# RECIPROCATING PUMP DYNAMIC CONCEPTS FOR IMPROVED PUMP OPERATIONS

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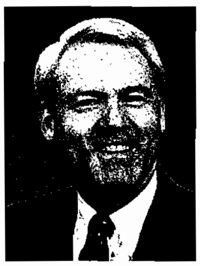

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## ABSTRACT

In order to ensure safe, reliable, efficient and cost effective operation of reciprocating pumps, pump dynamics should not be ignored. Pump problems due to dynamics are often difficult to diagnose and solve. Dynamic problems usually lie in one or more of the following areas:

- Pulsation control (piping acoustics)
- Vibration control (piping mechanics)
- Cavitation control (pulsation and pressure drop)
- Pump valve dynamics (valve motion)

All the areas can be interactive to such an extent that it is difficult to know where the most effective changes can be made. A clear summary of each area will be presented with a view toward understanding how these areas can interact. The topic will be approached from an engineering perception viewpoint (a spectral energy response overlay) with minimal emphasis on mathematical concepts.

Following the energy as it transfers though the whole system reveals how inefficient energy transfer and local energy magnification (resonance) is key to understanding how to prevent or solve everyday problems. Problems such as vibration, bladder failure, valve failure, poor performance, fatigue failure, and safety concerns are usually linked to one or more dynamic areas.

Along with a basic understanding of the physics involved in pump dynamic areas, it is also helpful to have a working knowledge of "rule of thumb" techniques that can be reliable and extremely cost effective.

Pulsation control will be viewed using mechanical analogies with cause and effect scenarios. Vibration control will be focused on reducing the shaking force through reducing acoustic-to-mechanical coupling mechanisms, instead of brute force mechanical modifications. Cavitation and NPSHR (the inlet pressure required to prevent cavitation) problems will be viewed from a simple understanding of local instantaneous pressure compared to vapor pressure. Valve dynamics will be viewed in terms of simple fluid pressure forces, as they are exerted on a pressure controlled valve element. This engineering perception has a firm theoretical foundation but requires only fundamental mathematics to employ.

Two short case histories will be presented illustrating pulsation/vibration control, cavitation control and pump valve problems. In almost all cases, piping vibration is the result of pulsation coupling into mechanical shaking forces yielding vibration with possible fatigue failures. Pump manifolds and external piping can be modified to ensure nonresonant acoustic systems that greatly reduce vibrational force. Once the shaking force is reduced, it remains to ensure that mechanical resonance is not present.

Although cavitation is generally thought to be pressure drop related, in many cases it can be traced to acoustical resonances associated with the pump manifold and connected piping. Pump valve problems are generally solved through increasing spring stiffness, decreasing lift and employing more rugged materials. The use of newly developed materials has had the tendency of appearing as a universal fix where high pulsation levels can be tolerated. The best solution for many valve problems is reducing the pulsations that induce excessive impact forces.

The use of a spectral energy response overlay or window is a concept that is in use by many who deal with dynamic problems. This type of approach is easily understood, but not as rigorous as the actual model approach. It should become apparent that most design quality can be determined by the overlay approach. It is not intended to replace the full model, but it will produce a good initial starting design and form the basis for analyzing existing field problems.

## INTRODUCTION

This presentation of ideas is very broad from the perspective that several complicated and involved issues will be discussed. It should be understood that many of the areas of interest, if covered fully, would require many times the effort. The intention here is to briefly cover a few important aspects of dynamic pump operation so that the reader can then pursue, in more detail, specific areas of individual interest.

Reciprocating pumps produce flow that is characterized by the motion of the moving elements (plungers, diaphragms, etc.). Although the fluid pumping elements act independently, there is very little phase shift between the elements at the frequencies of interest. Therefore, the time for the pressure/flow to propagate in the inlet or outlet of each valve passage is usually very small and the individual element flows add vectorially producing a composite dynamic flow. Figure 1 illustrates the additive effects of a three element reciprocating pump. The element motions will, in most cases, be simple sinusoidal dependent on the L/r (connecting rod length/crank radius) and crankshaft offset.

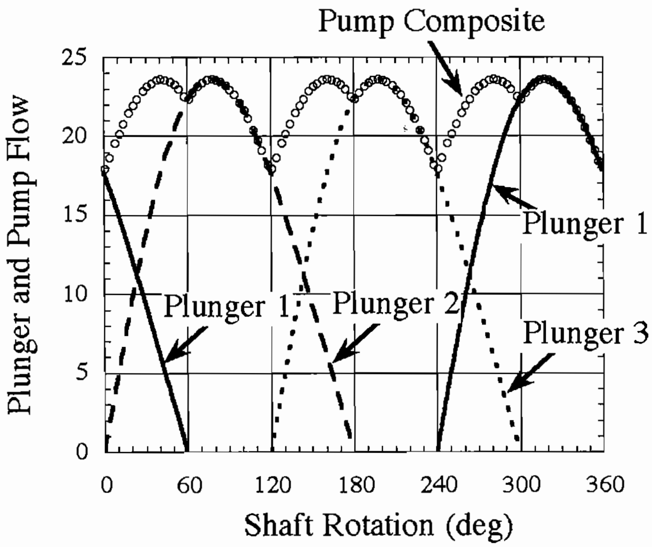


*Figure 1. Flow Rate From Each Plunger and Pump Composite.*

The significance of pulsation in pump systems is generally understated, due to a lack of information or a lack of understanding. It is important to view pulsation as dynamic energy in two forms, pressure and velocity. The pressure form of energy can be thought of as potential energy and the dynamic flow velocity can be viewed as kinetic energy. In fact, as the energy propagates through the piping system in the form of resonant pulsation, the energy is constantly being transferred from one form to another. An analogy (mechanical) to the energy conversion is a simple pendulum. As the pendulum oscillates at its natural frequency, the maximum height corresponds to maximum potential energy and the minimum height corresponds to maximum kinetic energy (velocity).

Since the system tends to see the composite energy, it is instructive to perform a spectrum analysis of the composite time wave. The results of such an analysis are shown in Figure 2. The bars represent specific multiples of the pump rpm. In this case, the pump rpm is 300, therefore the base frequency is 5 Hz. A three element pump will produce multiples of three, therefore, the only significant pulsation frequencies are 15 Hz, 30 Hz, 45 Hz, 60 Hz, etc. It is important to keep in mind that the amplitudes are dynamic flow, which can be used as the energy input of the pump into the manifold and piping system. This spectrum of the flow forcing function will be applied (actually multiplied by the transfer impedance) to the transfer impedance of the piping system to illustrate a simple method of analysis. In other words, the dynamic flow excitation is multiplied by dynamic pressure per dynamic flow to obtain dynamic pressure (or pulsation pressure) at the desired point. It is also instructive to recognize that in the simple dynamic flow composite, almost all the dynamic energy is contained in the 3×, 6×, 9×, and 12× components. If it were not for piping acoustics, these four components might be the only ones that need be addressed.

## PIPING ACOUSTICS

When a dynamic flow forcing function is applied to a liquid filled pipe, a time wave is transmitted down the pipe and the wave fluid interaction is defined by the wave equation. In the case of common pump piping, the frequency, propagation velocity, and

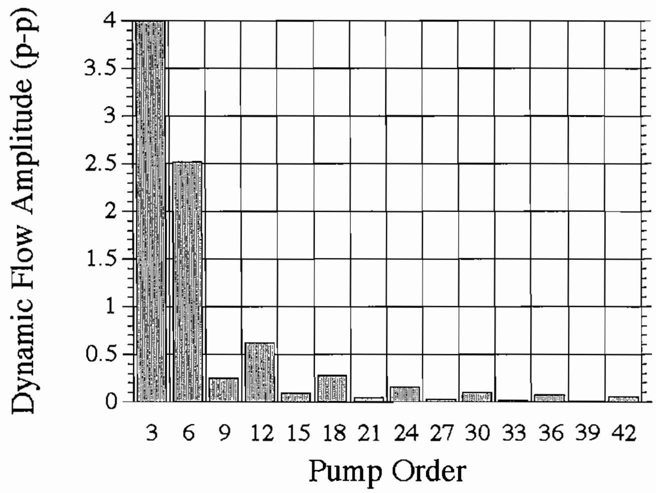


*Figure 2. Frequency Spectrum of Composite Pump Flow.*

pipe dimensions limit the type of wave to a plane wave. This simplifies the analysis considerably allowing fairly simple acoustic elements to be the object of our analysis. In the case of pumps, there are three acoustic elements that describe the basic combinations. These three elements are the quarter wave element, the half wave element and the simple volume element. Each of the wave elements has acoustic properties such as natural frequency, resonant mode shape, and energy transmission (transfer impedance).

### *Half Wave Acoustic Element*

The half wave acoustic element is simply a section of pipe that has the same end condition on both ends. Both ends are acoustically soft or hard, referring to their relative acoustic impedance. Simply stated, a soft (open) end is where the pipe flow area suddenly increases cross section by at least four times. This is the same as saying that the diameter changes by a factor of two. In like manner, an acoustical hard end is one where the area decreases by a factor of two or more. The natural frequency of a half wave section of pipe is:

$$f = n \frac{c}{2L} \tag{1}$$

Where $f$ is the natural frequency of the $n^{th}$ mode and $n$ has the values 1, 2, 3.... The velocity of sound is $c$ and the length of the element is $L$. A typical response of a half wave pipe element is shown in Figure 3. The ordinate is displayed in logarithmic scale to reveal the character of the pressure versus frequency profile. The frequency sensitive acoustic damping is deliberately not shown, so that the various modes respond equally. The scale and damping are intended to increase the clarity of content of the response.

### *Quarter Wave Acoustic Element*

The quarter wave is simply a section of pipe having opposite end conditions. One end of the element is soft (or open) and the other is hard (or closed). The simplified end condition criteria is the same as that stated previously. The natural frequencies of the quarter wave element are:

$$f = n \frac{c}{4L} \tag{2}$$

Where $f$ is the natural frequency of the $n^{th}$ mode and $n$ has the values 1, 3, 5.... The velocity of sound is $c$ and the length of the element is $L$. A typical response of a half wave pipe element is shown in Figure 4.

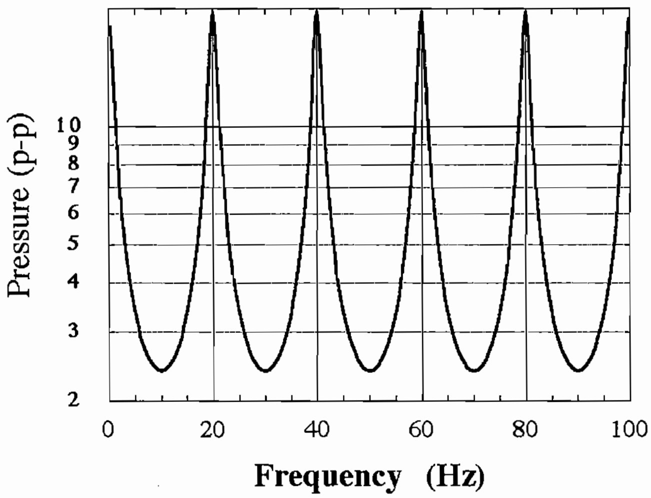


*Figure 3. Spectral Response of Pipe Acoustic Half Wave.*

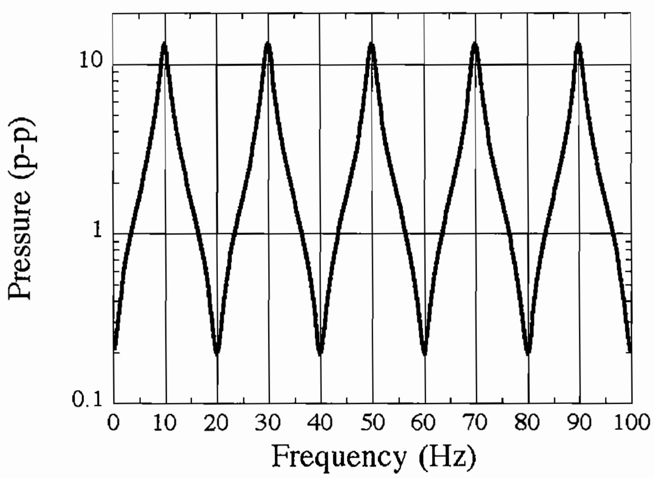


*Figure 4. Spectral Response of Pipe Acoustic Quarter Wave.*

*The Simple Volume Element*

The concept of a simple volume is different in contrast to the production of standing waves in previous elements. Its property is simply acoustic volume and its reaction to flow excitation is to convert the dynamic flow back into pressure energy. The volume can best be considered as a flow smoothing device. It also has relative properties and has been compared with several other volume parameters. Acoustic volume can be compared with the peak-to-peak composite flow, the total pump displacement or the volume of fluid between the valves and the volume element. When the acoustic volume is large (greater than five times, for example) in the comparison, the volume is considered to be large enough to be effective. Figure 5 illustrates the basic frequency response of a volume to velocity excitation. In Figure 6, the comparative volume could be the volume of the manifold plus the volume of each valve passage to the manifold.

It is now possible to view liquid piping systems as coupled elements (energy conveying) of acoustic length elements and simple volume elements. To serve as an example, the following simple piping system (Figure 6) will be analyzed to illustrate the principle of analysis. First, a few observations should be noted in order to get the proper perspective of the elements of the system. The passages from the valves to the manifold are usually too short to actually form quarter wave elements. Instead, they usually contribute a small volume pocket effect to the manifold that does form a quarter wave element. The volume is a simple volume. The

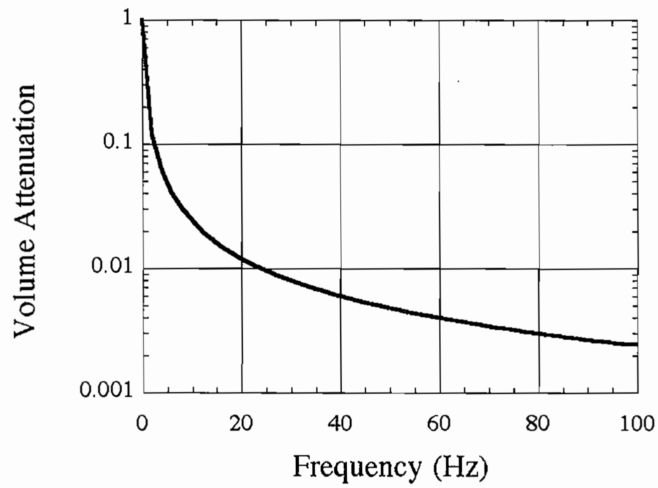


*Figure 5. Transfer Impedance of Simple Volume.*

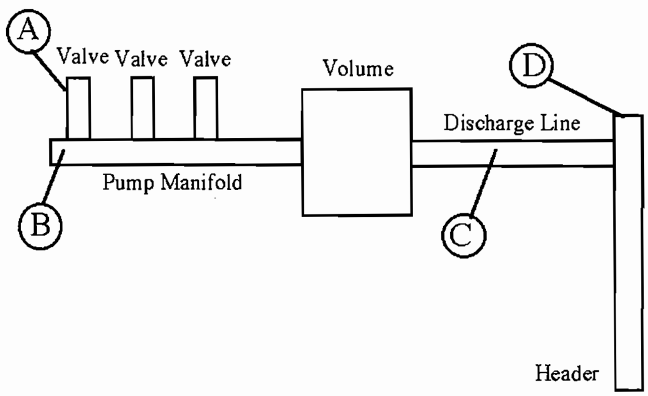


*Figure 6. Example Outlet Piping System.*

discharge line is a half wave element and the header is closed on the end with the test point D, and is nonreflective on the opposite end.

If the dynamic flow of the composite pump is placed at the valves of the pump, the pressure resulting from that dynamic flow can be determined by several different analytical methods (Blodgett, 1991, 1996). The transfer matrix will be used in these examples, but other methods could be used just as effectively to demonstrate the basic analysis process. Figure 7 illustrates the pulsation produced in the pump manifold at test point A with the imposed dynamic flow. The dominant acoustical response is the quarter wave response of the pump manifold. This type of response will be present when the pump employs a volume accumulator device on the inlet or outlet of the pump. In this particular case, the outlet is being studied.

Figure 8 is the transfer impedance of the pump manifold closed end and the composite dynamic flow. The most significant observation is the increase of the manifold response amplitude. The closed end of the manifold will always exhibit the maximum pulsation in relation to the quarter wave response, due to the standing wave or mode shape of the natural frequency. This is particularly significant when the instantaneous pressure is required to evaluate a system for sufficient NPSH (adequate inlet pressure to prevent cavitation). It is very common for the closed end of the manifold header to exhibit significant cavitation if the manifold acoustic response is excited.

Figure 9 illustrates the transfer impedance of the piping system at the midpoint of the discharge line (point C). The manifold response continues to propagate through the volume and continues to be the dominate response. However, the half wave response is also starting to become part of the response envelope.

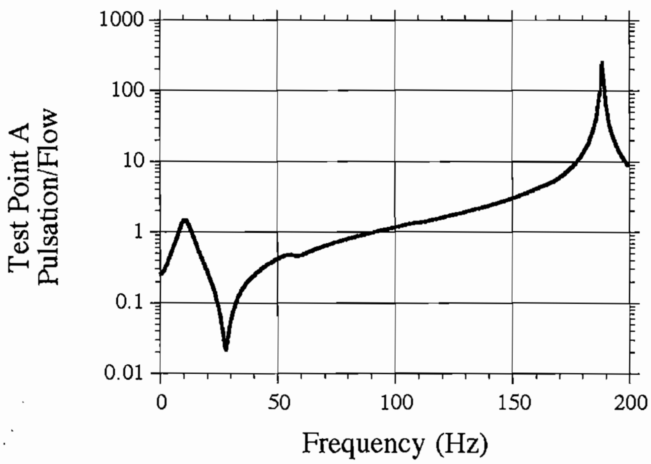


*Figure 7. Driving Point Impedance At Point A.*

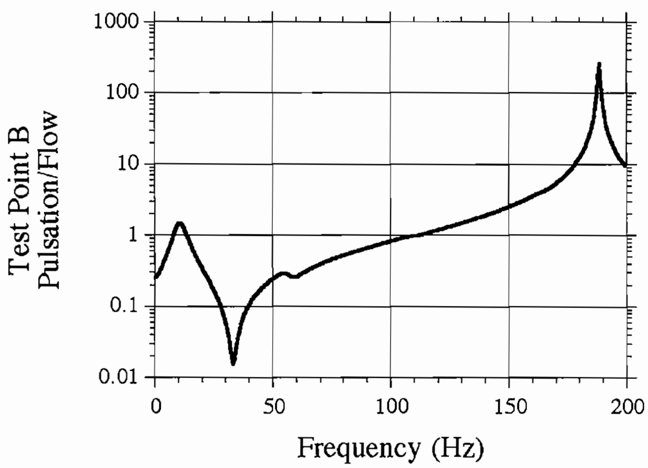


*Figure 8. Transfer Impedance At Point B.*

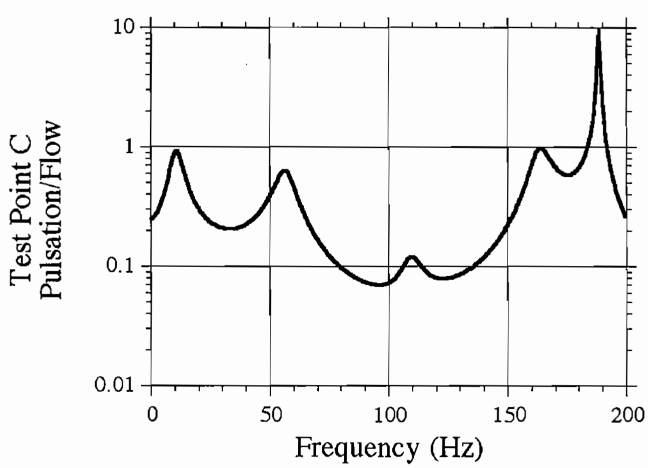


*Figure 9. Transfer Impedance At Point C.*

Figure 10 illustrates the transfer response at the closed end of the header. This response is composed of three elemental responses as noted in the figure. The quarter wave of the manifold, the half wave of the discharge line, and a response that is characterized by the "lumped response" composed of the volume element, the discharge line, and the volume character of the header. This type of acoustic response can be understood by visualizing a mechanical mass suspended between two springs. The mass will oscillate at a frequency dependent upon its mass and the composite stiffness of the two springs. In the case of acoustics, volumes are springs and restrictive elements such as a tube or line is a plug or mass of fluid. Therefore, the first volume is the volume element, the discharge line is the mass, and the second volume is the header element. This response has actually appeared in all of the response figures, but it appears most clearly in Figure 10.

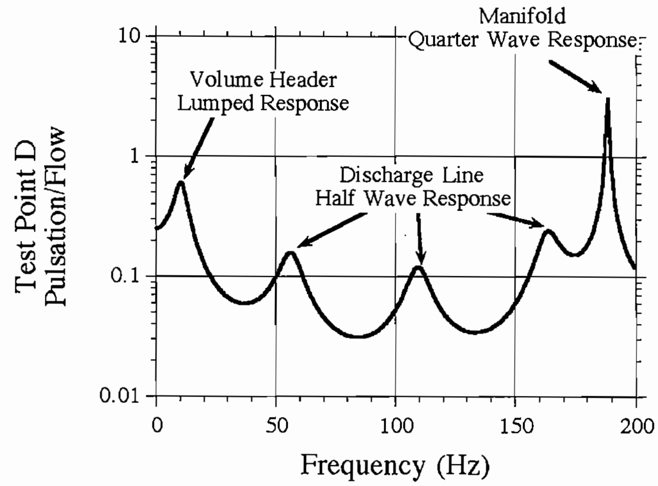


*Figure 10. Transfer Impedance At Point D.*

Figure 11 illustrates the idea of overlaying the spectral energy envelope and the excitation energy. This allows easy comparison of the frequencies of excitation with the natural frequencies to determine if the excitation energy is frequency coincident. When coincidence occurs, the energy is concentrated and magnified in the resonant element. When the excitation energy is not coincident with a natural response peak, the excitation wave travels through the element without magnification and exemplifies a simple nonreflecting or traveling wave. In systems where the dynamic pressures are not magnified by resonance, the pulsation is normally referred to as "residual or forced pulsation."

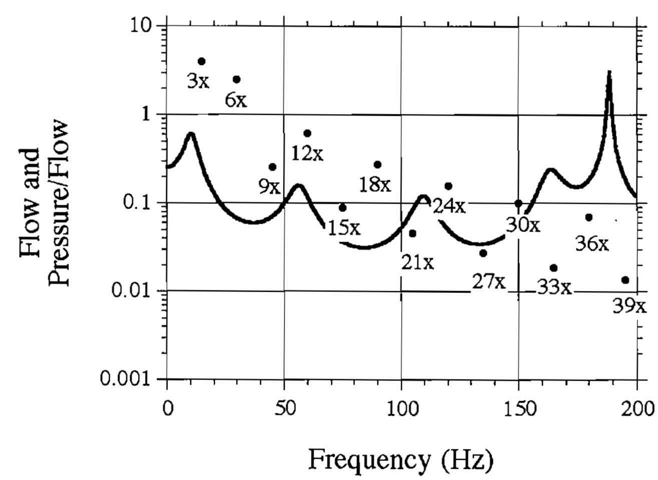


*Figure 11. Transfer Impedance At Point D with Composite Flow Overlay.*

Upon analysis of Figure 11, there are three flow excitation component frequencies that indicate proximity to natural response frequencies and, therefore, are expected to magnify. The near coincidences occur at 3×, 12×, and 24×. The results of the multiplication are in Figure 12 and, as expected, the flow excitation components near the acoustic response frequencies are significantly higher in the distribution than the original flow components. An ideal case would place the excitation components in the "valleys" of the response envelope. This can actually be done where sufficient control of element length and diameter is available. However, practical limitations of space or economics prohibit designing the ideal system. With the additional complication of variable speed pumps, the excitation energy at each of the component starts to cover a range of frequencies instead of being a single frequency. It is evident that placing all the energy in the "valleys" would become increasingly difficult. Therefore, a truism of engineering starts to become evident, "ideal systems cannot be developed due to limitations, instead a system must be optimized in view of all the limitations you must accommodate."

Another phrase that you may have heard suggests the same idea, "you must do the best you can with what you have." This actually makes engineering design a mixture of art and science.

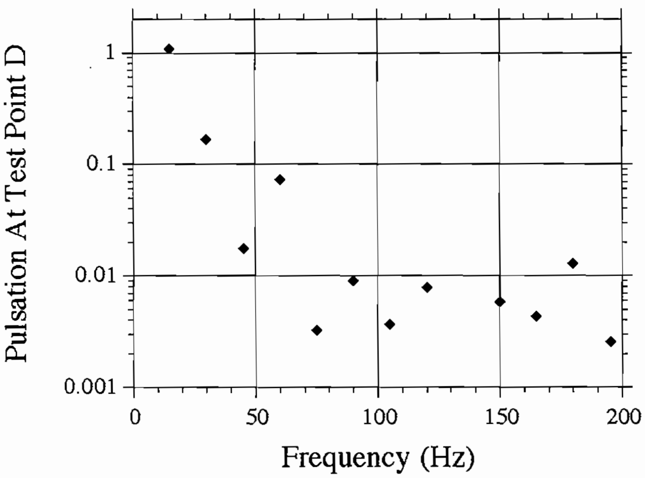


*Figure 12. Resultant Pulsation Levels.*

## MECHANICAL DYNAMICS

Just as acoustical resonance magnifies the dynamic pressure and flow, mechanical resonance will magnify piping vibration and dynamic stresses in the piping system. It is important to ensure that the mechanical natural frequencies are not coincident with significant pulsation-induced shaking forces in the piping. Pulsation energy couples or transfers into the mechanical system due to pressure coupling or momentum changes. Typically these coupling points are reflection points such as closed ends or piping bends. Mechanical natural frequencies can be calculated for simple piping configurations. The basic equation is:

$$f = \frac{\mu}{2\pi L^2}\sqrt{\frac{gEI}{w}} \qquad (3)$$

Where:

$f$ = mechanical natural frequency, Hz
$\mu$ = frequency factor
$\pi$ = 3.14
$L$ = length of span, in
$I$ = moment of inertia, $in^4$
$g$ = 386.088, in/sec $^2$
$E$ = pipe material modulus of elasticity, $lb/in^2$
$w$ = weight of pipe per unit length, lb/in

It is important to point out that the weight of the pipe per unit length must include the weight of the internal fluid and any other weight components. Failure to include the weight of the piping contents could yield natural frequencies that are 30 percent too high in frequency. The frequency factor is an important aspect of the calculation. It allows for compensation of the end condition or clamp. Table 1 illustrates the frequency factors for several end conditions and geometrical forms.

*Table 1. Frequency Factors for Calculating Pipe Natural Frequencies.*

| Beam Configuration | Frequency Factor | |
|---|---|---|
| | $\mu_1$ | $\mu_2$ |
| Cantilever | 3.52 | 22.0 |
| Simply supported span | 9.87 | 39.5 |
| Fixed supported span | 15.8 | 50.0 |
| Fixed - fixed span | 22.4 | 61.7 |
| Free - free span | 22.4 | 61.7 |
| | | |
| | **Out of Plane** | **In Plane** |
| Fixed end, Equal leg L bend | 3.74 | 15.4 |
| Fixed end, Equal leg U bend | 2.00 | 3.1 |
| Fixed end, Equal leg U bend | 2.26 | 2.8 |

The area of mechanical piping dynamics is so large that adequate coverage cannot be included in this work. For further information see publications that deal with this in detail (Miller, 1987, and Manual, 1992).

## CONTROLLING CAVITATION

Cavitation in reciprocating pumps occurs when the instantaneous pressure at some point in the inlet system drops below the vapor pressure of the liquid. It is important to understand the idea of instantaneous pressure in this statement. Cavitation usually occurs on the inlet side of the pump and most often in the pump manifold near the valve. Acceleration head was an effort to include the inertial effect of the liquid in the inlet line without real success. The use of inlet accumulators (and effort to supply liquid near the manifold) to reduce cavitation is also common practice, but this still does not address the real issue. There are several important aspects that must be addressed in order to prevent cavitation.

- Assure sufficient inlet pressure to overcome the frictional loss of the piping from a static flow viewpoint.
- Assure the pulsation level in the inlet manifold is not excessive, therefore contributing to lower instantaneous pressure.
- Assure normal dynamic valve operation to prevent late opening and the resultant lowering of pressure.

## VALVE DYNAMICS

The valves in reciprocating pumps operate similar to check valves, they are pressure differential controlled. Some valves have springs or a closing force while others rely on gravity. The effective flow area of the valve allows the normal flow to pass without excessive pressure drop and accompanying power loss. Experience has shown that there is a limit to the size of the flow area. As the area gets bigger, the moving element must also increase in size and weight, eventually canceling the benefit of the larger area. It is popular to include "sticktion" force as significant in valve motion, and in cases where viscous forces such as "sticktion" are present, grooves can be used to decrease such forces.

There are several factors that have been overemphasized, due to a lack of good experimental data. One of these is the idea of initial "area offset." Area offset is thought to occur due to the opening pressure area being generally smaller than the closing pressure area of normal valves. In some cases, this is called seat area or overlap area. There may be cases where "area offset" is significant, but in the vast majority it does not appear on field measurements.

Experience has shown that the lift area of the valve need not exceed the flow area of the valve. In some cases, if the lift area does exceed the valve flow area, unstable operation can result but not always.

It has been proposed that heavy valve elements increase the NPSH required. However, experience has shown that NPSH required is generally insensitive to valve weight.

Spring rate and preload do indeed contribute directly to the NPSH requirement. In some cases where the dynamic interaction of two or more pumps occurs, the spring preload or rate may need to be increased or decreased to aid in normal operation of the valve. In this special case, the NPSH required may not be consistent with normal logic. This is due to the dynamic pressure force imposed by the combination of pump dynamic flow and pulsation.

When all is said, the single most important factor that ensures normal valve operation and normal life of the valve is an acoustic nonresonant pump manifold. As mentioned previously, when the manifold has a significant standing wave, valve operation and life are at risk.

## CASE HISTORY 1— HIGH PRESSURE WATER INJECTION PUMPS

The following case history illustrates the type of performance and cavitation problems that occur when volume dampers are not properly implemented in piping systems.

The pump installation is composed of six triplex pumps that are being used in a saltwater oil field injection service. The pumps operate at the following conditions:

- Six triplex pumps operating in parallel
- The inlet piping has its origin at a common header, but the lines to each pump vary in length. There is a commercial gas-liquid damper on the pump inlet at the pump flange.
- The outlet piping has individual lines (different length) that merge also into a common line. There is liquid gas commercial gas-liquid damper on the pump outlet at the pump flange.
- The fluid pumped is saltwater.
- The pumps are fixed speed (180 rpm).
- The inlet pressure is 144.7 psia.
- The outlet pressure is 5814.7 psia.
- Pump unit flow is 171 gpm.
- The inlet and outlet temperature is 65°F.
- The piping from an acoustical viewpoint is simple.
- The piping from a mechanical viewpoint is simple with few elbows or changes in elevation.

The pulsation was measured on the inlet pipe before and after the damper device. The measurement indicated excitation of the manifold acoustic response, with 138 psi peak-to-peak at 108 Hz. Therefore, cavitation was also present in the piping and was audible.

The outlet piping pulsation was also measured at 713 psi peak-to-peak at 18 Hz. Pulsation downstream of the dampers was also high and vibration on the outlet lines was significant, although no fatigue failures had occurred. Since the second pump frequency (6× rpm) was not of a normal magnitude (it was many times greater than the first or 3× rpm), the pump was shut down, and the outlet bladders were inspected. The inspection indicated the bladder had failed and was not effective. Therefore, the acoustic natural frequency of the outlet line was being excited by the 6× pump component, causing significant magnification. The pump outlet damper bladder was replaced and properly charged, resulting in a normal spectrum dominated by a 168 psi peak-to-peak 3× pump component. This level was still considered excessive, so measures were taken to reduce the pulsation.

In this case, the same technique was successful in reducing the pulsation levels on both the inlet and outlet piping. A second gas liquid device was placed on the "back end of the pump" on both inlet and outlet pump manifolds. This was effective on the inlet due to the frequency shift of the manifold response to approximately twice the frequency. This shift was due to transforming the quarter wave response to a half wave response (open at both ends of the manifold, due to the two volume devices). The resultant pulsation level was only about 15 psi peak-to-peak. The cavitation that had previously occurred was no longer present. The outlet pulsation level was reduced to ~50 psi peak-to-peak with the additional effective volume. The outlet pulsation produced a corresponding reduction in the resultant vibration.

## CASE HISTORY 2— THREE AMMONIA PUMPS

To illustrate the type of pulsation and vibration problems that occur when volume dampers are not used in piping systems, the following case history is given. This case history is defined by the following elements and operating conditions.

- Three quintuplex pumps operating in parallel.
- The inlet piping has its origin at a common source, but the lines to each pump vary in length.
- The outlet piping has individual lines (different length) that also merge into a common line.
- The fluid pumped is ammonia.
- The pumps are variable speed (72 to 219 rpm).
- The inlet pressure is 275 psig.
- The outlet pressure is 5282 psig.
- The inlet and outlet temperature is 27°C to 30°C.
- The piping from an acoustical viewpoint is simple.
- The piping from a mechanical viewpoint is complex due to several changes in direction, elevations and associated elbows.

The pulsation was measured in the inlet and outlet lines of each machine in order to determine the level and spectral content. The maximum pulsation was consistently at 5× rpm with an amplitude of 1122 psi peak-to-peak in the outlet pump line of pump #1. The maximum pulsation occurred at 219 rpm, producing very significant vibration at 18.25 Hz. The next highest pulsation was measured on pump #2 in the outlet line with an amplitude of 1016 psi peak-to-peak at 5× (195 rpm) or 16.25 Hz. The pump #3 pulsation maximum was in the outlet line with an amplitude of only 117.6 psi peak-to-peak at 5× (120 rpm) or 10 Hz. Although the best place to measure pulsation in such systems is at the closed end of the pump manifold, it was not available. If measurements could have been made at the worse case, it would not be surprising to see twice the observed level. Due to budgetary restraints, there was no further analysis and recommendations were required in a trial and error fashion. Since it is clear that the pulsation is due to the acoustic response of the outlet piping, appendage type gas-liquid dampers were installed on the inlet and outlet of all the pumps. In cases such as this, it is always best to locate the dampers as close as possible to the pump inlet and outlet flanges. This forces the manifold response frequency as high as possible, thereby minimizing the energy for any pump order that will be coincident. When large rpm ranges are present, it is sometimes impossible to place the manifold response, so that it is noncoincident; therefore, the best alternative is to place it as high in frequency as possible.

## CONCLUSIONS

The dynamic flow and pressure components associated with pump systems are very significant and must be addressed in order to achieve an optimum design. The use of sophisticated models in this design process is required. 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. The value of personal engineering experience has recently been underestimated by the use of design systems that promise to be "expert systems." Ultimately the best, most efficient dynamic oriented designs are the result of sound engineering experience with good analytical tools.

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