

THE APPLICATION OF COMPUTER SIMULATION AND REAL TIME MONITORING TO MINIMIZE THE PRESSURE PULSATIONS IN COMPLEX PUMPING SYSTEMS

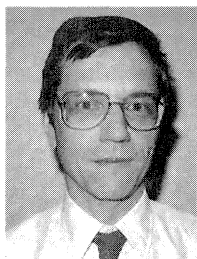
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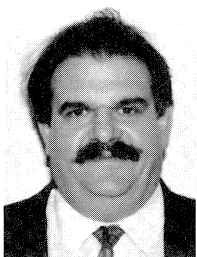
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pressures that can damage piping and components in the system along with the pumping machinery itself. These unsteady pressures can result in large forces that are reacted at the pump nozzles. These loads can lead to casing failures or extrusion of the flange gasket. The radial and axial thrust bearings are directly impacted and premature failure is a possibility well before the B10 life based on steady loading has been reached. There are several simple formulae that can be used to determine the order of magnitude of the pressure pulsations that are generated due to various system upsets. However, accurate modelling requires a numerical solution that can be performed on today's high end personal computers.

The numerical simulation is used together with a new remote monitoring system recently introduced by the authors' company. This technology has simplified the collection of real-time accurate field data during all phases of pump operation. The ability to retrieve the data remotely has provided a cost effective means of expanding the piping simulator's database. This has enhanced the simulator's ability to accurately predict a pump piping system response to anomalous flow conditions.

INTRODUCTION

Plant piping analyses carried out in the design stage are usually restricted to determining the piping losses in the system when the plant is operating at its rated condition. This is a situation where the system is "steady"—not changing in time. This analysis determines the pump head and flow requirements that are then specified in the bid process. It is unusual for the piping system, at this time, to be analyzed to determine the magnitude of potential pressure pulsations that might be caused by starting or tripping pumps or by valve movements. Yet it is important that the pumps be highly reliable, since the facility must run continuously because of the high capital cost of the plant and problems associated with restarting the process.

The study of unsteady flows in piping networks has usually been of interest mainly to civil engineers who are responsible for moving drinking and irrigation water over long distances. The pipe runs are extremely long, measured in tens or even hundreds of miles. The pipe diameters can be as large as five ft. This means that a considerable mass of water is in motion. Closing a valve in such a system means that there is the potential to create extremely large pressure pulsations in the piping as the kinetic energy of the fluid is converted into hydrostatic pressure. Mechanical engineers typically deal with piping system runs that are measured in tens of ft and pipe diameters that are less than one ft. Thus, a considerably

ABSTRACT

Chemical and process plants usually contain numerous valves and pumps in a complex network of pipe runs. Plant design and pump selection are usually based on steady state design conditions. However, the opening or closing of valves, or the starting or tripping of pumps, in the system can generate high unsteady

smaller mass of water is involved and the pressure pulsations experienced are typically not large enough to cause bursting failure. There are unique cases where severe problems exist even though pipe runs do not exceed the size of an industrial plant. This is typically a result of rapid system upsets, such as quickly opening or closing a valve. A plant in which pump trips or fast valving movements are more than a rare occasion should be considered a candidate for an unsteady pulsation analysis. The system can be properly designed to minimize pulsations at little additional cost if this analysis is done during the plant layout stage. Exactly what constitutes a "fast valve closure" depends on the length of pipe runs involved. Thus a valve closure that takes many seconds may still result in very high unsteady pressures when the associated pipe runs exceed 100 ft. The severity of the problem is increased if the pumps are multistage high pressure pumps.

Several simple formulae are given that are useful to obtain a rough approximation of the severity of the pulsations. These equations employ various simplifications that lead to an algebraic solution. Graphical approaches are also available. These approaches are more rigorous and systems with as many as nine or ten junctions have been analyzed graphically. Results are presented based on this method. This type of solution is based on solving the wave equation that results from simplifying the governing equations by dropping nonlinear terms. However, the inclusion of all the terms of the governing equations result in two coupled partial differential equations that must be solved numerically. These equations can be solved in a matter of minutes for a fairly complex piping network on today's high end personal computers.

The computer simulations were validated by data collected with a remote monitoring system. This system remains onsite for extended periods so as to capture the actual worst case operating conditions that various plant cycles may generate. The hardware consisted of an eight-channel controller that is configured to monitor pressures in the system along with bearing vibration levels, casing temperatures, lube-oil temperatures, and pump operating conditions. An important aspect of the instrumentation is that, once in place, it can be remotely monitored from the authors' engineering offices and at the plant. Data collected can also be remotely downloaded over phone modems. This is important for two reasons. First, some of the destructive pulsations are caused by upsets that may occur only occasionally—perhaps only once a month. Sending laboratory personnel to the plant for one shift will quite likely miss the worst plant transients. Another reason for acquiring the data remotely is that plant engineers are busy enough running the process without expecting them to take on the additional task of logging data.

Reviewed herein are the analytical tools available for analyzing a piping network to determine what potential dangers might exist as valves are opened or closed, or as pumps are tripped or started. Results are also presented for some of the processes that have been studied. Predictions and field data are compared. Recommendations are given so that plant designers from architectural, construction, and engineering firms can avoid some of the common errors and design a system that will minimize deleterious pulsations. A little additional thought and effort in the plant design stage can greatly increase the plant's availability over its useful lifetime.

Pump Remote Monitoring

The majority of the simulations done to date were performed only after a problem occurred in the plant process. The response was to instrument the pumps, and possibly, other parts of the piping network with the remote monitoring system and to begin a numerical simulation of the piping system. The data gathered were used to validate the simulation. The authors and the plant managers were confident that a proposed fix based on simulations of

alternate geometries would be successful if the predicted pressure transients for the plant system, as presently configured, agreed with the data obtained by the remote monitoring system. Thus, validation of the computer simulations has been one use of the remote monitoring system.

However, the true promise of the system lies in avoiding costly plant upsets. Previous to the remote monitoring packages now being offered, it seemed that the ability to predict the future was a required skill for every maintenance and operations manager. When machinery fails, it always seems to fail at the most inopportune time, usually at a critical juncture during the process. The pump OEMs utilize diverse tools and programs to minimize equipment failures. However, one of the most effective methods for anticipating a malfunction is a predictive maintenance program. An effective program requires the timely collection and analysis of trend data, recorded periodically from the equipment, followed by prompt action. Today, the challenge of maintaining equipment availability and efficiency is heightened by:

- Increased competition.
- Emphasis on increasing mean time between failures.
- Reduced plant operation and maintenance budgets.
- Decreased number of pump maintenance personnel.
- Less staff pump expertise available due to early retirement programs.

Part of the strategy with the enhanced monitoring systems is to improve predictive maintenance programs. For instance, vibration monitoring and spectrum analysis are routine practice today. Most rotating equipment is under some program of vibration monitoring to insure availability and prevent catastrophic failures.

In recent years, the availability of sophisticated monitoring hardware and software at an affordable price has made the monitoring of rotating equipment viable, but the diminution of inhouse expertise makes interpretation and maintenance planning based upon the collected data increasingly difficult.

The authors' company has developed a simple, cost effective remote pump monitor—that can be applied to monitor, alarm, trend, and provide data for failure prevention analysis. Developed four years ago, the system has been employed to solve pump warranty problems, determine causality of persistent maintenance issues and provide real time operational data to help optimize customers' processes and thus reduce their overall operating cost. The system is a complete monitoring and protection package for pumps, which reduces the need for manpower in data collection and analysis. The remote system can be utilized by inhouse personnel and/or networked, via its built-in phone modem, to pump OEM engineers and product specialists for expert analysis. This monitoring system provides valuable historical trend data required for effective maintenance planning, reducing the chances of a major pump failure.

The real time data recorded during these plant studies was collected, time and date indexed, and stored electronically. The rate of the data collections was varied during the experimentation period depending on the resolution required. Data collection rates varied from once per sec to once every 125 min. The reconfiguring of the data collection rate and the recovery of the data were simplified, because it was all done remotely from the authors' engineering offices. The actual downloading of the data was conducted autonomously by a desktop computer running a communications scheduler. This allowed for the timely collection of data during the once per sec scan rates which would fill the remote pump monitors database every four hr. The communications pathway was the conventional telecommunications networks and phone modem to modem communications at 2400 baud. A schematic of the remote monitoring system is shown

in Figure 1. The pump monitoring system box, which is located onsite, has a footprint of approximately one square foot. It is water-tight and can be placed in a harsh environment.

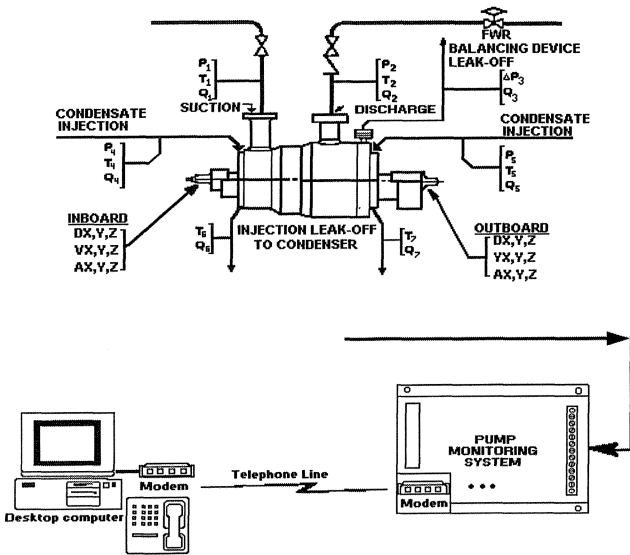


Figure 1. Schematic of Data Collection System.

This method of collecting and retrieving data was important to this study for two reasons:

- The ability to predict that a system is susceptible to anomalous flow transients is simpler than trying to predict exactly when a transit may occur. A destructive transient may only occur during certain system demand scenarios or may be the result of some random event. The ability to continuously monitor the system insures that if the system does experience an anomalous incident, the remote monitor will capture the data.
- The autonomous collection and retrieval of data, via the remote pump monitoring systems, also means that the authors were not dependent upon the subject facility operation personnel to record the data or required to station laboratory personnel at the plant.

The hardware used in the studies presented herein consists of an eight-channel controller that can be independently configured to alarm, trip, or simply monitor one or several pumps. Alarms can be localized and/or phone modemed to any location. The sensor input architecture is based on standard instrument signals, so a wide range of input configurations can be monitored. Typical inputs used were:

- Inner bearing housing vibration (two planes).
- Outer bearing housing vibration (two planes).
- Suction pressure.
- Discharge pressure.
- Flowrate.
- Balance line leakoff temperature.
- Lube oil pressure.
- Motor current.
- Suction temperature.
- Discharge temperature.
- Casing temperature.
- Lube oil temperature.

Since this monitoring system uses standard components, it was easily packaged to satisfy most environmental and safety demands. The monitoring hardware could also accept signals from existing installed instrumentation.

While the data collection process described above is important, it is only one part of the troubleshooting equation. The data must be analyzed, interpreted, and service recommendations made based upon the collected data. The interpretation of these data should be performed by expert analysts, either at the plant or at the OEM or by an outside consultant. The ability to perform a computer simulation of the process as described herein helps in understanding the data.

The monitoring system as currently used is a versatile device and powerful tool in determining the root cause of pump distress. Additionally, proactive programs help plant managers achieve their goals of increased reliability and efficiency of their pumping equipment.

FLUID MECHANICS OF FLOW IN PIPE NETWORKS

Steady Flow (No Change in Conditions with Time)

Plant designers are well versed in the simplest equation governing flow in pipe networks:

$$\frac{V_2^2}{2} + \frac{p_2}{\rho} + Z_2 = \frac{V_1^2}{2} + \frac{p_1}{\rho} + Z_1 - f \frac{L}{D} \frac{V^2}{2} \quad (1)$$

where:

- V = the pipe velocity
- p = the static pressure
- Z = the elevation
- L = the length of pipe between the first station, denoted by "1" and the second station, "2"
- D is the pipe hydraulic diameter

This permits the designer to properly determine the pumping head rise required to move fluid about the plant. Losses in straight runs of pipe, elbows, multiple turns, expanding and contracting sections, valves, etc., are included in the last term.

Unsteady Flow (Conditions Change with Time)

• *Oscillatory flow*—A periodic driving force which can result from:

- Reciprocating machinery—These problems are best solved by an impedance method (i.e., in the frequency domain).
- Resonance problems—An example would be a chattering valve at the natural frequency of the pipe network.
- *Transient flows*—Conditions resulting from moving from one steady flow geometry to another, such as opening or closing a valve. These problems are sometimes called "waterhammer" problems. They are best solved in the time domain. This is the class of problems considered herein.

THEORY OF HYDRAULIC TRANSIENTS

Hydraulic Analysis Techniques Available

- Rigid Water Column Theory

The fluid is assumed to be incompressible and the wave speed of propagation is taken as infinite. The pipe walls are assumed to be infinitely rigid. This is useful for slow changes in geometry.

- Elastic Water Column Theory

The fluid is acknowledged to be compressible and the wave speed of propagation is taken as finite. The pipe walls are recognized to be distensible.

Solution Techniques Available

- For the Rigid Water Column Theory:

Simple equations for the transient head generation, ΔH , can be derived from first principles, starting with Newton's Second Law of Motion: $F = \frac{d(mV)}{dt}$. The mass, m , is replaced by $\frac{\rho AL}{g}$. The term $\frac{dV}{dt}$ can be eliminated by using: $\Delta H = \frac{L}{g} \frac{dV}{dt}$. An algebraic equation is written to describe the valve motion or pump operation as a function of time.

- For the Elastic Water Column Theory:

- *Graphical methods*—These neglect friction and the nonlinear terms in the governing equations. These methods are based on solving the one-dimensional wave equation and have been in use for approximately 60 yr. The problem is solved in the H-q domain (where H is head and q is flow). Neglect of the frictional term can lead to underestimating the maximum transient.

- *Computer aided numerical solution*—These methods retain all the terms in the governing equations. These equations can be solved by various finite difference schemes, such as Lax-Wendroff and others. The equations can also be solved along specialized lines, called characteristics. This method is called "The Method of Characteristics." The results presented herein were computed with a modified version of the commercially available code (Flow Science Incorporated) SURGE which is typically applied in civil engineering applications.

The governing equations are:

Continuity

$$\frac{\partial h}{\partial t} + v \frac{\partial h}{\partial x} + \frac{c^2}{g} \frac{\partial v}{\partial x} - v \frac{\partial z}{\partial x} = 0 \quad (2)$$

Momentum

$$\frac{\partial h}{\partial x} + \frac{v}{g} \frac{\partial v}{\partial x} + \frac{1}{g} \frac{\partial v}{\partial t} + \frac{fv|v|}{2gm} = 0 \quad (3)$$

where:

h = pressure and elevation head = $\frac{P}{\rho} + z$
 v = velocity of fluid in pipe
 m = pipe area/perimeter
 f = Darcy-Weisbach friction factor

The algorithm used for the present studies solves the complete set of equations, i.e., Equations (2) and (3).

Some analyses neglect the term $v \frac{\partial h}{\partial x}$ as being small compared to $\frac{\partial h}{\partial t}$ since (by utilizing $x = ct$) $v \frac{\partial h}{\partial x} \approx \frac{v}{c} \frac{\partial h}{\partial t}$; they also neglect $\frac{v}{g} \frac{\partial v}{\partial x}$ as small compared to $\frac{1}{g} \frac{\partial v}{\partial t}$; then, if the frictional term is also neglected, the governing equations reduce to:

$$-\frac{c^2}{g} \frac{\partial v}{\partial x} = \frac{\partial h}{\partial t} \quad (4)$$

$$\frac{1}{g} \frac{\partial v}{\partial t} = -\frac{\partial h}{\partial x} \quad (5)$$

if $\frac{\partial(\text{Eqn4})}{\partial t}$ is added to $c^2 \frac{\partial(\text{Eqn5})}{\partial x}$, the One-Dimensional Wave Equation is obtained:

$$\frac{\partial^2 h}{\partial t^2} = c^2 \frac{\partial^2 h}{\partial x^2} \quad (6)$$

The graphical method is a solution of the one-dimensional wave equation, Equation (6).

Simplified Formulae For Estimating Transient Magnitude

Consider the problem of a valve downstream of a pump. The pump produces a total dynamic head, H_0 . Simple equations based on both the rigid water column and the elastic water column theories can be developed to predict the transient pressure pulsations, ΔH , generated when the downstream valve opens or closes. Let the valve open or close in time T .

- Rigid Water Column Theory:

$$\frac{\Delta H}{H_0} = \frac{K_1}{2} \pm \sqrt{K_1 + \frac{K_1^2}{4}} \quad (7)$$

$$\text{where } K_1 \equiv \left(\frac{LV}{gTH_0} \right)^2$$

(+ is for valve closure; - is for valve opening.)

Note that as the time to close the valve becomes shorter and shorter, i.e., T approaches 0, that the unsteady head becomes infinite!! This is because rigid water column theory assumes that the fluid in the entire length of the pipe is decelerated. This is obviously physically unreasonable.

- Elastic Water Column Theory:

Elastic water column theory will decelerate only the fluid in that length of pipe that a pressure wave can travel. This results in a finite head rise for the valve closure problem:

$$\Delta H = K \frac{cV}{g} \quad (8)$$

where:

K = function of $(N = T/(2L/c))$

c = wave speed

The value of K is given in Figure 2. This is the result of a graphical solution of the one-dimensional wave equation due to Kerr [1].

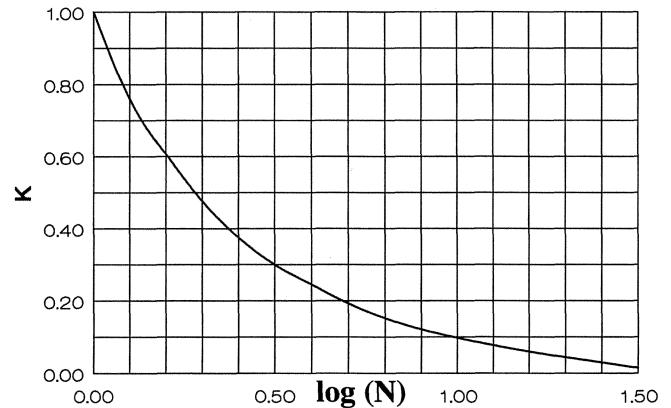


Figure 2. K vs $\log(N)$ for Valve Closure.

Computing the Wave Speed, c

The pressure pulsations travel down the network at a wave speed, c . The magnitude of the pressure pulsation depends strongly on the wave speed. If the valve closes in a time less than what is

required for the pulsation to travel down the pipe and return, the pressure pulsation generated will be as large as if the valve were closed instantaneously. This is because the reflected wave that returns to the valve would have decreased the original pulsation if it had returned while the valve was still closing. The reflected wave has the opposite sign (i.e., a rarefaction wave if the original wave generated was a compression wave caused by a valve closure) and, therefore, can diminish the original pulsation.

The wave speed depends not only on the speed of sound of the fluid but also on the stiffness of the piping and the air content. Actual piping is not perfectly rigid, and as the pressure pulsation passes, will distend. This causes the effective wave speed to decrease as the fluid must now fill a larger cross section. The wave speed is given by:

$$c = \frac{1}{\sqrt{\frac{\rho}{g} \left(\frac{1}{K} + \frac{d}{tE} \right)}} \quad (9)$$

where:

c = wave speed, ft/sec

ρ = density of fluid; for water with \times % air:

$$\rho = \left(1 - \frac{x}{100} \right) 62.4 + \left(\frac{x}{100} \right) 0.076 \text{ (lbm/ft}^3\text{)}$$

g = 32.174, ft-lbm/lbf-sec²

K = liquid bulk modulus, lbf/ft², [$= 39.3 \times 10^6$ for water with no air]

E = Young's modulus for pipe material, lbf/ft², [$= 44 \times 10^8$ for steel]

d = pipe average diameter, ft

t = pipe wall thickness, ft

The bulk modulus of water is greatly reduced with the addition of a small amount of air. One or two percent of air will result in a large reduction of the bulk modulus and, hence, of the wave speed. This will tend to increase pressure pulsation levels for a given valve stroke. The variation of the bulk modulus of water, K , with air content is shown in Figure 3. This figure is applicable at atmospheric pressure. There is a slight pressure effect. At 200 bar, the bulk modulus increases by 4.6 percent, leading to a 2.3 percent increase in the wave speed.

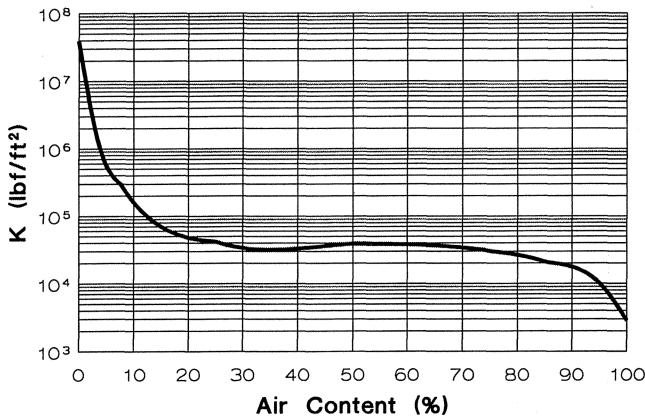


Figure 3. Bulk Modulus of Water vs Air Content.

The wave speed decreases as the ratio of pipe diameter to wall thickness increases. This also exacerbates the unsteady pressure

pulsation problem. The ratio of d/t is shown in Figure 4 for variations in pipe size and pipe schedule.

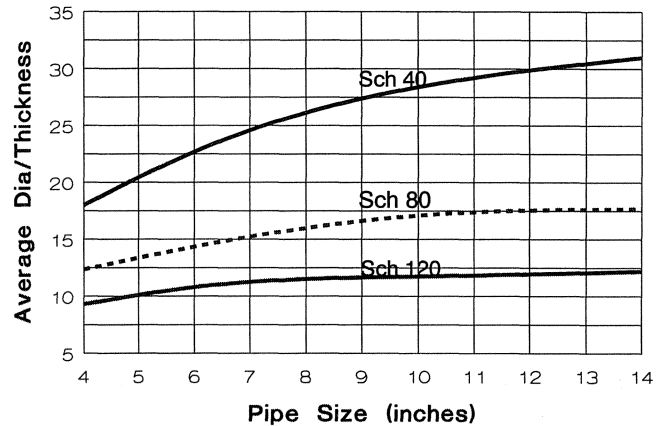


Figure 4. Pipe Mean Diameter to Thickness Ratio.

TYPICAL CALCULATION FOR PULSATION LEVEL

A severe test for both the calculation procedure and the remote monitoring system is a steel mill descaling section. A schematic is shown in Figure 5 of the system as originally configured. The descaling water system is the subsystem of a hot strip mill that achieves the required surface finish by removing scale. This is done by impacting the slab on its upper and lower surfaces with high pressure water forced through nozzles carried on upper and lower headers. Here, fast acting valves with strokes on the order of one second are located in descale boxes, which may be 300 ft from the pumps that supply the water. The rapid closing of the descale valves converts the kinetic energy of the flow into pressure energy in the form of a pressure pulsation, which travels back the piping run to the pump discharge. The combination of very fast acting valves and moderate pipe lengths can generate unsteady pressure pulsations with half amplitudes equal to the pump's output mean pressure. These pressure pulsations return to the "descaling pumps" (i.e., the pumps supplying the high pressure water—approximately 6000 ft head rise) and can severely damage thrust bearings and cause nozzle loads that exceed design limits. The descale pumps are multistage horizontally split pumps that resemble boiler feed pumps. They have conservatively designed components including thrust bearings. Recently the descaling pumps in a new mill experienced seal failures after only two months operation. Pump disassembly uncovered casing erosion that, along with the evidence of extrusion of the flange gasket, indicated excessive and abnormal nozzle loads. Subsequent hydraulic transient analysis of the system indicated severe problems in the layout of the piping system.

The mill descale pumps may receive their suction flow from a suction tank, which is usually located fairly closely or from a pump house that is often as far as 1500 ft away. In the latter case, the flow is provided by a single stage booster pump. When the descaling valves open, the flow in the suction line is accelerated. This results in a transient acceleration head which reduces the NPSHA. Obviously, it is preferable to feed the pumps from a nearby suction tank. If the transient is large enough to momentarily reduce the NPSHA to below the vapor pressure, the water in the suction pipe can flash to steam. When the steam pocket is pressurized again, it collapses and, in the process, generates a large transient pressure. This phenomenon is referred to as "column separation." The code used to simulate the transients can only handle single phase flow and is, thus, incapable of predicting the transients associated with

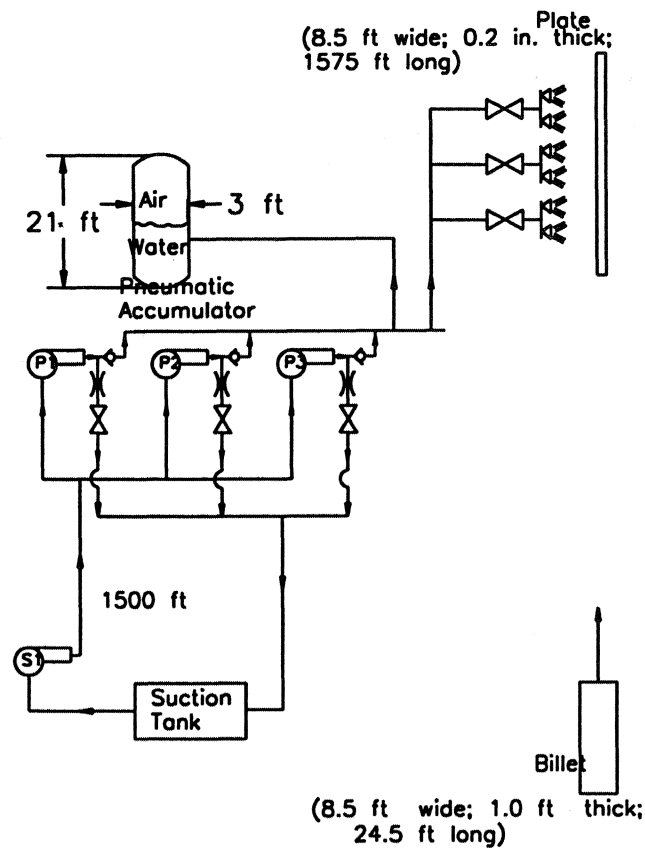


Figure 5. Typical Use of Accumulator.

this two-phase phenomenon. It can predict up to the point where "column separation" occurs, so it serves as a warning of a condition that must be avoided. Pump suction transients are best minimized by drawing the suction flow from a large suction pit located close to the pump inlet. Another alternative is to locate a special suction accumulator tank near the pumps suction line.

Results of Full Computer Simulation Compared to Data Gathered Via Remote Monitoring System

Pump discharge pressure pulsation data recorded by the remote-monitoring system is shown in Figure 6. The sampling interval was one min. This data chart indicates that transient pressure spikes of 670 ft were experienced at the pump discharge. The predicted time trace at the pump discharge is shown in Figure 7. A peak unsteady pressure pulse of 650 ft is predicted to be generated by the conversion of fluid kinetic energy into pressure energy. This compares very well with the levels measured in the discharge piping.

The corresponding suction transient data recorded for a 12 hr period taken at one min intervals is shown in Figure 8. Here peak transients are in the range of 35 to 60 ft. The predicted transients generated by the valve closure are somewhat higher—pressure pulsations that are approximately 100 ft peak-to-peak are shown in Figure 9. The predicted pulsations have a frequency of 0.73 sec, which corresponds to $2L/c$ of the pipe run between the pump house and the descale pump suction.

While the magnitude of these suction pressure transients is less than the discharge pressure transients, they serve notice that the far worse "column separation" transient may be a danger.

While the suction peak value is of very short duration (perhaps not long enough to cause "column separation"), there is sustained operation with an unsteady suction pressure 100 ft less than the

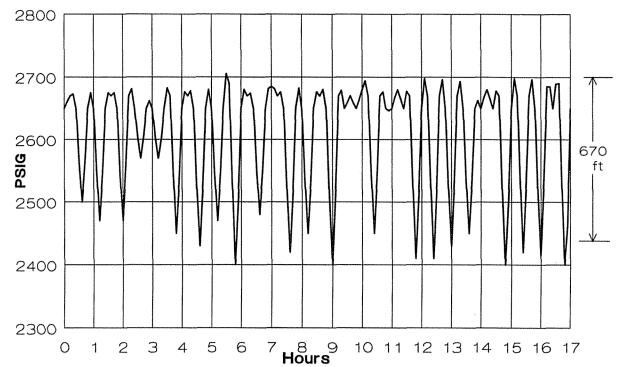


Figure 6. Pump Discharge Pulsations Recorded by Remote Monitoring System.

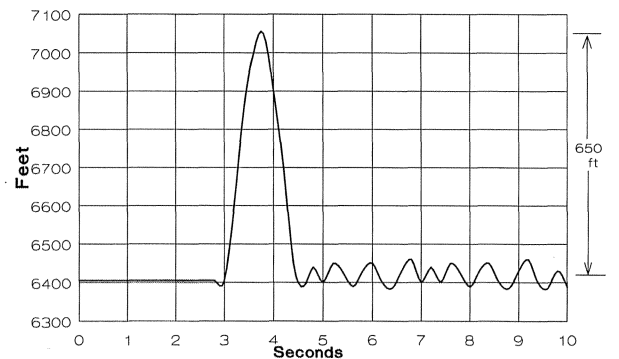


Figure 7. Pump Discharge Pulsations—Computer Simulation.

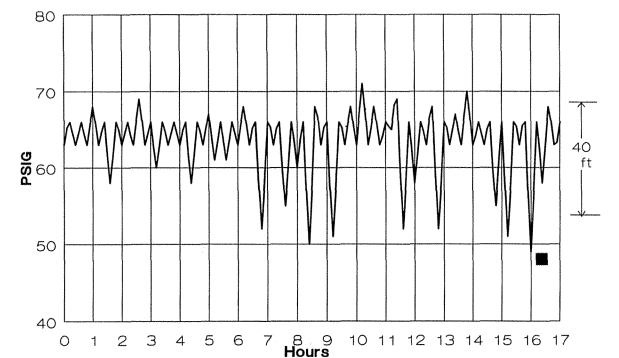


Figure 8. Pump Suction Pulsations Recorded by Remote Monitoring System.

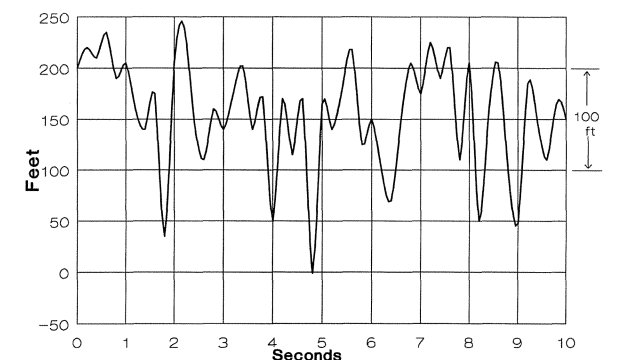


Figure 9. Pump Suction Pulsations—Computer Simulation.

nominal design level. Since the pump's NPSHR is 40 ft, it is clear that the actual system NPSHA, when the acceleration head is included, begins to get dangerously close to the pump's requirement to avoid a three percent drop-off in head production and may result in first stage cavitation damage.

Results of Rigid Water Column Theory

If Equation (7) is applied with pipe length of 125 ft; $H_0 = 5600$ ft; and schedule 80 pipe diameter = 6 in and valve stroke of one second:

$$\begin{aligned} \Delta H_{\text{Rigid Column Theory}} &= 96.3 \text{ ft (half amplitude) or} \\ &= 193 \text{ ft peak-to-peak} \end{aligned}$$

This prediction is lower than the recorded data peak value of 670 ft probably because the derivation of Equation (7) assumes a linear valve stroke that underestimates the initial rate of change of flow area and because the effect of friction has been neglected (called "line pack").

Results of Elastic Water Column Theory

If Equation (8) is applied to this problem with a wave speed of 3081 ft/sec and an initial pipe velocity of 22.13 ft/sec:

$$\begin{aligned} \Delta H_{\text{Elastic Column Theory}} &= 191 \text{ ft (half amplitude) or} \\ &= 381 \text{ ft peak-to-peak} \end{aligned}$$

This prediction is still lower than the recorded data peak value of 670 ft, but is a better approximation than that given by the rigid water column theory.

Thus, in general, the magnitude of both the pump discharge and suction pressure pulsations are predicted very well with the digital computer simulation of the system. The simplified formulae presented can be used to give some indication of the level of the transient.

**PREFERRED USE OF AN ACCUMULATOR—
TWO-WAY ACCUMULATOR**

The optimal location of the accumulator tank is in series between the pump discharge and the valves. Both the pump discharge transient pressure spikes and the pump suction side transients will be reduced if a two-way accumulator is used. This two-way location is very important in order to achieve a reduction in the pressure pulsations. Dead-ending the pressurized tank (i.e., a one-way connection) almost totally negates its effectiveness. Although the exact numbers depend on the overall characteristics of the plant, *dead-ending the accumulator will accomplish pressure pulsation reduction of only 10 percent to 30 percent of what the same size tank is capable if properly connected.* The proper means of implementing the pressurized tank is shown in Figure 10.

CONCLUSIONS

- The computer simulation predicted transient pulsations agree very well with the measurement data at the pump suction and discharge. This comparison serves to validate the numerical simulation of the system.
- The remote pump monitoring system deployed was an excellent cost effective means of gathering the empirical data that were used to qualify the prediction algorithms and calibrate the model. The empirical data were also used to plan remedial actions. Continuous use of a remote monitoring system provides processed data that can enable key personnel to make informed maintenance and operations decision resulting in less unscheduled downtime and more targeting of repairs.

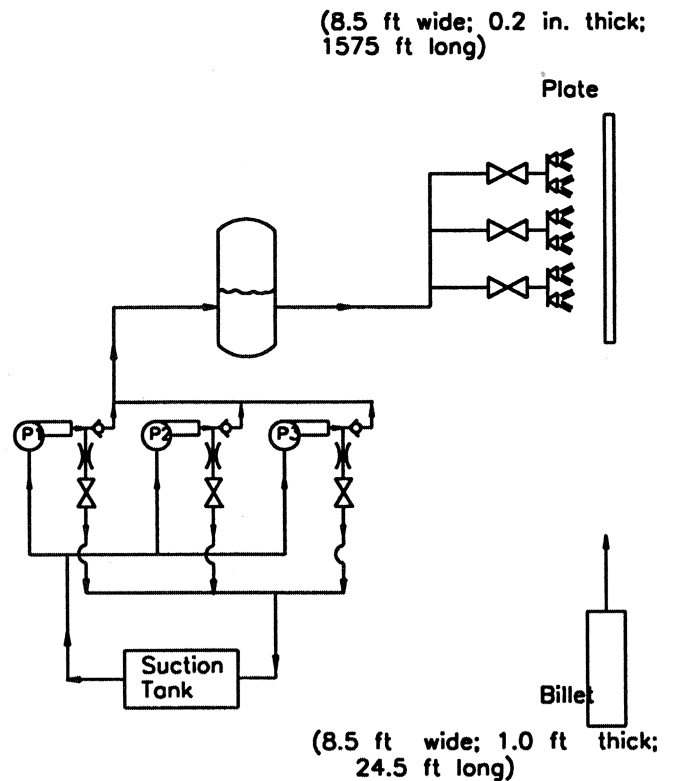


Figure 10. Preferred Use of Accumulator.

- Some of the simplified formulae presented are useful for a quick approximate calculation; however, modelling the governing equations is necessary to obtain accurate predictions.
- The predicted pump discharge transient pulsation of 650 ft, with the accumulator dead-ended as is presently the case, is predicted to be decreased to less than 30 ft with a two-way accumulator.
- Long pipe runs from a pump house to the suction of the pump can cause a large acceleration head when a discharge valve opens.
- The acceleration head results in a reduction of NPSHA possibly below the pump's NPSHR. This could result in cavitation damage to the first stage.
- A worse problem that could occur because of the lack of suction pressure would be "column separation." This would result in severe pump transient loading. Vapor bubbles will appear in the fluid; the bubbles will collapse when the flow reverses.

RECOMMENDATIONS

- It is extremely expensive to modify a complex piping network once operations begin. Thus, it is important that the numerical studies indicated herein be performed in the plant layout stage. Most of the recommendations will not result in additional costs if designed into the plant from the onset.
- Remote monitoring systems, such as the one used in this report to gather the pressure pulsation data, are an extremely cost effective tool. The data were used herein to verify computations and plan remedial actions. However, this monitoring system also improves the plant's predictive maintenance program. Problems are identified before they shut the plant down. Developing problems can be watched and scheduled maintenance can be performed before a component failure shuts the process down.

- Pressurized surge tanks should always be connected in series with the component which they are to protect—they must never be dead-ended off a line of their own.
- Pressurized surge tanks should be filled approximately one-third with water and two-thirds with air.
- The pumps should receive their flow from a suction tank located as closely as possible. Feeding the pumps from a pump house considerably distant may endanger the pump. Suction acceleration head can reduce the effective NPSHA and cause cavitation damage or extremely high pressure pulsations if the suction flow periodically flashes to vapor.

REFERENCE

1. Kerr, ASME Journal of Engineering for Power (1924).

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