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The Effect of Pulsations on Cavitation in Reciprocating Pump Systems

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Abstract

Cavitation in reciprocating pump installations is a major cause of piping and pump vibration and mechanical failures. Cavitation occurs in pump systems when the negative peak of the dynamic pressure wave, added to the steady state pressure, approaches the vapor pressure of the liquid. Even in those systems which have ample net positive suction head (NPSH) according to Hydraulic Institute Standards [1], cavitation can occur. These standards specify that, in addition to the net positive suction head required (NPSHR) by the manufacturer, an allowance should be made for the inlet piping pressure drop and acceleration head.

Acceleration head calculations are an attempt to account for the dynamic behavior of the system using quasi-static assumptions. In practice, these calculations can be inadequate since they ignore the dynamic acoustical response characteristics of the fluid.

Accurate calculations of the pulsation levels in pump systems must consider the dynamic flow by taking into account all the parameters which significantly influence the system, including the pump fluid end, the pump valves, and the associated suction piping.

This paper discusses the inadequacies of the acceleration head concept, and describes how pulsation can contribute to cavitation. Simulation of pump systems using a computer program to predict pulsation and the onset of cavitation is discussed. Using these techniques, cavitation and pulsation levels in the suction systems can be predicted and the need for additional suction head properly evaluated.

1. Introduction

Many reciprocating pump installations experience problems that are caused by cavitation in the suction piping system. These problems can cause increased maintenance and unreliable operation. Typical problems encountered are high amplitude vibration of the piping and pump and/or failures of the piping, valves, crossheads, plungers, connecting rods, working barrels, and crankshafts. An earlier paper discussed the effects of accumulators on pulsation amplitudes and shaking forces in the discharge of piping systems [2]. This paper explains how pulsation can cause cavitation in the suction piping of a pump.

The pump and its suction piping system form a complex acoustical system having numerous acoustical natural frequencies which can be excited by the flow modulations generated by the pump. A reciprocating pump generates pulsations at integer multiples of the pump speed with the highest amplitude components normally at the plunger frequency and its harmonics. These harmonics can be amplified by the acoustical natural frequencies of the system.

Amplification factors are typically 10–40 for pulsation resonances. Therefore, when there is a coincidence of an excitation frequency with an acoustical natural frequency of the system, amplification of the pulsation can occur which can result in severe cavitation in the pump manifold and suction piping.

By using a computer program specifically written to calculate the pulsation amplitudes at any point in a piping system, it is possible to evaluate the potential for cavitation in the system [2,3,4]. The effectiveness of any given accumulator or piping modification on cavitation and pulsation levels can be

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evaluated in the design stage. By examining the complex wave and the spectral analysis over the speed range, an accurate assessment of the required NPSH can be made before the system is installed.

This paper will address the required net positive suction head (NPSHR) of pumps and evaluate the concept of acceleration head, which is a procedure for accounting for the effects of pulsations based on quasi-static assumptions. The acoustic analysis of three parallel reciprocating triplex pumps is used to illustrate the effect of various accumulators in the system on cavitation.

2. Cavitation

Cavitation occurs in liquids when the local static (absolute) pressure falls below, or attempts to fall below the liquid vapor pressure. The liquid locally flashes, creating a vapor bubble. The extent of cavitation depends upon many factors including nuclei concentration [4]. The nuclei serve as seed for formation of the cavitation bubbles. Abundant nuclei are usually available in the form of dissolved gas, liquid impurities, and surface imperfections. The subsequent collapse of vapor pockets as the fluid is swept into the higher pressure regions of the pump may cause damage of pump parts, generate sound and vibration, and produce flow and pressure pulsations in the piping.

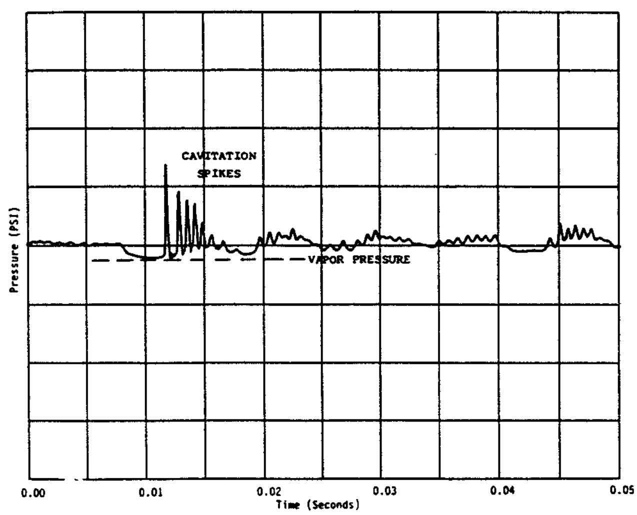

Figure 1: Measured Pressure-Time History During Cavitation

The occurrence of cavitation can be seen from field data obtained on a triplex pump [3,4] (Figure 1). As the pulsation caused the negative peak of the pressure-time wave to drop below its vapor pressure, cavitation occurred and the waveform flattened off (since the fluid can not support a negative pressure beyond the vapor pressure). As the pressure increased, the bubbles collapsed and high cavitation spikes occurred. These high frequency, high amplitude spikes can result in high dynamic forces in the system. In order to prevent cavitation, it is necessary to supply additional suction head, or to reduce the pulsations which may be causing the reduction of the available head.

Cavitation can also be caused by the dynamical effects of valves. Valve parameters such as lift, valve mass, spring rate, preload, valve lip area, flow areas, etc., can greatly influence local cavitation at the valve [4,7]. The valve effects on cavitation can be simulated with a digital computer program which includes these variables and also considers the system acoustics; however, this topic is beyond the scope of this paper.

The Hydraulic Institute Standards has recommendations and guidelines for pump systems to help prevent cavitation. Several definitions are reviewed below which are important to the understanding of pump design guidelines.

1. Net Positive Suction Head Required (NPSHR)

   The net positive suction head required (NPSHR) for a pump is usually specified according to the Hydraulic Institute Standards [1]. The NPSHR tests are typically conducted by throttling the suction while holding the discharge pressure and pump speed constant until either a three percent loss in capacity occurs or cavitation noise is clearly audible.

2. Net Positive Suction Head Available (NPSHA)

   The net positive suction head available (NPSHA) is the absolute pressure in the liquid (less vapor pressure at the pumping temperature) available at the pump inlet. The static pressure can be calculated by subtracting the pressure drop and elevation losses between the suction tank or source and the pump. Problems typically occur when the NPSHA is lower than the NPSHR at the pump. The component which most often causes the NPSHA term to be inaccurate is the dynamic pressure pulsation component. This component is difficult to calculate without a comprehensive simulation technique; therefore, investigators have attempted to take the dynamic pressure component into account by incorporating a conservative calculation of the acceleration head.

   Figure 2 shows a computed pressure-time history for one cycle superimposed onto the head available (dashed lines) for various tank levels. The NPSHA should be thought of as a time-varying parameter which is a combination of the static pressure component and the complex pressure pulsation component. The negative peak of the pulsation waveform subtracts from the NPSHA static value. Therefore, in systems with high pulsation amplitudes, the instantaneous value of the NPSHA can be below the NPSHR by the pump.

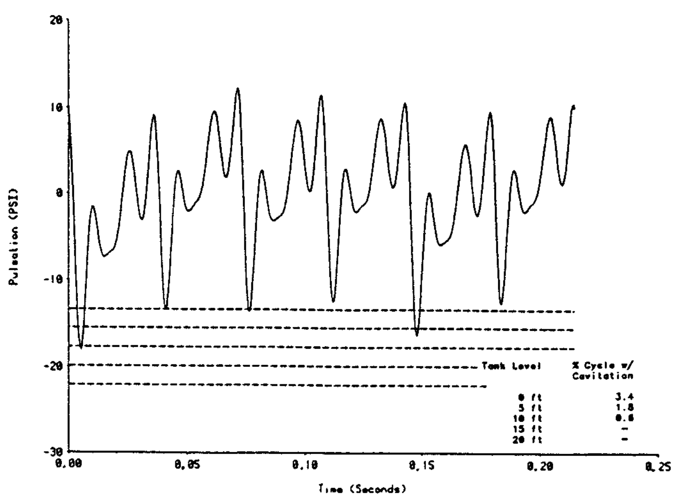

Figure 2: Computed Pressure-Time History and NPSHA (Dashed Lines Represent Available Head for 5 Noted Tank Levels)

3. Acceleration Head

   The acceleration head formula given in the Hydraulic Institute Standards [1] is a commonly used design criteria. The use of the acceleration head in the NPSHR calculation is an attempt to consider the effects of pulsations using static concepts and is not needed if an accurate model of the pump and piping is used to predict the pulsations.

3. Acceleration Head Calculations

Acceleration head may be calculated using several different methods, three of which are explained below.

3.1 Hydraulic Institute Standards

The equation from the Hydraulic Institute Standards can be used to calculate the acceleration head.

$$h_a = \frac{LvNC}{Kg}$$

where:

h_a = acceleration head, ft

L = actual length of suction line (not equivalent length), ft

v = velocity of liquid in suction line, $\frac{\text{ft}}{\text{sec}}$

N = rotative speed of crankshaft, $\frac{\text{rev}}{\text{min}}$

C = a constant, depending on the type of pump

= 0.400 for a simplex single-acting

= 0.200 for a simplex double-acting

= 0.200 for a duplex single-acting

= 0.115 for a duplex double-acting

= 0.066 for a triplex, either single- or double-acting

= 0.040 for a quintuplex, either single- or double-acting

= 0.028 for a septuplex, either single- or double-acting

= 0.022 for a nonuplex, either single- or double-acting

K = a constant which compensates for compressibility of the liquid

= 1.4 for deareated water (relatively incompressible)

= 2.5 for hydrocarbons with high relative compressibility

g = gravitational constant (32.2 $\frac{\text{ft}}{\text{sec}^2}$)

Figure 3 shows a single cylinder pump which runs at 200 rpm, having a 4 inch bore and a 4 inch stroke, and pumps pure water directly from a tank through a 25 foot long, 4 inch diameter inlet pipe. For these parameters, the acceleration head calculates to be 49.3 feet (21.4 psi). This means that in the design of the suction system, an additional head of 21.4 psi would be needed to provide the head necessary to "accelerate" the fluid into the plunger. Note that, if the pump were a multiplunger pump, such as a triplex, the coefficient (C) would be 0.066 rather than 0.4, and the calculated acceleration head would be 3.53 psi.

As can be seen, this acceleration head calculation does allow for the dynamic modulation of the fluid as it fills the plunger; however, it does not include the influence of resonant pulsations. To illustrate the effects of pulsations, it is necessary to consider both the generated pulsations and the acoustical interaction of the piping resonances.

3.2 Idealized Hand Calculations

The idealized pump and piping system shown in Figure 3 was also used to calculate the acceleration head using basic

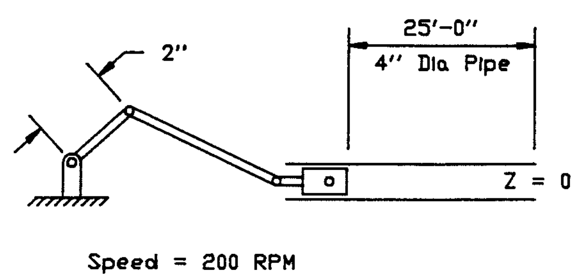

Figure 3: Idealized Single Plunger Pump System

quasi-static concepts. The forces involved in accelerating the mass of the liquid into the plunger were determined. To make the calculations equivalent to the static type of analysis (no pulsation), the bulk modulus of the fluid was assumed to be infinite so that the force would be equal to the mass times the acceleration. The force (F) is equal to the product of the maximum pressure (P) and the pipe area (A), which can be equated to the product of the density (ρ), the area, the pipe length (L), and the acceleration (a_p).

$$\begin{aligned}
a_p &= r\omega^2 \\
F &= PA \\
PA &= ma_p \\
PA &= (\rho AL)a_p \\
P &= \left(\frac{62.4}{386.4 \times 1728} \times 300\right)(877.3) \\
P &= \pm 24.6 \text{ psi}
\end{aligned}$$

Using the parameters above, the maximum pressure modulation calculates to be ± 24.6 psi. Therefore, the acceleration head calculated by the standard equation (21.4) compares with the idealized system.

3.3 Computational Method

A computer program that models the interaction of the pulsation energy generated by the pump with the acoustical characteristics of the piping system was also used to compute the instantaneous pressure versus time history at the plunger face [2,3]. As in the previous hand calculation, a rigid fluid column (infinite bulk modulus) was assumed. Figure 4 shows the results of these calculations. The minimum and maximum amplitudes were calculated to be +29.1 and −25.3 psi, which can be compared to the value obtained with the hand calculation, ± 24.6 psi. The small difference between the two is due to numerical effects resulting from the infinite bulk modulus assumption.

In this simple example, the fluid bulk modulus was selected so that no acoustical resonances existed. Calculations were then made for the same system using realistic values for the fluid properties, which resulted in a system with acoustic resonances. The speed of sound in the fluid was assumed to be 4000 feet per second. These calculations allow the dynamic effects of the system on the acceleration head to be demonstrated. Figure 5 gives the pressure wave resulting from this computation. Note that the wave shape is similar to that of Figure 4, but with significant pulsation superimposed. The resulting range of instantaneous pressure amplitude was +34.0 to −51.5 psi. The most important difference is the drop in instantaneous pressure since the negative peak was −51.5 psi compared to the value of −21.4 from the Hydraulic Institute equation. The difference in minimum pressure between the twc

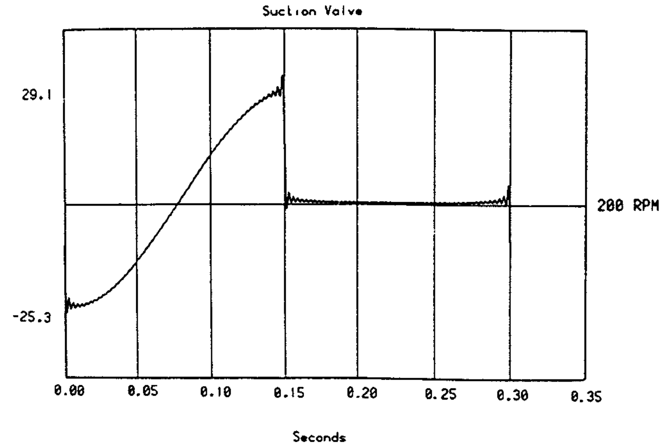

Figure 4: Computed Pressure-Time History for Example of Figure 3 — Infinite Acoustic Velocity

techniques is 30.1 psi or 140 percent. Therefore, the Hydraulic Institute acceleration head allowance would be deficient since dynamic effects are not considered.

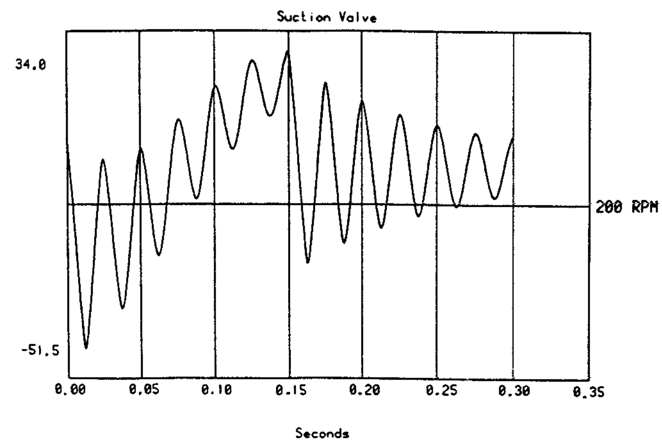

Figure 5: Computed Pressure-Time History for Example of Figure 3 — Acoustic Velocity = 4000 fps

4. Simulation of Typical Suction System

Most pulsation-caused cavitation problems in suction piping systems are caused by the system's acoustical characteristics. If no pulsation accumulator or filter is present in the system, the system will have numerous acoustical natural frequencies. If an accumulator or filter is introduced into the system, there will be a new set of acoustical natural frequencies. Depending on the location of the acoustic resonances and the plunger frequencies, the system may or may not have problems.

As discussed in Reference 2, when an accumulator or filter is installed near the discharge or suction flange, a resonance in the manifold is introduced which has the maximum pulsation amplitude at the closed end of the pump manifold and a minimum pulsation amplitude at the accumulator. This resonance is similar to a quarter-wave mode and is sometimes referred to as a manifold resonance. The acoustical natural frequency can be estimated by dividing the speed of sound by the product of 4 and the length from the end of the manifold to the accumulator. Typically, this frequency is greater than 100 Hz. When this resonance is excited, high pulsation amplitudes can occur and can cause cavitation in the pump, since the negative peak of the dynamic pressure subtracts from the steady state pressure.

To illustrate the influence on cavitation of various design configurations of suction piping systems, a simple system was analyzed with several designs, including acoustic filters and accumulators.

The pump piping system, shown in Figure 6, is typical of many pump facilities and consists of three triplex reciprocating pumps operating in parallel. The pumps are piped into a common header which continues into the supply tank. For the purposes of this paper, only the test points relevant to the particular point being emphasized will be shown.

Each of the pumps has 3.5 inch diameter plungers and a stroke of 5 inches, and operates over a speed range of 260 to 360 rpm. The maximum output from each pump is 224 gpm of water at 1000 psia. The suction pressure was controlled by the tank level and was a minimum of 16.6 psia (5 feet of head plus atmospheric pressure of 14.4 psi). The speed of sound of the water was assumed to be 4860 feet per second.

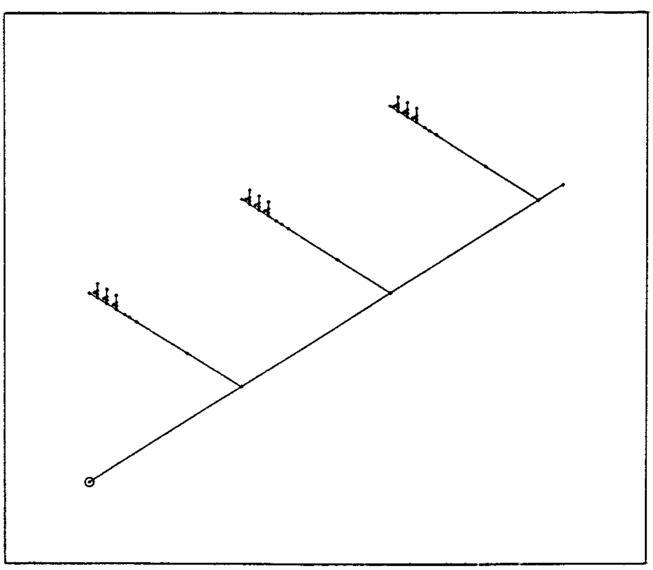

Figure 6: System Geometry Plot for Analysis

4.1 System with No Acoustic Treatment

The results of the acoustic analysis of the piping system with no acoustic treatment are shown in Figures 7–10. (Note that the vertical axis of each plot is independently scaled.) Figure 7 shows pulsation frequency spectra at three test points. Several acoustic modes are present in the system that cause resonant pulsation levels of 40–60 psi peak to peak. Significant "cross-talk" is also apparent. Cross-talk is caused by acoustic modes that involve more than a single pump, and therefore energy can be transmitted from one pump to another.

The dynamic pressure at the plunger, suction flange, and the closed end of the header are given in Figures 8–10. The minimum and maximum values of the pulsation are noted to the left of each complex pressure wave. The negative zero–peak pulsation is greater than the static head (NPSHA) at the plungers for most speeds. The NPSHA was 15.6 psi since the minimum tank level was 5 feet and 1 psi pressure drop occurred in the suction piping between the pump and the tank. At all speeds where the negative pulsation peak amplitude exceeds 15.6 psi, incipient cavitation is predicted. Therefore, cavitation would be expected at 270–281 rpm, and at 357 rpm.

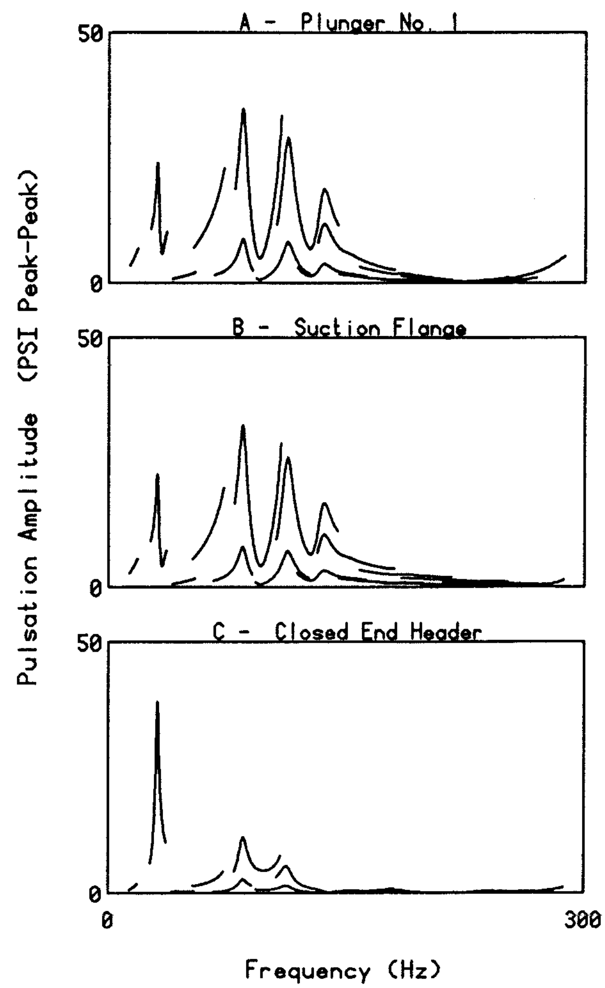

Figure 7: Pulsation Spectra — No Pulsation Treatment

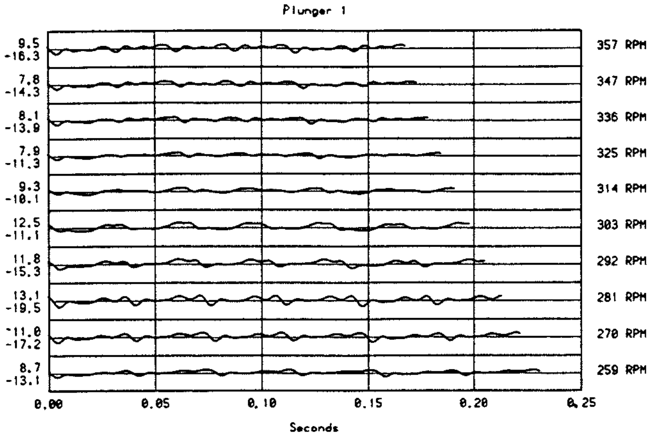

Figure 8: Pressure vs. Time Speed Raster
No Pulsation Treatment — Plunger Test Point

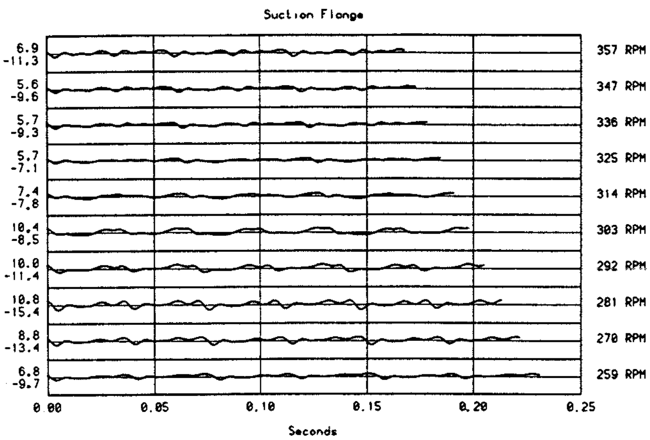

Figure 9: Pressure vs. Time Speed Raster
No Pulsation Treatment — Suction Flange Test Point

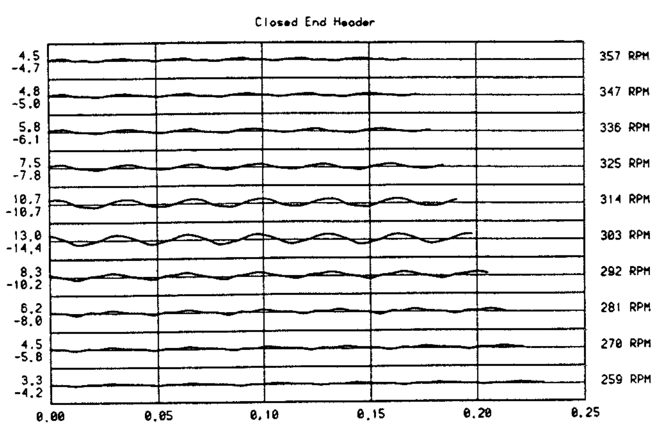

Figure 10: Pressure vs. Time Speed Raster
No Pulsation Treatment — Closed End Header Test Point

Note that at the plunger test point, the highest amplitudes are at 281 rpm, whereas for the header, the highest pulsation amplitudes are predicted at 303 rpm.

4.2 Effect of Accumulators on Cavitation

By adding an effective pulsation accumulator device, the suction pulsations can be reduced to more reasonable levels. There are basically five types of accumulators [2]: appendage with gas bladder, appendage with gas bladder and a diverter, flow-through with gas bladder, flow-through with gas blanket, and all-liquid filters.

From a pulsation standpoint, the job of the piping designer is to choose a pulsation control accumulator that will adequately protect the suction piping and the pump from cavitation and from pulsations that could cause excessive piping vibration. As will be shown, selection of a device without regard to the system acoustics can result in a system that is worse than if no pulsation dampener were used.

Generally, a good design practice is to locate the accumulator as closely as possible to the pump suction flange. As the distance between the end of the pump manifold and the accumulator increases, the manifold resonance frequency (quarter-wave) decreases. The higher energy content of the lower orders of pump speed will create a higher amplitude response at this resonance. This phenomenon can cause pulsation levels in the manifold of a quintuplex pump to be higher (because the manifold resonance frequency is lower) than those in a triplex pump, even though the generated pulsation energy of a quintuplex pump is lower than that of the triplex pump.

4.2.1 Gas-Filled Bladder Accumulator

A gas-filled bladder accumulator is usually effective in reducing the pulsations at the lower pump harmonics; however, due to the throat entrance effects, it may only be effective over a limited frequency range.

One disadvantage of gas-filled accumulators is that, if the bladder fails, the system is without pulsation control, and high pulsations can result and can cause cavitation. Another potentially troublesome characteristic is that the effectiveness of the accumulator can be quite sensitive to changes in the bladder gas volumes [2]. As the volume of gas in the bladder changes, system pulsations can change. In some cases, the system can be worse than if the accumulator were not present. To illustrate this point, the system was analyzed with a varying amount of gas volume in only one of the accumulators (Reference 2 discusses this problem encountered in discharge systems).

The normal gas volume for this accumulator is 231 in^3. The system was analyzed for gas volumes of 100% (Figures 11 and 12), 50%, 25%, 12.5%, 2.5% (Figure 13) and 0% (Figure 14).

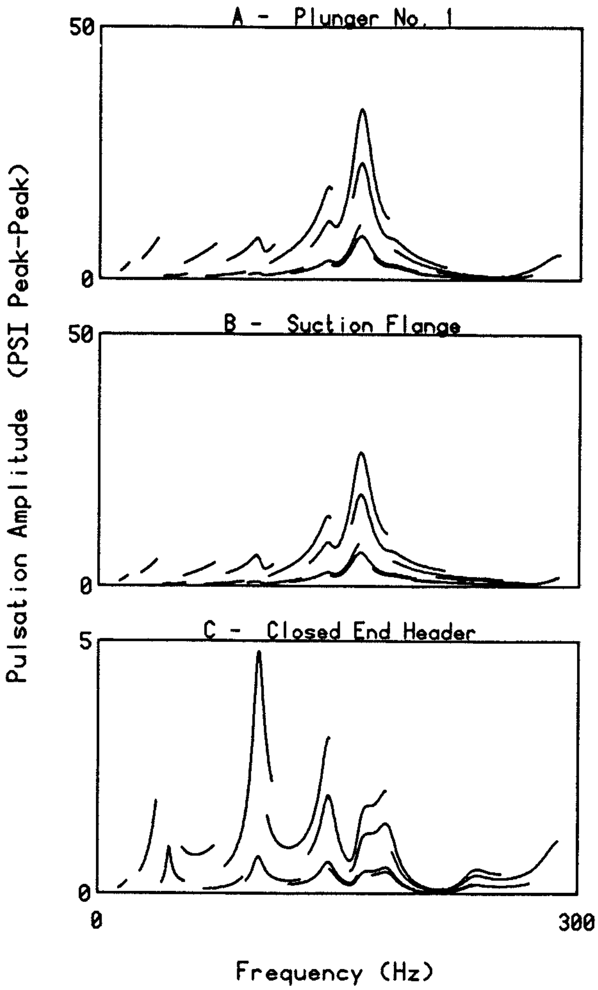

Figure 11: Pulsation Spectra — Bladder Accumulator

Figure 12 gives the complex wave for the system with the gas bladder accumulator. Note that cavitation is now predicted at all speeds since the negative peak pressure is always below the suction pressure (15.6 psia). For this system, the gas bladder accumulator causes the pulsations to be higher at the plunger and the cavitation to be worse. The maximum negative zero to peak pulsation is −35.4 psi at 326 rpm compared to −19.5 psi at 281 rpm (Figure 8) for the system with no accumulator.

The gas volume in the accumulator was decreased to 2.5% of full charge (Figure 13). A low frequency acoustic mode that was previously below plunger frequency became coincident with 1× plunger frequency. The mode was a function of the distance between the accumulators of the three pumps. As this acoustic mode frequency increased, it moved through the plunger frequency causing higher pulsations than the system without pulsation control.

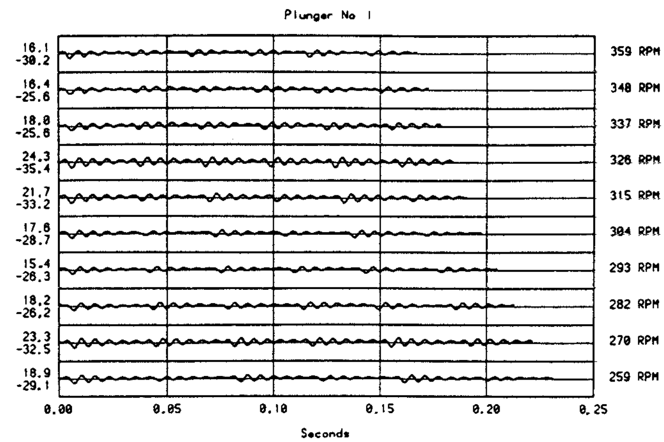

Figure 12: Pressure vs. Time Speed Raster Bladder Accumulator — Plunger Test Point

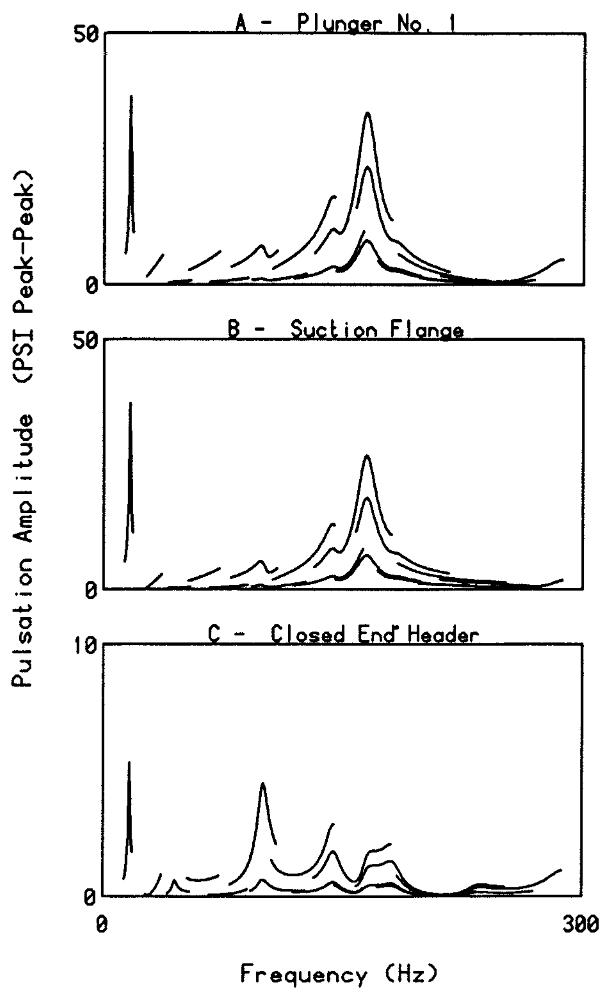

Figure 13: Pulsation Spectra — Bladder Accumulator 2.5% Charge

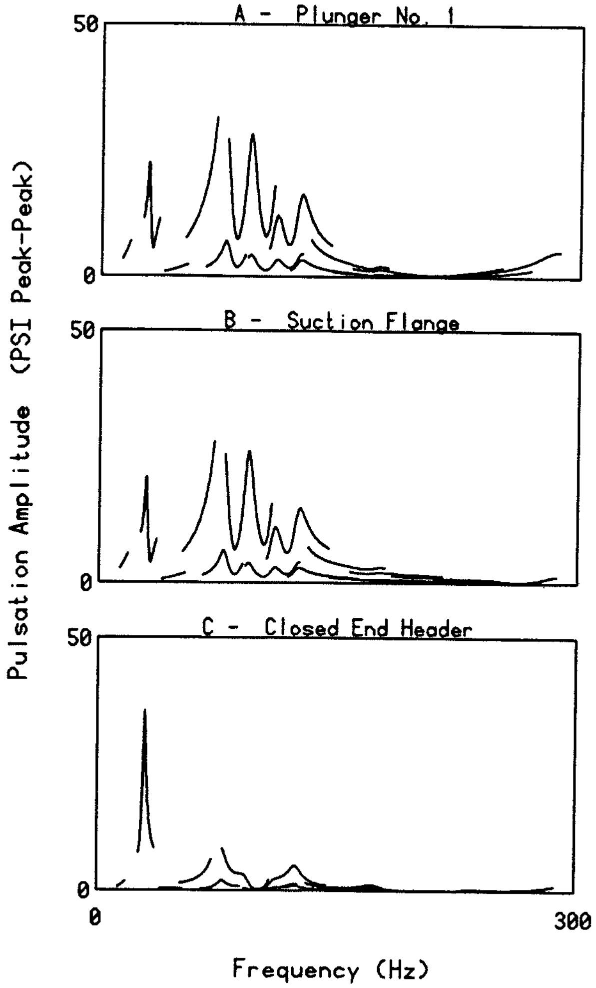

Figure 14: Pulsation Spectra — Bladder Accumulator No Charge

Note that when the gas charge is completely lost on one of the pumps (Figure 14) while the other two pumps have the normal volume, the predicted pulsations are different than for the system with no treatment (Figure 7). This is due to the interaction or cross-talk between the other pumps.

4.2.2 Flow-Through Accumulator

A flow-through accumulator, which has very little throat or neck restriction, was analyzed on the suction system. A comparison of the results of Figure 15 and Figure 7 shows that the pulsation amplitudes in the pump are lower since the attenuation at the higher frequencies is improved. The flow-through accumulator has characteristics very similar to the appendage type accumulator with no neck [2].

4.2.3 Gas Blanket Accumulator

Figure 16 gives the predicted pulsations for this system with the gas blanket accumulator. It can be seen that the pulsations are reduced at all harmonics except at the manifold resonance. This mode can be controlled by the using an orifice plate to obtain pressure drop (damping).

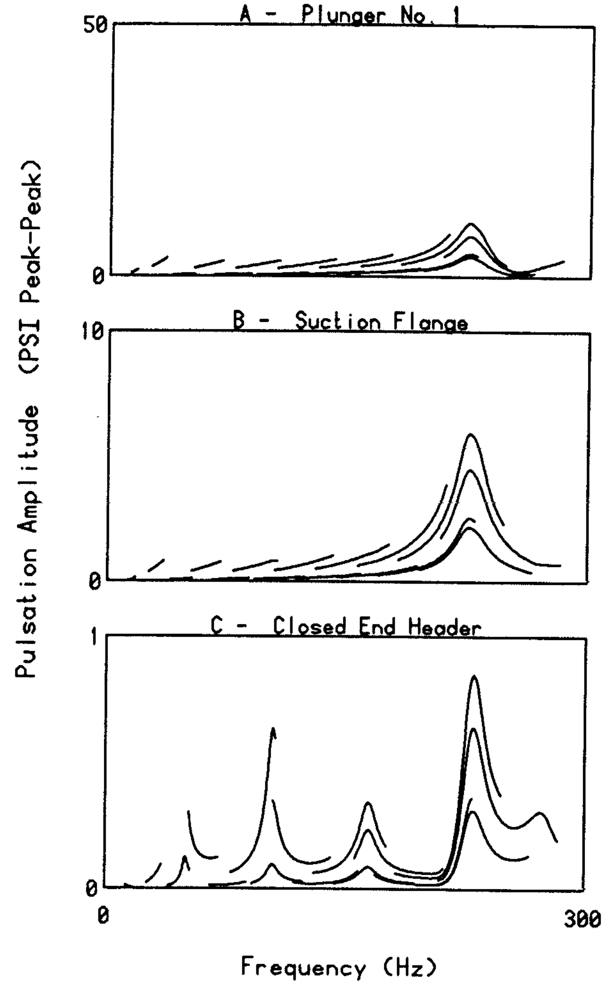

Figure 15: Pulsation Spectra — Flow-Through Accumulator

4.3 Acoustic All-Liquid Filter (Accumulator)

A properly designed acoustic filter, consisting of a large volume bottle and a small diameter choke tube, is capable of attenuating pulsation energy over a wide frequency range [2,3]. Acoustic filters on the suction are relatively immune to changes in line pressure and once installed, require little, if any, maintenance. However, these filters are generally much larger in size (primarily due to pressure drop limitations) and may have an initial cost higher than off-the-shelf accumulators.

To design an acoustic filter, a cut-off frequency is selected (usually one-half the plunger frequency), and pipe sizes are selected based on the allowable pressure drop, local site requirements, and cost.

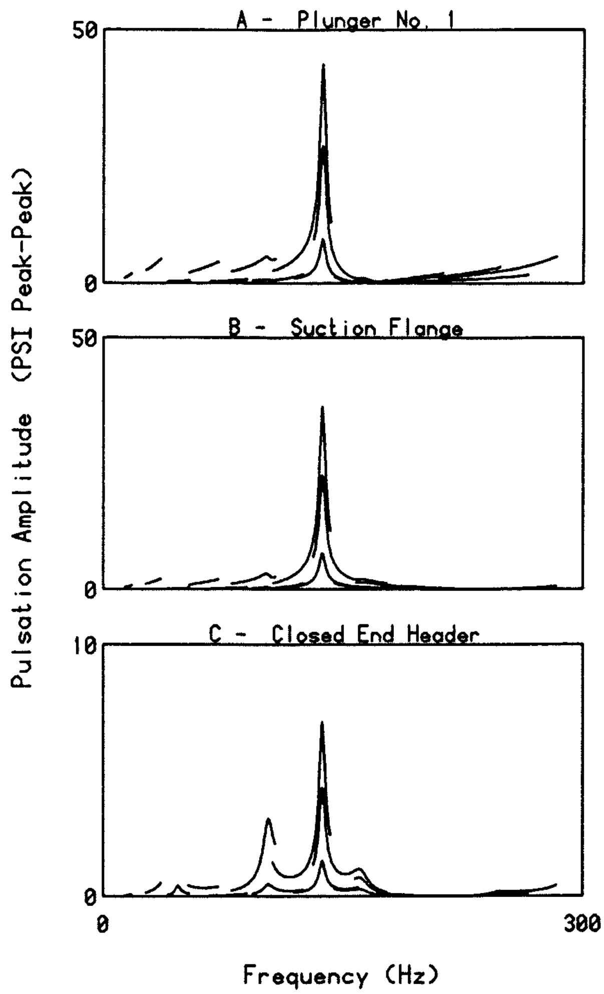

Figure 16: Pulsation Spectra — Gas-Blanket Accumulator

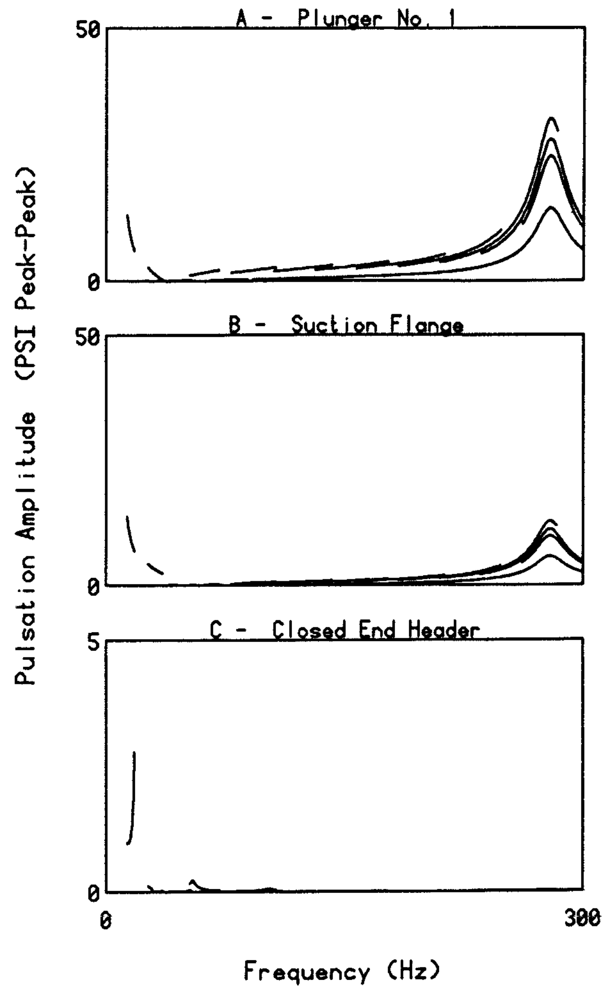

Figure 17: Pulsation Spectra — All-Liquid Filter

A volume-choke filter was designed and analyzed for this suction system (Figure 17 and 18). The final filter design consisted of a 5-foot-long bottle with an inside diameter of 30 inches and a 5-foot-long choke tube with an inside diameter of 1.94 inches. As shown in Figure 18, maximum pulsation amplitudes are calculated to be less than 5 psi in the piping. Cavitation is predicted to occur above 304 rpm at the plunger face. Since the maximum amplitude of pulsation is at the quarter-wave resonance, there is a trade off between the pressure drop of the orifice plate and the reduction of pulsation at the resonance. Therefore a 4 psi pressure drop orifice was simulated at the pump flange which was effective in attenuating the quarter-wave resonance (Figure 19).

5. Cavitation Potential Number (CPN)

The major reason for the use of an accumulator on the suction side is to protect the piping between the accumulator and

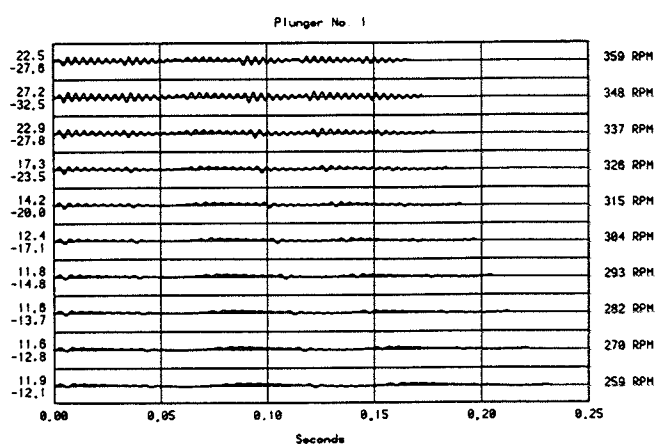

Figure 18: Pressure vs. Time Speed Raster All-Liquid Filter — Plunger Test Point

the supply source from pulsation and vibration. The manifold resonance caused by the addition of the accumulator can cause high pulsation, resulting in cavitation. As shown previously, this mode can be attenuated by a pressure drop at the accumulator entrance or at the pump flange.

To illustrate the effect on cavitation of increasing the pressure drop at the accumulator location, the system with the all-liquid filter (Figure 19) was also analyzed with a 1, 2, 4, 6, and 8 psi pressure drop. Figure 20 shows the effects of the orifice plate on pulsation amplitudes and NPSHA. Figures 18 and 19 give the pressure-time histories versus speed for the zero and 4 psi pressure drops. The analysis showed that the optimum pressure drop for this system was 4 psi at the volume bottle. The total pressure drop in the inlet piping and filter system

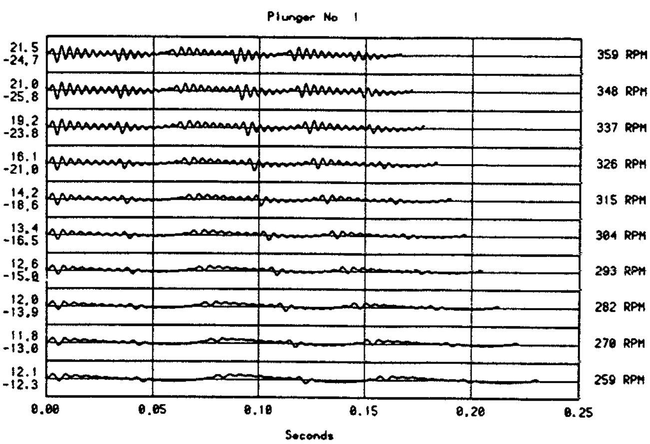

Figure 19: Pressure vs. Time Speed Raster All-Liquid Filter w/ 4 psi orifice — Plunger Test Point

was 6 psi; therefore, cavitation would be predicted whenever the pressure dropped to 9.6 psia. It can be seen that the system is predicted to cavitate at all speeds.

In this analysis, cavitation is defined whenever the complex zero to peak pulsation amplitude exceeds the NPSHA minus the vapor pressure; however, it does not consider other factors such as frequency, time of the excursion below the vapor pressure and other factors which have been shown to be important in cavitation. Actually, the predicted incipient cavitation does not necessarily mean that the pump will operate with a loss in capacity or exhibit cavitation noise. Our experience is that some pumps can have significant cavitation without a significant loss in capacity. The authors have tried to develop better methods to realistically judge the potential for cavitation in a pump system.

One method that has proven to be valuable is to plot the complex wave of the pulsation for one cycle and superimpose the NPSHA lines corresponding to the levels in the tank (or the charge pressure, if a charging pump is used). Figure 2 gives this type of plot for the system without pulsation treatment. It can be seen from this plot that, although cavitation is predicted during each cycle, the duration of the pressure pulse below the NPSHA is small and only occurs twice during the cycle. By

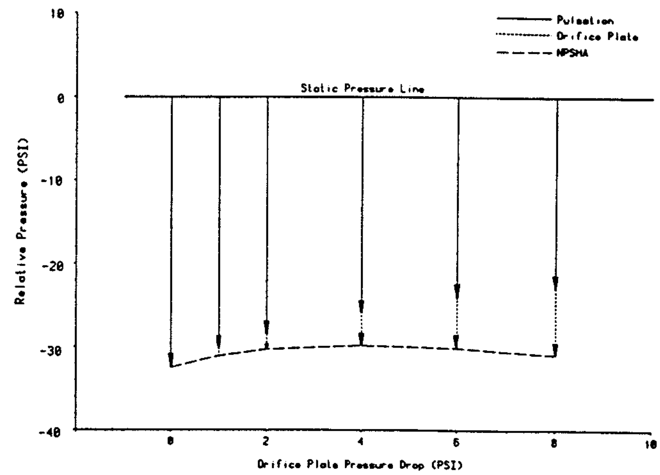

Figure 20: Effect of Orifice Plate on NPSHA

defining the NPSHA, it is possible to develop a Cavitation Potential Number (CPN) which gives the percentage of the time that the pulsations go below the vapor pressure. These CPN percentage numbers are noted alongside the tank levels. For this system without acoustical treatment or a charge pump, the CPN was only 1.8 for a tank level of 5 feet and only 3.8 for a zero feet tank level. Based on the authors' experience, a CPN value above 25 is required before the pump would have significant cavitation. Values above 50 are typically required to effect a 3 percent reduction in flow capacity. This is thought to be the result of the non-linear effects of bubble formation.

For the system with the gas-filled accumulator (Figure 21), the maximum pulsations were much higher and the CPN was 14 for the 5 foot tank level and 18 for the zero foot tank level.

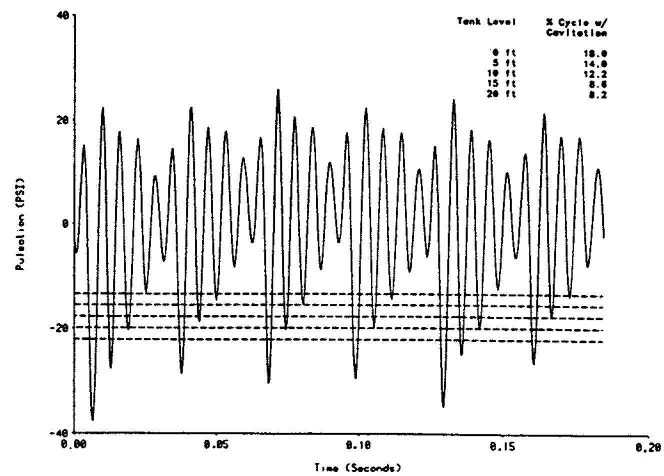

Figure 21: Computed Pressure-Time History and NPSHA Bladder Accumulator

For the system with the all-liquid filter system and the 4 psi orifice plate (Figure 22), the CPN was 12.6 for the 5 foot tank level and 18 for zero feet tank level.

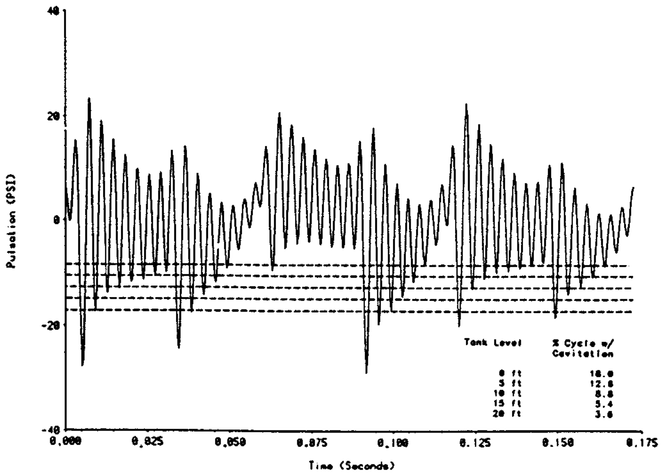

Figure 22: Computed Pressure-Time History and NPSHA All-liquid Filter

While the exact number needed to cause significant cavitation failures is not known, the use of the CPN has allowed us to better quantify and understand why some systems will work satisfactorily even though cavitation is predicted. Further research is needed to refine this approach or to develop one that is more applicable.

6. Conclusions

Many pump systems operate successfully without cavitation; however, some pump systems experience severe cavitation even when the acceleration head calculations indicate ample NPSHA. This paper has discussed the acceleration head equation and has shown that the generally accepted acceleration head equation as given in the Hydraulic Institute Standards may provide adequate pressure margin for simple systems which do not encounter acoustical pulsation resonances. Simple models were analyzed using a computer program which can simulate the acoustical dynamic characteristics of the pump and the fluid. These analyses showed that the calculated acceleration head for a non-resonant system was comparable to the computer results, but that when a pulsation resonance occurred, the acceleration head was not sufficient to prevent cavitation.

In addition, the effects of the pulsation control devices on a typical piping system were studied. Detailed modeling of a typical pump suction piping system was performed to develop a basic understanding of how pulsation causes cavitation. Based upon the calculations and the application of acoustic theory, the following conclusions can be made.

1. Cavitation in many pump systems occurs due to the coincidence of a pump speed harmonic with an acoustical resonance in the system. The high pulsation amplitudes reduce the pressure available at the plungers and, if high enough, can cause the instantaneous pressure to drop below the vapor pressure of the fluid.

2. Accurate calculation of pulsation amplitudes at any point in a pump piping system can be made using a computer program which can simulate the characteristics of all the acoustical elements in a piping system. Changes in the acoustical system can be studied to evaluate the potential for incipient cavitation.

3. Accumulators are necessary to control pulsation levels in pump piping systems. However, the majority of cavitation problems on suction systems are caused by an acoustic resonance which occurs in the pump manifold. This resonance results from the impedance discontinuity created by an accumulator. Using the computer program, the onset of cavitation can be calculated accurately and the need for a charge pump properly evaluated.

4. Addition of an orifice at the location of an accumulator can be effective in reducing the pulsation amplitudes and the potential for cavitation in the manifold. However, the orifice size should be optimized using a proven simulation technique.

Unit Conversions
1 psi = 6.895 KPa
1 in = 0.0254 m
$1\ \frac{\text{lb}}{\text{ft}^3} = 16.02\frac{\text{Kg}}{\text{m}^3}$

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