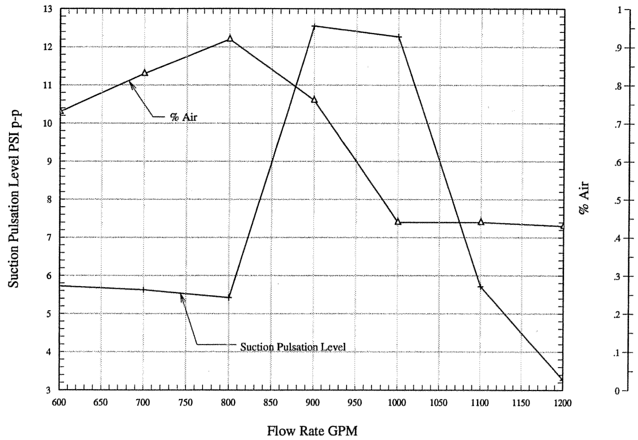

Figure 16. Suction Pulsation Level and Percent Air Versus Flowrate, Six Inch End Suction @ 3550 RPM, BEP = 1200 GPM, NPSH Margin Ratio = 2.0.

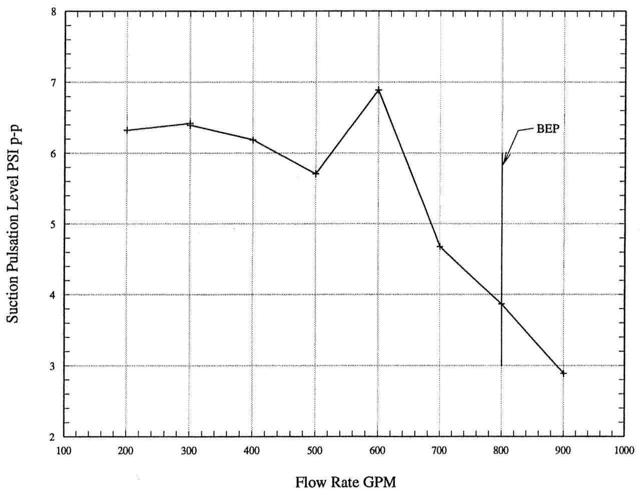

Figure 17. Suction Pulsation Level Versus Flowrate, Four Inch End Suction @ 5000 RPM, NPSH Margin Ratio Greater Than 3.0.

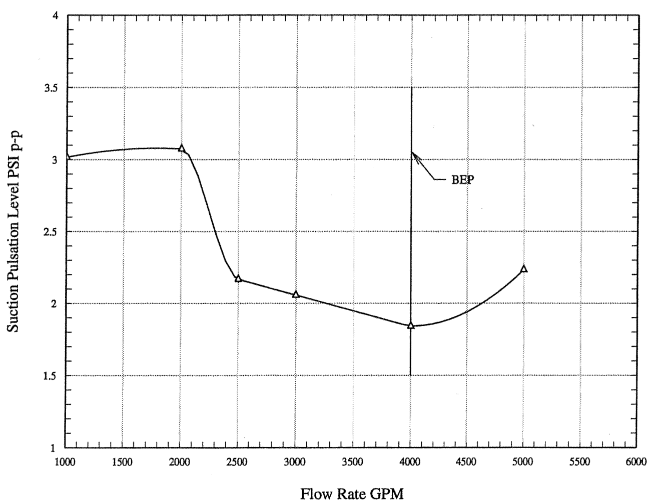

Figure 18. Suction Pulsation Level Versus Flowrate, 12 Inch Split Case Suction Pulsation Levels @ 1780 RPM.

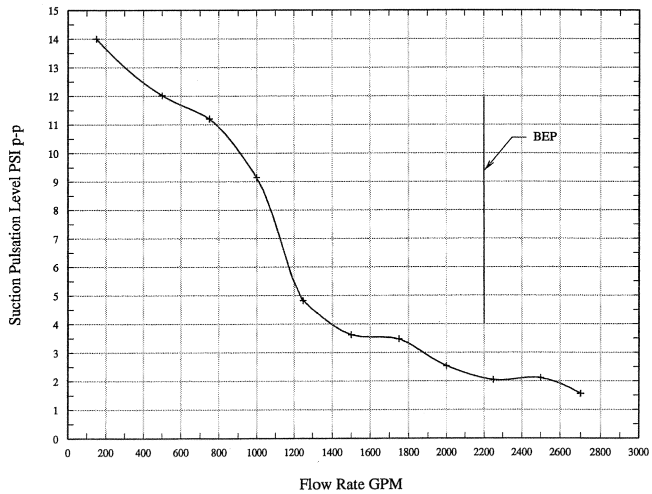

Figure 19. Suction Pulsation Level Versus Flowrate, Eight Inch Split Case Suction Pulsation Levels, 3550 RPM, 10.7 Diameter Impeller.

These tests demonstrate that the highest suction pressure pulsation levels are found in the suction recirculation region, even with different inlet piping to the pump, as discussed below. Because of this, many pump manufacturers restrict their high and very high suction energy pumps from operating in the suction recirculation region for extended periods of time.

INLET PIPING

While the Hydraulic Institute, and other pump experts recommend a minimum of five pipe diameters of straight pipe in front of a centrifugal pump inlet, many system designers do not follow this advice. In many cases, this does not cause a problem. However, when dealing with high suction energy pumps, especially with low NPSH margin ratios, unfavorable suction piping has caused field problems. The following tests were conducted to demonstrate the effects of poor inlet piping, and entrained air on high suction energy pumps.

Two high suction energy end suction pumps, a four inch with an energy level of 230×10^6, and a six inch pump with an energy level of 208×10^6, were tested with different inlet piping configurations (Figures 20, 21, and 22). The four inch pump was tested with three configurations, a straight inlet pipe, a short radius elbow, and two short radius elbows at right angles (Figure 20 and 21). The six inch pump was tested only with two inlet configurations (a straight pipe and one short radius elbow), but with an elevated air level. The short radius elbows were attached directly to the inlet nozzles of the pumps.

The four inch pump (Figure 21) demonstrated the highest pulsations at reduced flowrates in the recirculation region, with the two elbow configurations being the worst. This is to be expected, since it is the most disruptive to the flow. Elbows also significantly increased the pressure pulsations for most of the NPSH margin range tested with the four inch pump, at 75 percent of BEP flow (Figure 20).

Contrary to expectations, however, the lowest pressure pulsations were not always found with the straight inlet pipe configuration. The noisier straight (normal) inlet configuration, tested in Figure 22, has a much higher pulsation level between NPSH margin ratios of 1.2 and 1.5 than the short radius elbow inlet. The apparent answer is also shown in Figure 22, where the percent air curves for the two configurations are compared. The air level for the short radius elbow is .3 to .35 percentage points higher than for the straight pipe test, during most of the margin range, and we have continually seen how a small amount of air can make a large difference in the pulsation (cavitation intensity) level.

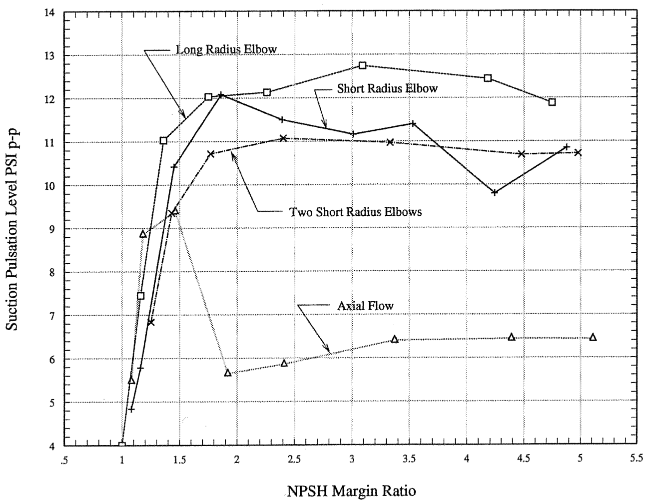

Figure 20. Suction Pulsation Level Versus NPSH Margin Ratio, Four Inch End Suction @ 5000 RPM and 75 Percent BEP, Various Inlet Configurations.

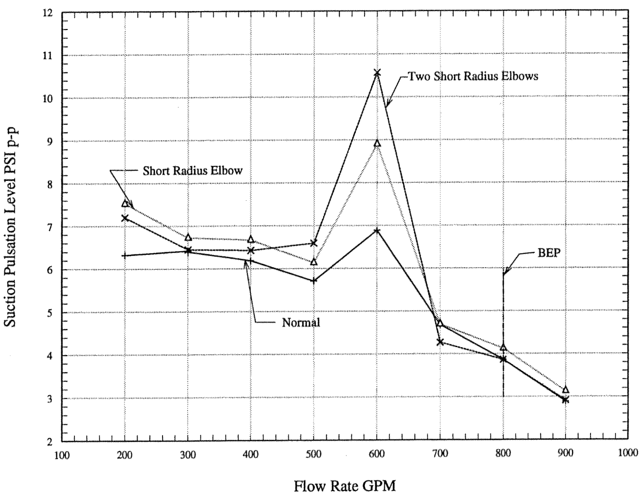

Figure 21. Suction Pulsation Level Versus Flowrate, Four Inch End Suction @ 5000 RPM, Various Inlet Configurations, NPSH Margin Ratio Greater Than 3.0.

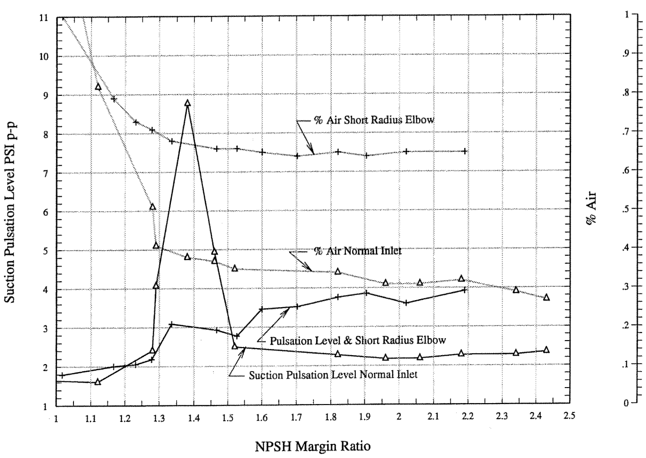

Figure 22. Suction Pulsation Level and Percent Air Versus NPSH Margin Ratio, Six Inch End Suction @ 3550 RPM 100 Percent BEP, Normal Inlet and Low Air, Short Radius Elbow and High Air.

The inlet piping tests in this study did not conclusively show the dangers of placing elbows close to the inlet of a pump when the pump is operated near the BEP flow. This can be contributed to the slightly elevated air levels in the test loop. However, in the recirculation region, poor suction piping was shown to significantly increase the pressure pulsation levels in these high suction energy pumps, even with a slightly elevated air content. It is important to follow good piping practices for high and very high suction energy pumps.

CONCLUSION

In summary, the following conclusions can be drawn from the tests and analysis presented in this study:

- Small amounts (.3 to 1.0 percent) of entrained air will generally quiet a pump dramatically, reducing the suction pressure pulsation levels by as much as a factor of five.

- After the level of entrained air reaches about one percent, higher levels have negligible effect on further reduction of the noise and pulsations. Levels above one percent can increase the apparent $NPSH_R$ of the pump.

- Cavitation exists at, and substantially above, the $NPSH_R$ of a pump, with the maximum suction pressure pulsation levels occurring between NPSH margin ratios of 1.3 to 2.3, for the high suction energy pumps tested in this study. The maximum values were dependent on the flowrate and interaction between the cavitation vapor bubbles and the entrained air.

- Cavitation, and the resulting suction pressure pulsation, increases as the NPSH margin is reduced below the point of incipient cavitation (the start of cavitation in a pump). The initial gradual pulsation increase continues until the suction pressure is lowered to the point where the cavitation vapor fully engulfs the impeller inlet, and at that point, the pulsation level rises sharply. If entrained air or other gas is present at this point, cushioning from the air takes over and the pulsations decrease as dissolved air comes out of solution. This quiets the pressure pulsations, with the reduction dependent on the air content, up to about one percent air.

- Elbows and other such pipe fittings located close to the inlet nozzle of a high suction energy pump can increase the cavitation and suction pressure pulsations, especially at reduced flowrates in the recirculation region.

- The highest NPSH margins are required at reduced flowrates, with the maximum margins required in the suction recirculation region. Further, the rate of increase in cavitation intensity with increasing suction energy is the greatest at reduced, recirculation, flowrates. Poor suction piping further aggravates this low flow condition. Air injection had the most effect in reducing pressure pulsations in the recirculation mode.

- The suction energy level of a centrifugal pump, as defined by the Hydraulic Institute, is a good guide for determining if the cavitation that frequently exists in a pump will cause noise, vibration, and/or damage. Low suction energy pumps can normally operate at or near their $NPSH_R$ with little or no problems from cavitation. High suction energy pumps under these conditions can be noisy and experience shorter life. Very high suction energy pumps are likely to experience cavitation damage if sufficient NPSH margin is not provided.

What does all this mean to the pump user? Well it means that it is important to identify the high and very high suction energy pumps in an installation, and to ensure that these pumps have sufficient NPSH margins and good suction piping. Higher cavitation noise and pressure pulsation levels can be expected if the liquid being handled is deaerated. Also, if the process can tolerate air, a small amount injected into the flow offers a simple way to quiet noisy pumps and cushion cavitation.

Further, according to the Hydraulic Institute (1997), although high suction energy pumps are likely to be noisy, they usually do not possess the energy to cause severe cavitation erosion damage, especially if erosion resistant materials, such as stainless steel, are used. This, however, is not the case with very high suction energy pumps, where the suction energy is often sufficient to do severe damage, if adequate NPSH margin and good piping practice are not provided.

Cavitation effects can vary for liquids other than water. Pumps used to handle petroleum for instance can usually survive with lower NPSH margin ratios, because hydrocarbons have a lower vapor to liquid volume, are often comprised of mixtures with different vapor pressures, and have a lower specific gravity.

NOMENCLATURE

BEP	=	Best efficiency point flowrate (no units)
D	=	Inlet casing nozzle diameter, inch
D_e	=	Impeller inlet eye diameter, inch
D_2	=	Impeller outer diameter, inch
HSE	=	High suction energy (inch $\times$ rev^2 $\times$ gallons$^{.5}$/(ft$^{.75}$ $\times$ min$^{2.5}$))
LSE	=	Low suction energy (inch $\times$ rev^2 $\times$ gallons$^{.5}$/(ft$^{.75}$ $\times$ min$^{2.5}$))
n	=	Shaft speed, rpm
NPSH	=	Net positive suction head, ft
$NPSH_A$	=	Net positive suction head available, ft
$NPSH_R$	=	Net positive suction head required, ft
Q	=	Pump flowrate, gpm
S	=	Suction specific speed, rpm $\times$ gpm$^{.5}$/$NPSH_R$(ft)$^{.75}$
SPP	=	Suction pressure pulsation, psi (peak-to-peak)
VHSE	=	Very high suction energy (inch $\times$ rev^2 $\times$ gallons$^{.5}$/(ft$^{.75}$ $\times$ min$^{2.5}$))

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