# ACOUSTICS IN PUMPING SYSTEMS

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## ABSTRACT

This tutorial will explain the pumping system speed of sound concept (how to account for acoustic velocity changes due to pipe wall flexibility and liquid properties), the definition of quarter-wave, half-wave, and higher order acoustic mode shapes, the importance of each mode shape, and examples of how to estimate these mode shapes and frequencies. The use of basic acoustics to identify and resolve pulsation and vibration issues in liquid pumping systems using acoustic filters will be described in this tutorial. Some simple examples of acoustic pulsation problems with the corresponding mode shapes and frequencies will be presented. Examples of pulsation control in liquid pumping systems using acoustic filters will be given. This tutorial will conclude with some recommended practices and guidelines for the application of acoustic theory and practice to the design, review, and problem avoidance or resolution in liquid pumping systems.

## INTRODUCTION

Acoustic pulsations are not as frequent or common a source of vibrations or dynamics problems in plunger pumps or other liquid pumping systems as they are for gas piping systems. However, when resonant pulsations do develop in a liquid pumping system they tend to be higher in amplitude and can cause more severe problems than in gas piping systems. Therefore, it is very important to know how pulsations behave in liquid systems, what pulsation frequencies plunger pumps and similar pumping devices develop, and how to predict the acoustic responses of simple liquid piping systems.

It is important to understand that the acoustic velocity in an actual pipe is affected by the pipe wall flexibility, and it is not the same as the acoustic velocity in an infinite (ideal) volume of the fluid due to the salinity of the liquid system. This tutorial will explain the speed of sound concept, changes with liquid properties, and the effects of pipe wall flexibility on the speed of sound. Methods for determining the speed of sound in the fluid will be described and some data will be presented. The definition of quarter-wave, half-wave, and higher order acoustic mode shapes, the importance of each, and examples of how to estimate these mode shapes and frequencies will be presented. An approach for estimating mode shapes and frequencies in order to identify possible pulsation problems will be discussed.

Some simple examples of acoustic pulsations problems with the corresponding mode shapes and frequencies will be presented. The use of basic acoustics to identify and resolve pulsation and vibration issues in liquid pumping systems will be described. The theory and concepts for the design and application of acoustic filters in liquid pumping systems will be discussed, and examples of pulsation control in liquid pumping systems using acoustic filters will be given. This tutorial will conclude with some recommended practices and guidelines for the application of acoustic theory and practice to the design, review, and problem avoidance or resolution in liquid pumping systems.

## ACOUSTIC VELOCITY CALCULATIONS

Determining the acoustic natural frequencies (or mode shapes) of fluid filled piping systems starts with being able to determine the appropriate speed of sound (or velocity of sound or acoustic velocity) that is to be used for the calculations. For liquid piping systems, the speed of sound of the fluid is not the same as the *effective* acoustic velocity. The acoustic velocity in liquid ($C_L$) can be expressed as stated in Equation (1).

$$C_L = \sqrt{\frac{E_L g}{\rho_L}} \quad (1)$$

where:
$C_L$ = Sound velocity, ft/s (m/s)
$E_L$ = Bulk modulus of the fluid, lb/ft$^2$ (kg/m$^2$)
$\rho_L$ = Fluid density, lb/ft$^3$ (kg/m$^3$)
g = 32.2 ft/s2 (9.8 m/s$^2$)

If the pipe walls were infinitely stiff, as they are relative to that of the gas in gas piping systems, Equation (1) could be used for calculating the speed of sound in liquid piping systems. However, the pipe wall compliance significantly affects the apparent or effective bulk modulus of the liquid. Changing the effective bulk modulus changes the effective speed of sound. The effective bulk modulus can be calculated as described in Equation (2).

$$E = \frac{E_L}{\frac{DE_L}{E_2 t} + 1} \quad (2)$$

where:
E = Effective bulk modulus of the fluid, lb/ft$^2$ (kg/m$^2$)
$E_2$ = Elasticity modulus of steel pipe, psi (Pa)
D = Inside pipe diameter, inch (m)
t = Pipe wall thickness, inch (m)

Therefore, the effective acoustic velocity can be presented as described in Equation (3).

$$c = \sqrt{\frac{E_L g}{\rho_L\left(\frac{DE_L}{E_2 t} + 1\right)}} \quad (3)$$

A graphical illustration of the significance of pipe wall compliance effect on acoustic velocity in a liquid medium is given in Figure 1. The figure illustrates the effective acoustic velocity variations for a water-filled piping system with 0.25 inch (6.4 mm) thick steel pipe walls and various pipe diameters. Pipe wall compliance can have a significant impact on the effective speed of sound.

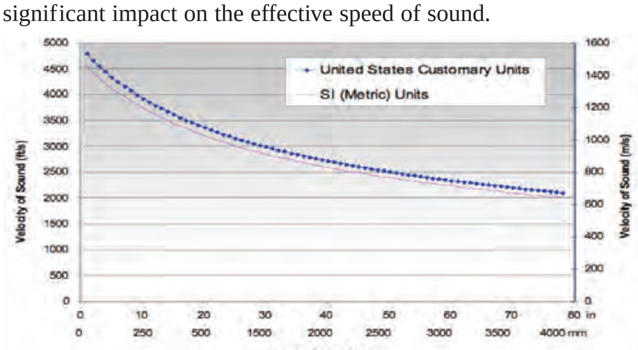



*Figure 1. Graph of the Effect of Pipe Wall Compliance on Acoustic Velocity in Water (Steel Pipe, WT=0.25 inch [6.4 mm]).*

## MODE SHAPES (ACOUSTIC NATURAL FREQUENCIES)

Acoustic velocity is a key variable that is needed to calculate the piping acoustic natural frequencies. Now that the proper equations for calculating the acoustic velocity in liquid piping systems have been presented, mode shape descriptions and equations used for calculating acoustic natural frequencies can be discussed. Mode shapes of simple piping spans can typically be broken into two categories common-boundary or opposite-boundary. Common-boundary spans of piping would be described as open-open or closed-closed spans. Opposite-boundary spans of piping are typically described as open-closed spans. Refer to Figure 2 for an illustration and example of each type of piping span.

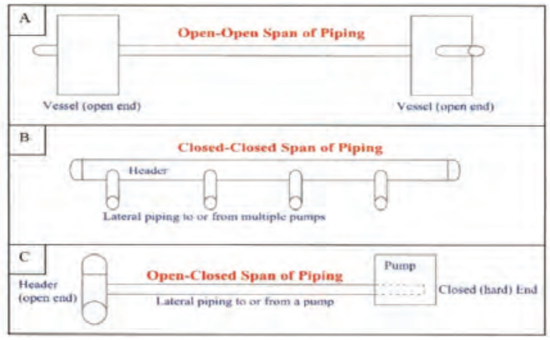



*Figure 2. Examples of Common-Boundary and Opposite-Boundary Spans of Piping.*

For open-open or closed-closed spans of piping, Equation (4) can be used to determine the acoustic natural frequency.

$$f = \frac{c}{2L_e} n \quad (4)$$

where:
f = Frequency, Hz
c = Acoustic velocity, ft/s (m/s)
$L_e$ = Equivalent length of pipe, ft (m)
n = Mode number—1, 2, 3, …

For open-closed spans of piping, Equation (5) can be used to determine the acoustic natural frequency.

$$f = \frac{c}{4L_e}(2n - 1) \quad (5)$$

The above equations allow for quick calculations of simple piping elements or sections of piping. A typical piping system is more complex than the simplified examples shown in Figure 2. More complex piping systems require modeling for accurate calculations of the system responses. Modeling also provides the opportunity to examine the predicted system pulsation amplitudes.

## LOW PASS ACOUSTIC FILTERS

As is the case of gas pipe systems, pulsation in liquid piping systems can often be effectively controlled by a two-chamber, low-pass acoustic filter as shown in Figure 3. The attenuation characteristics of such a filter can be shown as a passive response plot as is shown in Figure 4. It should be noted that frequencies below the filter cutoff frequency ($f_o$) pass through relatively unattenuated, whereas high frequencies can be significantly attenuated. As in the case of gas acoustic filters, the liquid filter is usually designed such that its cutoff frequency is at or below the lowest pulsation frequency that is to be attenuated.

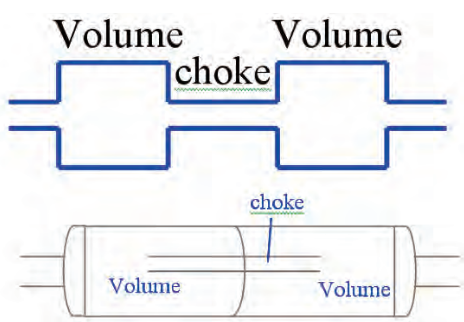



*Figure 3. Two Examples of Low-Pass Acoustic Filters.*

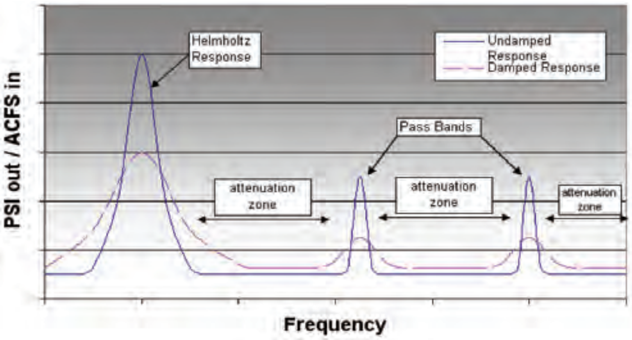



*Figure 4. Passive Response Plot Describing the Attenuation Characteristics of a Filter.*

The equation for predicting the resonant frequency for a symmetrical low-pass filter (i.e., where length of the volumes and choke are equal), Equation (6), is identical to that of a gas filter.

$$f_o = \frac{c}{2\pi}\sqrt{\frac{A}{L_e}\left(\frac{1}{V_1}+\frac{1}{V_2}\right)} \quad (6)$$

where:

c = Acoustic velocity in the fluid in the volume chambers, including the compliance effects of the chamber walls, ft/s (m/s)
A = Flow area of the choke tube, $ft^2$ ($m^2$)
$L_e$ = Equivalent length of the pipe, ft (m)
$V_1$ and $V_2$ = Volume of chamber 1 and chamber 2, $ft^3$ ($m^3$)

A more convenient form is described in Equation (7).

$$f_o = \frac{68.1}{2\pi}\sqrt{\frac{EA}{\rho L_e}\left(\frac{1}{V_1}+\frac{1}{V_2}\right)} \quad (7)$$

where:

E = The equivalent bulk modulus of the fluid, including pipe compliance effects of the chamber walls (i.e., the volumes), in psi (Pa)
ρ = fluid density, $lb/ft^3$ ($kg/m^3$)

In addition to the Helmholtz resonance of a two-chambered filter, other internal resonances may exist and will have the effect of "passing" particular frequencies virtually unattenuated or even amplified. The effects of such resonances are illustrated in the high frequency portion of Figure 4, and the resonances are usually length resonances of either the choke or volumes. Since these filter components normally consist of either open-open or closed-closed pipes, pass bands occur at half-wave length resonance and at all multiples thereof. For either open or closed end half-wave resonances, pass band frequencies may be calculated from Equation (4).

## SIDE BRANCH SUPPRESSORS

Because of the low compressibility of most liquids, conventional low-pass acoustic filters must often be quite large to provide a low cutoff frequency and even to provide high attenuation at frequencies above cutoff. One approach to providing the acoustic equivalent of a large liquid volume is to use gas-filled side branch elements often called "snubbers" or "dampeners." The design technology for such devices is developed below starting with quarter-wave stubs, progressing to side branch Helmholtz resonators, and finally gas-filled dampeners.

### *Quarter-Wave Stubs*

A typical quarter-wave stub configuration is shown in Figure 5. It consists of a tube joined to the primary (flow) piping at one end and closed at the other end (closed valve, capped end, etc.). The open-closed configuration exhibits resonances at frequencies where its length is an odd multiple of quarter-wave length. Thus, the stub resonant frequency can be calculated using Equation (5).

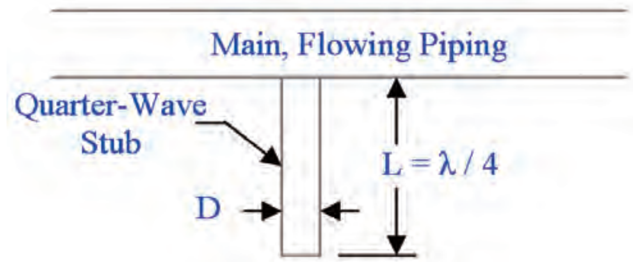



*Figure 5. Typical Quarter-Wave Side Branch Stub (A).*

The side branch will serve as an acoustic short circuit to all frequencies that correspond to its length resonances. Thus the stub can be effectively used to "detune" or absorb particular discrete pulsations.

In actual practice, a side branch stub is not a reliable means of controlling pulsation because of the following:

- The stubs work best when located at a point in the piping where a standing acoustic wave (pulsation resonance) exhibits a pulsation pressure maximum (velocity minimum). No effect will be realized when located at a pressure node.
- The side branch is most effectively used to eliminate particular resonant modes of the piping itself. It can be used to suppress particular pulsation source frequencies (e.g., a harmonic of pump speed) if the source is constant speed (or has a small speed range, i.e., 285 to 300 rpm) and if fluid temperature is relatively constant.
- Side branch stubs are most effective when their input diameter equals the pipe diameter. Although the reactive impedance of the stub is zero at resonance, its dissipational resistance remains and is inversely proportional to choke diameter cubed ($D^3$). Thus a small diameter side branch can still have appreciable impedance at resonance.

### *Side Branch Helmholtz Resonator*

The action, advantages, and disadvantages of a side branch Helmholtz resonator are identical to those of a side branch quarter-wave stub with the exception that it attenuates one frequency only and not the odd multiples thereof. Such a resonator is depicted in Figure 6. The resonant frequency of the resonator can be calculated using Equation (8).

$$f_h = \frac{1}{2\pi}\sqrt{\frac{c^2 A}{L_e V}},\ Hz \quad (8)$$

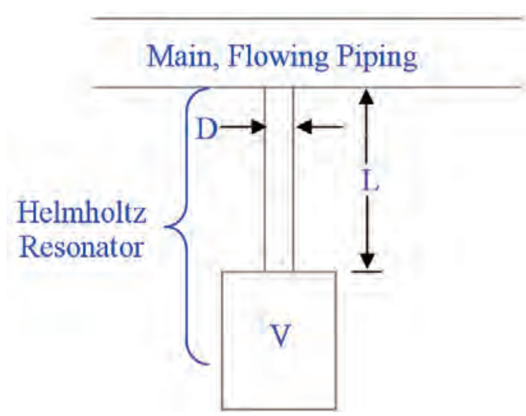



*Figure 6. Typical Quarter-Wave Side Branch Stub (B).*

*Dampener—Gas Charged Side Branch Element*

Either of the two devices above may be designed to use a partial air charge either as a free trapped volume or retained by a bag or bladder. The gas charge serves principally to reduce the required size of the side branch element as the gas charge supplies a considerable increase in compliance or equivalent volume.

When a constant diameter stand pipe is partially charged with gas as shown in Figure 7, its performance is basically that of a Helmholtz resonator where the liquid column is the choke tube and the gas column is the volume. The standard Helmholtz equation may be used, Equation (8), except that the gas volume ($V_g$) must be adjusted to give an equivalent liquid volume ($V_e$), where the subscripts g and L denote gas and liquid systems, respectively, in Equations (9), (10), and (11). Equation (11) has $V_e$ substituted into Equation (10) for the complete equation for $f_h$.

$$V_e = V_g \left( \frac{\rho_L \, c_L^2}{\rho_g \, c_g^2} \right) \quad \textbf{(9)}$$

$$f_h = \frac{1}{2\pi} \sqrt{\frac{c_L^2 \, A}{L_e V_e}}, \; Hz \quad \textbf{(10)}$$

$$f_h = \frac{1}{2\pi} \sqrt{\frac{\rho_g \, c_g^2 \, A}{V_g \, L_e \, \rho_L}}, \; Hz \quad \textbf{(11)}$$

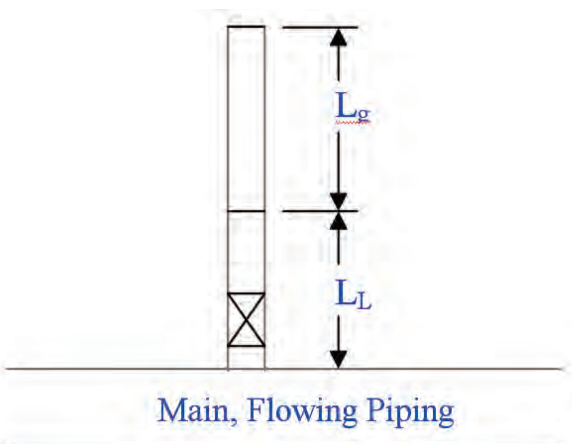



*Figure 7. Liquid-Gas Stand-Pipe Used as Side Branch Absorber.*

Normally such units are designed as stand pipes with a valve stem at the top to "tune" the gas charge.

For a side branch Helmholtz resonator partially charged with air as shown in Figure 8, an analytical description becomes complex. The liquid volume ($V_L$) may serve as a choke or volume, depending on diameter ratio and fluid parameters, while virtually all the compliance (effective volume) is supplied by the gas charge. Such systems should be analyzed with an acoustic design tool for accurate evaluation but an approximate definition of the Helmholtz resonant frequency can be obtained from Equations (11) and (12).

$$f = \frac{1}{2\pi} \sqrt{\frac{c_L^2 S}{L_e V_e}} \quad \textbf{(12)}$$

$$V_e = V_L + V_g \left( \frac{\rho_L \, c_L^2}{\rho_g \, c_g^2} \right) \quad \textbf{(13)}$$

where:
$V_L$ = Liquid-filled portion of the volume, ft$^3$ (m$^3$)
$V_g$ = Gas-filled portion of the volume, ft$^3$ (m$^3$)

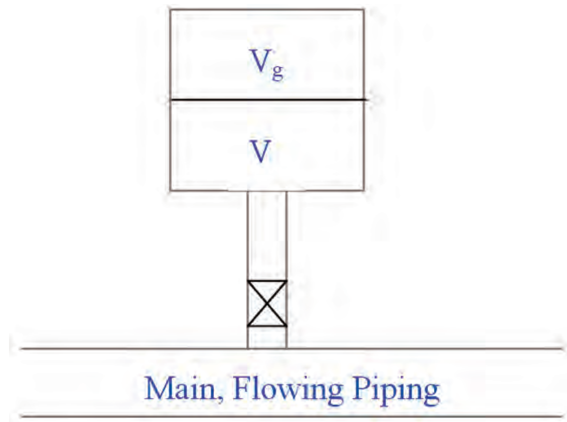



*Figure 8. Typical Gas Charged Side Branch Absorber or Snubber or Dampener.*

## FLASHING

The mechanism of flashing is described as the rapid change of a liquid to its vapor phase. By a sudden drop in pressure or increase in temperature, the state point of the liquid is rapidly moved to the vapor region and the liquid becomes vapor. In a confined piping system, the transformation can be abrupt and often damaging. An example is a feedwater regulator valve, which may drop the pressure of high temperature water several hundred psi. If the downstream pressure is below the vapor pressure (at that temperature) some of the liquid will flash into vapor. In a closed system (e.g., a piping system) enough of the liquid will flash to bring the system pressure up *to* vapor pressure. As these severe pressure fluctuations propagate up and down the line (often as a sloshing type of slug flow in a two-phase flow regime) local flashing may take place anywhere below the regulator valve, and the resultant force reactions often cause severe vibration of piping and pressure vessels and can seriously damage components such as valves and pumps. It can also play havoc with control systems either from:

- Severe mechanical vibrations,
- Making the downstream pressure set point unachievable (i.e., below vapor pressure), or
- Introducing transients into the control, which produce instability.

## CAVITATION

Cavitation is a familiar problem in liquid pipelines and hydraulic systems and a problem that can have severe consequences both in terms of efficiency of pumps, hydraulic actuators, etc., and in terms of cavitation erosion (a process of erosion or pitting of interior metal surfaces.) Such effects have been prominent in water pumps, water turbines, marine propellers, and in many other instances involving combinations of high impeller or piston velocities, low static pressures, severe flow constrictions, etc. Some example locations of potentially high cavitation are pointed out in Figure 9.

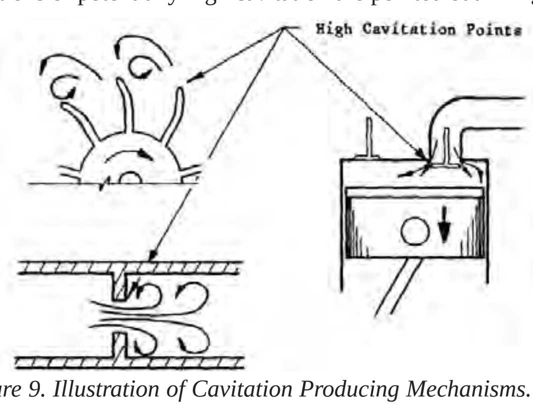



*Figure 9. Illustration of Cavitation Producing Mechanisms.*

The mechanism of cavitation starts with a situation that is similar to flashing in that the state point of the liquid (pressure, temperature, etc.) is suddenly forced into the vapor region. In cavitation however, the transition into the vapor region is of relatively short duration, as in the case of high velocity flow off a high speed impeller causing a drop in local pressure. Similarly, high velocity flow downstream of a severe constriction can cause local pressure drops, as can the passage of a rarefaction wave in severe pulsation conditions.

In either event, as the local pressure is dropped to local vapor pressure (it cannot go below), then voids or cavities filled only with vapor will appear within the liquid. As the negative pulse passes, pressure again rises above vapor pressure. The vapor then condenses and the bubble collapses with a virtually unrestrained collapse. The resultant implosion can cause severe local pressure comparable to thousands of g's. These high intensity pressure spikes will physically knock off or erode microscopic particles of nearby metal surface. Erosion will be most severe on hard (nonresilient) surfaces and on burrs, corners, or other sharp discontinuities that serve as nucleating points for formation of the cavitation bubbles.

The phenomena are illustrated in Figure 10. Consider a fluid filled pipe at pressure $P_1$, and a traveling pressure pulse of amplitude $\pm P_2$ proceeding as shown. If the negative half of this pulse drops pressure to the vapor pressure of the liquid (i.e., if $P_1 - P_2 < P_v$) then in the region where the negative peak is shown to dip below vapor pressure, cavitation bubbles will occur and tend to keep local pressure at $P_v$. As the negative pulse passes, the vapor condenses and the cavitation void collapses causing severe pressure spikes to propagate up and down the line. It is very common for significant cavitation to occur at the closed end of the pump manifold header if the acoustic response of the manifold is excited.

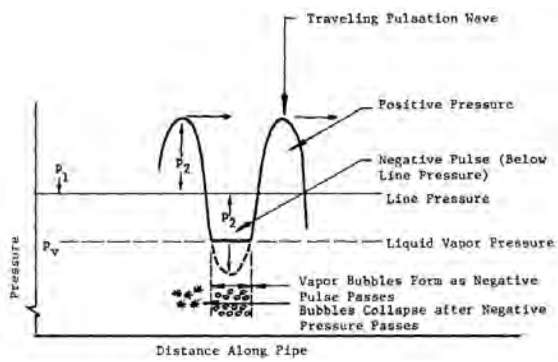
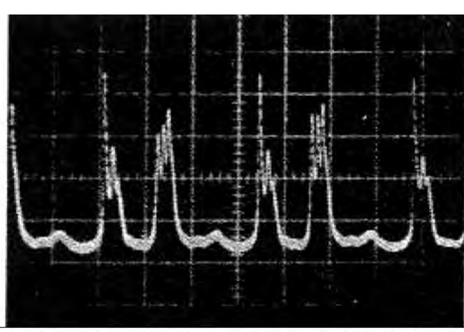



*Figure 10. Action of an Acoustic Pulse in Producing Local Cavitation in a Liquid-Filled Pipe.*

An example of field data showing signs of cavitation is shown in Figure 11. Note that the negative half of the cycle is flattened when vapor pressure is reached, and that very high amplitude pressure spikes are apparent.



Vertical Scale = 18 psi/div (124kPa/div)
Horizontal Scale = 50 ms/div

*Figure 11. Complex Wave Data Showing Cavitation Effects of Pressure Wave in a Liquid Piping System.*

For liquids with dissolved gasses, the above discussion is applicable with two modifications:

- The vapor pressure is increased. Therefore, a lower pulsation level is required to produce cavitation voids.
- The collapse of the void is less severe as gas is extracted to partially fill the cavitation void, which acts as a buffer when the void collapses. The gas, of course, does not redissolve immediately whereas pure vapor would condense almost instantaneously.

Any analytical or analog technique to simulate the actual phenomena of cavitation is virtually impossible, or at best impractical. Fortunately, such a process is not necessary to identify and solve an existing or potential cavitation problem. This contention lies in the fact one does not want to design a system *with* cavitation present, whereas conventional technology is adequate for analyzing cavitation-free systems. If the piping system of a reciprocating pump is adequately analyzed with an acoustic design tool, then an accurate prediction of pulsation amplitudes will result. If the negative portion of this pulsation wave drops below what has been established as the vapor pressure of the liquid (i.e., implying cavitation), the system should be redesigned to reduce pulsation amplitudes to an acceptable level. When this condition is achieved, an accurate simulation will have been achieved because cavitation no longer exists.

Cavitation can occur when system pulsations are excessive, and it can occur as a result of other scenarios. When the system inlet pressure is not great enough to overcome the frictional loss of the piping from a static flow viewpoint, cavitation can occur. Cavitation can also result from abnormal dynamic valve operation such as late opening, which can result in lowering the localized pressure.

## WATERHAMMER

Waterhammer is defined as the change in pressure in closed piping (positive and negative) caused by sudden changes in the steady-state flow. If liquid flow in a pipe is suddenly stopped, as by closing a valve, the kinetic energy of flow ($\frac{1}{2}\rho v^2$) is stagnated and converted to potential energy (excess pressure). The sudden increase in pressure acting against the valve is one aspect of waterhammer.

A second aspect of waterhammer occurs in the piping downstream of the valve. In this region, sudden valve closure does not necessarily stop flow suddenly and inertia of the flowing fluid may "stretch" the water column to the point that vaporization or separation of the fluid occurs just downstream of the valve. The liquid column is slowed by this vacuum (at liquid vapor pressure), stopped, and reversed until backflow fills the vapor void. The result is the backflowing column impacts the closed valve as the vapor pocket sustains an unrestrained collapse. The resulting impact from the downstream backflow is often far more severe than that upstream, and the backflow velocity is high, and the impact is extremely sudden.

The instantaneous pressure on each side of the valve is shown in Figure 12. Upstream pressure buildup is $P_u = P_{static} + P_{stagnation}$. Downstream pressure is $p_v$ or vapor pressure. When backflow occurs downstream, the collapse of the vapor void is extremely severe. Flow reversal or "fluid rebounding" can also occur upstream of the valve and cause vapor voids (flashing).

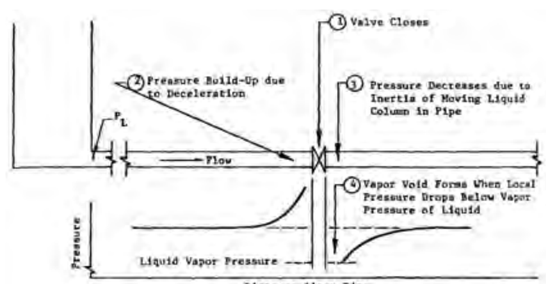



*Figure 12. Representation of Waterhammer Effects Due to Rapid Valve Closure.*

Figure 12 helps illustrate another reason why piping lengths can be critical with respect to pump speed and the opening and closing times of pump valves. Should resonance occur, i.e., the opening and closing times of valves coincident with the time for wave travel to piping discontinues, severe pressure surges can occur and be maintained.

## FIELD APPLICATIONS OF THE PUMP PIPING SYSTEM BASIC ACOUSTICS

Other methods of pulsation control that were not described in the two previous sections can also be applied to systems experiencing unacceptable pulsations. Some of those other methods include using a properly located orifice or pressure drop to damp pulsation amplitudes, changing pipe length such that a resonance is shifted off-speed, changing end conditions (such as installing a large volume [i.e., storage tank] in a pipeline), and changing the excitation frequency (i.e., pump speed). To keep this paper down to a reasonable length, many of the details of the following field applications will not be included. Field applications regarding high piping vibration, small line and tie-down clamp failures, and centrifugal pump shaft vibration will be described.

A plunger pump discharge piping system was experiencing high vibrations. The piping system was also experiencing small line and tie-down clamp failures. After analyzing the system, it was determined that the length of piping from the pump to the storage tank was being excited by the pulsations that are inherently generated by a plunger pump. This resonant length of piping could not be changed, so a section of large diameter piping was added to the piping system. Other potential solutions include the addition of a snubber volume or filter in the piping system or possibly the addition of an adequately placed and sized orifice.

A centrifugal pump was experiencing high shaft vibrations at a high frequency, which were causing bearing wear and failures. One of the observations that helped in the diagnosis of the problem was the fact that speed changes resulted in the increase and then decrease of vibration amplitudes. Different frequencies were noted on different days. It was determined that blade passing excited pulsations in the pump crossover passing caused by the impeller. The solution to this problem was to change the number of blades on the impeller.

A rotary lobe pump piping system was experiencing vibrations. Vibrations noted in the long, complex liquid piping system were all at a common frequency. The frequency corresponded to the rotary lobe passing frequency. The solution implemented to solve the problem was the installation of a Helmholtz side branch suppressor at the pump inlet and outlet.

An offshore shipping plunger pump experienced cracked piping and broken pipe tie-downs and clamps. High pressure pulsations and piping vibrations were observed in all directions (X, Y, and Z). These vibrations were at the pump plunger passing frequencies. It may be thought that Helmholtz resonators could be installed to solve this problem; however, Helmholtz resonators were already installed on the pump suction and discharge piping systems. It turned out that the discharge suppressor charge pressures were not sufficient for the entire operating range of the pump system. To cover the entire operating range, a second side branch absorber was installed such that its charge pressure overlapped that of the existing suppressors.

A centrifugal pump suction was experiencing cavitations. The average suction pressure was well above net positive suction head required (NPSHR) and vapor pressure. It was determined that the cavitations were being caused by resonant pulsations in the suction piping that were being excited by blade passing energy. The low pressures associated with the pulsations reached the vapor pressure and caused cavitations. Two potential solutions were proposed: change the piping to eliminate the resonance or install a pulsation filter or absorber.

There are many options available when trying to solve a problem that is associated with pulsations in a pump system. Sometimes the solution is as simple as the addition of an orifice, but sometimes the solution is more complicated, such as the redesign of a centrifugal pump impeller. Many of the acoustic phenomena described in this tutorial can serve as tools for determining the appropriate solution to problems that are associated with pulsations in a pump system.

## CONCLUSIONS

Pulsations in pump systems are not commonly noticed, but when they are noticed, they are typically causing a serious problem that needs to be fixed relatively quickly. Pulsations can cause cavitations in liquid pumping systems that can be eliminated by properly applied pulsation control. Proper system design (NPSH requirements and valve design) can also promote cavitation avoidance. Acoustic filters, particularly with gas charges, can be used to control pulsation in liquid piping. Other control methods such as pressure drop, length changes, and speed changes are possible. Keep in mind that pulsation frequencies are associated with the fluid acoustic velocity, and that velocity is not only a function of the fluid properties and operating conditions. It is also affected by the pipe wall compliance. Knowing the acoustic velocity and acoustic behavior of liquids in a piping system are key to being able to identify and solve pulsation problems.

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