# PRACTICAL DESIGN AGAINST PUMP PULSATIONS


**by**

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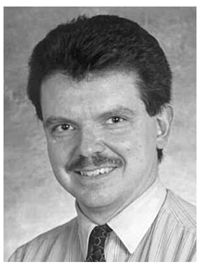

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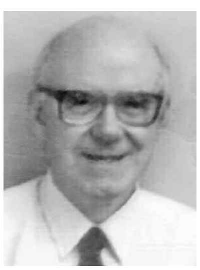

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## ABSTRACT

One of the foremost concerns facing pump users today is that of pulsation problems in their piping systems and manifolds. In cases where a fluid excitation is coincident with both an acoustic resonance and a mechanical resonance of the piping system, large piping vibrations, noise, and failures of pipes and attachments can occur. Other problems that uncontrolled pulsations can generate include cavitation in the suction lines, valve failures, and degradation of pump hydraulic performance. The potential for problems greatly increases in multiple pump installations due to the higher energy levels, interaction between pumps, and more complex piping systems involved.

The aim of this tutorial is to provide users with a basic understanding of pulsations, which are simply pressure disturbances that travel through the fluid in a piping system at the speed of sound, their potential for generating problems, and acoustic analysis and, also, to provide tips for prevention of field problems. The target audience is users who have had little previous exposure to this subject. Accordingly, the tutorial neglects the high level mathematics in the interest of presenting fundamental concepts in physically meaningful ways in the hope that benefit will be provided to the layman.


## INTRODUCTION

### *Overview*

One of the foremost concerns facing pump users today is that of pulsation problems in their piping systems and manifolds. In cases where a fluid excitation is coincident with both an acoustic resonance and a mechanical resonance of the piping system, large piping vibrations, noise, and failures of pipes and attachments can occur. Other problems that uncontrolled pulsations can generate include cavitation in the suction lines, valve failures, and degradation of pump hydraulic performance. The potential for problems greatly increases in multiple pump installations due to the higher energy levels, interaction between pumps, and more complex piping systems involved.

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Accordingly, the tutorial begins with a discussion of pulsations and why they are important in pumping systems. Since pulsation problems are almost always associated with the resonant excitation of acoustic natural frequencies, the fundamental concepts of acoustic natural frequencies, mode shapes, acoustic impedance, and resonance are described. For the many users who are familiar with mechanical systems, an analogy with mechanical natural frequencies is drawn and the basic acoustic elements of compliance, inertia, and resistance are compared to their mechanical equivalents (springs, masses, and dampers, respectively). Natural frequency equations and mode shapes are then given for the simplest piping systems, the quarter-wave stub and half-wave element. The dependence of acoustic natural frequencies on piping diameters, lengths, end conditions, and the local acoustic velocity (which, itself, is dependent on many parameters) is also discussed.

The three most common pulsation excitation sources in pumping systems are then described in detail. The first and probably best-known is the pumping elements, particularly those of reciprocating pumps. Although reciprocating pumps, and their characteristic pulsatile flows, are deservedly infamous in this area, pulsations at vane-passing frequency can also occur in centrifugal pumps, especially when running at off-design conditions and possessing positively-sloped head-flow characteristics. Second, excitations can arise due to vortex shedding arising at piping discontinuities such as tees and valves. Finally, transient excitations due to a sudden change in the piping system, such as the opening or closing of a valve, can lead to the so-called "water hammer" problems.

The tutorial then addresses the various pulsation control elements that are available and methods for sizing them and locating them within the system. Elements discussed in detail include surge volumes, accumulators (which include a gas-filled bladder to allow significant size reduction), acoustic filters (networks of acoustic volumes and inertia elements), and dissipative elements such as orifices. The advantages and disadvantages of each type, including the frequency ranges over which each is most effective, are discussed in detail. Emphasis is placed on proper location of these elements since a perfectly-sized element placed at the wrong point in the system can actually do more harm than good.

### *General Rules*

A pulsation is simply a fluctuation in pressure that occurs in the piping system of a pump. Pulsations are generated by all kinds of physical phenomena, including the action of a reciprocating pump, vortex shedding at a discontinuity in a pipe, and vane-passing effects in centrifugal impellers. These simple pulsations are usually not large enough to cause serious problems. However, if these pulsations are amplified through the action of resonance, which is identical to the idea of resonance in a mechanically vibrating system, they can become highly destructive.

There are two general types of pulsation sources, oscillatory flows and transient flows. Oscillatory flows are periodic driving forces that usually originate in reciprocating machinery but also can come from centrifugal pumps, vortex shedding at piping discontinuities, and resonances within the piping system, such as at a chattering valve. These types of sources lead to steady-state problems since they can typically last an indefinite amount of time. On the other hand, transient flows represent excitations that only last for a short period of time, usually no longer than a few seconds. These are the so-called "water hammer" problems that result from some rapid change in the flow path such as the opening or closing of a valve, the abrupt shutting off of a pump, etc.

All of the discussion provided in this tutorial assumes that the pulsations can be treated as one-dimensional plane waves. This means that at any given location in a pipe, the relevant oscillating properties, namely pressure and particle velocity, are assumed to be constant over the entire cross-section. In other words, variations in the radial direction are assumed to be zero. The assumption inherent in this treatment is that the pipe diameter is small compared to the wavelengths of the pulsations of interest. Fortunately, this assumption is valid for the vast majority of practical pump pulsation problems a user will encounter.

A distinction must be made between the fluctuating parameters and the steady-state parameters. In the general case of a fluid flowing through a pipe, its steady-state parameters are the flow rate and its static pressure along the length of the pipe. When pulsation occurs, fluctuating pressures and flows are superimposed on the steady-state values. For example, since the fluctuations are sinusoidal in nature, the pressure at a given point can be expressed as follows:

$$P(t) = P_{SS} + P_{CYCLIC} \bullet \sin(\omega t) \tag{1}$$

Where:

$P(t)$ = Pressure at a given point as a function of time
$P_{SS}$ = Steady-state pressure
$P_{CYCLIC}$ = Fluctuating pressure (pulsation)

The flow and velocity behave in exactly the same manner. Thus, when one speaks of the pressure or flow at a point, it could refer to one of two quantities—the steady-state value or the fluctuating value. However, since the main focus of this tutorial is on pulsations, the discussion almost always focuses on the fluctuating value, not the steady-state value. This should be kept in mind when a statement such as, "at a velocity node, the velocity is zero," is made. That type of statement is sometimes a source of confusion since the steady-state velocity at a velocity node is quite often not zero (although it could be). In any case, the point to remember is that, unless otherwise specified, the discussion applies to the fluctuating parameters, not the steady-state ones.

## FUNDAMENTALS

### *Mechanical Waves*

In order to understand how pressure pulsations travel through the piping systems of pumps, one must first have a basic familiarity with mechanical waves and their behavior. These fundamentals can be found in most elementary physics texts. One of the best treatments of this subject that the authors have seen is that of Resnick and Halliday (1977), which, not surprisingly, is a physics text. Accordingly, the following discussion follows their general treatment.

A mechanical wave is simply a disturbance that travels through a medium. Unlike electromagnetic waves, mechanical waves will not travel through a vacuum—they need a solid, liquid, or gas medium in order to propagate. Mechanical waves are normally initiated via displacement of some portion of an elastic medium from its equilibrium position. This causes local oscillations about the equilibrium position that propagate through the medium. It should be noted that the medium itself does not move along with the wave motion—after the wave has passed through a portion of the medium, that portion returns to rest. Waves can, and frequently do, transmit energy over considerable distances. A good example of this phenomenon is an ocean wave.

Regardless of the phenomenon that causes it, the speed that a mechanical wave travels through a particular medium is always the same. This is similar to the well-known mass-spring system that always executes free vibration at its natural frequency regardless of the origin of the vibration. Similar to the mass-spring system, the properties of the medium that determine the wave speed are its elasticity and its inertia. The elasticity gives rise to the restoring forces that cause a wave to be generated from an initial disturbance while the inertia determines how the medium responds to said restoring forces.

There are two primary types of mechanical waves of interest in physics—transverse waves and longitudinal waves. A transverse wave is a wave in which the motion of the particles conveying the wave is perpendicular to the direction that the wave travels in. A prominent example is the vibrating string shown in Figure 1. In the figure a horizontal string under tension is moved transversely at its left-hand end, thereby causing a transverse wave to travel through the string to the right.

Conversely, a longitudinal wave is a wave in which the motion of the particles conveying the wave is in the same direction that the wave is traveling in. To illustrate this, Figure 2 shows a vertical

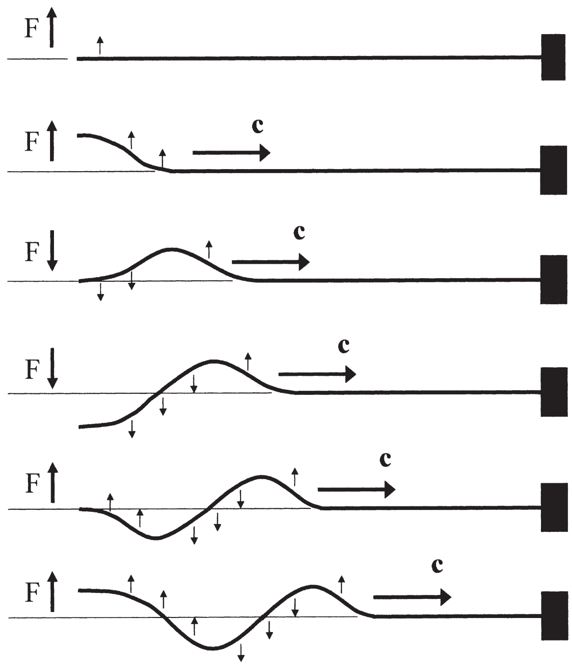


*Figure 1. Example of a Transverse Wave.*

coil spring that is subjected to an up and down motion at its top (free) end. As a result of this, the coils vibrate back and forth in the vertical direction and the wave travels down the spring. It should be noted that the acoustic waves that transmit pulsations in piping systems are longitudinal waves. However, since they often lend themselves better to visualization and understanding, transverse waves (specifically, the vibrating string) will be frequently used herein to illustrate various concepts.

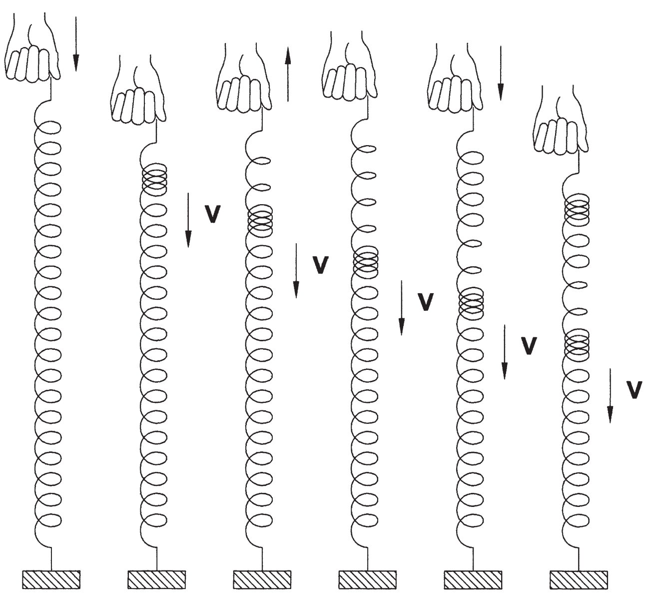


*Figure 2. Example of a Longitudinal Wave.*

*The Vibrating String*

One of the simplest forms of mechanical waves is the vibrating string, which is simply a string under tension that is subjected to a transverse displacement.

As was shown in Figure 1, if a single transverse movement is applied to the free end of a string under tension, a single "pulse" will travel through the string. Each individual particle in the string remains at rest until the pulse reaches it. At that point it moves for a short time, after which it again returns to rest. If, as is shown in the figure, the end of the string is subjected to periodic transverse motion, a "wavetrain" will move through the string. In such a case, every particle in the string executes simple harmonic motion.

Figure 3 shows a single pulse traveling to the right (in the positive x-direction) with velocity, c, in a string. Assuming that there is no damping present, the pulse retains its shape as it moves through the string. The general equation for a traveling wave moving in the positive x-direction is as follows:

$$y = f(x - c \bullet t) \tag{2}$$

Where:
y = Amplitude at any position and time
f = Any function
x = Position in direction of wave propagation
c = Wave velocity
t = Time

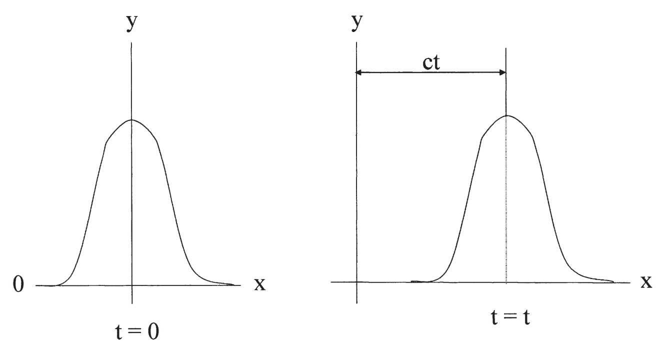


*Figure 3. Pulse Traveling to the Right in a String.*

Likewise, the general equation for a traveling wave moving in the negative x-direction is as follows:

$$y = f(x + c \bullet t) \tag{3}$$

Since, in both of these equations, y is provided as a function of x, the shape of the string at any given time (such as times, 0 and t, in the figure) can be obtained from these equations. In the words of Resnick and Halliday (1977), they provide a "snapshot" of the string at a particular moment in time. Additionally, for those interested in the behavior of a particular point on the string (i.e., at a given x value), these equations also give y as a function of time. This shows how the transverse position of any given point on the string varies with time.

As a specific example of Equation (2), the equation for a sine wave traveling in the positive x-direction in a string, is:

$$y = y_{MAX} \bullet \sin\left[(2 \bullet \pi / \lambda) \bullet (x - c \bullet t)\right] \tag{4}$$

Where:
y = Amplitude at any position and time (inch)
$y_{MAX}$ = Amplitude of sine wave (inch)
x = Position in direction of wave propagation (inch)
c = Wave velocity (in/sec)
t = Time (sec)
λ = Wavelength (inch)

This case is illustrated in Figure 4, which shows the string at two distinct times, t = 0 and t = t. This figure facilitates the definition of some of the most fundamental parameters associated with sine waves. The wavelength, λ, is the physical distance between two adjacent points having the same amplitude and phase. The period, T, is defined as the time required for the wave to travel a distance of one wavelength. The frequency, ν, is defined as the number of complete waves that propagate past a fixed point in a given time interval. The wave velocity, c, is the rate at which a given point on the wave travels in the direction of propagation. These parameters are all related by the following:

$$\lambda = c / \nu = c \bullet T \tag{5}$$

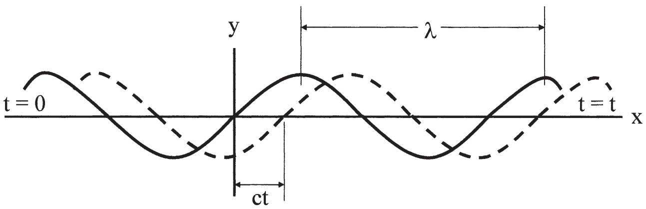


*Figure 4. Sine Wave Traveling to the Right in a String.*

Based on these definitions, at any given time, the amplitude at a position, x + λ, is equal to that at position, x. Additionally, at any given position, x, the amplitude at a time, t + T, is identical to that at time, t.

Two more fundamental parameters are the angular frequency, ω, and wave number, k, which are defined by the following:

$$\omega = 2 \bullet \pi \bullet \nu \tag{6}$$

$$k = 2 \bullet \pi / \lambda \tag{7}$$

It is a simple exercise to show that the following is also true:

$$k = \omega / c \tag{8}$$

Then, the general expression for a traveling sine wavetrain is as follows:

$$y = y_{MAX} \bullet \sin(k \bullet x - \omega \bullet t) \tag{9}$$

It is seen that any given point on the string undergoes simple harmonic motion about its equilibrium position as the wavetrain travels along the string.

*Standing Waves*

If two traveling wavetrains of the same frequency are superimposed on one another, they are said to be in a state of interference. There are two primary types of interest—constructive and destructive. Constructive interference occurs when two waves traveling in the same direction have about the same phase. In this case, the waves basically add together, as is shown in the top plot of Figure 5. On the other hand, destructive interference occurs when two waves traveling in the same direction are approximately 180 degrees out-of-phase with each other. As is shown in the bottom plot of the figure, for this case, the two waves essentially cancel each other out.

In most practical situations, interference occurs when wavetrains that originate in the same source (or in sources having a fixed phase relationship with respect to one another) arrive at a given point in space via different paths. The manner in which the waves interfere is entirely dependent on their phase difference, ϕ at the point of interference, which, in turn, is directly dependent on the difference in the lengths of the paths they took from their respective sources to the interference point. The path difference can be shown to be equal to ϕ/k or (ϕ/2π)λ . Thus, when the path difference turns out to be an integral multiple of the wavelength, λ (i.e., 0, λ, 2λ, 3λ, etc.), so that the phase difference is 0, 2π, 4π, etc., the two waves interfere constructively. Conversely, for path differences of 0.5λ, 1.5λ, 2.5λ, etc., the phase difference is π, 3π, 5π, etc., and the waves interfere destructively. All other path differences yield some intermediate result between these two extremes.

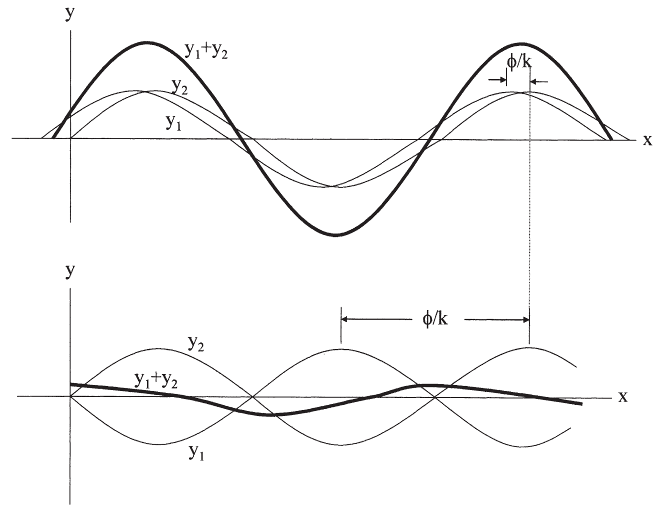


*Figure 5. Constructive and Destructive Wave Interference.*

The concept of interference is important in the analysis of the vibrating string because of the fact that traveling waves propagating through the string are reflected when they reach the end of the string. This reflection generates a second wave, having the same frequency and speed as the first, which travels in the opposite direction. Since the string now contains two waves, the initial (also known as incident) wave and the reflected wave, the two waves interfere. The resultant of these two traveling waves is called a standing wave.

Mathematically, two wavetrains of the same frequency, speed, and amplitude that are traveling in opposite directions along a string have the following governing equations, which directly follow from Equation (9):

$$y_1 = y_{MAX} \bullet \sin(k \bullet x - \omega \bullet t) \tag{10}$$

$$y_2 = y_{MAX} \bullet \sin(k \bullet x + \omega \bullet t) \tag{11}$$

The resultant of these two waves can then be shown to be as follows:

$$y = 2 \bullet y_{MAX} \bullet \sin(k \bullet x) \bullet \cos(\omega \bullet t) \tag{12}$$

This is the equation of a standing wave. It is seen that, similar to a traveling wavetrain, all particles in the string execute simple harmonic motion at the same frequency. However, in direct contrast to the traveling wave, the vibration amplitudes are not the same for all particles. Instead, they vary with the particle's location on the string.

The points of primary interest in a standing wave are the antinodes and the nodes. Antinodes are points that have the maximum amplitude ($2 \bullet y_{MAX}$ in the above example) while nodes are points that have zero amplitude. The antinodes are spaced one half-wavelength apart, as are the nodes.

Standing waves are generated whenever two traveling waves of the same frequency propagate in opposite directions through a medium. The manner in which this occurs is illustrated in Figure 6, which shows all three waves (the two traveling waves and the standing wave) at four moments in time. From the figure, it is clearly seen that energy cannot flow in either direction in the string since the lack of motion at the nodes prevents energy transmission. Thus, the energy remains constantly distributed or "standing" in the string and simply alternates form between potential and kinetic energy. Although, in the illustrated case, the two traveling waves had equal amplitudes, this is not a prerequisite for formation of a standing wave.

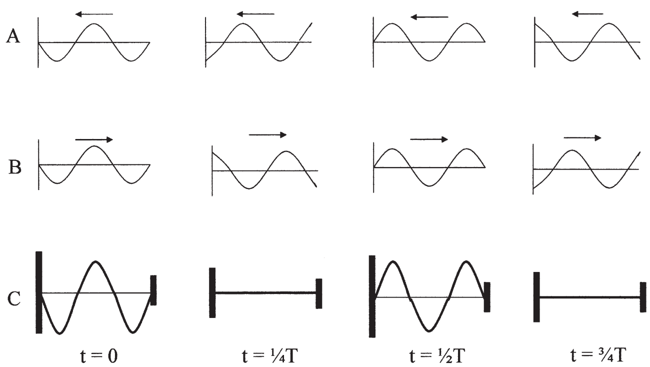


*Figure 6. Formation of a Standing Wave.*

Further examination of the figure reveals how the energy associated with the oscillating string shifts back and forth between kinetic and potential energy in a standing wave. At t = 0, every point in the string is at its maximum amplitude and the string is at rest. At this point, all of the string's energy is in the form of potential energy. At t = T/8 (not shown), an eighth cycle later, the tension in the string has forced the string to start moving back toward the equilibrium position and the potential and kinetic energies are equal. At t = T/4 seconds, another eighth of a cycle later, the string has no displacement but its particle velocities are at their maximum values. All of the energy at this point is, thus, in the form of kinetic energy. This cycling between kinetic and potential energy continues indefinitely in the same manner as that of the vibrating mass-spring oscillator.

*Vibrating String Reflection Laws*

The type of reflection that occurs when a traveling wave reaches an end of the vibrating string depends on the end conditions. As is shown in Figure 7, which is based on Resnick and Halliday (1977), if a pulse traveling along a string encounters a perfectly rigid end, it will exert an upward force on the support. Since the support is rigid, it does not move but it does exert an equal and opposite force on the string. This generates a reflected pulse that travels through the string in the opposite direction. A key item to note is that the reflected pulse is the exact reverse of the incident pulse.

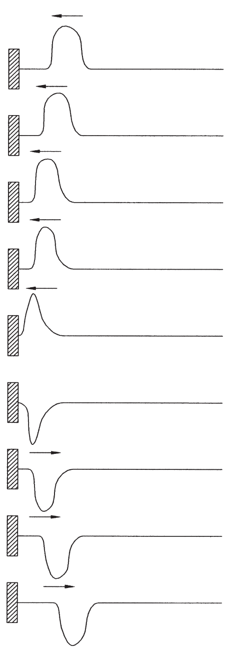

*Figure 7. Reflection of a Vibrating String Pulse from a Fixed End.*

The same effect occurs if a wavetrain traveling through a string encounters a fixed end. This will generate a reflected wavetrain and the combination of the two will set up a standing wave in the string. Since the displacement at any point in the standing wave is the sum of the displacements of the two traveling waves and since, by definition, the displacement at a fixed end must be zero, the reflected wave must exactly cancel the incident wave at the fixed end. This means that the two waves must be 180 degrees out-of-phase. Thus, one of the ground rules for a vibrating string is that reflection from a fixed end is accompanied by a 180 degree phase change.

The other extreme end condition that a vibrating string can have is a free end. As is shown in Figure 8, which mimics Resnick and Halliday (1977), a free end can be simulated by assuming that the string terminates in a very light ring that is free to slide without friction along a transverse rod. If a pulse traveling through the spring encounters the free end, it will exert an upward force on the ring. This causes the ring to accelerate upwards until it reaches the position where it has exactly the same amplitude as the pulse. However, when it reaches this point, the ring's inertia causes it to overshoot this position, such that it exerts an upward force on the string. This generates a reflected pulse that travels back through the string in the opposite direction. However, in direct contrast to the situation at the fixed end, the reflected pulse has the same sense as the original pulse.

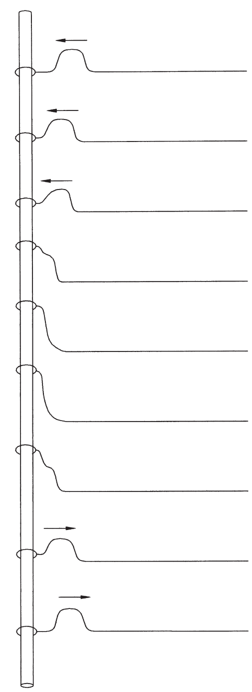

*Figure 8. Reflection of a Vibrating String Pulse from a Free End.*

The same effect occurs if a wavetrain traveling through a string encounters a free end. This will generate a reflected wavetrain and the combination of the two will set up a standing wave in the string. Since the displacement at any point in the standing wave is the sum of the displacements of the two traveling waves and since, by definition, the displacement at a free end must be a maximum, the incident and reflected waves must interfere constructively at the free end. Thus, the reflected wave must be in-phase with the incident wave at the free end. Accordingly, whenever a standing wave occurs in a string there must be a vibration node at all fixed ends and an antinode at all free ends.

The two cases just described represent the ideal cases where total reflection occurs at the end of a string. However, in the general case, there will be partial reflection at an end, along with partial transmission. For example, instead of the string being attached to a wall or being completely free, assume that its end is attached to that of another string under tension. An incident wave reaching the junction of the two strings would then be partially reflected and partially transmitted. In this case, the amplitude of the reflected wave will be less than that of the incident wave due to the energy that is lost to the adjacent string. Both the reflected and transmitted waves would have the same frequency as that of the incident wave. However, since the adjacent string would almost certainly have a different characteristic wave speed than the first string, the transmitted wave will travel at a different speed, and have a different wavelength, from the incident and reflected waves.

*Wave Speed in the Vibrating String*

As was stated previously, the velocity at which waves travel through the vibrating string is a characteristic property of the string that is determined by its elasticity and inertia. The equation for the wave speed is as follows:

$$c = \left(F / \mu\right)^{1/2} \tag{13}$$

Where:
c = Wave speed
F = Tension in string
$\mu$ = Mass per unit length of string

The tension represents the string's elasticity while the mass per unit length quantifies its inertia. In any given string, waves will always travel at the speed given by the above equation, regardless of their source, frequency, or shape. Additionally, the frequency of a wave will always be equal to the frequency of its source. Thus, if a given source is applied to two strings having different velocities, the generated waves will have equal frequencies but their wavelengths will be different.

*Resonance in the Vibrating String*

As is the case in most studies of oscillating systems, the concept of resonance is critical in pulsation analysis. In general, resonance can be defined as the condition where an elastic system is subjected to a periodic excitation at a frequency that is equal to or very close to one of its characteristic natural frequencies. At resonance, the system oscillates with amplitudes that are extremely large and are limited only by the amount of damping in the system.

For a vibrating string, any frequency that satisfies the boundary conditions at both ends is a natural frequency. For a string having two fixed ends, this occurs at all frequencies that give nodes at both ends. There may be any number of nodes in between or no nodes at all. Stated another way, the natural frequencies are those frequencies that yield an integral number of half-wavelengths, $\lambda/2$, over the length of the string. The natural frequencies for such a string are given by the following equation:

$$f_N = N \bullet c / \left(2 \bullet L\right) \tag{14}$$

Where:
$f_N$ = Nth natural frequency (Hz)
L = Length of string (inch)
c = Wave velocity in string (in/sec)
N = 1, 2, 3, etc.

The dependence of the natural frequencies on the wave velocity should be noted, as this is a fundamental concept in acoustics.

As stated previously, if a traveling wavetrain is introduced into one end of a string it will be reflected back from the other end and form a standing wave. If the traveling wave's frequency is equal or nearly equal to one of the string's natural frequencies, the standing wave will have a very large amplitude, much greater than that of the initial traveling wave. The amplitude will build up until it reaches a point where the energy being input to the string by the source of excitation is exactly equal to that dissipated by the damping in the system. In this condition, the string is said to be at resonance.

If the excitation frequency is more than slightly away from the string's natural frequencies, the reflected wave will not directly add to the excitation wave and the reflected wave can do work on the excitation source. The "standing" wave that is formed is not fixed in form but, instead, tends to "wiggle about." In general, the amplitude is small and not much different from that of the excitation. Thus, the string absorbs maximum energy from the excitation source when it is at resonance and almost no energy at all other conditions.

*Acoustic Waves*

When pulsations occur in a piping system, they are propagated through the system as acoustic waves or sound waves. Acoustic waves are mechanical waves that are highly similar to the vibrating string discussed above with one major exception—whereas the vibrating string represents a transverse wave, acoustic waves are longitudinal waves. The following treatment, which is also based on that given in Resnick and Halliday (1977), builds on the previous discussion of the vibrating string to describe the behavior of acoustic waves.

Figure 9, which is based on Resnick and Halliday (1977), illustrates how acoustic waves, or pressure pulsations, can be generated. The figure shows a piston at one end of a long tube filled with a compressible fluid. The vertical lines divide the fluid into "slices," each of which is assumed to contain a given mass of fluid. In regions where the lines are closely spaced, the fluid pressure and density are greater than those of the undisturbed fluid. Likewise, regions of larger spacings indicate areas where the pressure and density are below the unperturbed values.

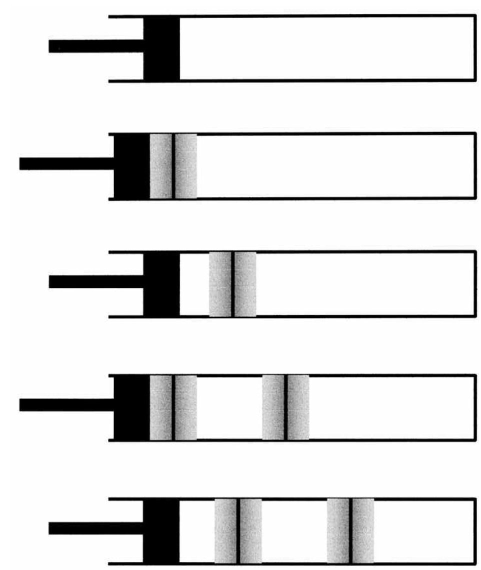

*Figure 9. Generation of Acoustic Waves by a Piston.*

If the piston is pushed to the right, it compresses the fluid directly adjacent to it and the pressure of that fluid increases above its normal, undisturbed value. The compressed fluid then moves to the right, compressing the fluid layers directly adjacent to it, and a compression pulse propagates through the tube. Similarly, if the piston is withdrawn from the tube (moved to the left in the figure), the fluid adjacent to it expands and its pressure drops below the undisturbed value. This generates an expansion, or rarefaction, pulse which also travels through the tube. If the piston is oscillated back and forth, a continuous series of compression and expansion pulses propagates through the tube. It should be noted that these pulses behave in exactly the same way as the longitudinal waves previously shown traveling in the mechanical spring of Figure 2.

A closer examination of the state of affairs in a compression pulse is provided in Figure 10, which is also based on Resnick and Halliday (1977). The figure shows a single compression pulse that could be generated by giving the piston of Figure 9 a short, rapid, inward stroke. The compression pulse is shown traveling to the right at speed, v. For the sake of simplicity, the pulse is assumed to have clearly defined leading and trailing edges (labeled "compression zone" in the figure) and to have uniform pressure and density within its boundaries. All of the fluid outside of this zone is assumed to be undisturbed. Following the lead of Resnick and Halliday (1977), a reference frame is chosen in which the compression zone remains stationary and the fluid moves through it from right to left at velocity, v.

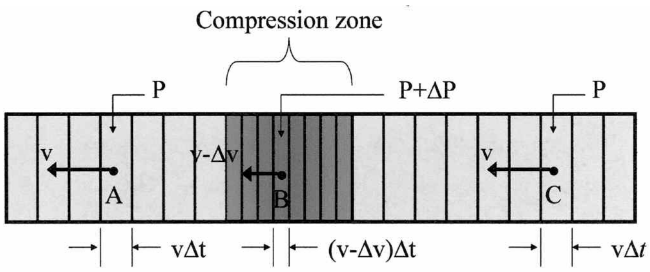


*Figure 10. Close-Up View of Compression Wave.*

If the motion of the fluid contained between the vertical lines at point C in the figure is examined, it will be seen that this fluid moves to the left at velocity, v, until it encounters the compression zone. At the moment it enters the zone, the pressure at its leading edge is P + ΔP while that at its trailing edge remains at P. Thus, the pressure difference, ΔP, between its leading and trailing edges acts to decelerate the fluid (i.e., F = ma) and compress it. Accordingly, within the compression zone, the fluid element has a higher pressure, P + ΔP, and a lower velocity, v − Δv, than it previously had. When the element reaches the left face of the compression zone, it is again subjected to a pressure difference, ΔP, which acts to accelerate it back to its original velocity, v (assuming no losses). After it has left the compression zone, it proceeds with its original velocity, v, and pressure, P, as is shown at point A.

Accordingly, an acoustic wave can be treated as either a pressure wave or a velocity (or displacement) wave. Since Resnick and Halliday (1977) show that, even for the loudest sounds, the displacement amplitudes are minuscule (approximately $10^{-5}$ m), it is usually more practical to deal with the pressure variations in the wave than the actual displacements or velocities of the particles conveying the wave. A very important point is that the pressure and velocity waves are always 90 degrees out-of-phase with each other. Thus, in an acoustic traveling or standing wave (which behave in the same manner as their vibrating string counterparts), when the displacement from equilibrium at a point is a maximum or minimum, the excess pressure is zero. Likewise, when the displacement and velocity at a point are zero, the pressure is either a maximum or minimum.

*Acoustic Wave Reflections*

When an acoustic wave traveling in a fluid-filled pipe reaches the end of the pipe, it will be reflected in exactly the same manner previously described for traveling waves in a vibrating string. Once again, interference between the incident and reflected waves gives rise to acoustic standing waves.

From the standpoint of reflection of a pressure wave, a pipe end that is completely open behaves in the same manner as a vibrating string's fixed end. Since a pulse impinging on an open end can generate absolutely no pressure (there is "nothing to squeeze the fluid against"), the acoustic pressure at an open end must be zero. Additionally, since the pressure at any point in the standing wave is the sum of the pressures of the two traveling waves and since this sum must be zero, the reflected wave must exactly cancel out the incident wave at the open end. This means that the two waves must be 180 degrees out-of-phase and have opposite signs (i.e., a compression wave is reflected as an expansion wave). Thus, one of the fundamental behaviors of acoustic waves is that reflection of a pressure wave from an open end is accompanied by a change in sign and a 180 degree phase change. Additionally, an open end is a pressure node in a standing wave.

Figure 11, based on Diederichs and Pomeroy (1929), illustrates the reflection. The top plot in this figure illustrates reflection of a pressure wave from an open end. The wave to the left of the dividing wall is the incident wave and that to the right is the reflected wave. The reflected waves are drawn in dotted line to indicate that they do not really exist in the position shown. Instead, the incident waves should be visualized as passing from left to right (as shown) but the reflected waves should be thought of as moving from right to left, starting at the partition.

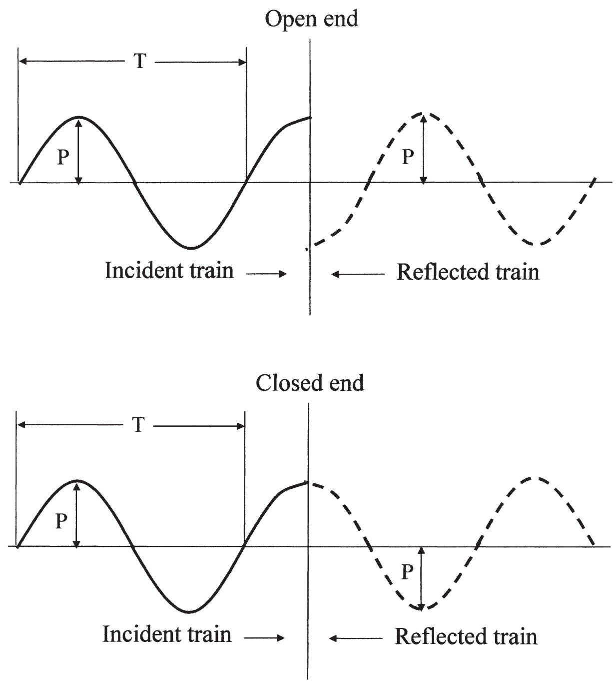


*Figure 11. Reflection of Pressure Waves from Closed and Open Pipe Ends.*

Examination of the figure indicates that the reflected wave has the same amplitude as the incident wave (a lossless system is assumed) and that it is 180 degrees out-of-phase with the incident wave. The sense of the reflected wave is also seen to be opposite that of the incident wave—that is, a positive incident wave generates a negative reflected wave. In other words, an incident compression wave will generate a reflected expansion wave. In order to determine the resultant pressure at the open end, the ordinates of the incident and reflected waves are simply added together. Thus, for the case shown, the pressure at the open end is seen to be zero. Similarly, the resultant pressure at any other point in the pipe can be obtained by adding the amplitudes of the two waves at that point.

On the other hand, reflection of a pressure wave at a closed end can be visualized by again referring to Figure 11, this time to the bottom plot. Examination of this plot indicates that the reflected wave has the same amplitude as the incident wave (a lossless system is assumed) and that it is perfectly in-phase with the incident wave. The sense of the reflected wave is also seen to be the same as that of the incident wave—that is, a positive incident wave generates a positive reflected wave. In other words, an incident compression wave will generate a reflected compression wave. The important thing to remember here is that a pressure wave encountering a closed end reflects with no change in sign and no phase change.

As was stated previously, the displacement and pressure waves are 90 degrees out-of-phase with each other. This means that the pressure variations (above and below the steady-state pressure) are maximum at displacement nodes and zero (i.e., constant pressure) at displacement antinodes. In other words, displacement nodes are equivalent to pressure antinodes and displacement antinodes are pressure nodes. Since most of the upcoming pulsation discussion will deal with the pressure wave, this means that closed pipe ends represent pressure antinodes and open ends are pressure nodes.

Resnick and Halliday (1977) provide a physical explanation for these relationships by pointing out that two small elements of fluid on either side of a displacement node are vibrating out-of-phase with one another. Thus, when they are approaching one another, the pressure at the node rises and when they withdraw from one another, the node pressure drops. On the other hand, two small fluid elements on opposite sides of a displacement antinode vibrate perfectly in-phase with one another, which allows the antinode pressure to remain constant.

As was the case with the vibrating string, the two reflection cases just described (i.e., closed end and open end) represent the ideal cases where total reflection occurs at the end of a pipe. However, in the general case, where the pipe end is neither completely closed nor open, there will be partial reflection at an end, along with partial transmission. For example, instead of the pipe being completely closed, assume that its end is attached to another pipe having a 25 percent smaller diameter. An incident wave reaching the junction of the two pipes would then be partially reflected and partially transmitted. In this case, the amplitude of the reflected wave will be less than that of the incident wave due to the energy that is lost to the second pipe. Both the reflected and transmitted waves would have the same frequency as that of the incident wave. However, if the second pipe had a different characteristic wave speed than the first, the transmitted wave would travel at a different speed, and have a different wavelength, from the incident and reflected waves.

*Acoustic Velocity*

As was the case with the vibrating string, acoustic waves traveling through a fluid in a pipe will always travel at the same wave velocity, which is referred to as the acoustic velocity. The basic equation for the acoustic velocity in a fluid is as follows:

$$c = \left(K_{BULK} / \rho\right)^{1/2} \quad (15)$$

Where:
c = Acoustic velocity
$K_{BULK}$ = Fluid adiabatic bulk modulus
ρ = Fluid density

This equation is seen to be in the exact same form as Equation (13) for the vibrating string. Specifically, the characteristic wave velocity is seen to be the square root of the ratio of the medium's elasticity (represented by the bulk modulus) to its inertia (characterized by the fluid density). Much more discussion on the acoustic velocity will be provided in upcoming sections of this tutorial.

*Acoustic Resonance*

The condition of acoustic resonance is responsible for the vast majority of pulsation-related problems experienced in piping systems. Similar to a vibrating string, a fluid within a pipe has certain acoustic natural frequencies. If the pipe is closed at both ends, resonance will occur for any frequencies that yield displacement nodes at both ends. There may be any number of nodes in between or no nodes at all. Stated another way, the natural frequencies are those frequencies that yield an integral number of half-wavelengths, λ/2, over the length of the pipe. The natural frequencies for such a pipe are given by the following equation:

$$f_N = N \bullet c / (2 \bullet L) \quad (16)$$

Where:
$f_N$ = Nth natural frequency (Hz)
L = Length of pipe (inch)
c = Acoustic velocity in pipe (in/sec)
N = 1, 2, 3, etc.

It is interesting to note that this is exactly the same as Equation (14), which gives the natural frequencies for a vibrating string having two fixed ends.

For a pipe having one end open and the other closed, the boundary conditions that must be satisfied are a displacement node at the closed end and a displacement antinode at the open end. These can be satisfied when the pipe length is exactly equal to a quarter-wavelength, λ/4, which yields a first natural frequency as follows:

$$f_1 = c / (4 \bullet L) \quad (17)$$

As stated previously, if a traveling acoustic wave is introduced into one end of a pipe it will be reflected back from the other end and form a standing wave. If the traveling wave's frequency is equal or nearly equal to one of the pipe's natural frequencies, the standing wave will have a very large amplitude, much greater than that of the initial traveling wave. The amplitude will build up until it reaches a point where the energy being input to the string by the source of excitation is exactly equal to that dissipated by the damping in the system. In this condition, the pipe is said to be at acoustic resonance.

Diederichs and Pomeroy (1929) provide a good description of the physics of acoustic resonance. As stated previously, acoustic wavetrains traveling in a pipe are reflected when they reach the end of the pipe, which generates a second wavetrain traveling in the opposite direction. This reflected wavetrain can, in turn, also be reflected once it reaches the opposite end of the pipe, thereby generating a third wavetrain that travels in the same direction as the initial wavetrain. These reflections can continue indefinitely until the system's damping is sufficient to finally cause the wavetrain to die out. As a result of all of these reflections and re-reflections, at any given time, there can be a multitude of wavetrains traveling throughout the pipe. All of these trains travel throughout the pipe as if the other trains were not there but, as described previously for the case of two wavetrains, all of the wavetrains combine to form a standing wave in the pipe. At any given point in the pipe, the pressure and velocity at any time are simply equal to the sum of the pressures and velocities of all of the trains passing that point at that instant.

Figure 12, which is based on Diederichs and Pomeroy (1929), illustrates how this multitude of wavetrains can rapidly combine to generate a standing wave having large amplitudes at resonance. Imagine a piston is being employed to excite a pipe having an open end opposite the piston. In the figure, horizontal distances represent time in seconds and vertical distances represent lengths along the pipe in feet. The two horizontal lines represent the two ends of the pipe—the upper line is the mid-plane of the piston (which acts as a closed end) and the lower is the open end. Since the system is assumed to be in resonance with the first acoustic mode, the pipe length, L, is equal to c × T/4, where T is the period in seconds.

In the figure, the pressure wave generated by the piston, designated wave A, is shown on the upper line beginning at time zero. In accordance with the physics of this system, this pressure wave will arrive at the open end as an incident wave, $A_1$, at time, T/4. The laws of reflection at an open end then yield a negative wave, $A_2$, which travels back through the pipe toward the piston. Upon reaching the piston at time, T/2, this wave is reflected without a change in sense and produces another negative wave of equal magnitude (the system is assumed to have no losses), $A_2$. However, at this exact same time, the piston is producing another negative wave, $A_3$. The fact that these two waves are perfectly in-phase with

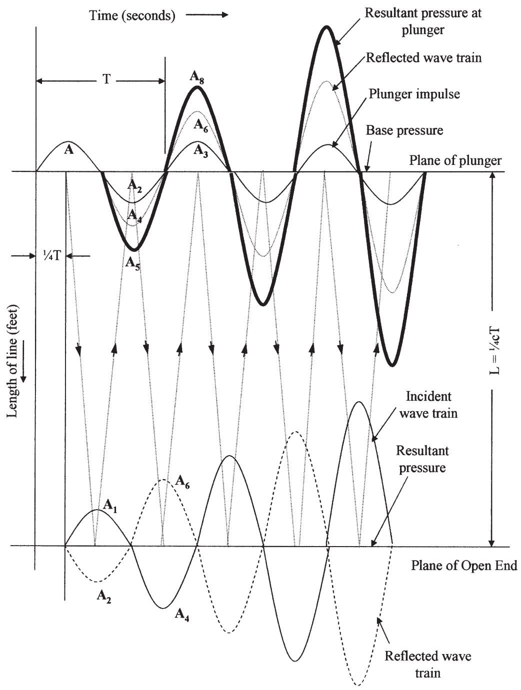


*Figure 12. Illustration of Acoustic Resonance.*

one another is the driver of the resonance phenomenon. Thus, these two waves are additive and the total pressure proceeding down the pipe toward the open end is the sum of $A_2$ and $A_3$, or $A_4$ (which has an amplitude of 2A). However, the total pressure at the piston is greater than that since, at any instant, it is the sum of the pressures in the incident wave, reflected wave, and piston-generated wave. Thus, the pressure at the piston is $A_2 + A_2 + A_3 = A_5$, which has an amplitude of 3A.

The negative wave, $A_4$, traveling down the pipe then encounters the open end T/4 seconds later and is reflected with a change of sense. The reflected wave is, thus, a positive wave, $A_6$ (which has an amplitude of 2A). This wave then travels back toward the piston and arrives T/4 seconds later. It again reflects from the piston without a change in sense at the same time that the piston is generating a positive wave, $A_3$. The pressure traveling back down the pipe is then the sum of $A_6$ and $A_3$, or $A_7$, which has an amplitude of 3A. However, the amplitude of the pressure at the piston is the sum of $2A_6$ and $A_3$, or $A_8$, which has an amplitude of 5A. Positive wave, $A_7$, then travels down the pipe and this phenomenon repeats itself indefinitely. At the open end, the reflected and incident waves always cancel so there are no pressure oscillations there. However, as has been shown, the pressure amplitude at the piston builds up rapidly. In fact, in the absence of damping, none of the waves would ever die out and the pressure at the piston would, theoretically build up to infinity. The potentially damaging ramifications of such a phenomenon on a practical piping system are quite easy to visualize.

## ACOUSTIC VELOCITY

It is hoped that the previous section made it clear that the acoustic velocity of a fluid in a piping system plays a large role in determining the system's pulsation behavior. In this section, the meaning of acoustic velocity is discussed more fully and equations are given for several situations of interest.

The technical definition of acoustic velocity is the rate at which a pressure disturbance (or noise) travels within a fluid. If a fluid is at rest, pressure disturbances will travel in any direction at the acoustic velocity. However, if the fluid is flowing in a pipe with a given average velocity, the upstream and downstream propagation velocities are different. If the disturbance is traveling upward, it travels at the difference between the acoustic velocity and the flow velocity—if it is traveling downstream, the velocities are summed. However, since in almost all practical pumping applications, the flow velocity is at least an order of magnitude smaller than the speed of sound, it is safe to treat pulsations as if they travel at the acoustic velocity in all directions. Per Resnick and Halliday (1977), typical acoustic velocities for air and water are 1087 and 4760 ft/sec, respectively.

Per Brennen (1994), the acoustic velocity for any fluid is given by the following:

$$c = \left(d\rho / dP\right)^{-1/2} \tag{18}$$

Where:
c = Acoustic velocity
P = Fluid pressure
ρ = Fluid density

In words, this means that the acoustic velocity of a fluid depends on how much a change in fluid pressure causes the fluid density to increase (or, alternatively, the volume to decrease). From this definition, it is seen that, for the purely theoretical case of a perfectly incompressible fluid, the acoustic velocity is infinite. However, since all real fluids have some compressibility, all have finite acoustic velocities. A measure of a fluid's compressibility is its bulk modulus, $K_{BULK}$, which is defined as follows:

$$K_{BULK} = \Delta P / \left(\Delta V / V\right) \tag{19}$$

Where:
ΔP = Change in fluid pressure (psi)
ΔV = Change in fluid volume
V = Initial fluid volume

Using this parameter, the general equation for a fluid's acoustic velocity is as follows:

$$c = \left(K_{BULK} / \rho\right)^{1/2} \tag{20}$$

If the walls of the pipe contain some flexibility, then an increase in pressure will impact the density of the fluid through two phenomena. First, the fluid's volume will change due to the compressibility of the liquid. Second, the flexibility of the pipe walls will also result in a change in fluid volume and density. Thus, the flexibility of the pipe walls acts as another "compressibility" in the system, which acts to further lower the acoustic velocity.

Accordingly, the above equation is strictly only valid when the pipe walls can be considered to be rigid. However, since gases have much smaller bulk moduli than liquids, pipe wall flexibility effects are seldom of importance when analyzing gases. Conversely, for liquids, these effects often have significant impact and are normally accounted for using the following equation, known as the Korteweg correction (which is valid for thin-walled pipes, having a wall thickness less than 1/10 of the diameter):

$$c = c_{RIGID} / \left[1 + K_{BULK} \bullet d / \left(E \bullet t\right)\right]^{1/2} \tag{21}$$

Where:
c = Actual acoustic velocity (ft/sec)
$c_{RIGID}$ = Acoustic velocity calculated using rigid pipe assumption (ft/sec)
$K_{BULK}$ = Fluid bulk modulus (psi)
d = Pipe diameter (inch)
E = Pipe elastic modulus (psi)
t = Pipe wall thickness (inch)

Obviously, the above correction only applies to pipes having circular cross-sections. However, Wylie and Streeter (1993) point out that when subjected to an internal pressure, a pipe having a noncircular cross-section deflects into a nearly circular section.

Since the increase in cross-sectional area for a given pressure change is greater than that for a circular pipe, the noncircular cross-section pipe has a larger effective compliance. Accordingly, employment of noncircular conduit can greatly reduce the acoustic velocity that, as will be shown later, is sometimes desirable.

If a liquid contains even minute amounts of entrained gas, or gas that has come out of solution, the acoustic velocity is drastically reduced. The presence of the gas has two effects—it modifies the effective bulk modulus of the fluid and it also decreases its effective density. Singh and Madavan (1987) provide the following equation for the acoustic velocity in a liquid-gas mixture containing small amounts of gas:

$$c_M = c_0 \bullet \left[1 + \left(V_{GAS} / V_L\right)\right] / \left[1 + K_{BULK} \bullet \left(V_{GAS} / V_L\right) / P_L\right]^{1/2} \quad (22)$$

Where:
$c_M$ = Acoustic velocity in mixture
$c_0$ = Acoustic velocity in liquid
$K_{BULK}$ = Liquid bulk modulus (psi)
$P_L$ = Line pressure (psi)
$V_{GAS}$ = Gas volume
$V_L$ = Liquid volume

Wylie and Streeter (1993) present a plot that shows how the introduction of entrained air into a liquid can result in a dramatic reduction in acoustic velocity. It is noteworthy that the introduction of an amount of air as small as 0.1 percent by volume can essentially cut the acoustic velocity in half.

## PIPING ACOUSTIC BEHAVIOR

### *Organ Pipe Resonances*

The two fundamental "organ pipe" resonances are the half-wavelength resonance and the quarter-wavelength resonance. A half-wavelength resonance occurs in a pipe of constant diameter that has the same condition at both ends—either both open or both closed. For a pipe having two closed ends (also known as "closed-closed pipe"), any vibrating frequency that yields displacement nodes at both ends is a natural or resonant frequency of the system. There may be any number of nodes in between or no nodes at all. Stated another way, the natural frequencies are those frequencies that yield an integral number of half-wavelengths, $\lambda/2$, over the length of the pipe. The natural frequencies for such a pipe are given by the following equation:

$$f_N = N \bullet c / (2 \bullet L) \quad (23)$$

Where:
$f_N$ = Nth natural frequency (Hz)
L = Effective length of pipe (inch)
c = Acoustic velocity in pipe (in/sec)
N = 1, 2, 3, etc.

Another way of looking at this is via the phenomenon of the buildup of a standing wave illustrated in Figure 12. Assume that a closed-closed pipe has a sinusoidal excitation source, such as the piston of Figure 9, located at one of its ends (a piston always behaves as a moving closed end). As the piston oscillates back and forth, it generates alternating compression and expansion pulses. Let us follow the motion of one of the compression pulses. Since the pulse travels at the acoustic velocity, c, it reaches the other closed end in a time of L/c seconds. Since a closed end reflects a wave with no change in sense, the reflected wave is also a compression wave. The reflected compression wave then travels back toward the piston and again takes L/c seconds to traverse the pipe. Thus, the total time elapsed from the generation of the initial pulse to the return of the reflected pulse is 2L/c seconds.

What happens at the piston is now highly dependent on the phasing between the pulse being generated at the piston and the reflected pulse. Since the reflected pulse is a compression pulse, it will add to the piston's pulse if the piston is just starting to generate another pressure pulse. By definition, the period of the piston's motion is the time between the generations of consecutive compression (or expansion) pulses. It is, thereby, seen that if the pulse's travel time, 2L/c, is exactly equal to the period of the piston, the newly generated wave will be perfectly in-phase with the reflected wave and resonance will occur. Thus, the resonant period is 2L/c or, in other words, the resonant frequency is c/2L, which is the value given by the above equation when N equals one.

However, that is not the only resonant frequency for the system. If the piston's period is one-half that of the previous case, or L/c, it will generate a compression pulse at the time, L/c, that the initial pulse reaches the other closed end and a second compression pulse at the time, 2L/c, the reflected pulse reaches the piston. Since this second pulse will be in-phase with the reflected pulse, this is also a resonant condition. Taking this to the general case, it can be seen that resonance will occur whenever the period of the piston is equal to 2L/Nc, as long as N is an integer. This translates into resonant frequencies of Nc/2L, as are given by the above equation.

In order to illustrate the system's physical behavior, mode shapes are plotted for acoustic standing waves in the same fashion as they are for mechanical vibrations. The only difference is that each acoustic mode has two distinct mode shapes—one for pressure and one for velocity. The pressure mode shape provides the amplitudes of pressure fluctuations at each point along the pipe. Similarly, the velocity mode shape shows the velocity sinusoidal amplitudes at each point within the pipe. As has been stated before, the two mode shapes are always 90 degrees out-of-phase with each other.

Figure 13 provides the pressure and velocity mode shapes for the first mode for the closed-closed pipe just discussed. Since this is the half-wave resonance, it comes as no surprise that both mode shapes are in the shape of a half-wave. The pressure mode shape has antinodes at both closed ends and a node at mid-length while the velocity mode shape is exactly the opposite. The ways that the pressure varies with time are also depicted for several points along the pipe.

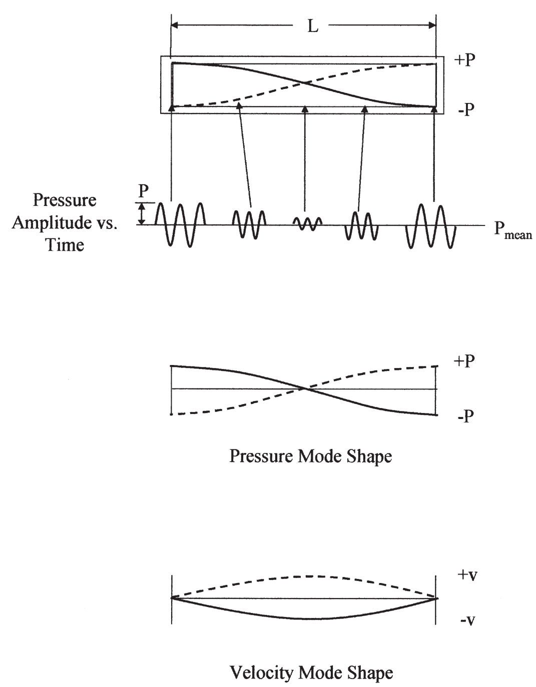


*Figure 13. Pressure and Velocity Mode Shapes.*

The first mode is known as the fundamental mode and the higher order modes are known as overtones. In cases such as this, where the higher order frequencies are simply integer multiples of the fundamental, these modes are referred to as harmonics. The pressure mode shapes for the first three modes are provided in Figure 14. The fundamental mode is seen to consist of a single half wave, the second mode contains two half waves, and so on.

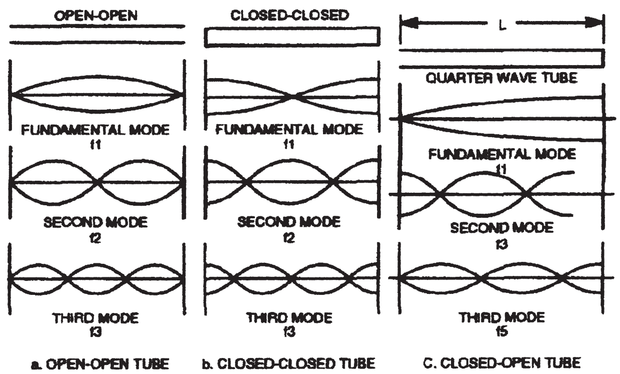


*Figure 14. Pressure Modes Shapes for Organ Pipe Modes. (Courtesy of Jungbauer and Eckhardt, 1997, Turbomachinery Laboratory)*

For an open-open pipe, the situation is very similar except the boundary conditions that must be satisfied are displacement antinodes at both ends. The natural frequencies for this case are again given by Equation (23) and the first three mode shapes are provided in Figure 14. It is seen that although the natural frequencies are identical to those for a closed-closed pipe, the mode shapes are different.

Kinsler, et al. (1982), state that the effective length of an open-open pipe is not the physical length but, rather, is given by the following:

$$L_{EFF} = L + 8 \bullet r / (3 \bullet \pi) \tag{24}$$

Where:
$L_{EFF}$ = Effective pipe length
L = Actual pipe length
r = Pipe radius

The other major organ pipe resonator is a pipe of constant diameter having one open end and one closed end, or an open-closed pipe. The boundary conditions that must be satisfied for this pipe are a displacement antinode at the open end and a displacement node at the closed end. The natural frequencies for such a pipe are given by the following equation:

$$f_N = N \bullet c / (4 \bullet L) \tag{25}$$

Where:
$f_N$ = Nth natural frequency (Hz)
L = Length of pipe (inch)
c = Acoustic velocity in pipe (in/sec)
N = All odd integers (i.e., 1, 3, 5, etc.)

The pressure mode shapes for the first three modes are again provided in Figure 14. Similar to the open-open pipe, the fundamental mode is seen to consist of a single quarter-wave. For this reason, the open-closed pipe is often referred to as a "quarter-wave stub." However, in direct contrast to the open-open and closed-closed pipes, all of the harmonics are not present. Specifically, all of the even harmonics are missing. This is because the even harmonics of the fundamental all consist of integral multiples of half-wavelengths that have already been shown to yield the same condition at both ends (i.e., either both nodes or both antinodes). Since the open-closed pipe requires opposite boundary conditions at the two ends, only the odd harmonics are present.

Similar to the open-open pipe, the effective length of the quarter-wave stub is also greater than the actual length. Rogers (1992) states that experimental measurements indicate that the effective length can be anywhere from 1 to 12 percent greater than the actual length.

An important point to remember is that the above simplified relations are only applicable to segments of pipe where the diameter is constant. If a segment contains a change in diameter, even a small one, reflections occur (as will be discussed shortly) and the above equations are invalidated. Another way of stating this is that the above equations only pertain to pipes where the characteristic impedance (defined in the next section) is constant.

*Impedance*

A highly important concept in the study of pulsations is impedance, which comes straight from electrical theory. In general, the impedance of an acoustic component is that component's resistance to acoustic particle flow. However, since there are several different definitions of impedance, this is a concept that is often misunderstood. In order to keep this as simple as possible, the only impedance referred to herein will be the acoustic impedance.

The acoustic impedance, Z, at any point, x, in an acoustic system is defined as the complex ratio of the oscillatory pressure, P, to the oscillatory volumetric flowrate, Q, at that point, as follows:

$$Z = P(x,t) / Q(x,t) \tag{26}$$

Where:
Z = Acoustic impedance at point x
P (x, t) = Oscillatory pressure at point x
Q (x, t) = Oscillatory volumetric flow at point x

Per Wylie (1965), the acoustic impedance is seen to be a complex number calculated from physical and dynamical flow properties of the system. In a particular pipe, it varies with x, boundary conditions, and frequency of oscillation. Per Kinsler, et al. (1982), in the metric system, the acoustic impedance units are Pa-sec/$m^3$, often termed an acoustic ohm.

The characteristic acoustic impedance for a fluid filled pipe is given by the following:

$$Z_0 = \rho \bullet c / A \tag{27}$$

Where:
$Z_0$ = Characteristic impedance
ρ = Fluid density
c = Acoustic velocity
A = Cross-sectional area

It is important to understand that, in general, the acoustic and characteristic impedances are not the same. The characteristic impedance refers to the simple case where a single traveling wave propagates through the pipe with no reflections. In a pipe of constant diameter, the characteristic impedance is constant everywhere in the pipe. It should be noted that in an infinite line, where there are no reflections, the acoustic impedance is equal to the characteristic impedance everywhere.

However, as will be discussed in the next section, most real pipe problems will involve two waves—an incident wave propagating in the positive direction and a reflected wave propagating in the negative direction. The acoustic impedance of a pipe containing both of these waves can then be shown to be:

$$Z = Z_0 \bullet (P_I + P_R) / (P_I - P_R) \tag{28}$$

Where:
Z = Acoustic impedance of pipe
$Z_0$ = Characteristic impedance of pipe
$P_I$ = Pressure amplitude of incident wave
$P_R$ = Pressure amplitude of reflected wave

In general, the acoustic impedance at any point is a complex number and can be expressed as follows:

$$Z_x = R_x + j \bullet X_x \tag{29}$$

Where:
$Z_X$ = Acoustic impedance at point x
$R_X$ = Resistive component of impedance
$X_X$ = Reactive component of impedance
j = Square root of negative one

The acoustic impedance at a constant head reservoir, or open end, is zero. The acoustic impedance at a closed end is infinity. A simple line that is connected to a reservoir or open end behaves as a fluid inertia. On the other hand a simple line that is connected to a closed end behaves as a hydraulic compliance.

*Reflections*

As has been stated previously, when an acoustic traveling wave propagating through a pipe encounters a closed or open end, it is reflected, and a traveling wave moving in the opposite direction is generated. These are specific cases of the more general rule that whenever a traveling wave moving through a piping system encounters a change in characteristic acoustic impedance, $Z_0$, a reflection occurs. Except in the extreme case of an open end, two new traveling waves are generated—a reflected wave and a transmitted wave. The reflected wave is the wave that travels back through the pipe in the opposite direction from the incident wave. Its frequency, propagation speed, and wavelength are always the same as those of the incident wave. With the exception of the special cases of complete reflection that occur at closed and open ends (where the pressure amplitudes are equal), the amplitude of the reflected wave is always smaller than that of the incident wave.

Up to this point, scant mention has been given to the second wave generated, the transmitted wave. The transmitted wave is another traveling wave that proceeds in the second pipe in the same direction as the incident wave. The transmitted wave will always have the same frequency as the incident wave but, depending on the fluid properties in the second pipe, its acoustic velocity and wavelength could be different from those of the incident wave. In stark contrast to the case with the reflected wave, the pressure and displacement amplitudes associated with the transmitted wave are often larger than those of the incident wave, with a maximum amplitude increase of 2.0 for a closed pipe.

Per Campbell and Graham (1996), physical elements that impose such characteristic impedance changes include the following:

- Closed ends (infinite impedance).
- Open ends (zero impedance).
- Changes in pipe diameter (expansions and contractions).
- Branches.
- Tees.
- Flow restrictions, such as orifices, valves, etc.
- Changes in density or acoustic velocity.

Conspicuous by their absence from the above list are elbows and bends. This is in accordance with Chilton and Handley (1952), who assert that laboratory tests have proven conclusively that elbows and bends do not form reflection points for pressure pulses since they do not represent a change in acoustic impedance.

In general, the ratios of the pressure amplitudes of the reflected and transmitted waves to those of the incident wave depend on the characteristic acoustic impedances and acoustic velocities in the two pipe elements. Two characteristic parameters associated with any reflection are the pressure transmission and reflection coefficients, which are defined by Kinsler, et al. (1982), as follows:

$$T = P_T / P_I \tag{30}$$

$$R = P_R / P_I \tag{31}$$

Where:
T = Pressure transmission coefficient
R = Pressure reflection coefficient
$P_I$ = Complex pressure amplitude of incident wave
$P_T$ = Complex pressure amplitude of transmitted wave
$P_R$ = Complex pressure amplitude of reflected wave

Another parameter that is of interest at a reflection, especially when the effectiveness of acoustic filtering devices is being evaluated, is how the incident wave's acoustic energy is divided between the reflected and transmitted waves. Although there are several parameters that measure the acoustic energy, Kinsler, et al.'s (1982), acoustic intensity will be employed in this tutorial. Kinsler, et al. (1982), define the acoustic intensity of a sound wave as the average rate of flow of energy through a unit area normal to the direction of propagation. Using this definition, the acoustic intensity can be shown to be given by the following equation:

$$I = P^2 / \left(2 \bullet \rho \bullet c\right) \tag{32}$$

Where:
I = Acoustic intensity ($W/m^2$)
P = Complex pressure amplitude
ρ = Fluid density
c = Acoustic velocity

Kinsler, et al. (1982), also note that the intensity transmission and reflection coefficients are real and are defined by:

$$T_I = I_T / I_I \tag{33}$$

$$R_I = I_R / I_I = |R|^2 \tag{34}$$

Where:
$T_I$ = Intensity transmission coefficient
$R_I$ = Intensity reflection coefficient
$I_I$ = Intensity amplitude of incident wave
$I_T$ = Intensity amplitude of transmitted wave
$I_R$ = Intensity amplitude of reflected wave

Kinsler, et al. (1982), give the following equations for the pressure transmission and reflection coefficients for a simple step change of characteristic impedance, from $Z_1$ to $Z_2$, in a pipe:

$$R = \left(Z_2 - Z_1\right) / \left(Z_1 + Z_2\right) \tag{35}$$

$$T = 2 \bullet Z_2 / \left(Z_1 + Z_2\right) \tag{36}$$

The intensity transmission and reflection coefficients for that same step change are as follows:

$$R_I = \left[\left(Z_2 - Z_1\right) / \left(Z_1 + Z_2\right)\right]^2 \tag{37}$$

$$T_I = 4 \bullet \left(Z_1 / Z_2\right) / \left[\left(Z_1 / Z_2\right) + 1\right]^2 \tag{38}$$

If the flow areas of the upstream and downstream pipes are $S_1$ and $S_2$, respectively, Kinsler, et al. (1982), and Diederichs and Pomeroy (1929) both show that the above equations can be written in terms of the areas as follows:

$$R = \left(S_1 - S_2\right) / \left(S_1 + S_2\right) \tag{39}$$

$$T = 2 / \left[ \left( S_2 / S_1 \right) + 1 \right] \tag{40}$$

$$R_I = \left[ \left( S_1 - S_2 \right) / \left( S_1 + S_2 \right) \right]^2 \tag{41}$$

$$T_I = 4 \bullet \left( S_1 / S_2 \right) / \left[ \left( S_1 / S_2 \right) + 1 \right]^2 \tag{42}$$

It should be noted that all of the above equations implicitly assume that no losses occur during reflection. The above equations have several important ramifications including the following:

- The reflection coefficient, R, is always real. Thus, the reflected wave must always be in-phase or 180 degrees out-of-phase with the incident wave.
- The reflection coefficient, R, is always less than or equal to unity. This means that the pressure amplitude in the reflected wave is always less than that in the incident wave, unless the reflection is from a closed end.
- For the trivial case of no area change ($S_1 = S_2$), the reflection coefficient, R, is zero and the transmission coefficient, T, is unity. This just states the obvious fact that there is no reflection if there is no impedance change.
- For a sudden contraction ($S_1 > S_2$), the reflection coefficient, R, is positive. This means that the reflected wave is in-phase with the incident wave and has the same sense (i.e., a compression wave is reflected as a compression wave). This is the same result that was obtained for closed ends earlier.
- In the limiting case for a contraction ($S_1 >> S_2$), the reflection coefficient, R, approaches unity and the transmission coefficient, T, approaches 2.0. This simply confirms that a closed end generates a reflected wave that is equal in amplitude to the incident wave and that it generates a pressure that is twice that of the incident wave.
- For a sudden expansion ($S_1 < S_2$), the reflection coefficient, R, is always negative. This means that the reflected wave is 180 degrees out-of-phase with the incident wave and has opposite sense (i.e., a compression wave is reflected as an expansion wave and vice versa). This is the same result that was obtained for open ends earlier.
- In the limiting case for an expansion ($S_1 << S_2$), the reflection coefficient, R, approaches −1.0 and the transmission coefficient, T, approaches zero. This simply confirms that an open end generates a reflected wave that is equal in amplitude but opposite in sense to the incident wave and a transmitted wave that has a pressure amplitude of approximately zero.
- The transmission coefficient, T, is always real and positive. Thus, the transmitted wave must always be in-phase with the incident wave and it also must have the same sense (i.e., a compression wave generates a transmitted compression wave).
- For a sudden contraction ($S_1 > S_2$), the transmission coefficient, T, is greater than unity. This means that the transmitted wave always has a larger amplitude than the incident wave.
- For a sudden expansion ($S_1 < S_2$), the transmission coefficient, T, is less than unity. This means that the transmitted wave always has a smaller amplitude than the incident wave.
- If the areas are significantly different (i.e., $S_1 >> S_2$ or $S_1 << S_2$), the intensity transmission coefficient, $T_I$, approaches zero. Thus, the system acts as an acoustic filter, protecting the downstream line from high energy pulsations. This will be addressed in greater detail in a subsequent section.

The above rules can be generalized by stating that whenever a compression wave encounters an increase in impedance, it is reflected as a compression wave and whenever it reaches a decrease in impedance, it is reflected as an expansion wave.

*Resonance*

It should be emphasized that the existence of quarter-wave and half-wave resonant modes in a piping system does not, by itself, necessarily mean that the system will encounter pulsation problems. In fact, all piping systems will have such modes, along with other resonant modes that are more complex. In order for these modes to represent a problem, there must be a means for exciting them. In most cases, the excitation is in the form of a flow or pressure variation generated by the pump. As has been stated before, resonance will occur when the total time it takes for the excitation to travel up the pipe, be reflected, and return to the source is such that the reflected wave is perfectly in-phase with a subsequent perturbation being generated by the source. Another way of saying this is that the frequency of the excitations must match one of the resonant frequencies of the system. When this happens, standing waves are formed in the piping system and the pressure and displacement amplitudes can build up to very large values in the manner described previously. Since the large pressure amplitudes are the element of most concern, the pressure antinodes are the areas of concern. It should be remembered that, even at resonance, the fluctuating pressures at a pressure node are zero.

If the system contained absolutely no damping, the pressure fluctuations at the antinodes would theoretically be infinite. However, all real piping systems contain acoustic damping due to the following mechanisms, taken from Wachel, et al. (1995):

- Viscous fluid action (intermolecular shearing)
- Piping resistance (pipe roughness, restrictions, orifices, etc.)
- Transmission (lack of total reflection) at line terminations, junctions, diameter changes, etc.

While the damping created by the first two items is probably easy to visualize, the effect of transmission may be less so. In order to understand this, it must be remembered that the previous description of amplitude buildup at resonance assumed perfect reflections at both ends of the pipe. That means that the reflected wave had the same amplitude as the incident wave. However, as was shown in the last section, the only cases where perfect reflections occur are when there are perfectly closed or open ends. In all other cases, there is only partial reflection and the reflected wave is smaller in amplitude than the incident wave. A glance at Figure 12 reveals that these losses would have the impact of retarding the amplitude growth at resonance.

In flowing piping systems, the damping is a direct function of the $\rho v^2$ pipe friction losses. For typical flows and flow velocities seen in pumps, these losses normally dwarf those that occur on the molecular level, such that the first item on the above list is almost always inconsequential. Since the pipe friction varies with the square of flow, the system damping is much larger for higher steady-state flows than for low flows. For this reason, Sparks (1983) observes that acoustic resonances are more prominent at low flows, where the system is very lightly damped.

Even when the flows are not low, most pump piping systems are relatively lightly damped from an acoustic standpoint. A measure of the amount of damping is the amplification factor at resonance, or Q factor, which is analogous to that associated with mechanical vibrations. The larger the amplification factor, the sharper the peak (corresponding to lighter damping) and the more dangerous the mode is. The acoustic amplification factor can be calculated using the amplitude bandwidth factor commonly used for mechanical systems. To implement this, the two half power points (frequencies where the pressure amplitudes are 0.707 times the amplitude at resonance) must be identified from a frequency response (also known as Bode) plot. The difference between these two frequencies is then designated the bandwidth and the amplification factor is obtained by simply dividing the resonant frequency by the bandwidth.

Although few would argue that pump piping acoustic systems are lightly damped, a review of the literature reveals some significant discrepancies in amplification factor magnitudes. Parry (1986) states that amplification factors can be as high as 15 while Beynart (1999) cites a value of 40. Wachel and Price (1988) effectively span these values when they provide a range from 10 to 40. At the extreme end of things are references, including Schwartz and Nelson (1984) and Singh and Madavan (1987), which cite a maximum possible amplification factor of 100. In general, the amplification factor is dependent on flow, line size, and frequency.

It should be remembered that resonance can only occur when reflections allow a standing wave to build up. Since an infinite pipe gives rise to no reflections, it cannot suffer from resonance. Accordingly, the pulsation levels at all locations in the pipe are simply equal to those introduced by the excitation source.

*Acoustic Behavior of "Real" Piping*

Per the authors' experience, the two organ pipe resonances most likely to be excited in turbomachinery and piping systems are the open-open and open-closed types. Although closed-closed modes are occasionally encountered in pulsation control bottles and acoustic filters, most practical piping is open at least at one end. As Diederichs and Pomeroy (1929) point out, since in practically all installations, the pump discharge line ends in a manifold connection or connections of larger cross-section, the pump discharge line almost always has an open end at its terminal end. Likewise, since the pump suction line usually begins at a reservoir or tank, the suction pipe almost always begins with an open end. Additionally, in a reciprocating pump, the pump ends of the discharge and suction lines normally behave as closed ends, unless a surge volume or accumulator is employed, since a pressure wave traveling toward the pump will either hit a closed valve or a piston (a piston behaves as a moving closed end).

Although there are many examples of perfectly closed and open ends in practical pump piping systems, Diederichs and Pomeroy (1929) state that there are also many cases where a configuration can be treated as a closed or open end. For instance, although the only condition where a valve represents a perfectly closed end is when it is completely closed, many times when a valve is only open a small amount, it acts like a closed end. A pipe also appears to be closed-ended when it ends in a significant decrease in area. According to API 618 (1995), if the diameter reduction is two-to-one or more, a contraction can be treated as a closed end.

On the other hand, when a pipe is connected to a reservoir, tank, or chamber, it behaves as if it is open-ended. Additionally, API 618 (1995) states that if a pipe is connected to a pipe having a diameter that is at least twice as large, it can be considered to be open-ended. Furthermore, per Diederichs and Pomeroy (1929), the dividing surface between a liquid and a lighter fluid, such as air, behaves as an open end, even when both fluids are pressurized.

Lewis, et al. (1997), provide a good illustration of how the resonance modes of concern are sensitive to the boundary conditions at the end of the pipe. They describe a case in which they had a control valve located at one end of a pipe whose other end was open. When the valve was open only 30 to 40 percent, it essentially behaved as a closed end, as was confirmed by the presence of quarter-wave resonances. On the other hand, when the valve was open 80 percent or more, the valve behaved as an open end, which changed the system to open-open, and half-wave resonances were observed.

Wylie and Streeter (1993) state that every series piping system has two different acoustic natural frequencies. The first is the theoretical frequency one obtains by using the organ pipe relations and making reasonable approximations in areas where there is a change in diameter. The second is the actual frequency observed in the field. Unfortunately, except in the simple case of a constant diameter pipe, the actual resonant frequency is usually quite different from the theoretical frequency. The difference between the theoretical and actual resonant frequencies results from the partial reflections that occur at each impedance change.

Even in the simplest configurations, the theoretical and actual resonant frequencies are often quite different. The authors demonstrated this by analyzing the two configurations shown in Figure 15, which are assumed to be filled with water, using their acoustic simulation code. The top configuration, referred to as the constant area pipe, is a classic quarter-wave stub whose fundamental frequency was found to be 58.7 Hz, which is very close to that obtained using Equation (25).

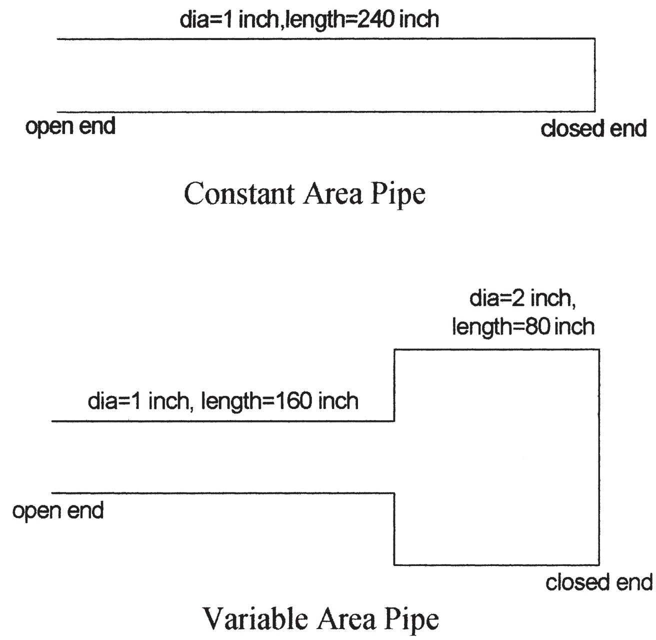


*Figure 15. Constant and Variable Area Pipes.*

The problem arises when it comes time to analyze the bottom configuration, referred to as the variable area pipe. Since it has the same total length as the constant area pipe and has the same diameter over two-thirds of that length, one would think that its fundamental frequency could be reasonably approximated. One thought might be to analyze the one inch diameter pipe as an open-open pipe using Equation (23), which yields a fundamental frequency of 176 Hz. Another reasonable approach might be to treat the two inch diameter pipe as an open-closed pipe using Equation (25), which also yields a fundamental frequency of 176 Hz. A third approach might be to ignore the change in diameter and simply look at the entire pipe as an open-closed pipe, which yields the same value obtained for the constant area pipe, 58.7 Hz. However, the actual fundamental frequency determined by the code is 36.8 Hz, which is far from all of the "theoretical" values. It should be noted that the variable area pipe does have a mode in the vicinity of 176 Hz (at 179.9 Hz) but it is the second mode, not the fundamental.

In more complex systems, such as those containing branches or parallel paths, the theoretical frequency does not even have any meaning since there is no simple manner of calculating it that can be reasonably justified. However, as in the preceding example, the actual resonant frequency can be calculated with the aid of a good acoustic analysis computer code. As is the case in simple pipes, higher order natural frequencies are also present in complex systems. However, unlike the situation with the organ pipe resonances, the higher order frequencies in complex systems are seldom harmonics of the fundamental.

When a piping system is subjected to an excitation whose frequency is not in resonance with any of the system's acoustic natural frequencies, the excitation wave simply passes through the system as a traveling wave without the formation of any appreciable standing waves. In this condition, the pulsation amplitudes everywhere in the system are essentially the same as those of the source. Blodgett (1998) refers to these nonresonant pulsations as "residual or forced pulsations."

Figure 16 illustrates how the response of an acoustic system varies with excitation frequency. The figure shows a piston operating in a pipe with a closed end opposite the piston. Since this system behaves as a closed-closed pipe, the resonant frequencies are Nc/2L, which for the given dimensions, occur at 20, 40, 60, 80, and 100 Hz. The plot in the figure represents the pressure amplitudes at point A, located at the piston, as the piston excitation frequency is varied from 0 to 100 Hz. Additional plots of the response at points B and C, at the midpoint and closed end, respectively, are provided in Wachel, et al. (1995). Examination of the three plots leads to the following observations:

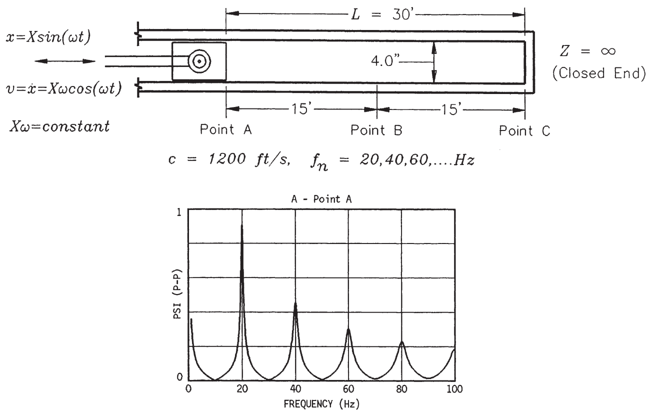


*Figure 16. Typical Frequency Response. (Courtesy of Wachel and Tison, 1994, Turbomachinery Laboratory)*

- The resonances at 20, 40, 60, 80, and 100 Hz are seen clearly in the response at points A and C.
- However, at point B, the only resonances observed are at 40 and 80 Hz, which are the even-numbered modes.
- Examination of the mode shapes for the closed-closed pipe (Figure 14) reveals that the midpoint is a pressure node for the odd-numbered modes and a pressure antinode for even-numbered modes. Thus, only the even-numbered modes can be detected.
- At points A and C, the peak pressures at resonance are approximately equal.
- As the frequency increases, the resonant amplitudes decrease. This is because the lowest modes, starting with the fundamental, have the most energy and, thus, are the most dangerous.
- In general, the responses at the nonresonant frequencies are small but nonzero.

## LUMPED ACOUSTIC SYSTEMS

In general, acoustic systems, like mechanical systems, can be broken down into two basic types—distributed systems and lumped systems. A distributed acoustic system is one in which all of the properties associated with flow, velocity, and pressure are continuously distributed along the length of the system. In mechanical engineering, a simple example of a distributed system is a cantilever beam. On the other hand, a lumped acoustic system is one in which the acoustic properties can be "lumped" together at various points in the system. In the field of mechanical vibration, a prominent example of a lumped system is the well-known mass-spring-damper system. Of course, all acoustic systems, as well as mechanical systems, are distributed. However, some are simple enough that the lumped approximation is reasonable. Use of a lumped system facilitates understanding of the physical system and lends itself to simpler solutions of the governing equations.

Per Kinsler, et al. (1982), the criterion that must be met in order to make the lumped approximation valid for an acoustic system is that the wavelengths of interest must be significantly longer than the physical dimensions of the components being studied. To and Doige (1979) state that the approximate cutoff point is when the dimensions are less than one-eighth of the wavelength. When this is true, each acoustic parameter remains time-variant but becomes almost independent of distance. An example of this is a bottle whose dimensions are all much smaller than the pulsation wavelength of interest. When this is the case, the bottle can be treated as a lumped element where the acoustic pressure is constant throughout.

*Lumped Acoustic Elements*

As is the case with mechanical and electrical systems, there are three basic elements in lumped acoustic systems. The first basic element is fluid inertia, which is also sometimes referred to as fluid inductance or the constricted element. This type of element is represented by a narrow or constricted passage that is short enough that its contained fluid behaves as a rigid body whose inertia must be overcome by the acoustic pressure difference across it. The governing equation for this type of element is as follows:

$$\Delta P = \left(\rho \bullet L / A\right) \bullet dQ / dt \tag{43}$$

Where:
$\Delta P$ = Pressure drop across element
$\rho$ = Fluid density
L = Effective length of passage
A = Cross-sectional area
Q = Volumetric flow rate

This leads to the definition of the fluid inductance of a given line, $I_F$, which is analogous to electrical inductance, as follows:

$$I_F = \left(\rho \bullet L / A\right) \tag{44}$$

This also leads to an equation for the acoustic impedance of a fluid inertia element as follows:

$$Z_I = \Delta P / Q = \left(\rho \bullet L / A\right) \bullet j \bullet \omega = I_F \bullet j \bullet \omega \tag{45}$$

Where:
$Z_I$ = Acoustic impedance for fluid inertia element
$\omega$ = Angular frequency of excitation (rad/sec)
j = Square root of negative one

The second basic element is the fluid compliance (also known as fluid capacitance) element. This type of element is represented by a chamber or tank and represents the ability of a fluid to "store" some of the volume flow entering it by converting it to pressure energy by virtue of its compressibility. For long wavelengths, the acoustic pressure is equal throughout the entire volume of the tank. The governing equation for this element is as follows:

$$Q = \left(V / K_{BULK}\right) \bullet dP / dt \tag{46}$$

Where:
P = Pressure (assumed constant throughout volume)
Q = Volumetric flow rate into element
V = Volume
$K_{BULK}$ = Fluid bulk modulus

This leads to the definition of the fluid compliance, $C_H$, also sometimes referred to as capacitance, for a volume, as follows:

$$C_H = V / K_{BULK} \tag{47}$$

This also permits the quantification of the acoustic impedance of a fluid compliance element, $Z_V$, as follows:

$$Z_V = K_{BULK} / (V \bullet j \bullet \omega) = 1 / (C_H \bullet j \bullet \omega) \quad (48)$$

The third basic element is the fluid resistance element. This type of element is represented by an orifice and represents any element, such as an orifice, valve, restriction, etc., having a fairly small flow area such that a relatively large pressure drop is required to generate fluid flow through the element. The governing equation for this element is simply the acoustic impedance equation:

$$R_F = \Delta P / Q \quad (49)$$

Where:
$R_F$ = Acoustic resistance
$\Delta P$ = Pressure drop across element (psid)
Q = Volumetric flow through element ($in^3/sec$)

It is easily seen that the acoustic impedance of a resistive element is simply equal to its resistance.

Per Yeow (1974), at very low frequencies, the fluid behaves as if it were incompressible so fluid compliance effects (which depend on compressibility) and inertia effects (which depend on frequency, ω) can safely be neglected. In this case, the only acoustic element that needs to be considered is the fluid resistance. Yeow (1974) also provides a mathematical discussion of when a given element should be represented as a fluid inertia and when it should be represented as a compliance.

*Mechanical and Electrical Analogies*

The authors are well aware that many pump users are far more familiar with mechanical systems than they are with acoustic systems. The authors are also acquainted with users that specialize in electrical engineering. Accordingly, in this section analogies are drawn between acoustic and mechanical and electrical systems in the hopes of facilitating understanding of basic acoustic concepts.

Now that the three basic acoustic elements have been defined, the analogies to mechanical and electrical systems can be drawn as follows:

- Acoustic volumetric flowrate = Mechanical displacement = Electrical current
- Acoustic pressure = Mechanical force = Electrical voltage
- Acoustic inertia = Mechanical mass = Electrical inductance
- Acoustic compliance = Mechanical compliance (spring element) = Electrical capacitance
- Acoustic resistance = Mechanical damping (dashpot element) = Electrical resistance

Some of the ways that the three systems are analogous include the following:

- The governing equations for the three basic elements in each system have exactly the same form.
- Each system has a "through" variable (flow, displacement, and current) and an "across" variable (pressure, force, and voltage).
- As has been done for the acoustic elements above, equations for the impedances of the basic mechanical and electrical elements can also be written and shown to be of equivalent form to those above.
- Each system has two elements that are incapable of energy dissipation. In the acoustic system, these elements are the fluid inertia and compliance. These elements are incapable of energy dissipation because their pressure and velocity are 90 degrees out-of-phase with one another. This is similar to the situation with mechanical masses and springs and electrical inductors and capacitors. All of these elements are referred to as reactive elements.
- Each system has only one element that is capable of energy dissipation—the acoustic resistor, the mechanical damper, and the electrical resistor. These elements dissipate energy because their through variables are in-phase with their across variables.
- Systems consisting of only the two reactive elements (a fluid inertia and compliance, a mass-spring system, and an L-C circuit) are one degree of freedom oscillators. The equations for the natural frequencies of these systems are essentially equivalent.
- When the purely reactive systems are excited at their natural frequencies, their amplitudes become infinite. All of these systems require the third element (acoustic resistance, mechanical dashpot, and electrical resistance) to provide damping to limit the resonant amplitudes to finite values.
- Each system has one element that acts to store kinetic energy (acoustic inertia, mechanical mass, and electrical inductor) and one that acts to store potential energy (acoustic compliance, mechanical spring, and electrical capacitor).

The above analogies have more usefulness than the facilitation of the understanding of acoustic systems. In the days before the advent of the digital computer, acoustics problems were often solved by converting the piping network into its equivalent electrical circuit and then either building the circuit or analyzing it on an analog computer. Chilton and Handley (1952) give a good illustration of these techniques.

## RECIPROCATING PUMP EXCITATIONS

As has been stated previously, in order for resonant pulsation problems to occur, an excitation source is needed to excite the acoustic system at one of its natural frequencies. In the pumping world, reciprocating pumps are probably the most notorious sources of these types of excitations. Although it is commonly believed that reciprocating pumps are the only type of positive displacement pump that can excite pulsation problems, that is far from the truth. For example, the authors have a great deal of experience with pulsation problems that were generated by gear and vane pumps. However, in the interest of keeping this tutorial to a manageable length, reciprocating pumps are the only type of positive displacement pump that will receive a detailed treatment. In general, the pulsations generated by positive displacement pumps are low frequency and high amplitude while those generated by centrifugal pumps (which are discussed later in this tutorial) are high frequency and low amplitude.

Figure 17, based on Wylie and Streeter (1993), shows a schematic of one cylinder in a reciprocating pump. Most reciprocating pumps are simple slider-crank mechanisms that consist of a rotating crankshaft, connecting rods, and pistons. They are essentially like an automobile engine running with the power transmission in the opposite direction. Specifically, the crankshaft, which is normally driven by an electric motor, transmits torque to the connecting rods, which convert the crankshaft's rotary motion into the reciprocating motion of the piston assemblies.

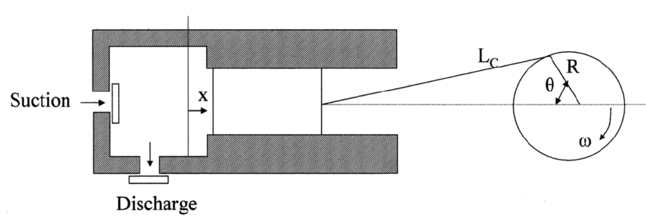


*Figure 17. Schematic of Reciprocating Pump.*

The reciprocating motion of the piston combines with the action of the suction and discharge valves to pump fluid from the suction line to the discharge line. When the piston is retracting from the cylinder (moving from left-to-right in the figure), the pressure in the cylinder drops, allowing the suction valve to open and the discharge valve to close. During this time the cylinder fills with fluid from the suction line. Once the piston reaches bottom dead center, it begins to move toward the cylinder, compressing the fluid within. This pressure rise forces the suction valve to close and the discharge valve to open. As the piston extends, it continues to force fluid out of the cylinder and into the discharge line until it reaches top dead center. At this point, the suction stroke again commences.

It should be noted that because the suction and discharge valves for any given cylinder are never simultaneously open (assuming that they are working correctly), the suction and discharge systems in a reciprocating pumping application are completely isolated from each other from an acoustics standpoint. Accordingly, the suction and discharge piping represent two completely distinct acoustic systems that must be analyzed separately. Additionally, pulsation problems occurring in the discharge system usually have no impact on the suction system and vice versa. It should be noted that the separation of the two systems only occurs with reciprocating pumps—in centrifugal pumps, the opposite is true.

*Flow Excitations*

Reciprocating pumps generate two distinct types of excitations that can generate pulsations in the suction and discharge piping—flow excitations and acceleration excitations. Since the flow excitations are almost always predominant in the discharge piping, these are the only reciprocating pump excitations discussed in many references on the subject. However, in some suction systems, the acceleration excitations are more dangerous than the flow excitations and, therefore, both types of excitations need to be considered in any pulsation analysis.

Typical pressure disturbances introduced into the suction and discharge systems by a reciprocating pump are shown in Figure 18, which is for a triplex pump and is taken directly from Miller (1988). Per Miller (1988), there are three apparently unrelated pressure disturbances, labeled “A,” “B,” and “C” in the figure. The “A” disturbances are the flow-excited disturbances that are the subject of this section. The “B” excitations, which occur at the beginning of each plunger stroke, and the “C” excitations, which occur at each point of flow velocity change (i.e., valleys in the figure) represent the acceleration excitations that are described in the next section.

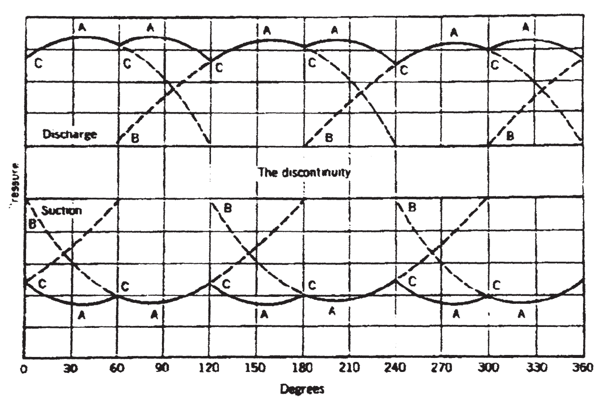


*Figure 18. Reciprocating Pump Pulsations.*

The flow excitations at “A” arise because the action of the pistons, along with the opening and closing of the valves, generates a pulsatile flow in both the suction and discharge systems. In fact, the flowrate generated by any reciprocating pump will vary from instant to instant. This is because during the discharge or suction stroke, the plunger begins at rest, accelerates to a maximum velocity at approximately midstroke, and then decelerates to rest again. Since the generated flow is the product of the plunger velocity and the plunger area, the flow associated with each cylinder varies in exactly the same manner. The magnitude of the pump’s overall flow pulsation is dependent on the number of plungers in the pump and will normally decrease as the number of plungers increases since the plungers can be phased to smooth out these effects. In the theoretical limit of an infinite number of plungers, the flow can be imagined to be uniform.

The flow in the discharge piping can be obtained from the following equations from Miller (1988):

$$Q_D = 0.50 \bullet A_P \bullet s \bullet \omega \bullet \left[\sin\theta + \left(s/\left(4 \bullet L_C\right)\right) \bullet \sin(2\theta)\right] if\ P_{CYL} > P_D \quad (50)$$

$$Q_D = 0\ if\ P_{CYL} < P_D \quad (51)$$

Where:
QD = Flow into discharge system ($in^3$/sec)
$P_{CYL}$ = Cylinder pressure at any given time (psi)
$P_D$ = Discharge pressure (psi)
$A_P$ = Cylinder area ($in^2$)
s = Piston stroke (inch)
θ = Crank angle
ω = Crankshaft rotating speed (rad/sec)
$L_C$ = Connecting rod length (inch)

Similar equations can be written for the suction flow. From the above equation, the presence of the ω term means that the magnitudes of the excitations increase with pump speed. It is also seen that for the ideal case where the connecting rod length, $L_C$, is infinitely long, the flow pulsations would be completely sinusoidal. However, the finite length of the connecting rod causes the actual flow to vary from a perfect sinusoid, as is illustrated in Figure 2 of Tison and Atkins (2001). Specifically, the sin (2θ) term means that the flow pulsations from a single cylinder will contain a significant 2× (two times per revolution) excitation as well as the 1× component associated with the perfect sinusoid.

Since those excitations occur in each cylinder, it stands to reason that the two largest excitations in a multiplunger pump would be obtained by multiplying those excitations by the number of plungers. In other words, the predominant excitations would be at the plunger frequency and at twice the plunger frequency, which would be 3× and 6× for a triplex pump. In Figure 21 of Wachel, et al. (1995), this is verified to be, indeed, true for an ideal pump having equal performing plungers and valves. In fact, for this ideal case, the only harmonics that are present are integral multiples of the plunger frequency (i.e., 3×, 6×, 9×, etc.)—all others are zero. However, in real pumps, the other harmonics are present in small amounts.

The resultant flow seen by both the suction and discharge systems in a real pump is shown in Figure 19, which is taken directly from Blodgett (1998). This profile is for a triplex pump and is fairly typical. It is seen that for a triplex pump, there are six points of maximum flowrate and three points of minimum flowrate per crankshaft revolution. This has the effect of generating excitations at 3× and 6×. In fact, Blodgett (1998) states that, in general, the only harmonics for which flow excitations are significant are integer multiples of the number of plungers (i.e., 2×, 4×, 6×, etc., for a duplex pump). Wachel and Price (1988) essentially agree when they state that even though a reciprocating pump generates excitations at all harmonics of pump speed (i.e., 1×, 2×, 3×, etc.), the only harmonics containing significant amounts of energy are the integral multiples of the plunger frequency. Naturally, if any of these harmonics coincide with an acoustic natural frequency of the discharge piping system (or, in some cases, the suction system), resonance occurs and pulsation-related problems may follow.

The situation in double-acting pumps is different. In Figure 18 of Wachel, et al. (1995), it is shown that in an ideal double-acting pump (perfectly symmetrical piston areas, ideal valves, and infinite length connecting rod), all of the odd harmonics cancel for a given cylinder such that the even harmonics are the only ones that are left. Thus, in an ideal double-acting triplex pump, the only harmonics present would be 6×, 12×, 18×, etc. However, as is shown in Figure 19 of Wachel, et al. (1995), the real effects of unequal area pistons (due to the presence of the connecting rod), nonperfect valves, and finite connecting rod length allow the odd harmonics (3×, 9×, 15×, etc.) to appear.

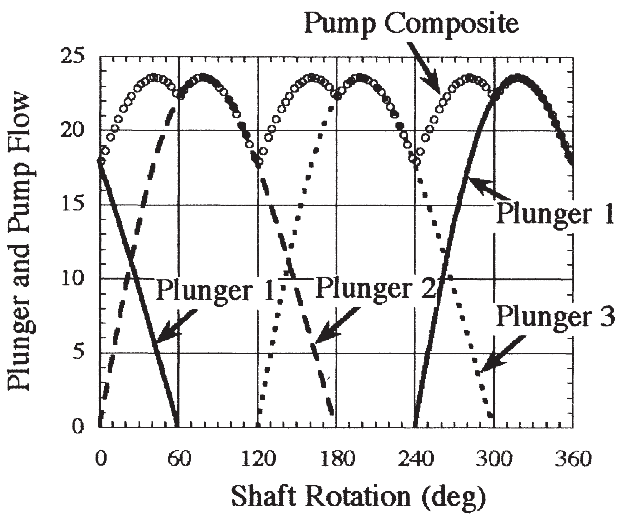


*Figure 19. Triplex Pump Flow Excitations.*

The flow pulsations convert to pressure pulsations based on the nature of the discharge and suction piping. Since the pump discharge pressure is normally set by resistance losses in the discharge piping (pipe friction, bends, expansions, etc.) which are normally a function of the square of the flowrate, the squared law causes the pressure pulsations to be larger (percentage-wise) than the flow pulsations. For example, consider a hypothetical system that has a discharge pressure of 500 psi that is totally generated by pipe friction loss at nominal flowrate. If the flow pulsation was 20 percent, that would translate into a pressure variation of 44 percent ($1.20^2 = 1.44$), or 220 psi, which is substantial.

*Acceleration Excitations*

The other type of excitation generated by reciprocating pumps is the acceleration excitation, sometimes also called an inertia excitation. These excitations can be understood by visualizing the liquid in the suction piping, which must be completely at rest during the discharge stroke. Once the suction stroke begins and the suction valve opens, the pump must accelerate the fluid in the suction line from rest to maximum piston velocity. At the end of the suction stroke, the suction valve closes and the fluid must then be decelerated to rest. The same phenomenon occurs with the liquid in the discharge line. The magnitude of the fluid acceleration can be determined from the equation for piston acceleration as a function of crank angle, from Miller (1988):

$$a_P = 0.50 \bullet s \bullet \omega^2 \bullet \left[\cos\theta + \left(s / \left(2 \bullet L_C\right)\right) \bullet \cos(2\theta)\right] \quad (52)$$

Where:
$a_P$ = Piston acceleration (in/sec$^2$)

All of these sudden accelerations generate pressure pulses in both the suction and discharge lines. Per Miller, these pulses are normally of the "water hammer" type and are often in the range of 75 to 200 Hz. Miller (1988) gives the following equation, which is based on Newton's law of motion, for calculating the pressure pulsation amplitude due to acceleration effects:

$$P = L \bullet \rho \bullet a \quad (53)$$

Where:
P = Pressure pulsation (psi)
L = Pipe length (inch)
$\rho$ = Fluid density (lbf-sec$^2$/in$^4$)
a = Fluid acceleration (in/sec$^2$)

This equation applies equally to the suction and discharge piping as long as the appropriate pipe dimensions are used. Miller (1988) states that even pumps with short suction piping (and, therefore, a fairly small mass of fluid to accelerate) can experience acceleration-excited pulsations of more than 25 psi. Warwick (1999) has observed these effects to be unexpectedly large, and to have caused problems in both suction and discharge systems, when pumping acids, because of their high densities.

As has been stated previously, increasing the number of plungers has the beneficial effect of reducing the flow-excited pulsations. However, Miller (1988) states that there is evidence that increasing the number of cylinders has no impact on the acceleration-excited pulsations.

Since Miller (1988) states that a typical acceleration-excited pressure pulsation is around 25 psi, the flow-excited pulsations, which were 220 psi in the example given above, are normally predominant in the discharge piping. Exceptions to this rule occur when the pump discharge pressure is generated by sources other than frictional losses. For example, if the pump is delivering into a low friction, high pressure system such as a short vertical pipeline as shown in Figure 10 of Miller (1988), the flow-excited pressure pulsations will be negligible. In these rare cases, the acceleration excitations can become the disturbance of concern in the discharge system.

The situation in the suction piping is quite different. Since pump suction systems normally employ short pipes of large diameter to meet net positive suction head (NPSH) requirements, the frictional pressure drop in suction lines is usually quite low. For this reason, flow-excited pulsations often have little impact on the suction system. For instance, if the pump described in the example above had a suction pressure of 25 psi, the 44 percent pressure pulsations only translate into 11 psi in the suction piping. This is hardly enough to generate piping vibrations or structural damage, although it could still be a concern from a cavitation standpoint.

Interestingly enough, Beynart (1999) states that there is evidence that flow and acceleration excitations cannot coexist during any given suction or discharge stroke of a pump. If the flow excitation is present, the acceleration disturbance will be absent, and vice versa. It appears that the dominant disturbance overwhelms and reduces the effect of the other. Beynart (1999) postulates that the reason for this is that a disturbance of either type will instantaneously reduce the volumetric efficiency sufficiently to make it impossible for the pump to reinstate the flow fast enough for another disturbance to occur in the short period of time (typically about 3 msec) available.

## CENTRIFUGAL PUMP IMPELLER EXCITATIONS

Although the reciprocating pump is the most notorious source of pulsation problems in the pumping world, the authors have substantial experience with such problems occurring in centrifugal pumping systems. Centrifugal pumping systems suffer from pulsation problems due to three very different phenomena. First, the pump can generate high frequency pulsations at vane-passing frequency that can excite acoustic resonances in the manner just described for reciprocating pumps. This is the subject of this section. Second, vortex shedding occurring at discontinuities in the piping system can also excite acoustic resonances. This is the subject of the next section.

The third phenomenon is complex interaction between the dynamic characteristics of the pump and those of the piping system. In some cases, the centrifugal pump can act as an amplifier or attenuator of excitations generated by other sources, such as reciprocating pumps. In other cases, the slope of the pump's head-flow curve can combine with the system characteristics to yield a system having negative damping, which is unstable. A discussion of all of these effects would be worthy of a tutorial all of its own. The interested reader is referred to Greitzer (1983), Sparks (1983), and Sparks and Wachel (1976) for more information on this highly complex and interesting subject.

As stated above, this section will focus on high frequency pulsations generated by centrifugal pump excitations at vane-passing frequency and its harmonics. Apparently the authors are not the only ones who have experience with pulsation problems created by these excitations as Wachel (1992) describes a case study in which a four-stage centrifugal pump suffered problems when a half-wave resonance in the crossover piping at 415 Hz was excited by pump vane-passing frequency (7×). Wachel (1992) also provides another case study where a centrifugal impeller having seven vanes excited an acoustic resonance at twice vane-passing frequency, 14×. Finally, Fraser, et al. (1977), experienced a case where a centrifugal pump having a tight tip-to-volute clearance generated significant pulsations at one, two, three, four, and five times vane-passing frequency.

These vane-passing excitations are generated by the action of the impeller vane passing stationary objects that are in close quarters to it. As a vane passes a stationary object, the wakes that are present on the suction surface of the vane impinge on the stationary object and generate a pressure pulse. Since each blade must pass the stationary object once each revolution, the pulses are generated at the vane-passing frequency. Of course, if there is more than one of these stationary objects (as in a double volute pump), the excitations are then generated at integer multiples of the vane-passing frequency. Per Guelich and Bolleter (1992), the pulsation spectrum for a centrifugal pump impeller typically shows peaks at vane-passing frequency and its harmonics (the second harmonic is often noticeably strong), as well as at 1×.

In direct contrast to the pulsations generated by reciprocating pumps, vane-passing pulsations occur at high frequencies. Price and Smith (1999) state that pulsation problems associated with these excitations normally occur at frequencies of 500 Hz or greater, although systems having large diameter vessels can experience problems at frequencies lower than this. As Howes and Greenfield (2002) point out, at these frequencies, the one-dimensional plane wave assumption that has been employed throughout this tutorial, becomes questionable. Since Howes and Greenfield (2002) cite the excitation of shell modes in piping and pulsation dampeners as problems that can be caused by pulsations at vane-passing frequency, it is obvious that three-dimensional effects are often important. This makes the analysis much more complex.

In general, the amplitudes of the vane-passing pulsations and their harmonics are relatively low. Lewis, et al. (1997), state that vane-passing frequency pulsations are normally about one-third of 1 percent of the pressure rise generated by the impeller. Price and Smith (1999) quote a figure of 0.5 psi peak-to-peak or less.

Guelich and Bolleter (1992) give the following equation for normalized pressure pulsations generated by the wake in an impeller:

$$\Delta P* = \Delta P / \left[ \left( \rho / 2 \right) \bullet u_2^2 \right] \tag{54}$$

Where:

$\Delta P^*$ = Normalized pulsation coefficient
$\Delta P$ = Pressure pulsation
$\rho$ = Fluid density
$u_2$ = Circumferential velocity at impeller exit

Based on a number of tests of single and three-stage pumps, Guelich and Bolleter (1992) state that virtually all pressure pulsations (95 percent confidence limit) have normalized coefficients below 0.015 at best efficiency point (BEP) flow and below 0.02 at 25 percent of BEP flow. These values are for the tightest radial clearances between impeller and diffuser or volute (ratio of collector diameter to impeller diameter of about 1.02). The test data, as functions of radial gap and flow, are given in Figures 7 and 8 of Guelich and Bolleter (1992).

Although the amplitudes of vane-passing pulsations are very difficult to predict analytically, the following factors, taken from Guelich and Bolleter (1992), Price and Smith (1999), Wachel (1992), Fraser, et al. (1977), and Schwartz and Nelson (1984), are known to have an impact on their magnitudes:

- Radial clearance between the impeller tip and the stationary collector (diffuser vanes or volute cutwaters)—In general increasing this clearance is the most sure way to reduce pulsation amplitudes. Guelich and Bolleter (1992) state that pulsations have been empirically found to vary inversely with this gap to the 0.77 power.
- Pump speed—Pulsations, like pressure generation, tend to increase with speed squared.
- Flow—Pulsations are normally minimized at BEP. As the flow moves away from BEP (in either direction), pulsation amplitudes increase.
- Geometry (thickness and form) of the vane trailing edge
- Vane loading (difference in velocity between pressure and suction sides of vane)
- Number of vanes—Increasing number of vanes normally reduces pulsations.
- Type of impeller—Pulsations tend to increase with increasing specific speed. Thus, pure radial impellers have the lowest pulsations, followed by Francis impellers and axial impellers.
- Reynolds number
- Combination of number of impeller and diffuser vanes
- Staggering of impellers in multistage pumps (obviously, staggering acts to reduce pulsations)
- Fluid properties, including content of free gases
- Fluid acoustic velocity
- Impeller tip speed—Higher tip speeds mean larger pulsations.
- Symmetry of the impeller and collector

Unlike the pulsations generated by a reciprocating pump, which are sometimes large enough to cause problems even in the absence of resonance, the centrifugal impeller excitations are so small that they can only do damage if they are amplified via acoustic resonance. Additionally, the impeller must be located at or near an antinode in the velocity mode shape in order for amplification to occur. If the impeller is located near a velocity node, it will have a very difficult time exciting that particular acoustic mode.

One other difference between a centrifugal pump and a reciprocating pump is the relationship between the suction and discharge piping systems. As stated previously, the valves in a reciprocating pump act to isolate the suction and discharge piping from one another. On the other hand, a centrifugal pump provides no such isolation. The suction piping, pump internal passages, and discharge piping all comprise a single acoustic system. It is not unusual to observe resonances generated by other sources in the system where the standing wave passes right through the impeller. When modeling these systems, the impeller's acoustic characteristics must be accounted for, normally through a specialized transfer matrix.

Table 2 of Guelich and Bolleter (1992) provides a list of design guidelines for keeping impeller pressure excitations in check. Probably the most basic are that the ratio of collector diameter to impeller diameter should exceed 1.04 and that the impeller and collector should never have the same numbers of vanes. Table 3 of the same reference provides design features that can be incorporated into centrifugal pumps to further reduce pulsation amplitudes.

In closing this discussion, it should be mentioned that another mechanism by which a centrifugal pump can excite pulsations in the piping system is rotating stall. Dussourd (1968) documents a case where rotating stall in a 10-stage boiler feed pump generated large pressure fluctuations and piping vibrations in the discharge system. Rotating stall occurs when a passage or group of passages in the impeller and/or diffuser stall out. The stall cells are unstable and, therefore, they rotate from passage to passage in the impeller in the direction opposite to rotation.

## VORTEX SHEDDING

The next pulsation excitation mechanism often present in centrifugal pumps is vortex shedding. Unlike the excitations previously described for reciprocating and centrifugal pumps, vortex shedding excitations occur within the piping system, not at the pump. Vortex shedding occurs when flow passing over an obstruction, side branch, or a piping discontinuity generates fluid vortices at a regular interval. Under certain conditions, these vortices can excite acoustic resonances and generate pulsation problems. Although vortex shedding can, theoretically, also cause pulsation problems in reciprocating pumping systems, it has been the authors' experience, and that of Howes and Greenfield (2002), that these problems are much more likely to occur in centrifugal pumping systems.

When a fluid flow passes over any body having a bluff (broad) trailing edge, such as a cylinder, vortices are shed from both sides of the body. The vortex shedding is not simultaneous—instead, vortices are alternately shed from one side and then the other. This alternating shedding causes pressure fluctuations to occur on both sides of the body. Under certain conditions, these fluctuating pressure forces can cause the body to vibrate.

A good physical description of how vortices are formed on a cylinder is given by Blevins (2001). As a fluid particle approaches the leading edge of the cylinder, its pressure is increased from the free stream pressure to the stagnation pressure. This high fluid pressure near the leading edge causes fluid to flow around both sides of the cylinder, generating boundary layers as it does so. However, at high Reynolds numbers, the pressure in the vicinity of the leading edge is not high enough to force the flow all the way around the back of the cylinder. Instead, at approximately the widest portion of the cylinder, the boundary layers separate from the cylinder and form two shear layers that extend beyond the cylinder and, thereby, bound the wake directly behind the cylinder. Since the no slip condition requires the innermost portions of the shear layers (which are in contact with the cylinder) to move much more slowly than the outermost portions (which are in contact with the free stream), the shear layers roll into the wake, where they fold on each other and generate a series of discrete swirling vortices. As is shown in Figure 20, a regular pattern of vortices, called a vortex street, forms in the wake.

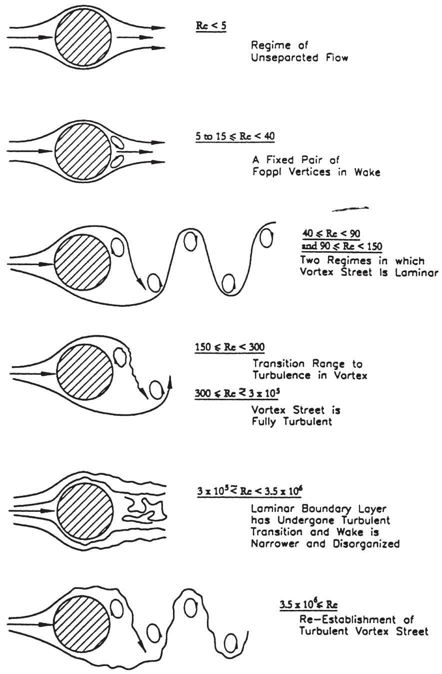


*Figure 20. Vortex Streets. (Courtesy of Price and Smith, 1999, Turbomachinery Laboratory)*

Although the above discussion was specifically related to cylinders, vortex shedding occurs with flow over bluff bodies of any cross-section. The vortex streets tend to be quite similar, regardless of the body shape. In general, the type of vortex street formed is a function of Reynolds number. Blevins (2001) defines seven distinct flow regimes, all shown in Figure 20, for vortex shedding from a smooth circular cylinder.

The frequency at which vortices shed from each side of a body has been empirically determined to be a function of a dimensionless characteristic parameter known as the Strouhal number, which is defined as follows:

$$f_s = S \bullet U / D \tag{55}$$

Where:

$f_S$ = Frequency that vortices shed from one side of body (Hz)
S = Strouhal number
U = Characteristic flow velocity (ft/sec)
D = Characteristic dimension of body (ft)

For internal flows, such as those in pipes, the characteristic flow velocity, U, is simply the mean velocity. For external flows, such as those over a cylinder, the characteristic flow velocity is the free stream velocity. Since the vortices are formed by the interaction of the two shear layers on either side of the body, the characteristic dimension, D, is simply the width between the two flow separation points. For all practical purposes, this can be assumed to be the maximum width of the structure perpendicular to the flow. For example, if the body is a cylinder, D is simply the diameter. For the case of flow past a tee or branch, one of the most common configurations of interest in pump piping systems, D is the inside diameter of the branch, with any rounding at the branch entrance accounted for. Per Chen and Florjancik (1975), for abrupt expansions, the characteristic dimension is simply the difference between the two pipe diameters. Au-Yang (2001) recommends that for "real" bodies, which often have tapered or nonconstant cross sections, the shedding frequencies should be calculated for both the smallest and largest widths to bound the problem.

It is customary to refer to the frequency obtained from the above equation, that associated with only one side of the body, as the vortex shedding frequency. Accordingly, it is easily seen that the fluctuating pressure force generated by the vortex shedding acting in the direction perpendicular to the flow (i.e., lift direction) acts at the vortex shedding frequency. On the other hand, since vortex shedding is occurring on both sides of the body, the fluctuating force acting in the direction of flow (i.e., drag direction) acts at twice the shedding frequency.

Blevins (2001) and Au-Yang (2001) both provide plots of Strouhal number versus Reynolds number obtained from tests for flow around circular cylinders. Au-Yang (2001) summarizes the results as follows:

- For 1000 < Re < 100,000—Strouhal number equals 0.2
- For 100,000 < Re < 2.0E6—Strouhal number lies between 0.2 and 0.47
- For 2.0E6 < Re < 1.0E7—Strouhal number lies between 0.2 and 0.3

Au-Yang (2001) continues to say that even though the above rules were obtained from testing on circular cylinders, they can also be employed for bodies having just about any cross-section. Additionally, Figure 3-6 of Blevins (2001) gives representative Strouhal numbers as functions of Reynolds number for various geometries. In Figure 3-7, Blevins (2001) also provides Strouhal numbers as a function of inclination angle for an inclined flat plate. In general, the more the plate deviates from being perpendicular to the flow, the larger the Strouhal number becomes.

In centrifugal pumping systems, the primary locations where vortex shedding occurs are at valves and other restrictions and at flow past tees and side branches. The types of valves where significant vortex shedding can occur include relief valves, throttling valves, and pressure regulators. Although Lewis, et al. (1997), state that typical Strouhal numbers provided by valve manufacturers range from 0.1 to 0.3, Wachel (1992) states that the Strouhal numbers for all of these types of valves can usually be assumed to be about 0.2. Baldwin and Simmons (1986) state that under resonance conditions, the relatively small vortex pulsations generated at a side branch connection can be amplified to levels of about 200 psi peak-to-peak at the closed end of the side branch (where a relief valve is often located).

Rogers (1992) provides the following formula for the Strouhal number for flow past a side branch, which is valid for main flow Reynolds numbers greater than $1.6 \times 10^7$:

$$St = 0.413 \bullet \left(d / D_P\right)^{0.316} \tag{56}$$

Where:
St = Strouhal number
d = Sidebranch diameter
$D_P$ = Diameter of main pipe

Wachel (1992) presents a case where vortex shedding at the opening of a dead-ended sidebranch generated a 25 to 30 Hz resonance with the quarter-wave frequency of the side branch, resulting in fatigue failures of valves, gauges, and instrumentation. Lewis, et al. (1997), report on a case where vortex shedding in a V-sector ball throttling valve, located immediately downstream of a centrifugal pump, excited a standing wave in the main discharge line, leading to pulsation problems.

Since the pulsation amplitudes of the vortices are always very small, they can only create problems if they excite an acoustic resonance. Additionally, vortices can only excite a resonance if they are located at or near a velocity antinode. One of the reasons that vortex-related problems frequently occur in side branches is that most side branches act as quarter-wave stubs that have a velocity antinode (open end) right at the location of vortex generation. Side branch problems can also be affected by the acoustics of the main line. If the main line is also in resonance and its standing wave also happens to have a velocity antinode at the side branch location, the problem can be exacerbated. On the other hand, if the main line has a velocity node at this location, side branch pulsations will be attenuated. For this reason, side branch resonance problems can sometimes be eliminated by making small changes to the length of the main line. Chen and Florjancik (1975) demonstrated this concept when they completely eliminated a side branch pulsation problem by merely cracking a valve, which had been shut, in the main line downstream of the branch.

Baldwin and Simmons (1986) state that vortex-excited side branch resonances normally occur in the 200 to 400 Hz range. In order to avoid such resonances, Baldwin and Simmons (1986) provide the following design guideline:

$$d / L > 2.4 \bullet U / c \tag{57}$$

Where:
d = Stub diameter
L = Stub length
U = Flow velocity in main pipe
c = Acoustic velocity

Baldwin and Simmons (1986) further state that in order to avoid side branch problems, the side branch should be designed to minimize its length-to-diameter ratio and to eliminate all sharp edges, which generate vortices. In Figure 5 of Baldwin and Simmons (1986), two side branch designs that they have found to be successful in eliminating vortex excitation problems are presented.

In one of their case studies, Lewis, et al. (1997), provide a good method for troubleshooting pulsation problems if vortex shedding is suspected to be the excitation source. Using the observed pulsation frequency, the known flow velocity, and the expected Strouhal number, they back-calculated the characteristic dimension, D, from Equation (55). They then searched their system for a flow obstruction or gap of about that size and pinpointed the stiffening rings on the strainer as the cause of their problem.

One of the best ways to minimize or eliminate vortex shedding is via streamlining of the downstream side of the flow obstruction. Per Blevins (2001), in order for this path to be effective the included angle between the two downstream surfaces cannot be more than 8 to10 degrees. Vortex shedding can also be reduced by adding a vortex suppression device to the obstruction. Figure 3-23 of Blevins (2001) shows several common configurations. All of these devices act to retard the generation of an organized, two-dimensional vortex street Blevins (2001) provides specific guidelines for the design of each of these devices.

If prevention of vortex shedding is not practical, the vortex shedding frequencies can be shifted away from acoustical and/or mechanical natural frequencies by making the effective width of the obstruction larger or smaller. Alternatively, the flow path can be modified to change the velocities across the obstruction.

## WATER HAMMER

All of the pulsation excitations discussed so far (i.e., reciprocating pumps, centrifugal impellers, and vortex shedding) can be classified as steady-state excitations in that they are capable of generating pulsations that can last an indefinite amount of time. However, those are not the only types of pulsation problems that can occur in pumps. Transient pulsations, which can be grouped together under the name "water hammer" are also possible and are the subject of this section.

The term "water hammer" is normally associated with the rapid changes in internal liquid pressure that occur in a pipe when the flow is suddenly interrupted via the closing of a valve and the "hammering" sound that may accompany those changes. A common, everyday example is the noise that sometimes occurs in the pipes of an old building when a water faucet is shut off. In spite of those connotations, water hammer can also occur under other circumstances such as the rapid opening of a valve or the sudden starting or stopping of a pump. In short, any transient pulsations that occur as the result of a rapid change in flow conditions can be referred to as water hammer. These phenomena are sometimes also referred to as surge or fluid shock.

Cornell (1998) provides a colorful analogy of what happens when a valve is closed instantaneously. At the instant of closure, the flow at the valve is stopped but the column of fluid behind it will continue moving forward. Cornell (1998) likens this to a speeding train where the engine abruptly hits a brick wall. Obviously, the wall stops the engine but all of the other cars in the train continue moving forward. The primary difference between a train and fluid flow is that the train is not contained and the cars, therefore, derail. Since the fluid is contained, it, instead, generates a large pressure spike. If the pressure spike is not sufficient to rupture the pipe or some other component, the compression wave created will reverse and travel back down the pipe toward the pump. When the wave hits the check valve or pump, it will again reverse and continue to reverberate until something finally breaks or the energy completely dissipates due to system damping. Even if nothing fails initially, this scenario subjects the system to repeated stresses, which could ultimately lead to fatigue failures.

In order to describe what physically takes place during water hammer, the treatment given by Wylie and Streeter (1993) will be followed. In Figure 21, which is based on a similar figure in Wylie and Streeter (1993), the fluid is assumed to be flowing from left to right at a constant velocity, $v_0$, and head, $H_0$, when the valve at the right-hand end of the line is suddenly closed. At the moment the valve closes (t = 0), the fluid immediately adjacent to it is decelerated from $v_0$ to rest. This causes its pressure to rise sharply, which causes the elastic pipe wall to stretch, as is shown in Figure 21a. As soon as that first fluid layer is brought to rest, the same action is then applied to the next layer, causing its pressure to rise and its pipe wall to stretch. In this manner, a compression wave of amplitude, H, travels from right to left at the acoustic velocity, c, and at a sufficient pressure to just bring the upstream fluid to rest.

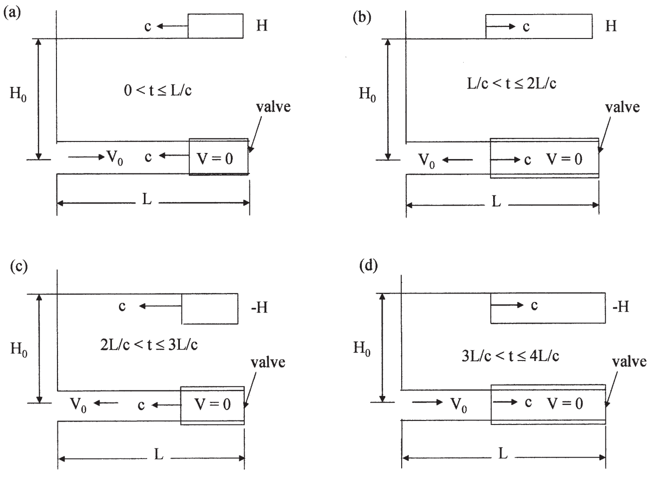


*Figure 21. Illustration of Water Hammer.*

The fluid upstream from the valve continues to flow downstream with velocity, $v_0$, until it encounters the compression wave. The compression wave continues to move from right to left, bringing the fluid to rest as it passes, compressing it, and stretching the pipe. When the wave reaches the reservoir at the end of the pipe (t = L/c), all of the fluid is at rest, at a greater head ($H_0$ + H) then it started at, and all of its initial kinetic energy has been converted into elastic energy (stored in both the fluid and the pipe wall).

When the compression wave reaches the reservoir, it is reflected. Since the reservoir behaves as an open end and since the laws of reflection require a wave to change sense upon encountering an open end, the compression wave is reflected as an expansion wave of the same amplitude, −H, moving from left to right at the acoustic velocity. As the expansion wave moves through the still fluid, the higher pressure on the right side of the expansion wave causes the fluid to start flowing from right to left at velocity, $v_0$ (refer to Figure 21b). The fluid remains at rest until it encounters the expansion wave, after which it flows to the left at $v_0$. The expansion wave returns the pressure to its value prior to valve closure, $H_0$, thereby leaving the fluid uncompressed and the pipe wall returns to normal. At the instant, 2L/c, that the expansion wave arrives at the valve, the pressures and pipe wall are back to normal everywhere in the pipe and the velocity everywhere is $v_0$ to the left. The system energy is then all in the form of kinetic energy.

When the expansion wave arrives at the valve, it is reflected. Since the closed valve behaves as a closed end and since the reflection laws state that reflection from a closed end brings no change in sense, the expansion wave is reflected as an expansion wave of the same amplitude, −H, moving from right to left. As the expansion wave passes through the successive layers of fluid, the pressure difference across it decelerates the fluid to rest (refer to Figure 21c). Each fluid layer that the expansion wave passes through has its pressure reduced to $H_0 - H$. This low pressure allows the fluid to expand (via its compressibility) and the pipe wall to contract. When the wave reaches the reservoir at time, 3L/c, all of the fluid is at rest and at a pressure of $H_0 - H$ and all of the energy is, once again, in the form of potential energy.

When the expansion wave reaches the reservoir, it is again reflected. Since the reservoir behaves as an open end and since the laws of reflection require a wave to change sense upon encountering an open end, the expansion wave is reflected as a compression wave of the same amplitude, H, moving from left to right at the acoustic velocity. As the compression wave moves through the still fluid, the higher pressure on the left side of the compression wave causes the fluid to start flowing from left to right at velocity, $v_0$ (refer to Figure 21d). The fluid remains at rest until it encounters the compression wave, after which it flows to the right at $v_0$. The compression wave returns the pressure to its value prior to valve closure, $H_0$, thereby leaving the fluid uncompressed and the pipe wall returns to normal. At the instant, 4L/c, that the compression wave arrives at the valve, the pressures and pipe wall are back to normal everywhere in the pipe and the velocity everywhere is $v_0$ to the right.

A check of these conditions reveals that they are exactly the same as at the instant of valve closure, 4L/c seconds earlier. Thus, the above chronology begins all over again and continues to repeat itself every 4L/c seconds until the damping in the system due to fluid friction and imperfect elasticity in the fluid and pipe wall dissipates all of the energy and allows the fluid to come to rest.

The transient behavior in the pipeline just described can be visualized in the x-t plane of Figure 22, based on Wylie and Streeter (1993). The distance values plotted as the abscissa represent the distance along the pipe measured from the reservoir at the left-hand end. The time values plotted as the ordinate represent the total time that has elapsed after the instant of valve closure. Since the water hammer compression and expansion waves take L/c seconds to travel from one end of the pipe to the other, the sloped lines in the figure may be visualized as the wavefronts of the compression or expansion waves as they traverse the pipe.

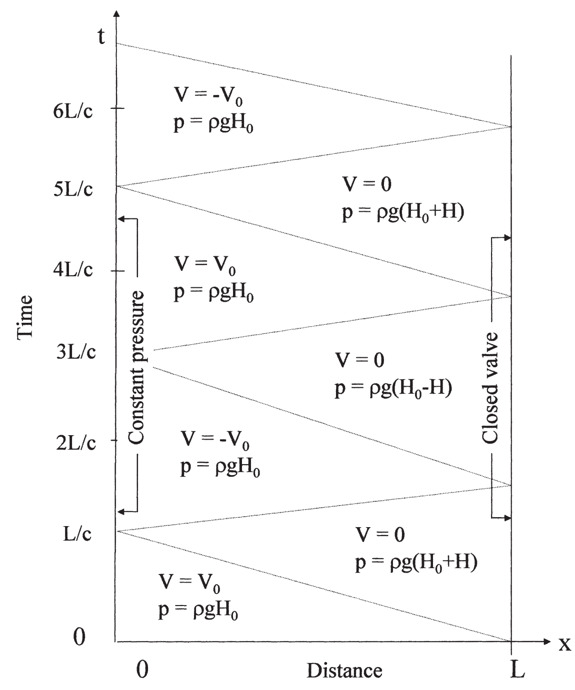


*Figure 22. Illustration of Water Hammer in x-t Plane.*

For instance, the lowest sloped line between t = 0 and t = L/c represents the initial compression wave that is generated as a result of valve closure. As stated previously, that wave travels from right to left and reaches the reservoir at time, L/c. Accordingly, that sloped line represents the position of that wave at any given time, t. Additionally, the sloped line divides the fluid into two segments, each having a different value of velocity and pressure. All of the fluid to the left of the wavefront has not encountered the compression wave yet and, thus, remains at the initial fluid velocity, $v_0$, and pressure, $H_0$. This is indicated by the velocity and pressure values immediately to the left of the sloping line in the figure. On the other hand, the fluid to the right of the wavefront has already been compressed by the compression wave and is now at rest and at a higher pressure, $H_0 + H$. This is indicated by the velocity and pressure values immediately to the right of the sloped line. The figure can, thereby, be utilized to determine the velocity and pressure at any point in the pipe at any given time.

It should be noted that the compressibility of the liquid and elasticity of the pipe serve to cushion the shock generated in the above scenario. In fact, if the pipe was rigid and the liquid was truly incompressible, the pressure spike would be infinite in the case of instantaneous valve closure.

The most important question associated with the previous example is what magnitude of pressure, H, is generated by the sudden valve closure. Wylie and Streeter (1993) provide the answer via the basic equation of water hammer, known as the Joukowsky relation, which is as follows:

$$\Sigma\ \Delta H = \pm(c / g) \bullet \Sigma\ \Delta v \tag{58}$$

Where:
$\Delta H$ = Change in head (ft)
$\Delta v$ = Change in fluid velocity (ft/sec)
c = Acoustic velocity (ft/sec)
g = Gravitational acceleration (ft/sec$^2$)

The above equation provides the magnitude of the pressure wave (in terms of head) generated by an instantaneous change in flow, such as that which occurs when a valve is opened or closed. The negative sign means that the pressure wave travels upstream in the pipe and the positive sign refers to downstream propagation. For the simple case given where a valve is initially flowing at a velocity, $v_0$, and a valve is suddenly closed, the amplitude of the pressure wave that travels upstream is simply $\rho \bullet c \bullet v_0$. Stepanoff (1949) points out that if the valve is only partially closed, so that the final velocity through it is given by $v_F$, the magnitude of the pressure wave is reduced to $\rho \bullet c \bullet (v_0 - v_F)$.

It is seen from the above equation that the amplitude of the induced pressure pulsation is strongly dependent on the acoustic velocity. However, the impact of the acoustic velocity goes even further than that since it determines the length of time that corresponds to an "instantaneous" change in valve position. This is because as long as the valve closes in a shorter time than the time it takes for the pressure wave to travel down the pipe to the other end, reflect, and then travel back up the pipe to the valve, the impact on the system is exactly the same as if the valve had closed instantaneously. As was demonstrated in the previous example, this critical time interval is 2L/c. As long as the valve closes completely in less than this time the impact on the system will be the same, regardless of how quickly the valve actually closes. Although all systems are different, Cornell (1998) states that a valve that closes in less than 1.5 seconds will normally behave as if instantaneous.

This critical time phenomenon occurs because if the reflected wave has time to get to the valve before it is finished closing, it will diminish the amplitude of valve closure seen by the system. When this occurs, the pressure rise at the valve becomes the sum of that created by instantaneous closure and the pressure carried by the reflected wave. Since it has already been shown that the reflected wave is an expansion wave, this results in a maximum pressure rise at the valve that is lower than that for the case of instantaneous closure. For this case of gradual valve closure, the maximum pressure rise occurs at the valve and the pressure rise reduces linearly along the length of pipe until it reaches zero at the reservoir. This is in contrast to the case of instantaneous closure where the maximum pressure rise, H, occurs at every point within the pipe.

Graf and Marchi (1997) provide a slightly different version of the above equation that allows determination of the head rise for cases where the valve closure cannot be considered to be instantaneous, as follows:

$$\Delta H = K \bullet (c / g) \bullet \Sigma\ \Delta v \tag{59}$$

It is seen that the only change to the equation is the introduction of the new term, K. Graf and Marchi (1997) provide a plot of this term as a function of the ratio of the valve opening or closing time, T, to the critical time, 2L/c. For all values of T less than 2L/c, the value of K is 1.0 and the case of the instantaneous valve is reproduced. However, if T is greater than 2L/c, the value of K is less than 1.0, and decreases as a logarithmic function of (T/(2L/c)).

For the case where the pipe is made up of a series of pipes of different diameters, Stepanoff (1949) provides the following equation for the equivalent initial velocity to be used in the above equations:

$$v_{EQUIV} = (L_1 \bullet v_1 + L_2 \bullet v_2 + L_3 \bullet v_3 + \ldots) / L \tag{60}$$

Where:
$v_{EQUIV}$ = Equivalent velocity
$L_I$ = Length of ith section of pipe
$v_I$ = Fluid velocity in ith section of pipe
L = Total length of pipe

In a typical pumping system, the events that can lead to water hammer problems include the following:

- Rapid closing of a valve.
- Rapid opening of a valve.
- Abrupt startup of a pump against a pipeline full of static fluid.
- Abrupt shutdown of a pump.
- Shutdown of one or more pumps due to power failure.

The potentially damaging effects of water hammer are obvious. If the pressure generated by the transient is high enough, pipes could rupture or valves and other components could fail. In general, the pipe's ability to withstand water hammer decreases as pipe diameter increases. Theoretically, the pipe could also fail by collapsing inwards during the negative pressure (expansion) wave but that rarely occurs since the pipe wall is normally stronger in compression than tension.

In general, since suction pipes tend to have short lengths and low flow velocities (due to the employment of large pipe diameters), water hammer problems seldom originate in the suction piping. However, in centrifugal pumping systems not employing check valves, if water hammer occurs in the discharge piping, its effects can pass through the pump and into the suction piping. Stepanoff (1949) describes several cases where water hammer originating in the discharge piping led to failures in the suction system.

Dodge (1960) states that sharp bends and elbows reduce the shock of water hammer and decrease the amplitude of the pressure wave. In general, the more branches there are in a circuit, the greater the damping. Thus, as a rule, complex circuits do not normally suffer from water hammer problems.

There are several methods for reducing the effects of water hammer. First, the valve or other quick-acting component can be modified to smooth out its flow and/or lengthen its opening or closing time. Examples provided by Dodge (1960) are the tapering of valve spools, nesting of springs to better control the closing force in a solenoid valve, addition of dashpots to valves, and the employment of cams to time valve or pilot operation.

Since, as was shown earlier, the severity of water hammer is highly dependent on the acoustic velocity, a reduction in acoustic velocity will always be beneficial from the standpoint of fluid transients. Since it has been previously shown that even minute amounts of gas can greatly reduce the acoustic velocity of a liquid, Wylie and Streeter (1993) state that entraining air into the liquid flow in a system is widely used to eliminate water hammer problems. This is normally done at a section in the system where the pressure is below atmospheric, such as at the entrance to a draft tube.

Finally, a system's transient performance can be improved by going to larger pipe sizes. Since the pressure spike is proportional to the initial flow velocity, doubling the pipe diameter has the effect of reducing the spike by a factor of four. This design change is often particularly helpful in suction piping since it can greatly reduce the likelihood of cavitation. If such measures are not feasible, a pulsation control device, such as an accumulator, can be added to the system to absorb the pressure spikes. Pulsation control devices are discussed in much greater detail in an upcoming section.

## PIPING VIBRATION

The previous sections have focused on the main sources of pulsation problems in pump piping systems—reciprocating pumps, centrifugal pumps, vortex shedding, and water hammer. Equally important are the adverse effects that pulsation can render. Although there are many of these, the biggest problem occurring in discharge piping is piping vibration and the biggest headache associated with suction systems is cavitation. Accordingly, these two troublesome areas are the subjects of this and the next section.

Of course, there are many other problems that can result from pulsation that space constraints do not permit detailed discussions of. First, if an acoustic resonance happens to coincide with the resonant frequency of one of the many valves in the piping system (relief valves, pressure regulators, etc.), that essentially behave as mass-spring oscillators, valve chatter can occur. Second, large pressure and/or flow fluctuations in the vicinity of the pump can impair pumping performance.

The subject of piping vibration is so vast that it is easily worthy of a tutorial of its own, which the authors may very well generate at some later date. In the interest of brevity, the treatment herein will only touch on how pulsations can lead to pipe vibration problems. Readers interested in a more comprehensive treatment of this subject are referred to Wachel, et al. (1990).

On its own, pressure pulsations are not capable of producing piping vibration. For instance, if an infinitely long straight pipe were subjected to traveling pressure waves of extremely large amplitudes, no vibration would occur (of course, the pipe could rupture from simple overpressure). In order for vibration to occur there must be some way for the pressure pulsations to translate into an oscillatory force that is applied to the piping. For instance, if the straight pipe referred to above were terminated in an elbow, the pulsations acting on the elbow's cross-section would generate the needed oscillating force and the pipe would probably vibrate excessively. The elbow is, thereby, referred to as a point of acoustical-mechanical coupling. In general, such points occur when the pulsating pressures act on unbalanced areas. Components in typical piping systems where this type of coupling occurs include elbows, sudden expansions and contractions, capped ends of bottles and manifolds, orifices, and valves.

In typical piping systems, the most common coupling point is at elbows or bends. The manner in which this coupling occurs at an elbow is illustrated in Figure 23 for the special case of a 90 degree elbow. If the pulsating pressure at the bend is P and the cross-sectional area of the pipe is A, it is easily seen that this will act to generate fluctuating forces in both the x and y-directions (where the x and y-directions are the directions of the legs and do not necessarily have to be horizontal and vertical) per the following:

$$F_X = F_Y = P \bullet A \tag{61}$$

Where:
$F_X$ = Cyclic force acting on elbow in x-direction (lbf)
$F_Y$ = Cyclic force acting on elbow in y-direction (lbf)
P = Amplitude of Pressure pulsations (psi)
A = Pipe Cross-sectional area ($in^2$)

Vector addition can then be used to show that the net resultant force, $F_{NET}$, for the 90 degree bend has a magnitude of 1.414 • P • A and acts at an angle of 45 degrees with respect to the x and y-axes. For the more general case where the angle between the two legs of the bend is θ, the resultant force can be shown to be:

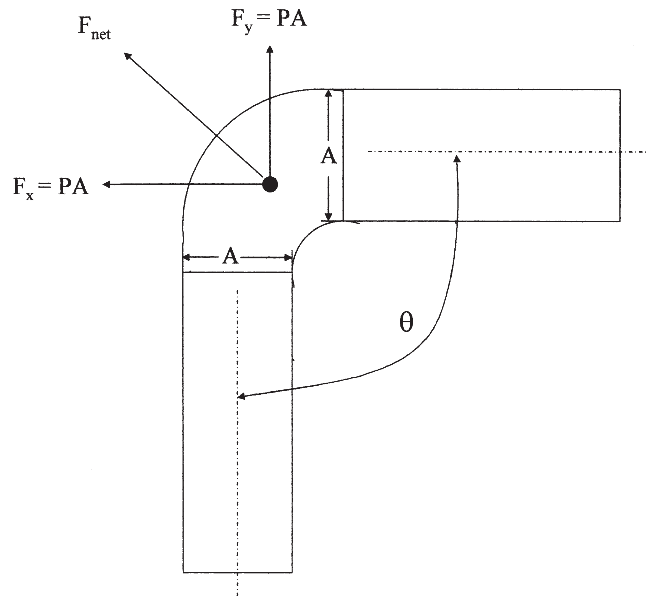


*Figure 23. Shaking Forces Generated in an Elbow.*

$$F_{NET} = 2 \bullet P \bullet A \bullet \cos(\theta / 2) \tag{62}$$

It is, thereby, seen that as the bend angle, θ, is made greater than 90 degrees, the magnitude of the shaking force is reduced. Thus, the worst case is also the most prevalent—the 90 degree, right angle elbow. Accordingly, if the piping must make a 90 degree change in direction, it is better, from a vibration standpoint, to employ two 45 degree elbows than a single 90 degree bend. Additionally, since elbows always convert pulsations into dynamic forces, it is good design practice to employ clamps in the vicinity of elbows to prevent them from applying shaking forces to the piping system.

Another source of dynamic coupling in a piping system is a change in piping diameter, either a contraction or expansion. It is easily shown that for either case, the shaking force is given by the following:

$$F_{DYN} = P \bullet (\pi / 4) \bullet \left(D_1^2 - D_2^2\right) \tag{63}$$

Where:
$D_1$ = Larger diameter
$D_2$ = Smaller diameter

Another source of coupling is any enclosed vessel such as a surge volume, bottle, manifold, etc. In such devices, the pulsation forces present at each end must be assessed, along with those acting on any vessel internals present such as baffles and supports, in order to determine the total unbalanced force. As will be shown shortly, there are some acoustic modes that generate large shaking forces on a vessel and others that generate no force, regardless of the amplitude of the pulsations. It all depends on the phasing of the forces with respect to one another.

This concept is illustrated for a typical vessel in Figure 24, which is taken directly from Wachel and Tison (1994). Since any vessel represents a closed-closed pipe, the first mode is a half-wavelength mode as shown in the top illustration in the figure. Since the pressure pulsations are 180 degrees out-of-phase, when the pressure on the left-hand end is positive, that on the right-hand end is negative, and vice versa. Accordingly, since the effect of both pressures is to generate a force acting from right to left, the two effects are directly additive. The resultant force is, thereby, given by the following:

$$F_{DYN} = 2 \bullet P_{DYN} \bullet A \quad (64)$$

Accordingly, whenever any vessel, such as a pulsation bottle or manifold, is in resonance with its fundamental mode, large shaking forces can result.

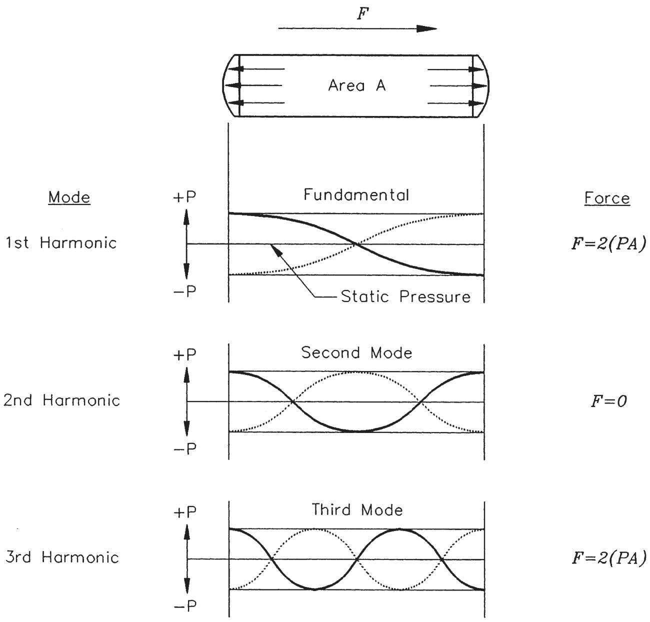


*Figure 24. Shaking Forces Generated in a Vessel.*

Examination of the figure reveals that the situation for the second (two half-wavelength) mode is entirely different. This time, the pressure pulsations at the two ends are perfectly in-phase. The forces acting on the left and right-hand ends, therefore, cancel each other out. Although they act to put the vessel in tension and compression, they generate no net force capable of exciting vibration, regardless of how large the pulsation amplitudes are. From the figure, it is easily visualized that all odd numbered modes generate large shaking forces (per Equation (64)) in vessels and all even numbered modes generate no shaking force at all.

A similar situation occurs when a volume-choke-volume filter's fundamental closed-closed mode is excited, as is shown in Figure 25. Again, the pressures acting on the two ends are 180 degrees out-of-phase but the shaking force is nowhere near as large as for the last case. The reason for this is that the pressures within the two volumes, $P_A$ and $P_B$, are assumed to be uniform everywhere throughout the volumes. Thus, if the right-hand volume is looked at, the force generated by pressure, $P_B$, acting on the right-hand side of the volume is almost totally counterbalanced by the force produced by $P_B$ acting on the left-hand side. The only difference between the two areas is the choke area, $A_{CHOKE}$, which generates a net force acting to the right. The situation in the left-hand volume is exactly the same—the differential area is again the choke area. Since the two volumes' pressures are 180 degrees out-of-phase (i.e., $P_B = -P_A$), the forces in the two volumes are additive so that the total shaking force is given by the following:

$$F_{DYN} = 2 \bullet P_A \bullet A_{CHOKE} \quad (65)$$

It is seen from all of the above equations that the shaking forces that are capable of exciting piping vibration are all proportional to the amplitude of the pressure pulsations. Since the pulsation amplitudes in suction piping are almost always a fraction of those in discharge systems, piping vibration problems are much more common in discharge systems.

A piping system behaves in the manner of any elastic mechanical system—it has natural frequencies that are characteristics of the

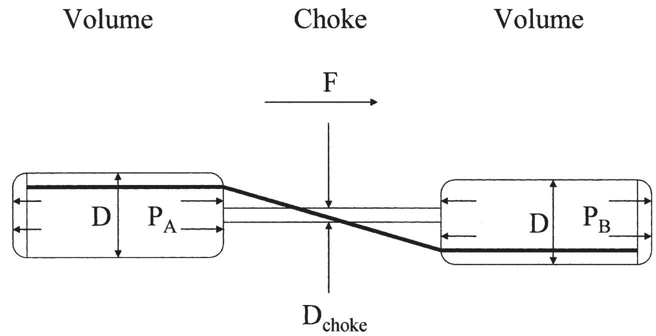


*Figure 25. Shaking Forces Generated in Volume-Choke-Volume Filter.*

system. If the piping system is excited by an oscillatory force having a frequency equal to or close to one of its natural frequencies, mechanical resonance occurs and the vibration amplitudes, forces, and stresses can become excessive. This is exactly the same situation as the piping acoustic resonance discussed earlier. Per Parry (1986), in typical piping systems, mechanical resonance can be associated with amplification factors as high as 20, a number verified by Beynart (1999). This means that the vibration amplitudes are 20 times what they would be if the same force were applied statically.

Of course, the worst of all worlds is when a piping system has an acoustic resonance that is coincident with or close to a piping mechanical resonance. In this case, the acoustic amplification factor, which Parry (1986) states can be as high as 15, combines with the mechanical amplification factor (20, from above) to generate an overall amplification factor that can be as much as 300. It is not hard to visualize how such a situation could lead to problems.

The most important thing in piping system design is, therefore, to ensure that acoustic resonances are not capable of exciting any piping mechanical resonances. In order to ensure that a piping system will operate troublefree in the field, a thorough acoustic analysis should be combined with a good piping vibration analysis. Although the details of the latter are well outside the scope of this tutorial (interested readers should refer to Wachel, et al., 1990), there is one aspect of the analysis process that should be brought up—the relative accuracy of the piping acoustic analysis versus that of the mechanical vibration analysis.

In an ideal world, a piping system would be designed using both analysis types in a complementary fashion. However, in reality, many engineers put much more emphasis on one discipline or the other. For instance, some engineers believe that if they employ a rigorous pulsation analysis to ensure that pulsation levels are kept low at all frequencies of concern, they can get by with only a rudimentary piping vibration analysis. On the other hand, other engineers prefer to utilize a rigorous piping vibration analysis to show that the piping is theoretically sound and not worry about the pulsation analysis.

In the authors' experience, and that of Tison and Atkins (2001) and Grover (1966), the two approaches are not even close to being equally effective. Because of the accuracies that the two analyses can be performed to, the first approach is highly preferable to the second. Unfortunately, the second approach is much more common, for two primary reasons. First, most organizations have at least a few engineers who are highly proficient at using finite element analysis computer codes for structural analysis. Second, acoustic simulation codes are nowhere near as prevalent as finite element codes and, more importantly, the number of engineers skilled at using such codes is relatively low.

The primary reason for the preference for the first approach is the large amount of error that is present in even the best piping vibration analyses. Based on their field experience, Tison and Atkins (2001) state that even under the best of circumstances, piping mechanical natural frequencies can only be calculated to an accuracy of ± 20 percent. This is under the optimum situation where the piping's boundary conditions are accurately known and the piping system and supporting structure are modeled in extreme detail. Under more typical circumstances, Tison and Atkins (2001) estimate the error in piping natural frequency calculations to be ± 50 percent, or even higher.

The biggest reason for these large inaccuracies is the difficulty in accurately modeling the piping boundary conditions. In the words of Tison and Atkins, a piping/structural support system is not a "polished machined part," for which finite element models can be accurately prepared. As Tison and Atkins (2001) point out, in many real world installations, pipe supports become loose or break, leaving the piping unsupported at that location. If the analysis had counted on that support to keep the mechanical natural frequencies above a certain threshold, the loss of that single support invalidates the analysis. Tison and Atkins (2001) provide a list of several other factors that add to the inaccuracy of such analyses.

In addition to natural frequency analyses, the other major type of piping vibration analysis, the forced response analysis, is not likely to be more accurate, and is probably even less so. Per Tison and Atkins (2001), all of the items that create inaccuracies in natural frequency calculations have the same effect on response analyses. In addition, since the exact amount of damping present in a piping system is almost impossible to calculate, an amplification factor at resonance must be assumed. Per Tison and Atkins (2001), this can vary all the way from 10 to 100. Since, in steady-state response analyses, the piping displacements and stresses vary inversely with the assumed amplification factor, this can introduce an uncertainty of as much as 10-to-one to the results. Tison and Atkins (2001) even go so far as to say that response analyses are almost useless unless test data from the field is available to anchor the model so that it yields the correct natural frequencies and damping.

Another shortcoming in most piping vibration analyses is that the models often do not include the components that are most vulnerable to vibration-related failures. Tison and Atkins (2001) state that many vibration-related problems are not associated with the main process piping itself, but with other attached components such as valve actuators, tubing, conduit and cable trays in rack systems, instrument connections (thermocouples, pressure transducers), and small branch connections (for instruments, vents, and drains). Other common problems cited by Lovelady and Bielskus (1999) include failures of electronic instrumentation, sight gauges, thermal wells, pressure taps, and brackets. It is not at all uncommon for the main process piping to have low vibrations but to also act as an exciter for resonances in the attached components. However, most models do not account for any of these components.

All of the above problems are in stark contrast to the accuracies achievable in pulsation analysis. For instance, Tison and Atkins (2001) state that acoustic natural frequencies can normally be calculated to accuracies of ± 5 percent. Accordingly, Tison and Atkins (2001) recommend overcoming the inaccuracies inherent in piping vibration analyses through robust acoustical design. The authors are in complete agreement with this approach.

Although the previous discussion questions the value of detailed piping vibration analysis during the design stage, the authors do advocate the calculation of the pulsation-induced shaking forces, which are relatively straightforward as long as a good pulsation analysis has been performed, during this stage. While acknowledging the difficulty in determining an acceptable shaking force level, Tison and Atkins (2001) state that there is no question that much higher forces are permissible at lower frequencies (defined as frequencies that are significantly below the lowest piping mechanical natural frequency). Table 4 of Tison and Atkins (2001) provides allowable forces in terms of the total shaking force in a straight run of pipe divided by the number of supports that resist axial vibration of that particular run. The allowable forces are given as multiples of the nominal pipe size (thereby, larger pipes have larger allowable forces) and are provided for nonresonant and potentially resonant frequencies (with regards to piping mechanical resonance). The allowable levels for elevated piping, such as in pipe racks and offshore platforms, are given as one-half of those for ground level pipes.

In spite of the problems inherent in analyzing piping vibrations, there are some general rules that can be employed during design to minimize the chances of piping vibration problems occurring. These include the following, taken from Wachel, et al. (1995), Lovelady and Bielskus (1999), Cornell (1998), Grover (1966), API 618 (1995), Tison and Atkins (2001), and Wachel and Tison (1996).

- The number of elbows should be minimized.
- In the areas where elbows must be employed, 45 degree bends are preferential to 90 degree versions.
- All elbows and concentrated masses such as valves should be rigidly supported with clamps.
- Avoid using threaded connections for instrumentation (susceptible to vibration-related failures at thread roots).
- All welded connections should be stress relieved.
- Pipe braces, not hangers, should be employed to support the piping.
- Pipe supports should be attached to relatively rigid anchor structures.
- The stiffnesses of piping clamps and supports should be at least twice the stiffness of the basic pipe span.
- Avoid using gravity-based supports.
- If the piping natural frequencies in reciprocating pumping systems can be accurately determined, they should be kept well above (Tison and Atkins, 2001, suggest a margin of 50 percent) the frequencies of significant pulsation-induced forces

## CAVITATION

Although piping vibration is often a troublesome problem in pump discharge piping, in suction piping, the pressure levels are normally too low for significant vibration amplitudes to arise. Instead, the most likely adverse consequence of large pulsation levels in the suction system is cavitation. That is not to suggest that piping vibration problems do not occur in suction systems—they do occasionally occur. However, when they occur, the triggering mechanism is usually pressure spikes due to cavitation, not pressure spikes due to pulsation alone.

Cavitation occurs when the local static pressure in a liquid falls below its vapor pressure. The liquid flashes locally and a gas bubble is formed. The formation of cavitation bubbles relies on the presence of microscopic gas nuclei within the liquid. Nuclei are normally present in the form of impurities, dissolved gas, and surface imperfections. The presence of these nuclei is the reason why liquids cannot withstand a tensile force. In the initial stages of cavitation, the nuclei give rise to the formation of gas bubbles in regions of low static pressure. The pump's flow then carries many of these bubbles to regions where the static pressure is above the vapor pressure. This causes the bubbles to collapse and produce high intensity pressure waves. This process continues as long as regions of pressure that are below the vapor pressure persist. Per Beynart (1995), although this repeating cycle normally lasts for only a few milliseconds, the localized pressures and temperatures can be as high as 60,000 psi and 1500°F.

Cavitation can occur at a pump's inlet, inside the cylinders of a reciprocating pump, in the suction manifold, and in other places where changes in fluid velocity occur. The repeated pulses generated during bubble collapse can generate pulsations, noise, and vibration of pump components or piping. They also can do significant damage to the pump internals. If the collapses occur in the vicinity of any metal parts, microscopic jets of liquid impinging on the metal can eventually lead to erosion damage. Cavitation also can degrade pump performance, sometimes drastically, if enough bubbles are present.

Figure 26 shows how pressure pulsations can cause cavitation. It is seen from the figure that the mean static pressure for this hypothetical pump, $P_{MEAN}$, is above the liquid's vapor pressure, $P_{VAPOR}$. Accordingly, if there were no pulsations in this suction system, there would be no cavitation. However, the presence of a pulsating pressure, $P_{CYCLIC}$, means that the minimum static pressure in the system is now $P_{MEAN} - P_{CYCLIC}$. If, as is shown in the figure, this minimum pressure is below the vapor pressure, the liquid will theoretically cavitate. It is, thereby, seen that higher pulsation amplitudes increase the likelihood of cavitation.

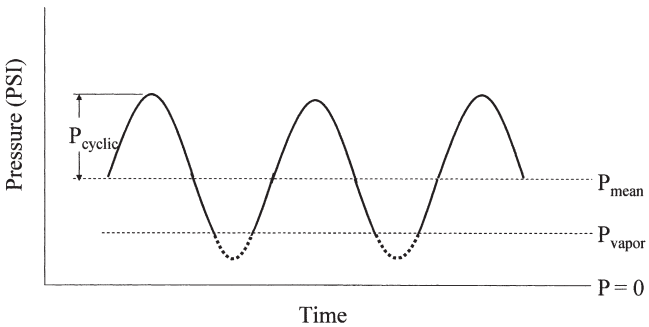


*Figure 26. Pulsation-Generated Cavitation.*

As most pump designers will recognize, the treatment shown in the figure is conservative. Although whenever the negative pressure peak manages to drop below the vapor pressure for an instant, cavitation will occur in theory, a more severe condition is required to generate any noticeable effects. As Wachel, et al. (1989), note, this simplistic theory of when cavitation occurs neglects factors such as frequency, duration of time that the pressure is below vapor pressure, and other factors that have been proven to be important in cavitation.

In an attempt to rectify this shortcoming, Wachel, et al. (1989), define a cavitation potential number (CPN), which gives the percent of the cycle that the static pressure is below the vapor pressure. Wachel, et al. (1989), further state that based on their experience, a CPN value above 25 percent is required before significant cavitation will occur. Additionally, they state that values above 50 percent are normally required to generate the 3 percent reduction in flow capacity that is conventionally used to define the onset of cavitation. Wachel, et al. (1989), further state that while the exact CPN number associated with significant cavitation failures is not yet known, the use of the CPN method has permitted them to better understand why some pumps operate satisfactorily even though cavitation is theoretically present. While the authors do not have any personal experience using this methodology, it certainly appears superior to the overly simplistic method of Figure 26.

If pressure pulsations are successful at generating significant cavitation, it can usually be easily seen in a pressure versus time plot obtained from the field. Whereas normal pulsation generates a sinusoidal wave, the presence of cavitation will cause a squaring off of the bottom portions of each of the waves, as is shown in Figure 1 of Wachel, et al. (1989). This squaring off occurs because a fluid cannot support a negative pressure below the vapor pressure. After the initial signs of cavitation, the pressure signature also often shows a series of sharp pressure spikes, indicative of cavitation bubble collapse.

The manner in which pump designers traditionally attempt to avoid cavitation is via specification of the net positive suction head at the pump inlet flange. The NPSH is simply the difference between the absolute total pressure at the pump inlet flange and the liquid's true vapor pressure at the inlet temperature. There are two types of NPSH—net positive suction head required ($NPSH_R$) and net positive suction head available ($NPSH_A$). The $NPSH_R$ is the amount of head that is required at the pump inlet flange in order to fully charge the pumping elements without causing the liquid to cavitate. $NPSH_R$ is normally specified by the pump manufacturer based on tested pump performance. In general, higher values of $NPSH_R$ represent more stringent conditions. On the other hand, the $NPSH_A$ is the actual amount by which the total pressure at the inlet flange exceeds vapor pressure. In general, $NPSH_A$ is a measure of how much margin the suction system has against cavitation (higher $NPSH_A$ values represent larger margins).

In order to determine how much $NPSH_A$ a given pump needs to have in order to avoid cavitation, the Hydraulic Institute provides a standard calculation procedure, as follows:

$$NPSH_A = h_s - \left(h_F + h_A\right) - \left(TVP + 7\right) \tag{66}$$

Where:
$NPSH_A$ = Available net positive suction head at the pump inlet flange (ft)
$h_S$ = Total head at source (ft)
$h_F$ = Frictional head loss in suction lines (between source and pump inlet) (ft)
$h_A$ = Acceleration head (ft)
TVP = Absolute vapor pressure at pumping temperature (ft)

The way that this equation is utilized is that, starting with the $NPSH_R$ value obtained from the pump manufacturer, the system designer decides how much $NPSH_A$ is desired. The $NPSH_A$ must be at least equal to the $NPSH_R$ but is often made somewhat greater in order to provide margin. Using this desired value and calculated values for the friction head, $h_F$, and acceleration head, $h_A$, the required total head at the source can then be back-calculated.

The term of interest from the standpoint of pulsation is the acceleration head, $h_A$. The acceleration head is supposed to account for the pressure pulsations occurring in the suction line. However, inspection of the equations used for its calculation reveals that the quantity being calculated is basically the amount of head required to accelerate the mass of fluid in the suction line into the pumping elements. Accordingly, this is basically an attempt to account for a dynamic phenomenon with a static calculation. Not surprisingly, there are many situations, especially if any acoustic resonances occur in the suction system, where this method underpredicts the pulsation amplitudes, often by a substantial amount. Thus, in such cases, the calculated $NPSH_A$ could very well meet the required value but the pump could still suffer from severe cavitation.

Parry (1986) suggests that the above flaw be corrected by subtracting the actual pulsation amplitude in the suction system from the value of $NPSH_A$ calculated from the above equation to obtain the actual $NPSH_A$. However, as Wachel, et al. (1989), note, knowledge of the actual pulsation amplitude requires that an indepth pulsation analysis be performed, the likes of which is often not available at the time of suction system design. Thus, Wachel, et al. (1989), state that the unknown pulsations have traditionally been accounted for by using conservative values for the acceleration head term.

Miller (1988) states that one of the first expressions for acceleration head is the following, which was empirically developed from test data:

$$h_A = L \bullet v \bullet N \bullet C_2 / \left(K \bullet g\right) \tag{67}$$

Where:
$h_A$ = Acceleration head (ft)
L = Actual length of suction line (ft)
v = Velocity of liquid in suction line (ft/sec)

N = Crankshaft rotating speed (rpm)
g = Gravitational acceleration (ft/sec$^2$)
K = Constant that compensates for liquid compressibility
$C_2$ = Constant, depending on type of pump

Miller (1988) states that the first constant, K, is equal to 1.4 for cold water. Wachel, et al. (1995), expand on this by stating that the value of 1.4 is representative for relatively incompressible liquids, such as deaerated water, while a more representative value of K for liquids having high compressibility (such as hydrocarbons) is 2.5. Table 3 of Miller (1988) provides values for the second constant, $C_2$, for different types of single and double-acting pumps. In general, $C_2$ decreases as the number of cylinders increases. Additionally, for a given number of cylinders, $C_2$ for the double-acting version is less than or equal to that for the single-acting pump. It should be noted that while the above equations account for dynamic modulation of the liquid as it fills the pump cylinder, they are incapable of accounting for resonant pulsations.

When pulsation-induced cavitation problems occur in the field, the natural response of raising the system's suction pressure will sometimes eliminate the cavitation problem. However, obviously, it has no impact on the pulsation amplitudes. Since pulsation-induced piping vibration and other structural problems become more likely at higher values of system pressure, this remedy will sometimes result in the simple exchange of one problem for another. Once again, the best remedy is the employment of proper pulsation control elements, designed on the basis of a good acoustic simulation.

## PULSATION CONTROL ELEMENTS

At this point, the reader should have some appreciation for how much trouble uncontrolled pulsations can cause. In order to assist in the control of pulsations, there are many commercially available devices that can be installed in suction and discharge piping systems. Although they go by many different names, such as pulsation dampeners, surge suppressors, acoustic filters, accumulators, stabilizers, snubbers, and desurgers, most of them operate on very similar principles. In fact, the large number of devices available can almost all be placed into one of three basic categories—energy absorbing devices (surge volumes), acoustic filters, or dissipative components. Of course, some devices incorporate features from more than one of these basic categories.

### *Helmholtz Resonator*

One of the fundamental elements in acoustics, which is strongly related to surge volumes and accumulators, is the Helmholtz resonator. A Helmholtz resonator consists of a rigid-walled cavity of volume, V, that is fed by a neck of area, S, and length, L. If a lumped approach is permissible, the neck acts as a fluid inertia, the volume behaves as an acoustic compliance, and the neck opening radiates sound as a simple source does and, therefore, behaves as an acoustic resistance. Of course, the neck also generates viscous losses. However, per Kinsler, et al. (1982), if the neck is larger than about 0.4 inch, the viscous losses are usually less than the radiation losses and can, therefore, be ignored.

If all of the preceding lumped element assumptions are valid, the natural frequency for a Helmholtz resonator is as follows:

$$\omega_N = c \bullet \left[ S / \left( L_{EFF} \bullet V \right) \right]^{1/2} \quad (68)$$

Where:
$\omega_N$ = Natural frequency (rad/sec)
c = Acoustic velocity (in/sec)
S = Neck cross-sectional area (in$^2$)
$L_{EFF}$ = Effective length of neck (inch)
V = Volume of cavity (in$^3$)

Kinsler, et al. (1982), explain that acoustic radiation effects render the effective length of the neck to be greater than its actual length, L. If the outer end of the neck is flanged, Kinsler, et al. (1982), suggest adding 85 percent of the neck diameter to the actual length, L, and if unflanged, they suggest 75 percent.

The acoustic impedances for the fluid compliance and inertia elements can be calculated using the equations given previously. With viscous effects ignored, the acoustic resistance of the neck is given by Kinsler, et al. (1982), as follows:

$$R = \rho \bullet \omega^2 / \left( 2 \bullet \pi \bullet c \right) \quad (69)$$

Where:
R = Acoustic resistance
c = Acoustic velocity
ρ = Fluid density
ω = Excitation frequency (rad/sec)

Whenever an incident acoustic wave has a frequency at or near the resonator's natural frequency, greatly amplified acoustic pressure will be produced within the cavity. Although this is probably of interest to the designer of stereo speakers, the relevance from a pulsations standpoint is that, at the resonant frequency, the Helmholtz resonator behaves like an electrical short circuit and transmits almost zero power to the downstream pipe. Thus, if such a device were placed at a side branch between a reciprocating pump and the main discharge line, at the resonant frequency, the discharge piping would be almost completely isolated from the pulsations generated by the pump. The transmission loss of a Helmholtz resonator is provided as a function of frequency in Figure 2.4 of Junger (1997).

### *Energy Absorbing Devices (Surge Volumes)*

The first class of pulsation control devices is energy absorbing devices or surge volumes. A surge volume is a relatively large, empty bottle that is connected to the suction or discharge of a pump. The bottle behaves as an acoustic compliance that can isolate the fluid in the suction and discharge lines from the pulsations generated by the pump. A mechanical analogy is the use of a low stiffness coupling in a torsional system to isolate the vulnerable elements in the system from the torsional excitation source (normally, a reciprocating engine or synchronous motor). Since the compliance associated with well-designed surge volumes is usually quite high, the presence of the surge volume normally results in the lowering of the system's acoustic natural frequencies.

In addition to their isolation capabilities, surge volumes can also be looked at as flow smoothers. In a reciprocating pump application, the surge volume acts to absorb the peaks of the flow pulsations and returns the flow to the main line during the "valleys." In this manner, the fluid in the suction and discharge lines see a much smoother flow.

In general, the effectiveness of a surge volume is directly proportional to its acoustic compliance. Since referral to Equation (47) reveals that the compliance is directly proportional to volume, larger volumes tend to be more effective, consistent with intuition. Cost and space constraints normally place an upper limit on the bottle volume that can be used in a given application.

In general, surge volumes are most effective at damping out pulsations at low frequencies, such as those generated by a reciprocating pump. Miller (1988) states that the frequency at which these devices start to lose their effectiveness is at about 50 Hz. This means that they should be effective at attenuating the plunger frequency excitations of reciprocating pumps as long as the pump does not run faster than 500 rpm or have more than six cylinders.

Although all surge volumes operate on the same basic principles, they can basically be broken down into two distinct categories, liquid volumes and accumulators. Liquid volumes are precisely what the name suggests—surge volumes in which the working fluid is liquid. On the other hand, accumulators employ some type of inert gas, usually dry nitrogen, as the working fluid.

Although some accumulators permit the gas to directly communicate with the pumped liquid, most utilize some form of bladder or diaphragm to keep the two separate.

Beynart (1999) suggests that surge volumes for reciprocating pump suction systems be equal to10 times the total displacement of the pump, where the displacement is simply the number of cylinders multiplied by the cylinder bore area and the piston stroke. Blodgett (1998) goes further by suggesting that in addition to the total pump displacement, the surge volume should also be compared to the total volume of fluid between the valves and the surge element. Blodgett (1998) suggests that the surge volume be at least five times larger than those two volumes.

In addition to the volume, the physical lengths of all dimensions within the tank need to be kept small enough to make the lumped parameter assumption valid. Per Chilton and Handley (1952), this means that all surge volume dimensions must be less than about 1/12 of the expected wavelength.

*Accumulators*

As stated previously, accumulators employ a gas as the working volume. Although there are some accumulators where the gas and liquid are in direct contact with each other (sometimes called gas-over-liquid types) still in use, most accumulators employ a bladder to separate the two. Accumulators are commonly used on both the suction and discharge sides of reciprocating pumps.

The reason that accumulators use gas as the working fluid is that it takes a much smaller volume of gas (compared to liquid) to generate a desired acoustic compliance. The equivalent volume of liquid provided by a specified volume of gas is given by the following:

$$V_{EQUIV} = \left(K_{LIQUID} / K_{GAS}\right) \bullet V_{GAS} \tag{70}$$

Where:
$V_{EQUIV}$ = Equivalent volume of liquid
$V_{GAS}$ = Actual volume of gas
$K_{LIQUID}$ = Liquid bulk modulus
$K_{GAS}$ = Gas bulk modulus

Since the bulk modulus of a gas is directly dependent on the gas pressure, this ratio is dependent on line pressure. In some systems, this ratio can be as large as 10,000 to one although Singh and Chaplis (1990) indicate that a 2000 to one ratio is probably more typical. In other words, to achieve the same pulsation attenuation, a liquid surge volume would need to have a volume 2000 times that of the gas in an accumulator. The advantages provided by using gas are obvious.

Accumulators are highly effective at controlling pulsations at low frequencies, such as the flow-excited pulsations generated by reciprocating pumps. However, since the bladder must flex every time the liquid pressure changes, their response is greatly degraded at higher frequencies. Parry (1986) states that the effectiveness limit for most bladder-type accumulators is around 50 to 75 Hz.

The precharge pressure of an accumulator is the pressure that the gas in the bladder is set at during installation. The performance of the accumulator can be dramatically affected by the precharge pressure. If this pressure is set too low, the bladder will be in a perpetual state of motion, which can lead to early bladder fatigue. Low precharge pressure also reduces the effective volume of the accumulator. On the other hand, if the precharge pressure is made too high, the accumulator will not respond whenever the line pressure drops below the precharge pressure.

Although the optimum precharge pressure is somewhat dependent on the application, in general, Miller (1988) recommends that it be set at 60 to 70 percent of average discharge pressure. Singh and Chaplis (1990) state that most accumulator manufacturers recommend somewhere between 50 and 70 percent of line pressure. Cornell (1998) agrees with the 50 percent number for suction accumulators but recommends that discharge accumulators be precharged to 80 percent of discharge pressure.

Unfortunately, the ratio of charge pressure to line pressure can vary in field installations for several reasons. First, since the discharge pressure in a reciprocating pumping system is normally set by the flow resistances in the system, as was discussed earlier, changes in flow result in changes in discharge pressure. Additionally, when that resistance is generated by one or more variable area valves, changes in valve position also affect discharge pressure. Finally, bladders can develop leaks over time, creating a loss in charge pressure.

Fortunately, Singh and Chaplis (1990) report on tests that showed that as long as the precharge pressure stayed between 10 and 90 percent of discharge pressure, the discharge accumulator's performance was not seriously affected. They believe that the reason for this is that, even at very small gas volumes, the high compliance of gas maintains the accumulator's effectiveness. Of course, if the precharge pressure ever exceeds 100 percent of the discharge pressure, the bladder would expand to its maximum value and the system's level of protection would be minimal, if anything. Wachel, et al. (1985), report on a case where pressure pulsations increased dramatically whenever the discharge pressure dropped below the accumulator's charge pressure.

The effectiveness of accumulators can be reduced by bladder stiffness, bladder permeability to certain liquids, and restriction of bladder expansion and contraction by accumulator internals such as constraining cages, mandrels, etc. Additionally, since their performance depends on their acoustic capacitance, which in turn depends on volume and bulk modulus, their effectiveness can be degraded by large variations in line pressure, upon which both volume and bulk modulus depend. All of these complicating factors make prediction of the performance of accumulators more difficult than that of all-liquid surge volumes.

There are two basic configurations by which accumulators can be incorporated into a piping system. The first is with the accumulator attached to the main line via a side branch. Such an accumulator is referred to as an appendage accumulator, or side branch accumulator. In the other configuration, the accumulator is actually right in the main line so that all of the flow has to pass through it. Such a device is called a flow-through accumulator.

In an appendage accumulator, the gas-filled bladder is separated from the main flow via a neck and a volume of liquid beneath the bladder. The factors that affect the acoustic performance of an acoustic accumulator include the neck, the compliance of the liquid that is in contact with the bladder (usually insignificant), the elastic and mass properties of the bladder, and the compliance of the gas within the bladder. Since the neck behaves as an acoustic inertia, these devices are usually most effective at low frequencies. However, since the presence of the neck renders the accumulator highly similar to a Helmholtz resonator, it can improve the attenuation at the resonant frequencies, compared to designs that do not employ necks.

The dynamics of an appendage accumulator can be understood by referring to its mechanical analogy. Since the neck behaves as an inertia and the gas behaves as a compliance, the mechanical analogy is a simple mass-spring system. This system has a resonant frequency that Wachel and Price (1988) state will normally be less than 100 Hz. At frequencies much above this frequency, a mass-spring system and, by analogy, an accumulator, have serious response limitations.

Some appendage accumulators are designed to be used with a "diverter," which Figure 1E of Wachel and Price (1988) shows schematically. Basically, a diverter is a device that is placed in the main line opposite the accumulator neck, with the purpose of directing the main flow right at the bladder. Since, as stated above, appendage accumulators are limited at high frequencies, accumulator manufacturers claim that the use of a diverter will improve the high frequency performance of the accumulator. Test data have shown this to be true.

While acknowledging the better high frequency performance that a diverter provides, Wachel and Price (1988) dispute that the reason for this improvement is the diversion of the flow. Instead, they believe that since the presence of the diverter in the main line increases the pressure drop, the diverter is providing additional acoustic damping. Wachel, et al. (1985), corroborate the lack of benefit of directing the flow at the accumulator by reporting a case study where, at the suggestion of the accumulator manufacturer, the piping was changed to direct the flow at the bladder and the accumulator's performance showed no change. Wachel and Price (1988), therefore, recommend that the diverter be modeled as an acoustic resistance and they state that models of that type have satisfactorily explained the observed improvement in high frequency performance.

The primary advantage of using a flow-through accumulator, compared to an appendage design, is that the elimination of the neck allows it to provide significant attenuation over a larger frequency range. Thus, these devices behave very similarly to appendage accumulators without necks. The expanded frequency range makes it less critical that the accumulator's maximum attenuation frequency be matched to the main system excitation frequency than in appendage accumulators. According to Beynart (1999), these devices also carry the advantage of quicker bladder response time. However, for a given gas volume, flow-through accumulators are somewhat larger than their counterparts and, therefore, are normally more expensive. Additionally, as was shown in the discussion of the Helmholtz resonator, the presence of the neck in an appendage accumulator provides damping via acoustic radiation and viscous dissipation.

Chilton and Handley (1952) state that flow-through accumulators will normally outperform appendage designs, unless the latter can be designed to be essentially neckless. Graf and Marchi (1997) are even more emphatic, claiming that appendage accumulators will only reduce pulsations by 10 to 30 percent of the attenuation of the same size accumulator in a flow-through arrangement.

Although larger gas volumes are normally associated with higher accumulator capability, the two do not necessarily go hand-in-hand. Wachel, et al. (1995), provide a case study where increasing the bladder volume actually increased the pulsation amplitudes. The explanation for this was the larger volume (and, therefore, higher compliance) lowered the acoustic natural frequency sufficiently to put it in the vicinity of the excitations coming from the triplex pump. Furthermore, Wachel, et al. (1989), state that a system having an incorrectly sized accumulator can experience higher pulsations than one having no accumulator at all. Thus, accumulator sizing, like the design of all pulsation control components, needs to be performed using a rigorous acoustic analysis of the system.

Per Warwick (1999) metering pumps in chemical and water services normally employ an appendage-type accumulator in the discharge line. These accumulators are normally installed at a tee in the discharge line and should be no further than 40 pipe diameters to the pump discharge flange. The pipe between the tee and the accumulator should be straight, the same diameter as the accumulator or larger, and no longer than 15 pipe diameters. As a rule of thumb, the effective volume should be at least 15 times the total displacement of the pump. Such accumulators will normally be effective in minimizing the effects of both flow and acceleration excitations.

Cornell (1998) states that accumulators can also be effectively deployed to alleviate water hammer problems originating at quick-closing valves. These devices absorb the pressure spike by providing a secondary outlet to the flowing liquid after the valve closes. Because of the quick response required, Cornell (1998) recommends that the dampener be installed immediately upstream of the valve and no more than 10 pipe diameters away. Since the full capacity of the accumulator must be available for this type of service, Cornell (1998) recommends that the bladder be precharged to 95 to 98 percent of line pressure.

Because of the large precharge, accumulators employed to control water hammer essentially operate at the local system pressure. Figure 27 shows a schematic of such a device—although no bladder is shown, the behavior is the same whether a bladder is used or not. If the valve is closed abruptly, a situation that normally leads to water hammer, the pump flow enters the accumulator, compresses the air in the bladder, and the flow in the line is reduced as the pressure builds up (the pump is assumed to be a centrifugal pump—no sane design would ever result in the deadheading of a reciprocating pump). Although the system pressure rises, the peak pressure is much less than for the case with no accumulator.

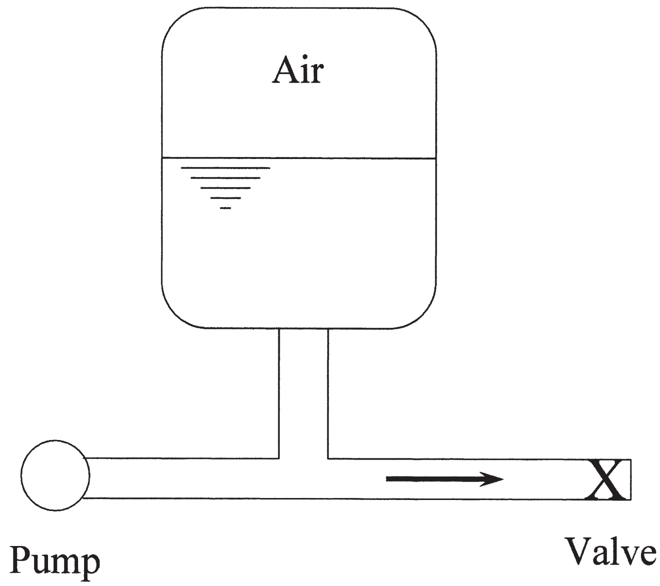


*Figure 27. Use of Accumulator to Alleviate Water Hammer.*

Of course, if an accumulator is to be used to control water hammer, the side branch connection and accumulator neck must be designed to minimize accumulator response time. The gas volume needs to be sized to handle the expected flow and pressure and the accumulator's liquid volume needs to be sized to avoid emptying the accumulator during the water hammer low pressure cycle.

There are some applications where the conditions are such that a bladder-type accumulator cannot be employed. Since bladders are made from elastomers, they are not appropriate for high temperature applications. Miller (1988) sets the limit at about 300°F. Additionally, applications where the pumped liquid would chemically attack an elastomer are also off limits. In such cases, a gas-over-liquid design, which is just as efficient from a dynamic standpoint, must be employed or alternate pulsation control strategies must be implemented. If the gas-over-liquid route is selected, there are two main problems that must be overcome. First, the maintenance of a constant gas volume is much more difficult without the benefit of a bladder. Second, the gas can enter the liquid and introduce two phase flow problems.

*Acoustic Filters*

Acoustic filters use combinations of the three basic acoustic elements, inertia, compliance, and resistance, to prevent the transmission of pulsations. In most pumps, these filters are placed in the main line so that their primary function is to isolate the piping from the pulsations generated by the pump. Since, unlike accumulators, acoustic filters do not require the employment of any gas, they are often referred to as "all liquid filters" when utilized in pumps.

In direct contrast to accumulators, acoustic filters can be designed to be effective at either low or high frequencies, although not usually both at the same time. However, the size of the filter is inversely proportional to the frequency that has to be controlled. Thus, filters required to be effective at frequencies below about 50

Hz are usually extremely large and cost-prohibitive. Additionally, since acoustic filters are liquid-filled devices, they can be very expensive in high pressure systems.

The basic principle behind acoustic filters is one that has come up again and again in this tutorial—the fact that when a traveling pressure wave encounters a change in impedance, part of it is reflected and part of it is transmitted. An acoustic filter acts as an impedance change whose goal is to reflect as much of the incident energy as possible and, thereby, transmit as little as possible. The intensity transmission coefficient, introduced previously in Equation (33), is, thus, a good measure of the effectiveness of an acoustic filter. For instance, if an acoustic filter is placed between a reciprocating pump and the main discharge line, if it transmits no acoustic energy (i.e., intensity transmission coefficient of zero), than it has completely protected the discharge pipe from the pulsations generated by the pump. Unfortunately, there are no filters that can do this at all frequencies.

One of the most basic acoustic filters is the simple low-pass filter. As its name suggests, this type of filter is highly effective at preventing the transmission of high frequency acoustic waves and highly ineffective at dealing with low frequency waves (i.e., it allows them to "pass"). Although low-pass filters come in many different configurations, one of the simplest is achieved by inserting an enlarged section of pipe of cross-sectional area, $S_1$, and length, L, in a pipe of cross-section, S, as shown in Figure 10.6 of Kinsler, et al. (1982). Kinsler, et al. (1982), provide the following equation for the intensity transmission coefficient for this filter:

$$T_I = 4 / \left[ 4 \bullet \cos^2 kL + \left[ \left( S_1 / S \right) + \left( S / S_1 \right) \right]^2 \bullet \sin^2 kL \right] \qquad (71)$$

Where:
$T_I$ = Intensity transmission coefficient
k = Wave number

The frequency dependence in the above equation is due to the presence of the wave number, k. Per Kinsler, et al. (1982), this equation predicts that at low frequencies, the transmission coefficient is 100 percent (thus, not a very good filter) but it gradually decreases to zero at high frequencies. Figure 10.6 of Kinsler, et al. (1982), provide a plot of this coefficient versus frequency for an expansion chamber that is 1.965 inches long and has a cross-sectional area that is four times that of the main pipe.

From the above equation, the transmission intensity coefficient reaches its minimum value (i.e., location of optimum filtering) when the value of kL is π/2, which corresponds to the length of the filter being equal to a quarter wavelength. Following this minimum, this coefficient gradually increases with frequency until it again reaches a value of 100 percent when kL is equal to π. After that, the behavior fluctuates through a series of minima and maxima. The important point to note is that this device is not a low-pass filter in the truest sense since it does not attenuate all high frequency waves.

Another simple approximation of a low-pass filter may be achieved by inserting a constriction of cross-section area, $S_1$, and length, L, into a pipe. Kinsler, et al. (1982), state that the above equations can also be applied to this filter since their derivation makes no assumption of whether S or $S_1$ is larger. It should be noted that these two simple filters are commonly used in the design of automobile mufflers, gun silencers, and sound-absorbing plenums used in ventilating systems.

The counterpart to the two low-pass filters just described is the high-pass filter. As its name suggests, this filter is effective at attenuating low frequencies but allows high frequency waves to be transmitted. One of the simplest high-pass filters is a simple side branch having a radius, r, and length, L. Per Kinsler, et al. (1982), the side branch has a power transmission coefficient given by the following:

$$T_I = 1 / \left\{ 1 + \left[ \pi \bullet r^2 / \left( 2 \bullet S \bullet L_{EFF} \bullet k \right) \right]^2 \right\} \qquad (72)$$

Where:
$T_I$ = Intensity transmission coefficient
r = Radius of side branch
S = Cross-sectional area of pipe
k = Wave number
$L_{EFF}$ = Effective length of sidebranch

Figure 10.8 of Kinsler, et al. (1982), shows that this yields a transmission coefficient that is very nearly zero at low frequencies but which rises to nearly 100 percent at higher frequencies. Another manner of constructing a high-pass filter is by placing a single orifice in a pipe. As the diameter of the orifice is increased, the low frequency attenuation is improved, and the frequency corresponding to 50 percent power transmission is raised. If a number of orifices are placed in series and are spaced at appreciable fractions of the wavelength, the attenuation at low frequencies can be greatly increased. It should be noted that the filtering provided by an orifice has nothing to do with viscous dissipation—instead, it is due to its ability to reflect incident waves.

A third basic type of filter is a band-pass filter. Unlike the previous two types, instead of being effective at only low or high frequencies, it has a specific band of frequencies over which it is effective. The side branch, which is a high-pass filter, can be converted into a band-pass filter by simply adding a fluid compliance element to it. One way of achieving this is to simply make the side branch extremely long and terminate it with a closed end. Another, more common, method is to terminate the side branch with a volume, which would form the Helmholtz resonator discussed earlier. Per Kinsler, et al. (1982), the power transmission coefficient for a Helmholtz resonator is as follows:

$$T_I = 1 / \left\{ 1 + c^2 / \left[ 4 \bullet S^2 \bullet \left( \omega \bullet L_{EFF} / S_B - c^2 / \left( \omega \bullet V \right) \right)^2 \right] \right\} \qquad (73)$$

Where:
$T_I$ = Intensity transmission coefficient
c = Acoustic velocity
S = Cross-sectional area of pipe
ω = Excitation frequency
$L_{EFF}$ = Effective length of neck
$S_B$ = Neck cross-sectional area
V = Chamber volume

Figure 10.9 of Kinsler, et al. (1982), provides a plot of this for a representative resonator. When the excitation frequency, ω, is equal to the resonator's natural frequency, $\omega_N$, obtained from Equation (68), the transmission coefficient goes to zero, as was discussed previously. It should be remembered that since all equations given herein for the Helmholtz resonator neglect viscous losses in the neck, these equations need to be modified to account for those effects for the case of long, narrow necks.

*Volume-Choke-Volume Filters*

From the above discussion, it can easily be imagined that acoustic filters having all kinds of diverse characteristics can be built using various combinations of the three basic acoustic elements. In most systems these elements are in the form of volumes, constrictions, orifices, and enlargements. Many acoustic filters are built via analogy to electrical filters whose performance has already been demonstrated.

A particular combination of the basic acoustic elements is employed in one of the most common types of acoustic filters used in reciprocating equipment, the volume-choke-volume filter, which is also known as a π-type filter. In the context of pump applications, these filters are sometimes also referred to as all liquid filters since they do not require any gas. As is shown in Figure 25, as its name suggests, this filter consists of two bottles (volumes) that are connected by a pipe of relatively small diameter (choke). If properly designed, the filter's components behave as basic lumped acoustic elements—i.e., the bottles behave as acoustic compliances and the choke acts like an acoustic inertia. These lumped characteristics are valid as long as the excitation frequencies are below the open-open resonant frequency of the choke tube and the closed-closed resonant frequencies of the bottles.

Volume-choke-volume filters are basically low-pass filters, similar to those described previously. Typical transmission characteristics as a function of frequency are given in Figure 28, which is based on Tison and Atkins (2001). It is seen that above a certain frequency, known as the cutoff frequency, this filter is highly effective at attenuating all pulsations. At lower frequencies, the filter is nowhere near as effective and at the filter's resonant frequency, known as the Helmholtz frequency (shown at about 8 Hz in the figure), pulsations will actually be amplified. The Helmholtz frequency can be calculated from the following equations from API 618 (1995):

$$f_H = \left(c/(2 \bullet \pi)\right) \bullet \left(\mu/V_1 + \mu/V_2\right)^{1/2} \tag{74}$$

$$\mu = A/\left(L_C + 1.2 \bullet D_C\right) \tag{75}$$

Where:
$f_H$ = Helmholtz frequency (Hz)
c = Acoustic velocity (ft/sec)
$V_1$ = Volume of bottle nearest pump ($ft^3$)
$V_2$ = Volume of bottle remote from pump ($ft^3$)
$\mu$ = Acoustical conductivity (ft)
$L_C$ = Actual length of choke tube (ft)
$D_C$ = Choke tube diameter (ft)
A = Area of choke tube ($ft^2$)

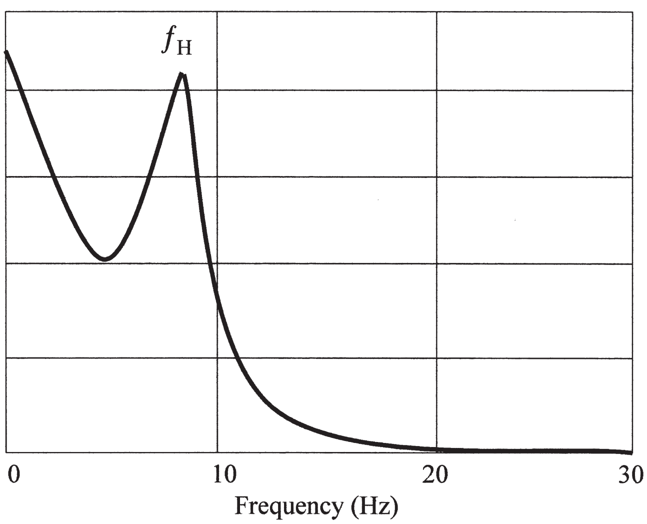


*Figure 28. Typical Frequency Response of Volume-Choke-Volume Filter.*

If the filter's design can be made symmetrical (i.e., both bottles identical and bottle length same as choke tube length), API 618 (1995) states that Equation (74) can be simplified to the following:

$$f_H = \left[c/\left(\sqrt{2} \bullet \pi \bullet L\right)\right] \bullet \left(D_C/D\right) \tag{76}$$

Where:
$f_H$ = Helmholtz frequency (Hz)
c = Acoustic velocity (ft/sec)
L = Acoustic length of bottles and choke (ft)
$D_C$ = Choke tube diameter (ft)
D = Bottle diameter (ft)

Although, in some ways, the symmetrical design is optimum, layout constraints usually preclude its use. It is typical to design the filter so that the Helmholtz frequency is significantly below the lowest pulsation frequency that needs to be attenuated. In general, although API 618 (1995) defines the cutoff frequency as 1.414 times the Helmholtz frequency, the authors consider them to provide adequate attenuation at all frequencies above approximately twice the Helmholtz frequency. It is, therefore, obvious that the lower the Helmholtz frequency is, the greater the capability of the filter. As a point of reference, API 618 (1995) provides the following equation for the preferred Helmholtz frequency for filters used in reciprocating compressor systems:

$$f_H = rpm/85 \tag{77}$$

Where:
$f_H$ = Helmholtz frequency (Hz)
rpm = Compressor speed (rpm)

In comparison, Wachel, et al. (1995), recommend that the Helmholtz frequency be kept below the 1× excitations, which is equivalent to using a value larger than 60 in the denominator of the above equation. However, there are many pump applications where achieving this is impossible and one has to settle for placing the Helmholtz frequency between two harmonics of running speed, usually 1× and 2×. This is not an easy task if the pump has to run over a considerable speed range. If the application is for a multiple cylinder reciprocating pump the authors recommend simply setting the Helmholtz frequency at one-half of the plunger frequency.

In order to reduce the Helmholtz frequency, one must either increase the volumes of the bottles or reduce the diameter of the choke tube. Unfortunately, neither of these alterations come without a price. As was stated previously for surge volumes, increasing volumes of bottles is often in direct conflict with space and cost constraints. Additionally, since the steady-state flow must pass through the volume-choke-volume filter, a reduction in choke diameter means the system must sustain additional pressure drop.

The mechanical analogy for the volume-choke-volume filter is a low stiffness spring (volume) in series with a large mass (choke) and another low stiffness spring (volume). If the excitation source is placed on one side of the system and the components to be protected on the other, it is easy to visualize that at frequencies above the mechanical system's natural frequency, the system isolates the two from each other, with the isolation increasing with frequency due to the inertia of the large mass. Likewise, in the acoustic system, the filter acts to isolate the fluid in the discharge line from the pulsations generated at the pump.

In order to demonstrate the performance of a volume-choke-volume filter, the authors performed an acoustic simulation on the system shown in Figure 29. The filter is placed in the main discharge line between a reciprocating pump and a throttling valve. The system is flowing water at 50 $in^3$/sec and 250 psig discharge pressure. The bottle volumes were chosen to be 100 $in^3$, the tube diameter was 0.40 inch, and the tube length was 24 inches. The system was analyzed both with and without the filter and the frequency responses, in terms of the valve flow fluctuations divided by the pump flow fluctuations, converted to decibels, are plotted in Figure 30.

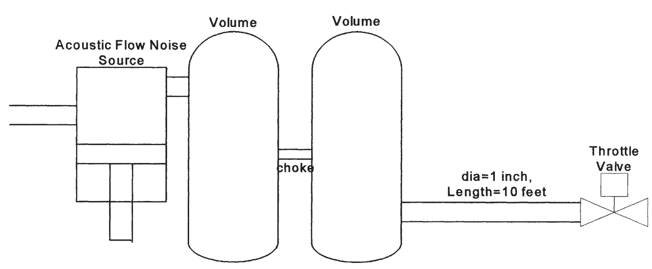


*Figure 29. Schematic of Volume-Choke-Volume Simulation Case.*

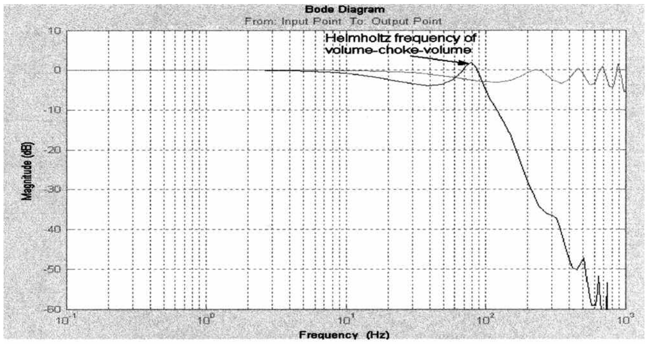


*Figure 30. Volume-Choke-Volume Simulation Results.*

Examination of the figure reveals that the filter performs in the same general fashion described above. The filter is seen to be a true low-pass filter since it attenuates the pulsations extremely well at all frequencies above a certain cutoff frequency (approximately 160 Hz in this case). At low frequencies, the filter is seen to be highly ineffective, being barely better than no filter at all. In the vicinity of the 80 Hz Helmholtz frequency, the resonance condition causes the performance with the filter to be worse than that with no filter at all. However, once the frequencies reach the authors' threshold level of twice the Helmholtz frequency, at 160 Hz, the attenuation is seen to be good and it continues to improve as the frequency increases.

Figure 31, which is patterned after Wachel and Tison (1994), schematically shows several volume-choke-volume arrangements that are employed in the field. The middle arrangement illustrates that a volume-choke-volume filter can be created by simply adding a baffle with an integral choke tube to a surge volume to divide the volume into two separate chambers.

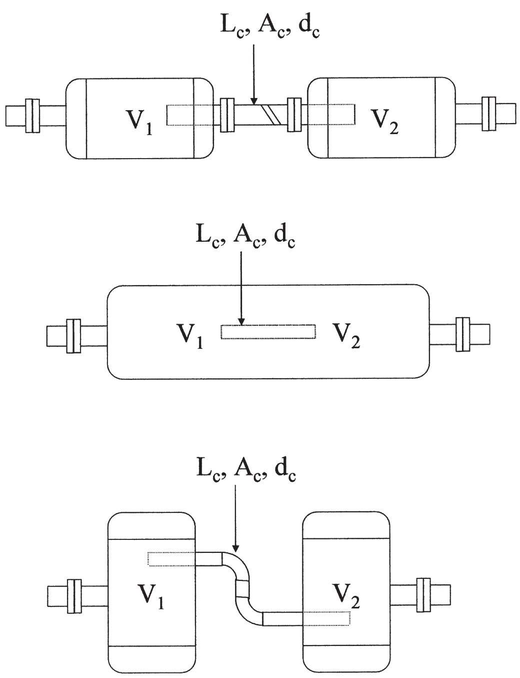


*Figure 31. Possible Volume-Choke-Volume Filter Arrangements.*

In addition to the Helmholtz resonance of the entire filter, the filter's internal components also have resonant frequencies. Accordingly, when designing volume-choke-volume filters, as is the case with all acoustic filters, the dimensions must be selected to avoid acoustic resonances within the filter. Specifically, since the choke tube behaves as an open-open pipe and the bottles act as closed-closed pipes, the lengths of the bottles and choke tube must be selected so as to avoid resonances with the expected excitation frequencies. One of the reasons that the perfectly symmetric arrangement is advantageous is that the bottles and choke tube have the same length in that design. Accordingly, API 618 (1995) specifies that as the preferred arrangement and also recommends that, in the event that a perfectly symmetrical design cannot be employed, the choke tube length should be made equal to one of the bottle lengths, if possible.

While properly designed filter internals (i.e., choke tubes and baffles) provide filtering that can greatly reduce unbalanced shaking forces, the mechanical design of these components must always be robust enough to withstand these forces, themselves. Wachel, et al. (1995), provide some good design guidelines for choke tubes and baffles in their Appendix. Although API 618 (1995) applies to compressors, not pumps, it should be noted that it requires the use of dished baffles, which are more robust, instead of flat baffles.

API 618 (1995) recommends that the diameter of the bottle closest to the pump (referred to as the "cylinder bottle") be greater than or equal to the flow passage diameter at the pump flange. Additionally, API 618 (1995) recommends that the diameter of the bottle remote from the pump (referred to as the "filter bottle") be greater than or equal to three times the line piping diameter. If two separate bottles are used to create a volume-choke-volume filter, Wachel and Tison (1996) recommend that the external portion of the choke tube between the two bottles be made straight in order to minimize shaking forces.

Although volume-choke-volume filters are frequently used in reciprocating compressor applications, their complexity is often not needed in pump applications. Instead, many pumps use a derivative of this called a volume-choke all liquid filter, as is shown in Figure 32. This filter is seen to employ a single bottle (volume) near the pump flange, followed by a choke tube that may be either internal or external to the bottle. The choke tube connects directly to the main suction or discharge line through a reducer. This device does not result in as sharp a cutoff of higher frequency pulsations as does the volume-choke-volume filter but it is effective in pumps since the allowable pressure drops in reciprocating pumping systems are almost always higher than those in reciprocating compressor applications. This allows pumps to reap the additional filtering benefits provided by the smaller choke tubes that can be employed and permits the elimination of the second bottle.

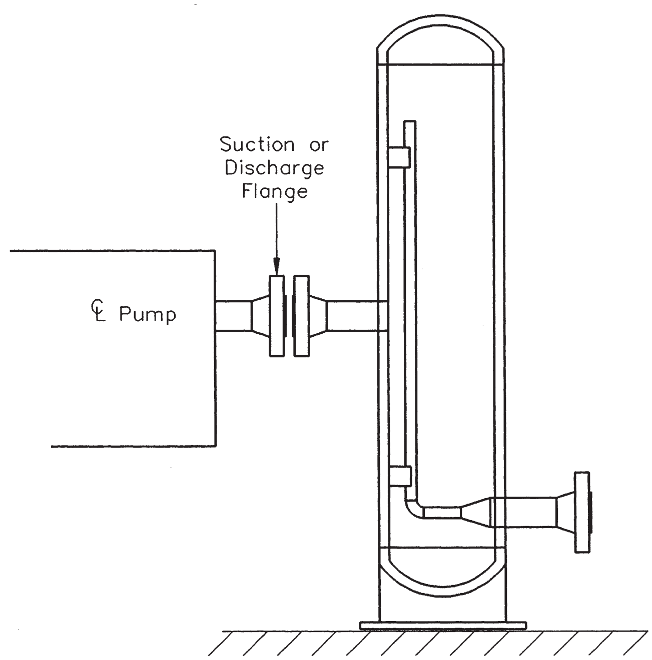


*Figure 32. Volume-Choke All Liquid Filter. (Courtesy of Wachel and Tison, 1994, Turbomachinery Laboratory)*

In addition to the obvious cost and space benefits, elimination of the second bottle brings a couple of other changes. First, as stated earlier, deleting this bottle reduces the attenuation of the filter, a deficiency that is made up for by the employment of smaller choke diameters in pumps. Second, removing the second bottle reduces the Helmholtz frequency, which is an added benefit.

In the opinion of most experts, acoustic filters provide better pulsation protection than accumulators and surge volumes. Whereas, the effectiveness of the former is limited to low frequencies, the latter's ability to handle all frequencies above a certain cutoff value means that they cover a much greater frequency range than do the former. This greater range means that acoustic filters will normally be effective over relatively large pump speed ranges. Unlike accumulators, their performance is also relatively insensitive to changes in line pressure since the acoustic velocities of most liquids are virtually independent of system pressure. Additionally, unlike accumulators, since they do not use any gas, they require almost zero maintenance once they have been installed. On the down side, acoustic filters are normally much larger than accumulators due to the necessity of minimizing their pressure drop. Additionally, they are also usually more expensive.

Of course, as Chilton and Handley (1952) point out, the above advantages of acoustic filters over accumulators assume that the main objective is to attenuate the pulsations within the lines, not those within the pump itself. In general, accumulators do a better job of attenuating the pulsations within the pump internals. Chilton and Handley (1952) illustrate this by comparing the performance of a single surge volume with that of an equivalent size volume-choke-volume filter in a reciprocating compressor application. They found that although the filter provided about twice as much attenuation of the pulsations in the piping, it also resulted in pressure fluctuations in the compressor that were more than twice as large as those of the surge volume.

*Dissipative Devices*

Dissipative devices (also known as resistive devices) operate by introducing damping into the system, similar to the function of a viscous dashpot in a mechanical system. This is accomplished by forcing the flow through small openings, thereby generating substantial pressure drop. The most frequently employed device of this type is the orifice plate.

Of the three major types of pulsation control devices, orifices are, by far, the cheapest and the one most amenable to "quick fix" solutions. If a pulsation problem is uncovered in the field, it is relatively easy to add an orifice to the system, compared to adding an accumulator or acoustic filter. Of course, the likelihood that simply placing an orifice in the most available location will solve the problem is not terribly high. As with accumulators and acoustic filters, the sizing and placements of orifices need to be guided through an accurate acoustic simulation of the entire system.

Like energy absorbing devices and acoustic filters, the effectiveness of dissipative devices is frequency-dependent. Normally, their performance is better at high frequencies. Additionally, one of the most important things about using orifices is that, in order for them to be effective, they must be placed at or near the location of a velocity antinode in the mode shape of the mode to be attenuated. If, instead, the orifice is located at a velocity node, it will have no impact on that particular mode. Since orifices are normally installed at the pump flange and since a reciprocating pump normally acts as a closed end (i.e., a velocity node), their effectiveness is usually quite limited.

In general, a good starting point for sizing an orifice is to make the orifice diameter equal to about one-half of the pipe diameter. In fact, in order to provide any appreciable damping to the system, the orifice diameter should not be any larger than one-half of the pipe diameter. Thus, its best to start at the maximum size and work downward's, in the interest of minimizing steady-state pressure drop.

In addition to adding damping to a system, orifices can also change the natural frequencies. For instance, if an orifice is placed at a velocity antinode (open end), it can change the end condition from open to partially closed.

Per Vetter and Seidl (1993), in addition to providing damping by viscous dissipation, orifices can also provide an isolating effect since, since they represent a change in impedance, they generate reflections.

Some engineers believe that the sizing of orifices is not as critical to successful operation as the sizing of other pulsation control devices. Although orifices undoubtedly lend themselves better to trial and error in the field than other devices, a case study reported by Lewis, et al. (1997), showed the impact the size of an orifice can have on system acoustic behavior. In this study, they were adding an orifice to the open end (i.e., velocity maximum) of a pipe suffering from a quarter-wave resonance and they found:

- With an orifice sized to yield 2 psi pressure drop, pulsations were reduced from 50 to 14 psi.
- With a 10 psi pressure drop orifice, pulsations were completely damped out.
- With a 20 psi pressure drop orifice, the system changed to a closed-closed pipe and half-wave resonances came into play.

*Practical Considerations*

When pulsation dampeners are employed in a system, the question always arises as to what level of attenuation is sufficient to avoid problems. Although nothing substitutes for the performance of a rigorous acoustic analysis, coupled with analyses of potential problems, experience has proven that there are some general pulsation levels that can be aimed for. For instance, Chilton and Handley (1952) give a rule of thumb limit of 2 percent of line pressure. They elaborate on this by stating that figure should be decreased somewhat for high pressure lines and can probably be increased for suction lines. Similarly, for a reciprocating compressor installation, Grover (1966) recommends a limit of 2 percent of line pressure for the peak-to-peak pulsations in the line and 7 percent of line pressure for the peak-to-peak pulsations at the compressor. Grover (1966) continues by noting that certain parties have advocated reducing the 2 percent figure to ½ percent for high pressure systems and increasing that figure to 5 percent for very low pressure designs.

Parry (1986) and Wachel, et al. (1995), both state that pulsation dampener manufacturers normally size these devices based on pump speed, flowrate, and operating pressure levels. Since they do not normally consider the pulsation frequencies that need to be controlled, this method is often inadequate. In addition to the size of the dampener, its location in the system is critical. Many references report on cases where a properly sized dampener placed in the wrong location actually acted to increase pulsations, not attenuate them. In general, in reciprocating pump applications, the dampeners should be located as close to the plungers as possible—thus, they are normally attached to the pump inlet and discharge flanges. Regardless, in order to determine the proper size and location for pulsation dampeners, a thorough pulsation analysis must be performed.

Miller (1988) states that in multiple reciprocating pump installations, it is almost mandatory to employ properly designed pulsation control devices in both the suction and discharge systems for each and every pump. Even though multiple pumps are usually designed to run at slightly different speeds, it is almost impossible to prevent them from occasionally reaching a condition where all of the individual pumps' excitations act in-phase with one another and are, thereby, directly additive. The amount of pulsation energy and resulting pipe vibrations are then increased by a factor equal to the number of pumps compared to their typical values.

When variable speed pumps are operated at speeds other than their design speed, the effectiveness of most pulsation control equipment is degraded. This is because, as stated earlier, the dimensions of such devices are selected to provide optimum attenuation at the normal excitation frequencies and wavelengths. Since changing pump speed results in alteration of excitation frequencies under most circumstances, the performance is degraded. Lovelady and Bielskus (1999) suggest that the performance of acoustic filters at off-design conditions can be improved by changing the end conditions of choke tubes via tapering or perforations but such practices usually hurt performance at the design speed.

Miller (1988) recommends that two or more accumulators (having a total gas volume equal to that required for a single accumulator) be considered for the discharge piping in long pipeline service for two reasons. First, since pipeline system pressure is brought up to operating levels quite gradually (over a period of several minutes to an hour), there is a time where the discharge pressure is lower than the precharge pressure that would be employed in a lone accumulator, thereby leaving the system without pulsation control. Employment of two accumulators with different precharge pressures would alleviate that situation. Second, it is often necessary to pump alternate batches of liquids with greatly differing viscosities. Since the pipe frictional losses for the two liquids will be greatly different, their discharge pressure levels will also be different. The advantages of using dual accumulators having different precharge pressures are apparent.

Singh and Chaplis (1990) report that due to the differences in various manufacturers' products, they have occasionally solved field pulsation problems by simply switching the pulsation dampener to an equivalent device from a different manufacturer.

In addition to money and size, the other cost almost always associated with employing pulsation control elements is that they introduce an additional parasitic pressure loss to the system (this is particularly true for flow-through devices). Although the allowable pressure loss is dependent on the system in question, API 618 (1995) gives a good ballpark number of 0.25 percent of the line pressure at the location of the element.

Of course, adding an accumulator or filter to a system is not always 100 percent beneficial from an acoustics standpoint, either—the accumulator or filter can introduce pulsation problems of its own. One common problem reported by both Blodgett (1998) and Wachel and Price (1988) occurs when an acoustic filter or flow-through accumulator is placed in the vicinity of a pump's suction or discharge flange (which is where they are located the great majority of the time). This has the effect of introducing a quarter-wave mode between the closed end of the pump manifold and the entrance to the pulsation control element. This mode, which Wachel and Price (1988) state normally has a resonant frequency somewhere between 50 and 300 Hz is sometimes referred to as an acceleration head mode. Per Wachel and Price (1988), this mode, which is normally excited by higher plunger harmonics, is responsible for the majority of cavitation problems occurring in suction systems. In order to avoid problems, the resonant frequency should be made as high as possible by locating the element as close to the pump flange as possible. This mode can also sometimes be controlled by installing an orifice at the pump flange or accumulator entrance.

Another potential problem, courtesy of Wachel and Price (1988) is that, in systems having multiple pumps, the units can interact with each other to create severe problems if the gas charges in the various accumulators are not identical. Additionally, if only one of the accumulators loses its gas charge, the pulsations in all units can be affected.

Although the preceding treatment is hardly exhaustive, it is hoped that it provides the user with a flavor for the types of pulsation control devices that are available. Probably the most important concepts are that, regardless of what type of device is used, it should be located as close to the excitation source (pump or valve) as possible. Additionally, the selection and sizing of a pulsation control device needs to be guided by a thorough system pulsation analysis. In the words of Parry (1986), "Selection of required pulsation dampeners should not be done until after the acoustic analysis has been completed. The cost of this additional analysis will pay for itself in increased production and reliability of the piping system."

## ACOUSTIC SIMULATIONS

Since throughout a good portion of this tutorial, the authors have continually emphasized the need for a good, thorough acoustic simulation, this tutorial would not be complete without at least a mention of that subject. However, since the authors recognize that most pump users are not likely to ever need to run a pulsation analysis of their own, this section is purposely limited to a short overview.

There are two basic types of computer simulations for acoustic problems, analog and digital. Naturally, the earliest simulations, which began in the reciprocating compressor industry in the 1950s, were all analog. Analog simulations rely entirely on the breaking down of an acoustic system into its lumped approximation. Although analog computer simulations can be done for the actual acoustic system, the electrical analogies discussed previously are often used to convert the acoustic system into an analogous electrical network. The network is then analyzed on the analog computer to determine the electrical system's currents and voltages as functions of time and the results are then converted to their acoustic equivalents, flows and pressures.

Naturally, the advent of the high speed digital computer has made digital simulations more popular and analog simulations relatively obscure. In digital simulations, the governing equations of fluid mechanics, the continuity equation, the Navier-Stokes equations, and the thermodynamic equation of state, are directly solved for the pressures and flows as a function of time. Since digital simulations are fully capable of handling distributed parameter systems, the limitations of lumped approximations that are inherent in analog simulations are not an issue.

Although there are many different methods used in digital acoustic simulation codes (finite volume, finite element, transfer matrix or four-pole, and method of characteristics, just to name a few), all of them are faced with the same task—the solution of the two basic equations from fluid mechanics, the momentum equation and the continuity equation. Normally, this is accomplished by combining these two equations to obtain the one-dimensional wave equation and then generating a numerical solution for that.

The authors' acoustic simulation code employs a finite volume scheme (similar to those used in CFD codes) to model the transient or steady-state behavior of pump piping systems having any number and arrangement of pumps, valves, pulsation control elements, and piping. The one-dimensional wave equation is solved using a staggered grid of pressure and flow nodes.

Regardless of the type of code employed, there are three basic types of acoustic analyses that are normally performed—passive, active, and transient. A passive analysis is used to determine the system's acoustic natural frequencies and associated mode shapes. It is directly analogous to an undamped natural frequency analysis in the field of mechanical engineering. Likewise, an active analysis is analogous to a steady-state response analysis in the field of mechanical vibrations. A continuous excitation, such as that provided by a reciprocating pump, is applied to the system and the system's steady-state pulsations are computed. In an active analysis, transient effects are assumed to quickly die out and are not considered. A transient analysis models the behavior of the system when it is exposed to a transient stimulus, such as the closing of a valve. All water hammer problems require employment of a transient analysis.

Similar to a picture being worth a thousand words, a sample analysis is probably more illuminating than the world's greatest description of one. Since the typical networks that the authors analyze are far too complex to describe in the space available, the authors have invented the relatively simple, fictional case of Figure 33 to illustrate a typical analysis of a hydraulic transient or water hammer problem. The system is seen to consist of a triplex reciprocating pump that is augmented by a charging boost pump of centrifugal design. Both the suction and discharge of the triplex pump are protected by surge volumes that span the three cylinders. The pump provides flow to a branched line. Each branch contains a throttling valve that permits adjustment of the flow split between the two branches. Valve number 1 is a simple throttling valve that is positioned mechanically while valve number 2 is an electromechanical proportional valve.

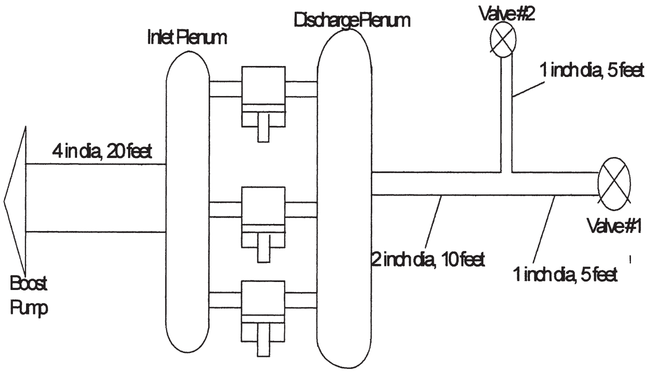


*Figure 33. Schematic for Transient Simulation Example.*

At the time the transient begins, the valve areas are both wide open such that the 200 gpm of water being output by the pump is split evenly between the two lines, 100 gpm in each. The transient consists of an abrupt failure in the power supply to valve number 2 at a time of 0.50 seconds, followed by an abrupt recovery 0.04 seconds later. Since the proportional valve is designed to fail closed, the power failure and recovery result in the valve closing and then reopening. Upon power failure, the valve's flow area does not close off instantaneously due to the mechanical response time of the valve. Instead, the power failure and subsequent recovery cause the flow area to vary in accordance with the top trace of Figure 34.

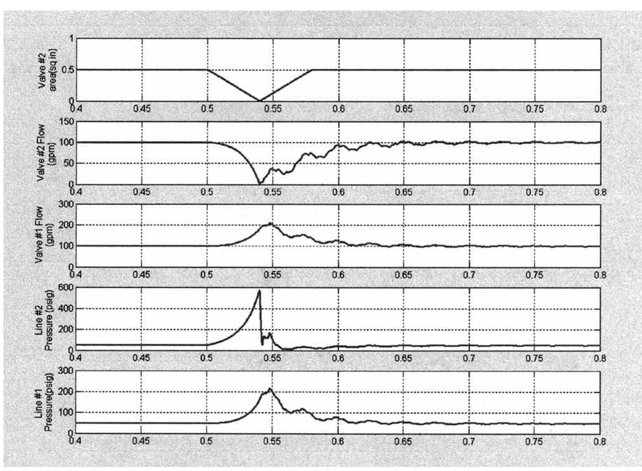

*Figure 34. Transient Simulation Example Results.*

The results of the acoustic simulation are provided in the time traces of Figure 34. Inspection of these traces reveals that the flow through valve 2 drops to zero as it closes, causing the flow through valve 1 to increase to 200 gpm. The sharp pressure spike corresponding to water hammer is clearly seen in the line 2 pressure trace. Additionally, line 1 also experiences a milder spike due to its valve being abruptly forced to handle twice its initial flow.

It is hoped that this simple example provides the user with a flavor for what goes into an acoustic analysis. Wachel, et al. (1995), provide several other good examples of such analyses for readers interested in seeing more.

All of this discussion of the value of pulsation analyses is probably leading the user to question if such analyses are needed in the design phase of every single pump and, if not, which applications can they be omitted on. Although the answer to the first question is no, the dividing line between which pumping applications should be analyzed up front and which ones do not need to be is in a highly gray area. In a lot of cases, it is a simple judgment call based on how much risk the user is willing to take versus how much up-front cost they are willing to incur. The authors believe in erring on the side of caution since they are familiar with many cases where the decision to forgo these analyses in the design phase proved to be in the "penny wise, pound foolish" category. In any event, the authors agree with Singh and Chaplis (1990) that, in the very least, acoustic simulation in the design phase is imperative for critical pump applications, high energy pumps, and configurations employing multiple reciprocating pumps.

## CONCLUSION

A relatively comprehensive look at the field of pressure pulsations in pumping applications has been provided. Although a great deal of material has been covered, the key points that the user should take away from this tutorial are as follows:

- Pressure pulsations are sources of significant problems in many pumping applications.
- A large percentage of pulsation problems are associated with acoustic resonance. In most situations, this condition needs to be avoided.
- Systems employing reciprocating pumps, especially if there is more than one, need to be handled with care.
- Although reciprocating pumps have a well-deserved notorious reputation for causing pulsation problems, they are not the only pump types that are culpable. Pulsation problems can also often occur in centrifugal, gear, vane, and other pump types.
- In many applications, a thorough pulsation analysis should be performed during the design phase.
- Most field pulsation problems are preventable.
- Pulsation control components (acoustic filters, accumulators, orifices, etc.) should not be selected or sized without the guidance of a good acoustic analysis.
- Pulsation analysis and the design of pulsation dampeners are not exact sciences that can be performed by just anybody. The importance of the skill, judgment, and experience of the engineer involved should never be underestimated. In the words of Blodgett (1998), "It is also important to understand that the value and quality of a design is more dependent on the quality of the engineer who makes design decisions than on the analytical tools."

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