

# CONTROL VALVE INDUCED PIPELINE VIBRATIONS IN A PAPER PULP PUMPING SYSTEM

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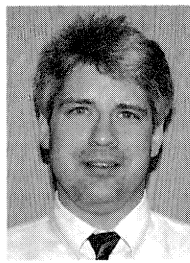
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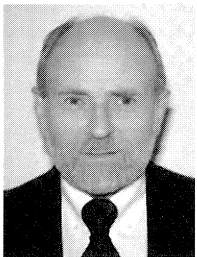
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## ABSTRACT

A case history is presented pertaining to piping vibrations in a medium consistency paper pulp pumping system at a large North American integrated pulp and paper mill that ultimately resulted in pipe rupture. The system was designed to transfer pulp at 12 percent oven-dry consistency (12 percent fiber-88 percent water by mass) from a washer stage to a storage tower through nearly 700 ft of 20 in diameter pipe. Testing and analyses were performed on the system to diagnose the problem and develop a solution to reduce the piping vibrations and prevent further failures.

Initial tests on the system indicated the piping had pressure pulsations of between 50 psi and 100 psi peak-to-peak which were exerting a shaking force of approximately 15,000 lb to 30,000 lb peak-to-peak at each of seventeen 90 degree elbows in the piping system. The frequencies of the major pressure pulsations were at

1.3 Hz and 3.8 Hz, which corresponds to the one-quarter and three-quarter (closed-open piping system) standing pressure waves in the piping system. These forces produced by the pulsations caused the piping system to respond at the natural frequencies of the pipe and support structure at each section of the pipe system.

Attempts to reduce the amplitudes at the standing wave frequencies with attenuators (valve or shear plates) at the end of the pipeline were not successful due to the nature of the pulp and flow pulsation problems. Some success was achieved by changing the standing wave (acoustic) frequency with the addition of gas to the pulp in order to change the bulk density, but this was deemed an impractical solution.

The problem was diagnosed as an interaction between the control system and the piping system. The source of the pulsation energy was traced to the control valve located immediately downstream of the centrifugal pump. Strouhal vortex shedding frequencies were generated at the exit of the control valve ball which excited acoustic standing waves in this particular piping system and amplified the pressure pulsations. Evidence is presented that shows that the pulsation amplitude was dependent upon the valve ball position and motion; therefore, the valve was removed and an alternate method of system control with a different mechanism was used that eliminated the source of the harmful pulsations.

The authors present application guidelines intended to assist the reader in evaluating pipe system designs for the potential of standing wave excitation. The Strouhal vortex energy from the control valve is discussed. Also, the calculation of acoustic velocity is covered including the effects of temperature, entrained air, and pulp consistency.

## DESCRIPTION OF PUMPING SYSTEM

The system consists of two nearly identical pumps and piping systems located in a pulp mill. The purpose of the systems is to transfer medium consistency paper pulp at 12 percent oven-dry consistency from the brown stock washing stage to a 500 ton storage tower to await bleaching. Typically, in the making of chemically prepared wood pulp suspensions, wood chips are cooked in a digester with suitable cooking "liquors" to convert the chip into the wood fibers from which paper is made. The pulp is then put through a series of brown stock washers in order to displace the cooking liquor with water and to recover the cooking liquor partly for fuel and partly for reuse. The decker is a device that serves as the final stage in the brown stock washing process, along with thickening the pulp from between one and one-half and three percent to 12 percent consistency. The systems here are referred to as the "east" and "west" decker pump systems and were each designed to deliver 360 ADST/D (air-dried short-tons per day) at 450 gpm of 12 percent consistency pulp to the storage tower. The pumps are 4 × 8–18 size and produce 650 ft of head at the rated flow in this system at a running speed of approximately 2500 rpm using a four-pole motor and a sheave V-belt drive.

Paper pulp, or paper "stock" as it is sometimes referred to, is a suspension of wood fibers in water or other aqueous solution. At low fiber consistencies (typically one percent or less) the fibers move freely in the suspension. Above this consistency the fibers tend to agglomerate into fiber bundles, or flocs, in which the fibers are mechanically entangled. Rheologically, wood pulp suspensions behave as non-Newtonian, viscoelastic materials as a result of the fiber network that is formed. Pipe friction losses for chemical pulps are higher than for water, except at very high velocities, and increase with increasing consistency [1]. In the medium consistency range (from approximately 8 percent to 15 percent oven-dry consistency), the fiber network is extremely strong with an apparent viscosity reported in the range of 0.33 lb-sec/ft<sup>2</sup> [2]. This results in relatively high pipe friction loss. Typically,

discharge pipe system designs are kept in the flow velocity range of 0.3 to 1.0 ft/sec in an effort to minimize friction losses, which are usually between 0.5 and 1.0 ft of water per ft of pipe length.

The pump applied to this service is a medium consistency centrifugal pump as shown in Figure 1. The pump is equipped with an inducer to assist in feeding pulp to the impeller. Medium consistency pulp contains large amounts of entrained air. As a rule of thumb, the amount of entrained air by percentage volume is numerically equal to the consistency by mass percent, but this can vary widely depending on fiber type, process, temperature, and several other variables. As a result, the pump is also equipped to separate and remove the entrained air through a vacuum air removal system. The suction of the pump is attached to a "suction standpipe," which is a cylindrical vessel four ft diameter by 15 ft tall. The vessel is gravity fed from the east or west decker, located on the floor above the pumps. The flowrate of the pulp is controlled by a V-sector ball valve located at the pump discharge flange which receives a signal from a level sensor mounted within the standpipe. An auxiliary control system (the dilution control) is also mounted on the suction standpipe, but is not required under normal operating conditions. Typical components that make up a medium consistency pump system are shown in Figure 2.

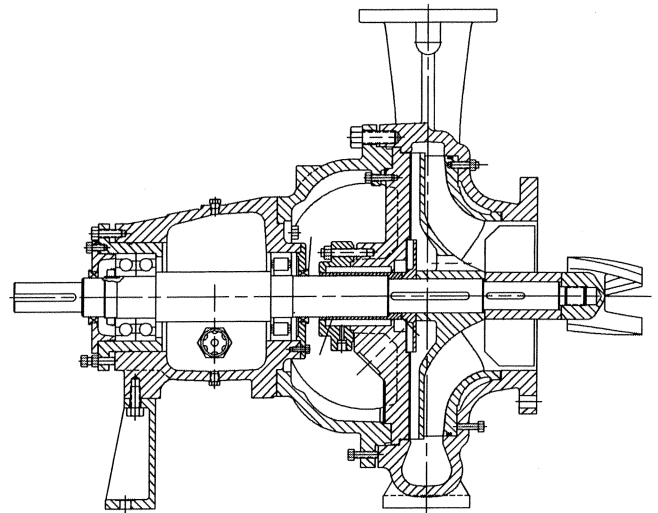


Figure 1. Cross Section of a Medium Consistency Centrifugal Paper Stock Pump.

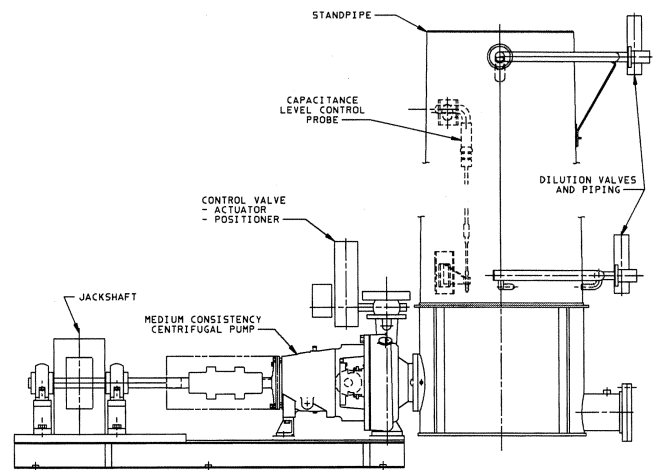


Figure 2. Medium Consistency Centrifugal Pump and Typical Control Components.

The piping system shown in Figure 3 is an isometric drawing of the east decker pump piping. The west piping system is identical except for the first few feet at the pump. The systems are hung side-by-side from points C to I. The pipe itself is 20 in diameter, 0.25 in wall thickness stainless steel (Schedule 10) and extends some 685 ft in length with an elevation increase of 120 ft. There are seventeen 90 degree elbows distributed along the length of the piping with several 45 degree elbows. The pipe is supported at point B by a short column fixed to the masonry floor. The piping was originally hung with weight hangers from the existing building structure which resulted in low mechanical natural frequencies of the piping and structure. Braces have been added to the pipe to resist the vibrations; however, the braces cannot completely restrain the large forces generated by the pressure pulses.

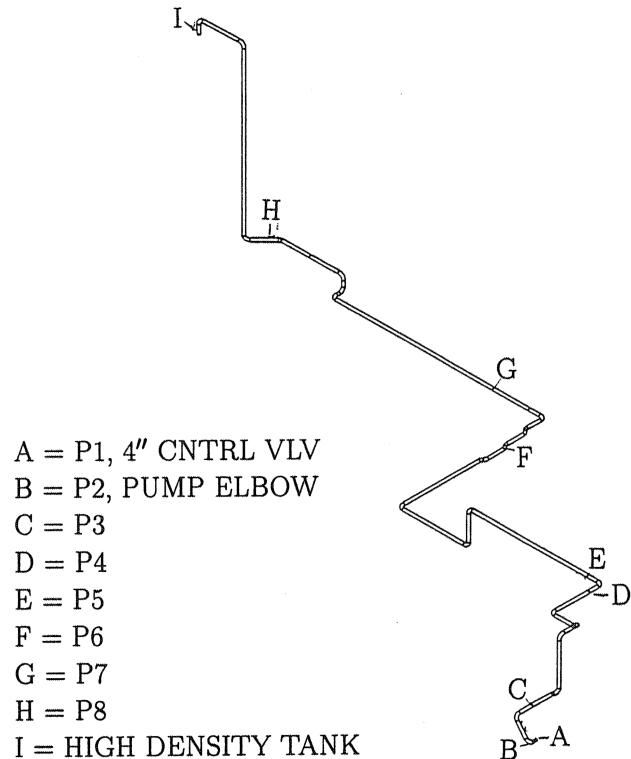


Figure 3. East Decker Piping System Isometric Drawing.

The west decker standpipe, the pump, and the first pipe anchor at point B on the isometric are shown in Figure 4. From points C to E, the pipe is hung on a preexisting masonry building that houses the brownstock washers, deckers, and pumps. A photograph of point C on the isometric drawing is presented in Figure 5. The hanger on the left is the original pipe hanger; the one on the right was added in an attempt to stiffen the pipe system. Note the signs of relative movement between the pipe and the original hanger indicating a large amount of pipe motion. From points E to F, the pipe is hung from a preexisting column and beam structure.

A photograph is shown in Figure 6 from point E on the isometric looking downstream as the piping enters the beam and column structure. This was the site of several pipe ruptures in both systems. The original pipe hangers can be seen in the background while the beam truss shown in the foreground was added after problems arose. From the 90 degree elbow (Figure 7) between points F and G to point H, the pipe is supported in a pipe rack. Finally, from point H to the conclusion at point I, the pipe is supported on hangers mounted to the side and top of the masonry storage tower, as shown in Figure 8.

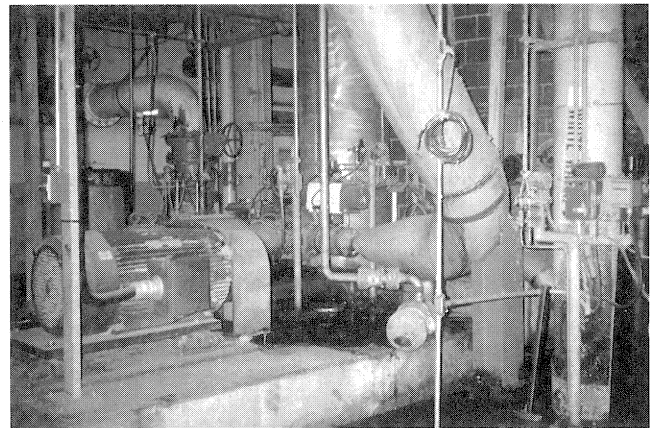


Figure 4. West Decker Pump, Standpipe, Control Valve, and Discharge Pipe.

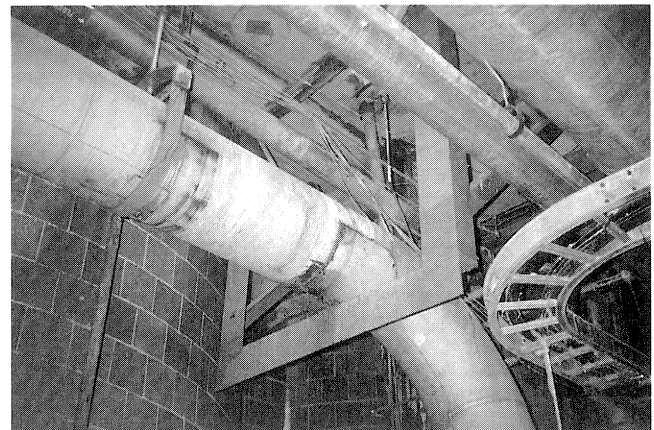


Figure 5. East Decker Piping System—Point C.

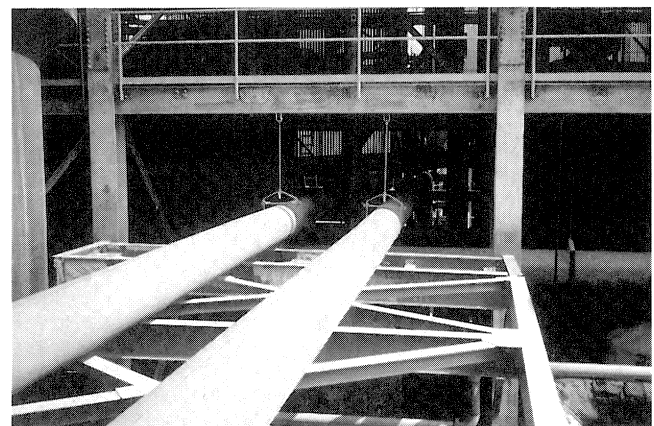


Figure 6. East and West Decker Piping Systems from Point E.

## DESCRIPTION OF PROBLEM

During startup and commissioning of these pumps, extremely large and violent vibrations were observed in both piping systems. Over time, damage occurred to the piping and its supporting structures, including ruptured pipe, cracks in masonry supports, failures of diagonal cross bracing in the column, and beam support structures, and fastener failures on the pipe rack. Obviously, these events resulted in production delays and safety issues that were of great concern to the owner of the systems. Initial analysis indicated

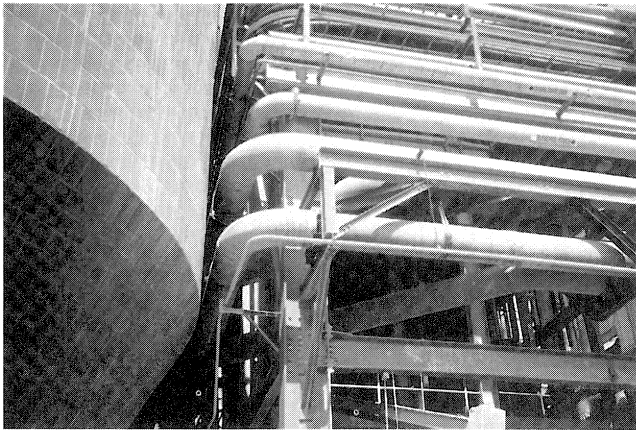


Figure 7. East and West Decker Piping Systems—Elbow Between Points F and G.

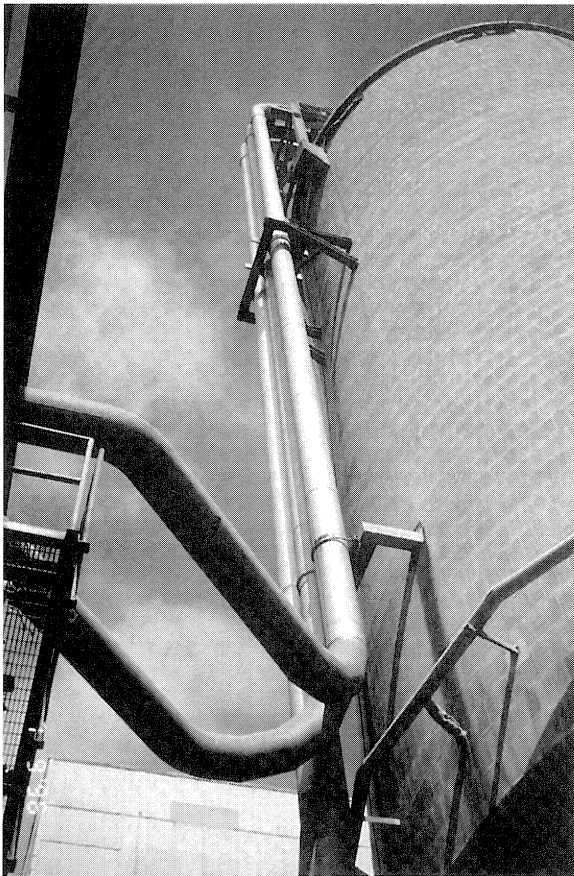


Figure 8. East and West Decker Piping Systems from Point H.

that the pumps were performing properly; therefore, attention turned to the piping system. Several modifications had been made to the piping system that were designed to support the pipe and stiffen its support structures. None of these modifications resulted in any significant reduction in the vibrations.

At this point, a field test and analysis of the nature of the vibrations was undertaken to determine their source. Several types of transducers were used to acquire the field data including accelerometers (piping vibrations), static/dynamic pressure transducers (pressure pulsations) and strain gages (pipe strain). The data that were acquired during the field study are presented in frequency spectra and complex wave shape formats. A complete

listing of the instrumentation along with signal processing equipment is presented in APPENDIX A.

The pulsation frequencies measured were well below any of the normal pump-related frequencies; therefore, the pump itself was not considered to be the source of the problem. The pulsation at vane passing frequency was measured at one psi peak-to-peak (approximately one-third of one percent of developed head) which is normal for a pump of this type.

The preliminary tests of the vibration and pulsation in the east decker piping system confirmed a standing pressure wave (resonance) that produced pulsation peaks of 80 psi to 110 psi peak-to-peak. A plot of pressure vs time (Figure 9) for test points P1 and P3 for a flowrate of 373 gpm and 12.4 percent consistency indicates the severity of the pressure pulsations. The pressure waves are reflected at the change in flow area (control valve) at the pump and at the end of the pipe at the storage tank. Pressure pulsations were measured at several locations along the length of the discharge pipe, as shown in Figure 3, for the purpose of confirming the existence of standing waves. A plot is shown in Figure 10 of pressure vs frequency for each of the eight test point locations taken at a flowrate of 524 gpm and 11.4 percent consistency. The pulsation amplitudes produced a large shaking force on the pipe at the elbows that caused the observed vibrations.

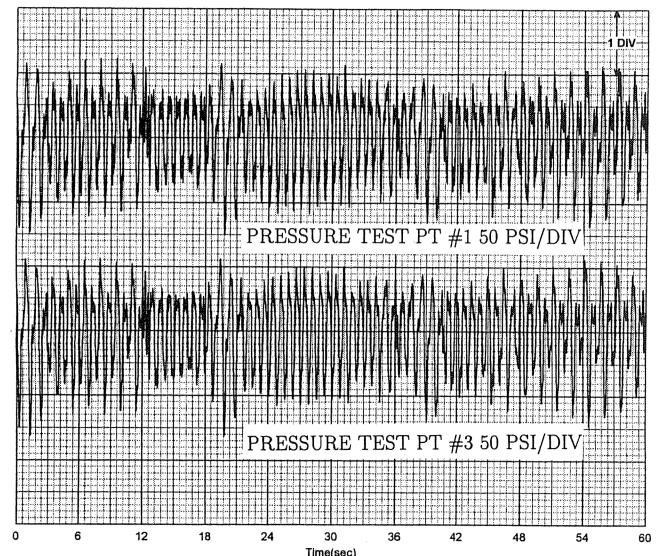


Figure 9. Pressure vs Time at P1 and P3.

The pulsations were predominantly at 1.3 Hz and 3.8 Hz, which are characteristic of a one-quarter and a three-quarter standing pressure wave, respectively, in the pipe [3]. The mode shape of a standing wave is characterized by dynamic pressure amplitudes that vary in a sinusoidal pattern along the length of the pipe. The solid curve shown in Figure 11 is a quarter sine function and the dashed line is a three-quarter sine function for the length of the pipe from the pump to the storage tank. The solid circles are the pressure pulsation amplitudes at 1.3 Hz at the test point locations. The solid squares are the pressure pulsation amplitudes at 3.8 Hz at the test point locations. The phasing of the pulsations at 3.8 Hz for the first three plotted points (zero to 225 ft) was out of phase with the other pulsations (225 ft to 685 ft) that matched the theoretical pattern of the three-quarter wave. The close match of the pulsation amplitudes and phase with the theoretical amplitudes confirms the existence of the standing waves.

The acoustic velocity of the medium consistency pulp ranged from 2900 ft/sec to 3700 ft/sec. For comparable conditions, the acoustic velocity of water is 5000 ft/sec to 5100 ft/sec. The pipe

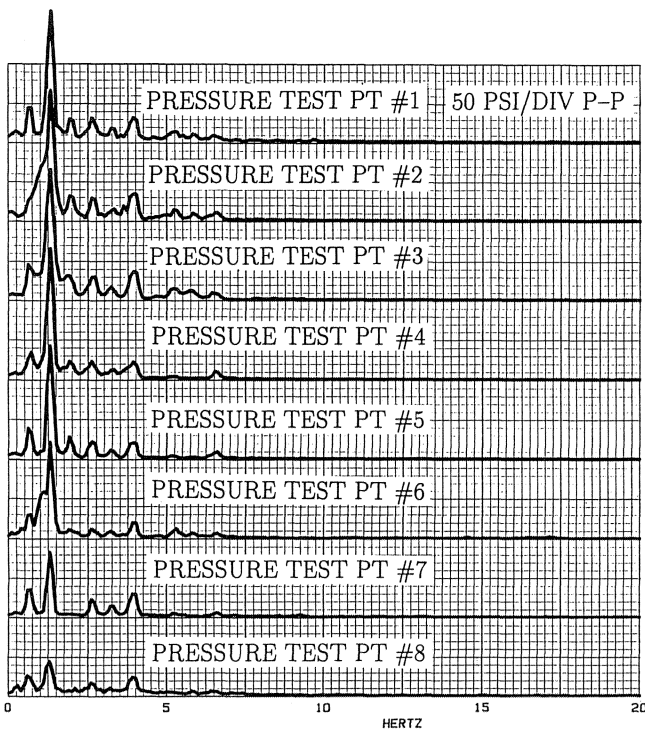


Figure 10. Pressure vs Frequency at All Test Locations.

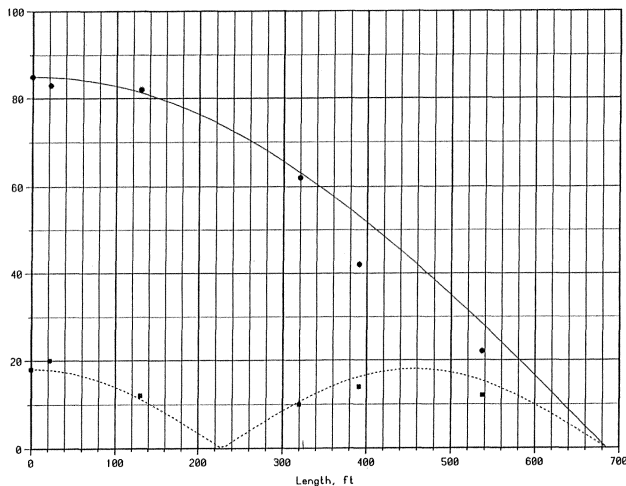


Figure 11. Quarter and Three-Quarter Sine Mode Shapes for East Decker Piping System.

wall flexibility reduces the effective acoustic velocity to 2200 ft/sec to 2800 ft/sec. The following equations for a Newtonian fluid were used as guidelines in evaluating the acoustic response of the medium consistency pulp [4]:

The relation for a quarter wave resonance:

$$f = (2N - 1) (a/4L) \tag{1}$$

where:

N = 1, 2, 3... = harmonic,

and:

a = the velocity of sound in an elastic container (acoustic velocity), ft/sec

L = total pipe length, ft  
 f = quarter wave resonance frequency, Hz

The relation for the velocity of sound of a fluid in pipe:

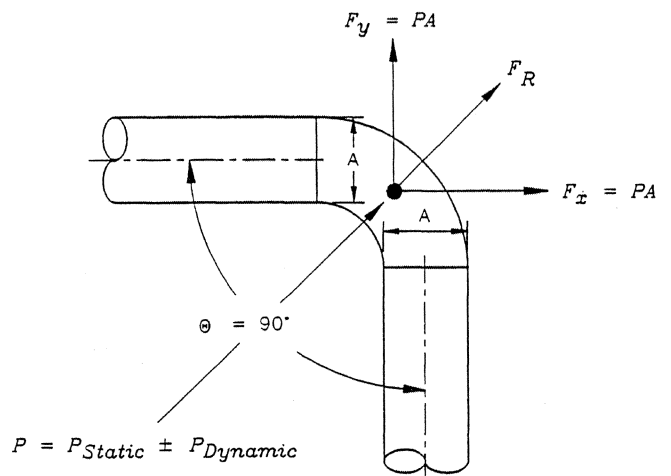
$$a = 12(gE/\rho)^{1/2} (1/(1+DE/tE_s))^{1/2} \tag{2}$$

where:

- g = acceleration of gravity, ft/sec<sup>2</sup>
- ρ = density of fluid, lb/ft<sup>3</sup>
- E = bulk modulus of fluid, lb/in<sup>2</sup>
- D = mean pipe diameter, in
- t = pipe wall thickness, in
- E<sub>s</sub> = elastic modulus of pipe wall, lb/in<sup>2</sup>

Note: The authors acknowledge that the quarter-wave resonance and acoustic velocity relations presented here are most properly applied to a Newtonian fluid, while medium consistency paper pulp is a complex, non-Newtonian material. However, by making corrections for density, temperature, and the effects of entrained air (as shown in APPENDIX B), these equations yield reasonable approximations.

Standing pressure waves are the result of reflected pressure pulses traveling back and forth along the pipe. A pulsation of 100 psi peak-to-peak is a pressure variation of ± 50 psi from line pressure. As the pressure wave passes an elbow, the 50 psi pressure wave acts on the projected area of the 20 in pipe elbow (299 sq in) and causes a force of about 15,000 lb [9]. This calculation is illustrated in Figure 12. The pressure wave may produce a “shock” on the piping system which can cause it to vibrate at its natural frequency, not necessarily at the major pulsation frequency. This effect may be seen in Figure 13 that shows that the major pulsation frequency (trace 1) and the pipe vibration (trace 2) occur at different frequencies. If the pulsation frequency matched the pipe mechanical resonant frequency, the vibrations could be much larger.



A = Projected Area

$$F_{Dynamic} = P_{Dynamic} A = F_x = F_y$$

$$F_R = 2 P_{Dynamic} A \cos(\theta/2)$$

Figure 12. Dynamic Force at Pipe Elbow.

Since “stick-slip” flow is a possibility in medium consistency pulp pumping systems at low, nonsteady flowrates [5], tests were made at net pipe flowrates of 0.34 ft/sec to 0.86 ft/sec and consistencies from 9.4 percent to 12.6 percent in an effort to

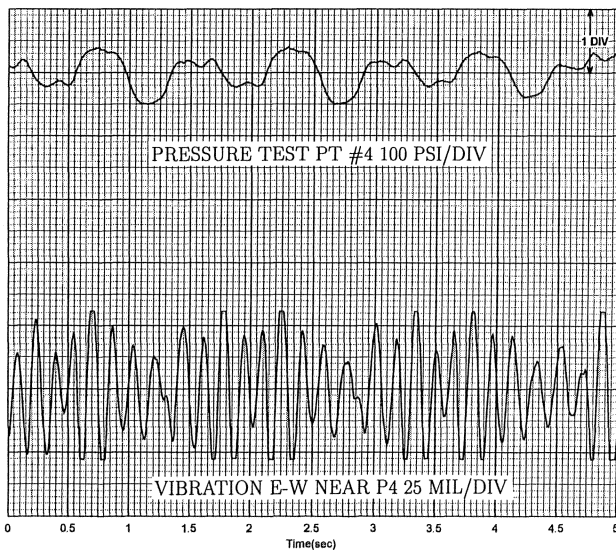


Figure 13. Pressure at P4 and Vibrations Near P4 vs Time.

confirm or refute this phenomenon as the cause of the pulsations. “Stick-slip” flow is a momentary stoppage of flow caused by the friction between the pulp and the pipe wall. The pressure pulsations were approximately 100 psi peak-to-peak, independent of these test conditions. Variations in the pipe flow velocity and stock consistency did not have a noticeable effect on the pulsation amplitudes. These flow velocities were above the threshold velocity for “stick-slip” flow for consistencies above 10 percent and the pressure pulsations were unchanged. If “stick-slip” flow were the cause, the pulsations would be significantly reduced above the threshold flowrate (0.3 ft./sec. for 11 percent consistency). On the basis of these data, “stick-slip” flow has been eliminated as the cause of the pulsations in this system.

During the testing, a modulation in the pulsation frequency and amplitude was observed. The pressure signal in Figure 9 shows frequencies varying from 0.8 Hz to 1.4 Hz. The frequency modulation repeated after 15 to 20 sec. This repeat rate correlated to the valve position that changed as much as 10 percent at times. This frequency and amplitude modulation was related to the control valve changing position in response to standpipe level control sensor signal to the control valve. The pulsation amplitude also varied from 80 psi to 110 psi peak-to-peak as the frequency changed. These data were the first to point to the control valve as the possible source of the pulsations since changes to the valve resulted in changes in the nature of the pressure pulses. As a result of these data, tests were performed, whereby the proportional gain of the valve was changed and the pulsation frequency and amplitude were observed, in order to further confirm the control valve operation as the source of the pulsations. The changes in amplitude of the major pulsation frequencies with different gain settings are illustrated in Figures 14, 15, and 16. The data indicated that the amplitudes and frequencies of the pulsations changed as the gain setting changed. The one-quarter and three-quarter standing wave frequencies were predominant, while a 0.5 Hz pulse was exhibited and was particularly strong at the lowest gain setting of 0.6. The lowest overall amplitudes were found to be at the (highest) 1.6 gain setting, but no gain setting would eliminate the pulsations. The change in preferential frequency with proportional gain indicated that the control valve was instrumental in determining the acoustical frequencies that are excited.

#### PROBLEM RESOLUTION

The most plausible hypothesis for the generation of the pressure pulsation is broadband turbulence generated as the pulp passes

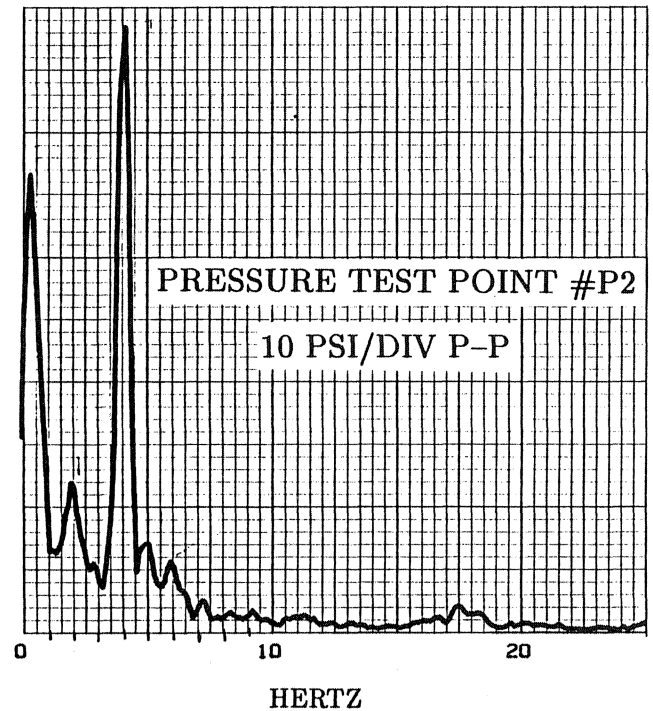


Figure 14. Pressure vs Frequency at P2–0.6 Proportional Gain on Control Valve.

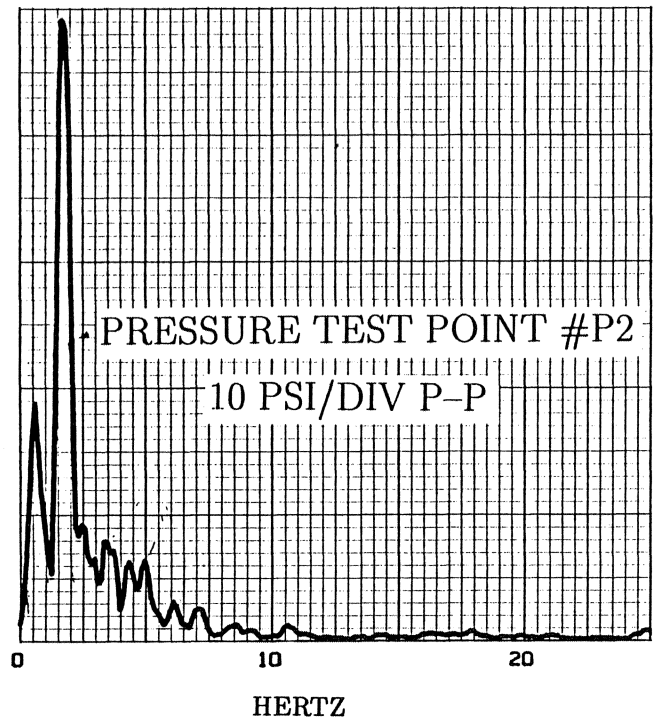


Figure 15. Pressure vs Frequency at P2–1.3 Proportional Gain on Control Valve.

through the restriction (V-notch) of the control valve. As stated by Blevins [6]: “Any structure with a sufficiently bluff (broad) trailing edge sheds vortices in a subsonic flow. The vortex streets tend to be very similar regardless of the tripping structure. Periodic forces on the structure are generated as the vortices are alternately shed

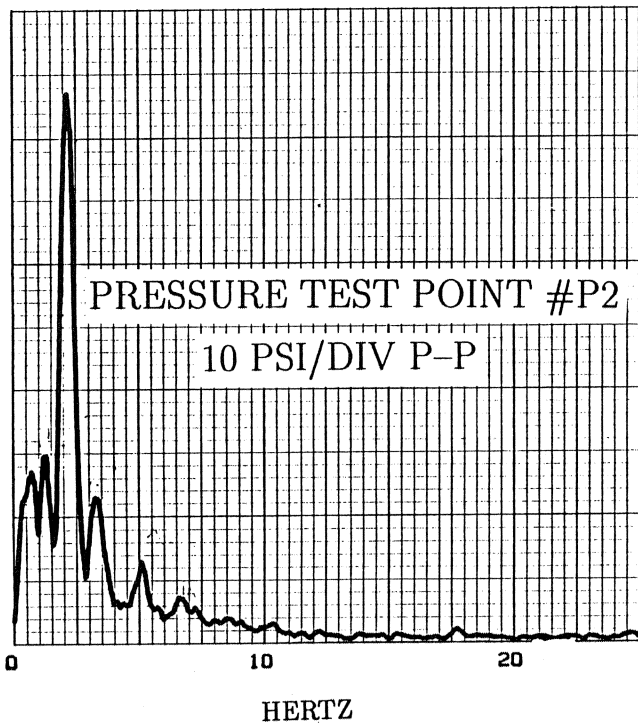


Figure 16. Pressure vs Frequency at P2-1.6 Proportional Gain on Control Valve.

from each side of the structure.” These relationships also hold true for restrictions in pipe flow. The dimensionless Strouhal number ( $S$ ) is the name given to the constant of proportionality between the vortex shedding frequency ( $f_s$ , Hz), and the free stream velocity ( $U$ , ft/sec) divided by the restriction width ( $D$ , ft) in the following relation:

$$f_s = S (U/D) \quad (3)$$

The Strouhal number,  $S$ , is a function of geometry (shape of the flow obstruction or restriction) and Reynolds number for low Mach number flows. Strouhal numbers have been analytically or experimentally determined for a number of geometric shapes such as circles (cylinders), squares, and triangles along with inline and staggered tube banks as may be found in a heat exchanger. They have also been determined for a number of structural shapes such as flanged beams, channels, and angles. Strouhal numbers for V-notch ball valves passing medium consistency pulp, however, have not been published.

At production rates of 360 ADST/D and 12.5 percent consistency, the control valve operates in the range of 30 percent to 40 percent open and has a large pressure drop (85 psi). For small openings, the V-notch ball acts as a restriction in the flow to generate periodic vortex patterns and turbulence. A small pressure disturbance in a liquid system can be amplified as much as 50 times or more if it is coincident with the acoustic resonant frequencies of the piping system. The hypothesis then is that the broad-band vortex shedding (Strouhal) frequencies of the control valve excited several modes (acoustic natural frequencies) or standing waves of the piping system resulting in a large amplification of the valve turbulence to as much as 110 psi peak-to-peak pressure pulsations. These pulsations then resulted in large piping system vibrations due to shaking forces at each elbow at widely distributed piping mechanical frequencies. Note also that the valve is at the antinode, which is the most efficient location for maximum pulsations for the quarter waves.

Based on the acoustic analyses of the piping system, variation of the control valve position changed the boundary condition at the valve and resulted in shifting the system acoustic natural frequencies which would explain the observed frequency modulation of Figure 9. The broad-band turbulence (Strouhal frequency) generated by the valve depends on the velocity through the restriction. Variation of the control valve position would affect the flow velocity at the V-notch which would have a minor effect on the Strouhal center frequency of the turbulence.

A number of design changes to the system were considered to eliminate the pulsations:

1. Reduce the size of the control valve and/or the speed of the pump in order to reduce the “energy level” of the vortices and/or change the Strouhal frequency spectra from the control valve. These changes could reduce the level of pulsations generated due to flow through the valve, thereby reducing the amplified acoustic resonance; particularly if the frequency spectra can be changed in this manner. These changes could, however, affect the pump output and impact the system requirements.
2. Attenuate the quarter-wave resonances and their multiples by the addition of a pressure drop at the outlet or open end of the pipe (the node) [7]. Using an acoustic analysis [9] with a normalized forcing function and the attenuation (damping), a parametric study of the effect of pressure drop at the outlet of the pipe system was performed. It showed that for a two psi drop, a reduction in the pressure pulsation from 50 psi zero-to-peak to 14 psi zero-to-peak could be expected. When the pressure drop was increased to 10 psi, the resonances were damped out. A 20 psi pressure drop at the outlet is equivalent to a closed-closed system and the half-wave resonances and their multiples were predicted.
3. Detune the system by changing the pipe diameter over a significant portion of its length. The effect on the acoustic resonant frequency by changing 350 ft of the piping to 40 in diameter was calculated and an acoustic analysis was performed similar to the pressure drop analysis. The results of this analysis indicated that lower pulsation amplitudes would result, but the resonant frequencies could not be moved completely out of the range of the frequencies generated by the control valve.
4. Reduce the amplitude of the vortices and/or change the Strouhal frequencies by modifying the geometry of the restriction in the flow; that is, the V-ball notch of the control valve. A change in the width of the restriction may change the Strouhal frequency sufficiently to prevent the resonance with the piping acoustic frequency. An orifice plate mounted in the pipe immediately after the valve may accomplish this by introducing additional losses which would act to open the valve.
5. Detune the system by changing the acoustic velocity of the medium consistency pulp. Since it is known that the addition of a gas into a liquid has a dramatic effect on its acoustic velocity, this could be done by introducing air or another suitable gas into the pulp in the pipeline downstream from the pump and control valve. This would have the additional benefit of reducing the system friction losses and, therefore, reducing total system energy. The disadvantage is that it introduces another system requirement (gas source) with its reliability and maintenance issues. The introduction of a gas into the pulp is not possible in every medium consistency system, as it would interfere with some bleaching processes.
6. Remove the V-notch control valve and use another method to control the flow, such as a variable speed motor drive. The drive would receive the same signal from the level sensor, but would advance or retard pump speed in response to that signal rather than restricting flow by partially closing the control valve. The removal of the level control valve would eliminate the pressure drop across

the valve, which should be beneficial by increasing overall system efficiency.

In determining the best methods from the above choices, numbers one and three were discounted as being either too invasive to the system operation or too costly. Number two (introduction of a pressure drop at the end of the pipeline) was tried, since the computer analysis indicated complete damping of the resonance if a 10 psi drop could be achieved. A 16 in V-notch gate valve was installed at the end of the pipe and pulsations were recorded while the valve was partially closed in steps. Unfortunately, only a small amount of pressure drop could be achieved due to fear of plugging this valve, due to the physical characteristics of medium consistency pulp. This may be a viable solution to problems of this sort that do not have the physical constraints of medium consistency pulp. Number four was also attempted (orifice plate after valve), but also failed to contribute significant pressure drop at the orifice diameters, which it was felt, could be safely applied with this pulp. A restriction after the control valve has been successful in controlling the pulsing at one other similar mill location.

Subsequent to these failed attempts to resolve the problem, an attempt was made to change the acoustic frequency of the piping system (item number five) by introducing a gas into the system, thus detuning the resonant acoustic frequencies of the piping from the forcing Strouhal frequencies at the valve. This resulted in some success, but was not considered by this system owner to be a viable permanent solution for reasons already mentioned. The before and after effects are shown in Figure 17 of injecting 0.66 acfm of nitrogen gas (approximately 1.15 percent by volume) into the piping immediately downstream from the control valve (test point location P2) while pumping 10.5 percent consistency pulp. The upper trace depicts the pressure pulsations vs time at test point location P2 before the addition of nitrogen. The pulsations before nitrogen was added were 100 psi to 120 psi peak-to-peak at a period of about one sec corresponding with the quarter-wave resonant frequency. The lower trace depicts the pressure pulsations vs time after the nitrogen had been added to the system for 30 min and was discharging at the pulp storage tower. As the nitrogen was being added, the period of the pulsation response progressively increased to about 10 sec, while the amplitude of the pulse was reduced to about 30 psi to 50 psi peak-to-peak. This did make a significant decrease in the piping vibration severity, but did not eliminate it altogether.

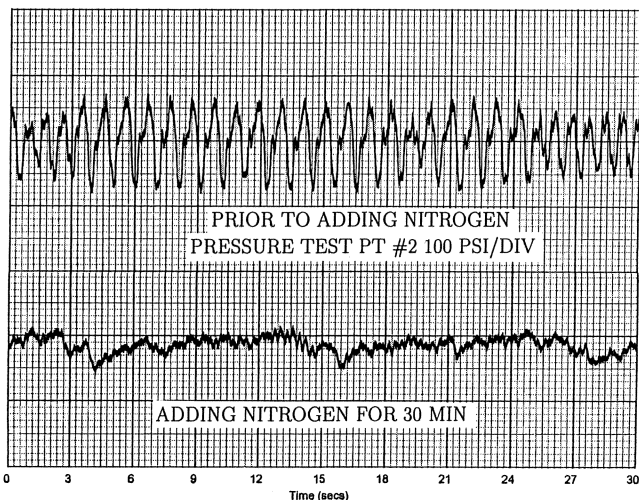


Figure 17. Pressure vs Time at P2 Before and After Injection of Gas Into Discharge Piping.

A special test was arranged by temporarily replacing the fixed-speed drive system with a variable frequency motor drive (VFD) and removing the control valve from the piping. Pulsation data were measured during a two hr test period, as pulp was pumped at consistencies ranging from 10 percent to 12 percent with a flow range of approximately 525 gpm to 425 gpm. Data recorded throughout this test included static and dynamic pressure, piping vibration, motor current, and pump speed. These data were then plotted vs time as shown in Figures 18 and 19. Note that with the VFD, the system experienced sudden pressure drops instead of pressure pulsations as with the control valve. The pressure drops were not as severe as the pressure pulsation with the control valve. Also, the period of the pressure drops was directly proportional to the ramp rate (the rate of change of motor speed increase) of the motor variable speed controller. A fairly short ramp rate is depicted in Figure 18, while a ramp rate about three times as long is depicted in Figure 19. The occurrence of these pressure drops is due to the fact that the variable speed drive ramp rate had not been tuned to respond properly to the level controller signal. It is also important to note that, with the variable speed control, the system acoustic resonance was changed because of the removal of the V-notch valve restriction. Consequently, the quarter wave acoustic frequencies of the piping were no longer being excited. The pulsation frequency response at the 1.3 Hz and 3.8 Hz harmonics were eliminated as shown in Figure 20. The elimination of the harmonic standing waves in the pipe significantly reduced the vibrations of the piping system and supporting structure and the harmful damage that was occurring. In conclusion, the removal of the control valve with its 85 psi pressure drop removed the harmful pulse from the system. Operation of the pump at lower speeds (below 2000 rpm) with the variable speed motor may have contributed to the improvement.

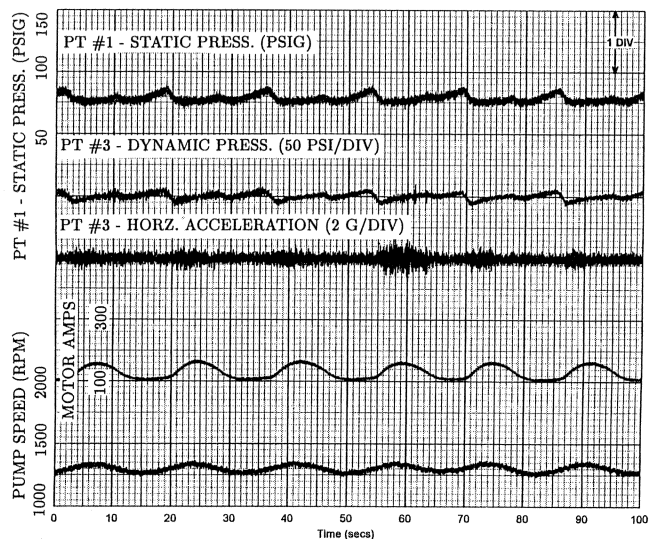


Figure 18. Effect of Replacing Control Valve with VFD: Static Pressure at P1, Dynamic Pressure and Acceleration at P3 vs Time for a Short Motor Ramp Rate.

Just prior to conducting the variable speed drive test, the piping system was flushed with water. A typical frequency spectrum obtained during this period is shown in Figure 21. At the time these data were recorded, the flow was approximately 750 gpm and the control valve was 80 percent open. It is interesting to note that the mode shape of the harmonic has changed to half-wave (3.5 Hz) and full-wave (7.1 Hz). This correlated with acoustic analyses of the piping system with the control valve open at a high percentage,



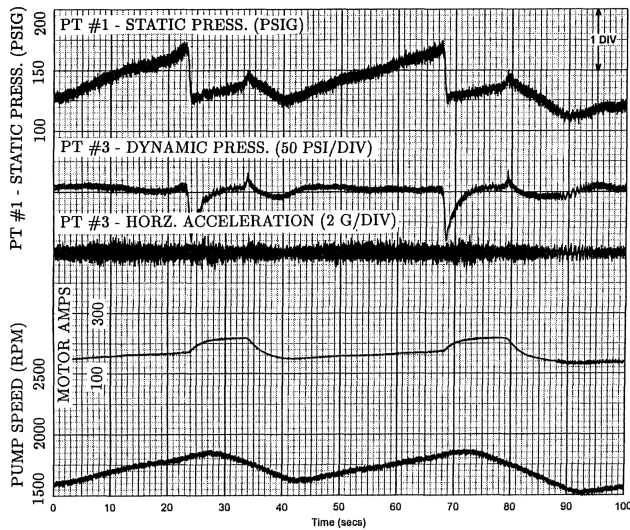


Figure 19. Effect of Replacing Control Valve with VFD: Static Pressure at P1, Dynamic Pressure and Acceleration at P3 vs Time for a Long Ramp Rate.

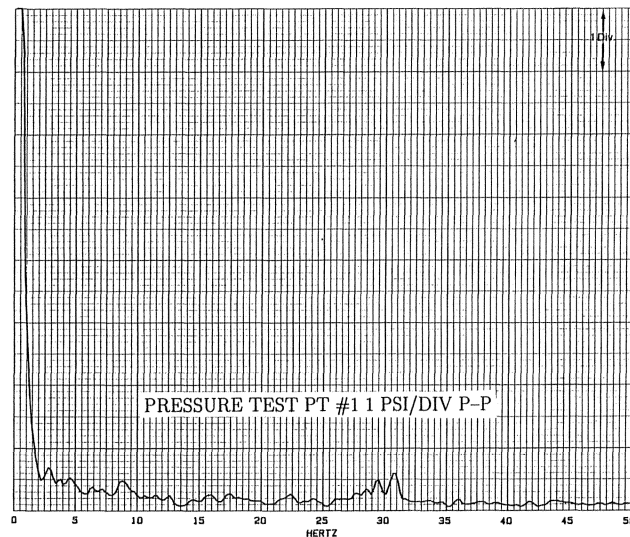


Figure 20. Effect of Replacing Control valve with VFD: Pressure at P1 vs Frequency.

which changed the boundary condition to an open-open pipe system. The amplitude of the pulsations varied while pumping water from three psi peak-to-peak to about 25 psi peak-to-peak depending on the valve position and motion.

## CONCLUSIONS

The following conclusions can be drawn from this case history of a medium consistency pulp pumping system:

- For this piping system with a control valve at the upstream end and an open discharge, the valve generated vorticity/turbulence at the Strouhal frequencies which excited quarter wave standing waves (dependent on the piping length). These acoustic resonances amplified the pressure pulsations and caused large piping vibrations. While a comparison of different control valve brands was not conducted as part of this case history, the authors are aware of several other medium consistency pumping systems which exhibit similar pressure pulsations, but that use different manufacturers V-ball or reduced-port ball control valves.

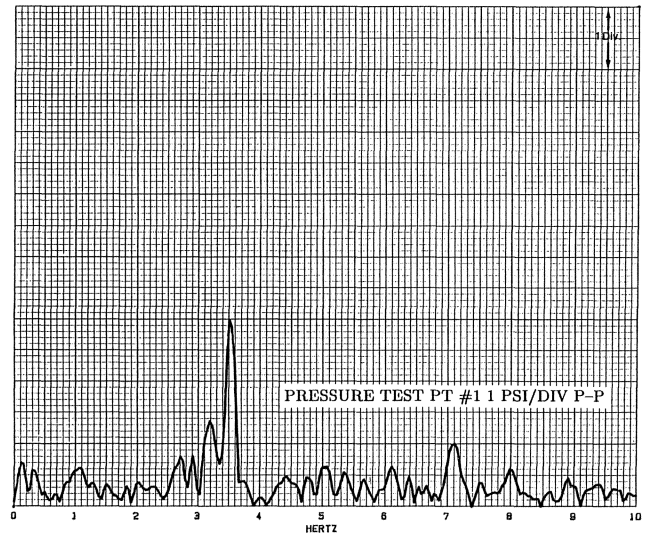


Figure 21. Pressure vs Frequency at P1 when Pumping Water.

- The “stick-slip” flow phenomenon was not the cause of pulsations in this medium consistency pulp centrifugal pumping system, nor was the centrifugal pump itself considered at fault.
- It is difficult to detune pulsing medium consistency pumping systems with traditional means of attenuation or flow manipulation because of the nature of the pulp and its propensity for plugging.
- Injection of air, or a suitable gas, into the pulp stream to lower the acoustic velocity reduced the amplitude of the pulsations and their frequencies, but did not eliminate them. Gas injection may, or may not be, a practical solution to problems of this type.
- Removal of the control valve from this system changed the resonant frequencies of the system, eliminated the exciting force, and minimized the excitation of the piping acoustic natural frequencies.

On the basis of these investigations, the owner of these medium consistency pump systems agreed that the best solution to this pulsation problem was to utilize a different control mechanism that did not rely on a control valve. Therefore, a variable frequency motor drive was installed in late 1995. The systems have been operating since that time at all flow and pulp consistency ranges without the harmful pulsations and vibrations caused by the standing waves.

## APPENDIX A

### Testing Procedures and Instrumentation

Several types of transducers were used to acquire the field data, including accelerometers, dynamic pressure transducers, and strain gauges. The data that were acquired during the field study are presented in frequency spectra and complex wave format.

#### Frequency Spectra

The transducer signal was processed with an FFT analyzer. The data were then plotted in the frequency domain, amplitude vs frequency. Peaks in the data typically indicate discrete frequency energy or harmonics of the signal.

#### Complex Wave Shape

The transducer signal was processed with an analog to digital converter in the computer. The time domain data were then plotted on the digital plotter.

Table A-1. Data Acquisition System.

| COMPONENT                             | DESCRIPTION / FUNCTION  |
|---------------------------------------|---|
| Austin Portable Microcomputer-486/100 | 100 MB hard disk, 2 MG RAM—acts as system controller to obtain and process data.  |
| HP 3582A—RTA                          | Two channel FFT analyzer, controlled via IEEE 488 bus.  |
| HP 7470A—Digital Plotte               | Two pen digital plotter for hardcopy, controlled via IEEE 488 bus.  |
| HP 2316B—Counter                      | Frequency counter to provide tachometer information. Controlled by computer via IEEE 488 bus.                                   |
| PCB/EDI 483A02                        | Six channel transducer amplifier for pressure transducers and accelerometers. Double integrators designed and installed by EDI. |
| TEAC XR-70—Data Recorder              | 21 channel FM tape recorder.  |
| Trans-Metric P27MH-500                | Pressure Transducer to measure static and dynamic pressure.   |
| PCB SH111A22                          | Pressure Transducer to measure dynamic pressure.  |
| PCB 308M218                           | Accelerometer to measure acceleration, integrated to displacement.  |

The data acquisition system was set up in the building near the east decker pump. The system included a portable microcomputer that controlled the instruments via an IEEE 488 interface bus. An FFT analyzer was used to obtain frequency analyses of the signals from the transducers. During the stages of the testing, a 21 channel FM recorder was used to obtain some of the data simultaneously. A digital plotter was used to generate hardcopy output. The primary components of the data acquisition system are described in Table 1.

Pump/System flowrates were determined from data provided from the mill control system (DCS). Field samples of pulp consistency were also obtained from the mill laboratory facilities.

## APPENDIX B

### Application Guidelines

While this pump user decided to install a permanent variable frequency drive system to resolve their pump system pulsation problem, the best way is, of course, to prevent such a system from being installed by predetermining the acoustic piping frequencies and valve Strouhal frequency and providing for a sufficient separation margin. The following application guidelines are intended to aid the reader in performing such a calculation:

The two formulae previously defined are used to determine the acoustic velocity of the piping system and the quarter wave resonance frequency.

$$a = 12(gE/\rho)^{1/2} (1/(1 + DE/tE_s))^{1/2} \quad (B-1)$$

and:

$$f = (2N - 1) (a/4L) \quad (B-2)$$

where:

N = 1, 2, 3... = harmonic

The second term in the equation for acoustic velocity is a modifying factor for the influence of the elasticity of the container (pipe) on the acoustic velocity that is also affected by the pulp air

content, pulp consistency (density or specific gravity), and temperature. The effect of each of these modifying factors on the acoustic velocity are discussed individually.

*Note:* The authors acknowledge that the quarter-wave and acoustic velocity relations presented here are most properly applied to a Newtonian fluid, while medium consistency paper pulp is a complex, non-Newtonian material. However, by making corrections for density, temperature, and the effects of entrained air these equations yield reasonable approximations.

### Variation of Acoustic Velocity with Air Content

Medium consistency centrifugal pumps do not, in general, remove all of the air from the pulp. A small amount is retained in the fiber network or perhaps in the fibers themselves. This air passes through the pump and enters the discharge pipe system. Based upon the acoustic velocity data from several mill sites, the graph in Figure 22 was developed showing the percent air remaining in the pulp after air removal in the pump at suction pressure conditions vs percent OD consistency. A graph is shown in Figure 23 [8] that can be used to correct the pipe system acoustic velocity once the percent entrained air is known. Note that entrained air has a significant effect on the acoustic velocity as 0.1 percent air by volume can halve the acoustic velocity. The percent air by volume from Figure 23 must be corrected to pipe system pressure before using Figure 22 to determine the  $K_a$  factor.

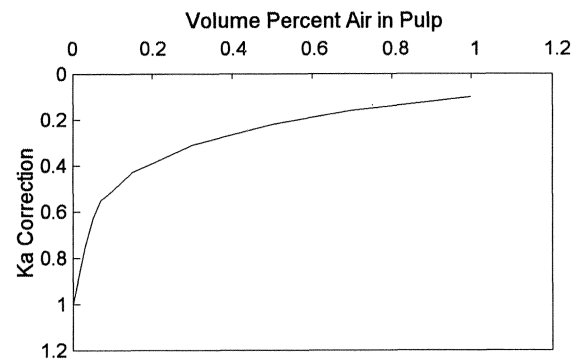


Figure 22.  $K_a$ -Correction to Acoustic Velocity for the Effect of Entrained Air.

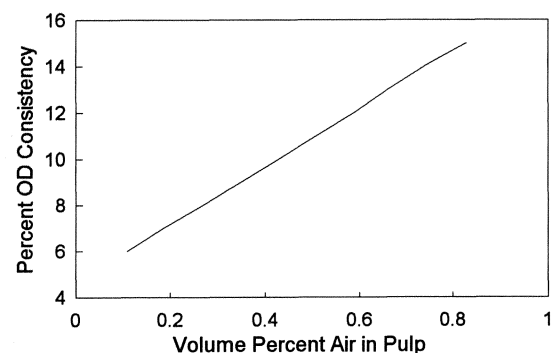


Figure 23. Retained Air Content in Medium Consistency Pulp After Air Removal—Softwood Kraft.

### Variation of Acoustic Velocity with Pulp Consistency

The acoustic velocity is inversely proportional to the square root of fluid density. As the consistency of pulp increases, its density does likewise. The following relationship can be used to calculate pulp density at various consistencies for use in the above equation for acoustic velocity.

$$\rho = 62.4 (1 + 0.004C) \quad (\text{B-3})$$

where:

C = percent OD consistency

$\rho$  = density lb/ft<sup>3</sup>

#### Variation of Acoustic Velocity with Temperature

The graph shown in Figure 24 relates the effect of temperature on acoustic velocity and is based on information contained in Wylie [8]. This graph can be used to determine the temperature correction factor ( $K_t$ ) at various temperatures common to medium consistency pulp processing.

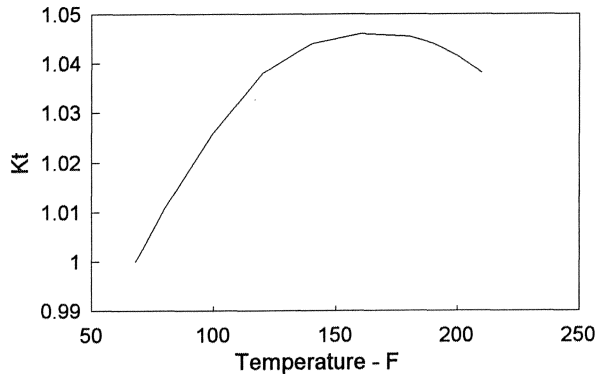


Figure 24.  $K_t$ -Correction to Acoustic Velocity for the Effect of Temperature.

#### Variation of Acoustic Velocity with Piping Geometry (Elasticity)

In noncompressible (liquid) systems, the flexibility of the pipe wall affects the apparent acoustic velocity of the liquid. The Korteweg correction factor ( $K_f$ ) can be applied to the acoustic velocity of the liquid to account for the pipe wall flexibility. This factor is limited to thin wall pipes where the wall thickness is less than 10 percent of the pipe diameter.

$$K_f = (1/(1 + DE/tE_s))^{1/2} \quad (\text{B-4})$$

where:

E = 318 × 10<sup>3</sup> psi, Bulk Modulus for water

E<sub>s</sub> = 29 × 10<sup>6</sup> psi, Elastic Modulus for stainless steel pipe

D = mean pipe diameter, in

t = pipe wall thickness, in

The equation for acoustic velocity can now be restated as:

$$a = 12 (gE/\rho)^{1/2} \times K_a \times K_t \times K_f \quad (\text{B-5})$$

The acoustic velocity in a medium consistency pulp pipe system should be determined by calculating the acoustic velocity in each pipe section (i.e., each horizontal run and each vertical run for each pipe diameter). After the acoustic velocity is known for each section, the period for each section can be determined by dividing the length of each section by the acoustic velocity for that section. The average acoustic velocity for the pipe system would be the total pipe length divided by the sum of the periods for each section. This average acoustic velocity is then used in the equation for quarter wave resonance using the total pipe length.

#### Control Valve Forcing (Strouhal) Frequency

The testing performed in this case history confirmed that the control valve was the source of the forcing frequency. It was also

confirmed that the center frequency of the broad-band turbulence generated by the level control valve follows the equation:

$$f_s = S (U/D) \quad (\text{B-6})$$

Typical values of Strouhal numbers for water from valve manufacturers range from 0.1 to 0.3. Based upon the testing conducted, a wider range of Strouhal numbers was found to exist for medium consistency pulp tending towards lower values. There are a number of variables that affect the calculations. It could be theorized that the Strouhal numbers for valves in medium consistency pulp service are significantly different than the common values because the Reynolds number for valves passing pulp are much lower than for valves in water service. (Note: The apparent viscosity of medium consistency pulp is much higher than water and increases with consistency) The authors could not find good data for Strouhal numbers that would apply to flow restrictions such as a V-ball or for any valve in pulp service. However, based on experience with several pulsing medium consistency piping systems, the predominant forcing frequency generally falls between one and two Hz.

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