PULSING AND VIBRATION CAUSED BY CONTROL VALVE INDUCED ACOUSTIC RESONANCE

ANDREW L. LEWIS AND DANIEL R. ROLL

DESIGN CONSIDERATIONS CAN LIMIT THE PROBABILITY OF PULSING IN NEW INSTALLATIONS AND CAN SOLVE PULSING PROBLEMS IN EXISTING INSTALLATIONS.

PULSATION AND VIBRATION FOR MEDIUM-CONSISTENCY pulp of 8–16% in pipe systems has become a common and serious problem. The pulsation forces in the pipe system can cause rupture and can weaken and damage the piping system supports. The most widely accepted cause of pulsation in medium-consistency pulp pipe systems has been "stick-slip" flow. An earlier work by V. B. Bodenhiemer in *Southern Pulp and Paper Manufacturer* described this as pulsing and plugging flow.

The authors reviewed pulsing and vibration in several medium-consistency pulp pipe systems and extensively analyzed a pipe system in an integrated kraft pulp mill in the southeastern United States. The most common cause of the pulsation and vibration was acoustic resonance. This produced large periodic pressures surges in the pipe system. This paper describes the pump system, the problem, and the resolution of the problem. It also presents application guidelines to help the pipe system designer when evaluating pipe systems for potential acoustic resonance with subsequent periodic pressure surges. There is also a discussion of possible solutions for existing pulsing pipelines.

HISTORY

In the past 15 years, the number of medium-consistency pipelines that pulse and vibrate has increased. They have become a major concern for designers and operators. To minimize the probability of pulsing and vibration in a pipe system, there has been more attention to the pulp flow velocity and the diameter of the discharge pipe. Specification of minimum-flow velocities and maximum pipe diameters have ensured that the pulp flow velocity and the diameter flow velocity agree with the diameter-to-velocity relationship predicted by the Bodenhiemer equations or charts.

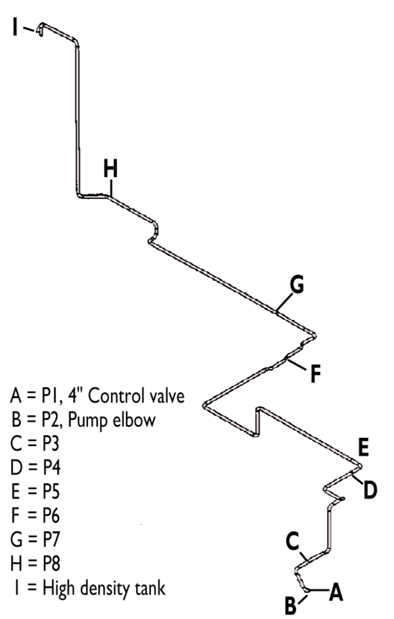
A review of many medium-consistency pulp pipe systems installed since the advent of centrifugal medium consistency shows that most existing systems are operating in a flow velocity or diameter-to-velocity range well below the minimum values at which pulsing or plugging flow would occur. Most systems do not pulse. Some systems that are operating at flow velocities above the "critical" flow velocity range do pulse and vibrate. All the systems reviewed that exhibit pulsing and vibration are delivering pulp to high-density storage tanks.

Review of the data showed that stick-slip flow does not always cause pulsing in medium- consistency pulp pipelines. Our company had an opportunity to study a pulsing and vibrating pipe system in an integrated kraft pulp mill in the southeastern United States in 1994 and early 1995. The results of this study showed that shedding downstream from a v-sector control valve excited the acoustic frequency of the pipe system. This caused a large-scale periodic pressure wave in the pipe system.

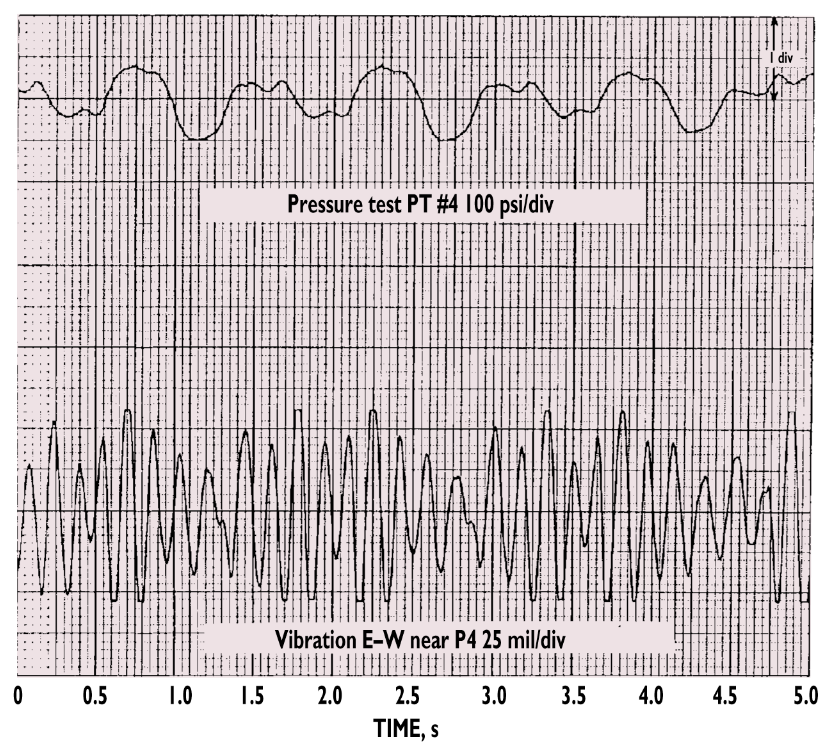
DESCRIPTION OF PUMP SYSTEM

The pump system in the mill under study can transfer 360 a.d. metric tons/day of southern pine kraft brown stock at 12% o.d. consistency from a decker through approximately 690 ft of 20- in. schedule 10-type 316 stainless steel pipe to a high-density storage tank of 130-ft height. **Figure 1** shows the pipe system that includes seventeen 90° elbows, thirteen 45° elbows, and one 30° elbow.

Pulp flows from the repulper of the decker to a small suction vessel usually called a standpipe. A commercially available medium-consistency stock pump bolts directly



1. Piping system diagram



2. Pulse frequency compared with pipe vibration frequency

to the standpipe to transfer the pulp to the high-density storage tank. The pump produced 670 ft of head at the rated flow and was driven at approximately 2500 rpm with a 4-pole motor and v-belt drive.

A 4-in. v-sector ball valve mounted directly to the discharge flange of the pump controls flow through the pump. The standpipe level controls the valve position to control the flow through the pump.

The discharge piping system was subject to large-scale pulsing and vibration at startup. Pulsing was present with water at low flow rates and with pulp over a consistency range of 6–12% o.d. and at flow velocities of 0.35–1.0 ft/s.

The range of pulse frequencies was 0.75–1.4 Hz. The predominant range was 1.0–1.25 Hz. These frequencies are well below any rotational frequency of the pump. The pump was therefore not the source of the pulsing. Initial studies of the system showed that the pulse would disappear or decrease with rapid changes in the level control valve position.

PUMP SYSTEM ANALYSIS

Initial testing to determine the source of the pulsation and find the problem revealed a large-scale pressure wave in the pipe system with peak-to-peak amplitudes as high as 110 psi. The pulses were predominantly at 1.25 Hz and 3.75 Hz. This is characteristic of a quarter-wave acoustic resonance in the pipe system.

Figure 1 shows the test stations for recording piping pressures as points A–H. Accelerometers at specific locations recorded the piping vibration.

A standing pressure wave in the piping system with a peak-to-peak amplitude of 110 psi can exert shaking forces as high as 18,000 lb_f acting at the 90° elbows near the source of the pulse.

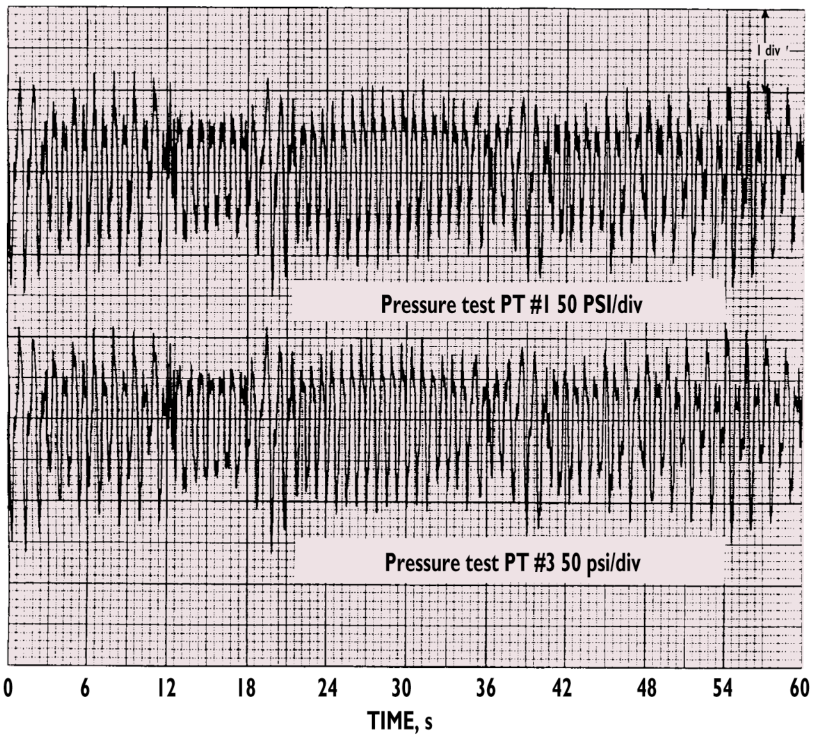
Figure 2 compares the pulse frequency with the pipe vibration frequency. It clearly shows that the forces in the pipe system created by the pressure wave caused the pipe system to vibrate at the pipe system's natural mechanical resonance frequency. Subsequent testing confirmed that the control valve was the source of the energy that excited the acoustic resonance of the pipe system to create the standing pressure wave.

The pulse frequency followed the relationship that predicts the middle-range frequency of the vortex shedding downstream of an obstruction in the flow path. The following equation shows this relationship:

$$f = SV/D \qquad (1)$$

where

f = *middle range frequency of vortex shedding by valve, Hz*
V = *flow velocity through the valve seat, ft/s*



3. Variation of pressure with time

D = valve closure width, ft
S = dimensionless constant— Strouhal number.

The Strouhal number, S, depends on the shape of the flow obstruction and the Reynolds number for low Mach number flows. Another work presents a detailed discussion of the testing and conclusions.

PROBLEM RESOLUTION

Attempts to reduce the pulsing with attenuators—valves and shear plates—at the end of the pipeline and immediately downstream of the level control valve were unsuccessful due to flow velocity changes in the pipe system. Adding nitrogen to the pulp stream immediately downstream of the valve with a second test injecting air approximately midway in the pipe system did provide some success. With the nitrogen and the air addition, the pulse levels decreased. Some pipe movement still occurred.

An experiment involved replacing the fixed-speed drive with a variable-frequency drive and removing the level control valve from the system. **Figure 3** shows the pressure test data before removal of the level control valve. **Figure 4** shows the pressure test data after removal of the control valve and installation of the variable- frequency drive. Comparing the figures shows that removal of the level control valve eliminated the high-amplitude periodic pressure pulses. This confirms that the level control valve was the source of the periodic energy that excited the acoustic resonance to create the standing pressure wave.

Note in Fig. 4 that the system experienced sudden periodic pressure drops at a very low frequency instead of the pressure pulsations experienced with the level control valve. The amplitude of the pressure change with the variable frequency drive was below 15 psi and did not cause any pipe vibration or movement. The period of the pressure drops was directly proportional to the ramp rate—rate of change of the motor speed increase—of the motor variable- speed controller. The pressure drops are a result of the variable-speed drive ramp rate not being properly tuned to the standpipe level controller signal. Following these tests and the success of the variable-frequency drive, the pump user installed a permanent variable-frequency drive on the pump system.

NEW PIPE SYSTEM GUIDELINES

The authors acknowledge that the following relations for the Strouhal frequency, quarter-wave resonance frequency, and the acoustic velocity more properly apply to a Newtonian fluid. Pulp is a non-Newtonian material. By correcting the acoustic velocity for specific gravity, temperature, and the effect of entrained air, the relationship provides a close approximation of the values for the acoustic velocity for pulp measured during the testing.

The published Strouhal values for water for a valve are 0.1–0.3. From testing and review of data from other systems that pulse, the Strouhal values for pulp passing through a v-notch ball valve are lower than the published water values.

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

$$a = 12(gE/\rho)^{1/2}\,[1/(1 + DE/tEs)]^{1/2} \quad (2)$$

where

a = velocity of sound in the fluid (acoustic velocity), ft/s
g = acceleration of gravity, ft/s^2

** = density of fluid, lb/ft^3*

E = bulk modulus of fluid, lb/in.^{2} (318 x 10^3 psi for water)

D = inside pipe diameter, in.

t = pipe wall thickness, in.

Es = elastic modulus of the pipe wall, lb/in.^{2} (29 x 10^6 psi for stainless steel pipe).

The second term in Eq. 2—the Korteweg correction factor, Kf, corrects for the elasticity of the pipe. The relationship for the quarter-wave resonance frequency is as follows:

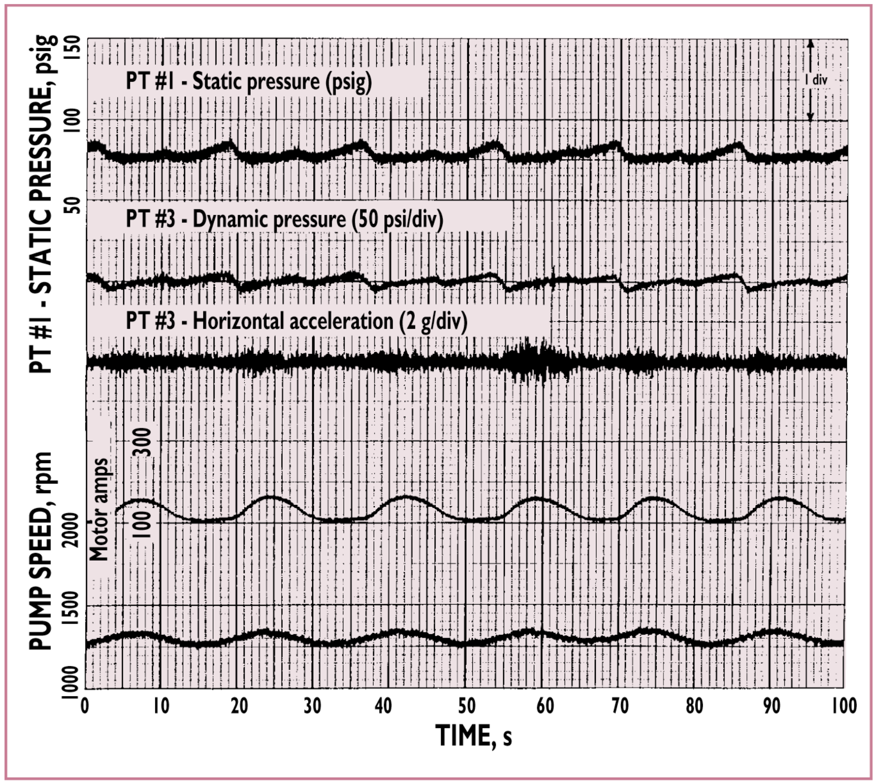
$$fa = (2N - 1)\,(A/4L) \qquad N = 1, 2, 3\ldots \text{hamonic} \qquad (3)$$

where

a = system acoustic velocity, ft/s

fa = quarter-wave resonance frequency, Hz

L = total pipe length, ft.



4. Pressure test data after removal of the control valve and installation of a variable frequency drive

Variation of acoustic velocity with air content

Medium-consistency centrifugal pumps often do not remove all the air from the pulp before the pulp discharges into the pipeline. **Figure 5** uses acoustic velocity data from several mill sites. The graph shows the volume percentage of air remaining in kraft brown stock after air removal in the pump at suction pressure conditions vs. percentage of o.d. consistency. The information in **Fig. 6** can determine Ka to correct the pipe system acoustic velocity after knowing the percentage of air at pipe system pressure. Note that entrained air significantly affects the acoustic velocity. Only 0.1% air can reduce the acoustic velocity by 50%.

Variation of acoustic velocity with pulp consistency

As Eq. 2 shows, the acoustic velocity is inversely proportional to the square root of the pulp density. The following relationship can calculate pulp density at various consistencies. (Using it in Eq. 2 can correct for pulp consistency.):

$$\rho = 62.4(1 + 0.004C) \qquad (4)$$

where

C = pulp consistency, % o.d.

ρ = density, lb/ft^3.

Variation of acoustic velocity with temperature

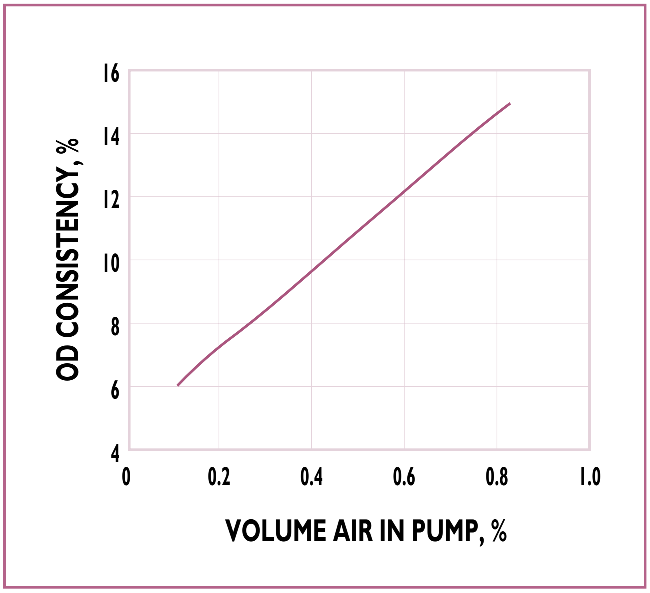
Figure 7 shows the effect of temperature on the acoustic velocity in a pipe system. This graph used earlier information. This information can determine the temperature correction factor, Kt, at the pipe system temperature.

Rewriting Eq. 2 to include Kf, Ka, and Kt gives the following:

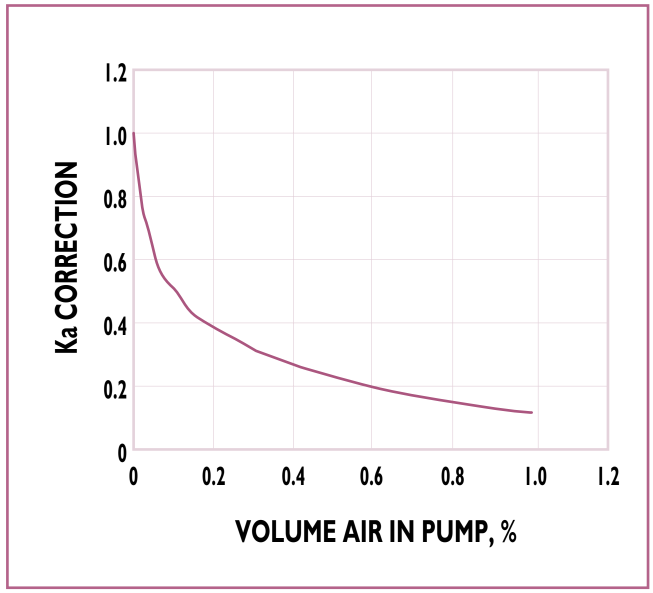
$$a = 12(gE/\rho)^{1/2}(KaKtKf) \qquad (5)$$

One can calculate the acoustic velocity in the pipe system by determining the acoustic velocity in each pipe section. Knowing the acoustic velocity for each pipe section, it is possible to calculate the period for each section by dividing length of each section by the acoustic velocity for each section. The average acoustic velocity for the complete system is the total pipe length divided by the sum of the periods for each section. Knowing the average acoustic velocity, one can calculate the acoustic quarter-wave frequency using Eq. 2.

Equation 1 can calculate the Strouhal frequency for



5. Retained air content in softwood kraft pulp after air removal



6. Ka correction to acoustic velocity for the effect of entrained air

the standpipe level control valve. It is possible to determine a good approximation of the Strouhal frequency by substituting the square root of the valve seat area minus the effective valve open area for D and using the published Strouhal values for water.

If the valve Strouhal frequency is less than 7 Hz and the pipe system acoustic resonance frequency is 1–3 Hz, there is a good probability that high-pressure periodic surges will be present in the pipe system.

The amplitude of the pressure surge depends on the pressure drop and the change in velocity through the valve and the "magnification factor." As the pressure drop and the change in velocity through the valve increases, the energy released by vortex shedding increases.

The following is the relationship for the "magnification factor":

$$M.F. = 1/[1 - (ff^2/fa^2)] \qquad (6)$$

where

M.F. = magnification factor
ff = forcing frequency, Hz
fa = acoustic resonance frequency, Hz.

As the forcing frequency approaches the acoustic frequency, the magnification factor can become very high. It becomes infinite if the forcing frequency equals the acoustic frequency. Inherent damping usually existing in the pipe system will reduce the magnification factor. This work estimated that the magnification factors in the systems studied were 8–12. Inherent damping in the pipe system will account for these lower than expected values. One can expect damping because all the systems reviewed used a minimum 1:2.5 ratio to increase the piping from valve diameter to pipeline diameter. Increasers or reducers can attenuate or reduce the amplitude of the reflected wave.

Some pipe systems with pressure drops of 3–10 psi will also pulse if the flow velocity through the valve seat is lower than 10 ft/s. In such cases, the middle-range Strouhal frequency is usually below 7 Hz.

DESIGN CONSIDERATIONS

Piping design and the piping support system can significantly reduce the damage from pressure pulses in medium-consistency pipe systems. The authors have reviewed one pipe system that had 100 psi peak-to-peak pressure pulses in the system. There was no visible pipe movement. For the piping design, increasing the pipe wall thickness will result in an increase in the acoustic velocity with a concomitant increase in the acoustic frequency. Increasing the pipe diameter will lower the acoustic velocity and reduce the acoustic frequency. Either consideration could change the acoustic resonance frequency so it would be outside the critical range of the forcing frequency. This would reduce the amplitude of the pressure wave.

Using two or three different pipe diameters in the pipe systems could provide for attenuation of the pressure wave through the reducers and increasers. This could change the acoustic frequency so it would be outside the critical range of the forcing frequency.

SUGGESTIONS FOR REDUCING PULSATION LEVELS IN EXISTING PIPE SYSTEMS

The following suggestions may not provide the solution to pulsing pipe and vibrating pipe systems for all cases. Reviewing the pipe system and test data may be necessary to decide the best solution for a particular pipe system.

Installation of a valve or orifice at the end of pipe system

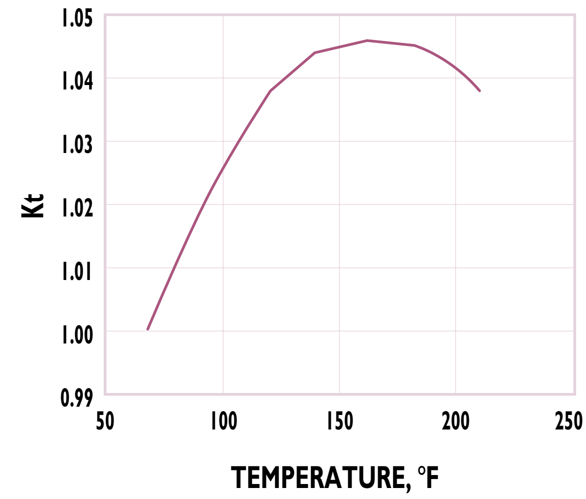
In some locations, the addition of a valve or orifice near the end of the pipe run has successfully attenuated the pressure pulse generated by a standing wave. One must size the valve or orifice to provide a damping effect that depends on the pressure drop across the valve or orifice and the flow velocity through the valve opening or the orifice. The addition of a valve is often preferable because one can modulate the valve opening to handle different flow rates. If the feed to the pipe system is erratic and there is a flow velocity variation in the pipe system, then the valve at the end of the pipeline may not be a successful solution.

Addition of a valve downstream of the level control valve

Installing a valve 1–2 ft downstream of the level control valve has reduced pulsations in existing pipe systems to acceptable levels. The second valve reduces the pressure drop across the level control valve to reduce the amplitude of the forcing frequency. Another benefit of the valve downstream of the level control valve is that the amplitude of the reflected wave can be attenuated as it passes through the valve to reduce the magnification factor of the pressure pulse. This option does not fit all systems. One must size the valve to meet the specific requirements of the system.

Addition of air to the pipe system

Addition of air to the pipe system can reduce pulsations to acceptable levels. Air entrainment reduces the acoustic frequency of the pipe system by reducing the acoustic velocity of the system. As the acoustic frequency moves away from the forcing frequency, the magnification factor decreases. This reduces the amplitude of the pressure surge. Caution is necessary when injecting air into the pipe system. If there is too much air addition, the point of air addition may behave as an open end. This could cause acoustic resonance to occur between the point of air addition and the source of the pulse energy. Air addition should be at several locations circumferentially around the pipe. The recommended volume of air addition is approximately 1% by volume at the pipe system pressure and temperature.



7. Kt correction to acoustic velocity for temperature

Note that the addition of air will reduce the piping friction loss by approximately 20% from the point of air addition to the end of the pipe. This reduction of friction loss will result in higher pressure drops across the control valve. The pulse amplitude may increase, and the pulse frequency may decrease. One should also determine the acoustic frequency of the system as noted earlier to be certain that the addition of air will reduce the pulsation levels.

Dilution downstream of the level control valve

The addition of dilution after the control valve will cause the acoustic velocity and the acoustic frequency of the system to increase. Dilution reduces the friction loss in the pipe system. This increases the pressure drop across the control valve. At the smaller valve opening, the frequency of the downstream disturbance will decrease. The separation of the acoustic frequency from the forcing frequency can cause a reduction in the amplitude of the pressure pulse. Dilution will increase the flow velocity downstream of the valve. This can attenuate the amplitude of the reflected wave to reduce the overall pulse levels in the pipe system. The usual dilution required is 30–50%.

Installation of a variable-frequency drive

Removal of the level control valve from the system and replacing it with a variable-speed control to control the standpipe level eliminates the source of the forcing energy that excites the acoustic resonance of the system. This eliminates the large periodic pressure surges in the pipe system.

One should consider variable-frequency drives in applications that have several different operating requirements. Besides reducing the risk of pulsing pipe systems, substantial power savings can accrue.

CONCLUSION

All medium-consistency pulp transfer systems that deliver pulp to high-density storage and have devices to restrict or control the pulp flow can have low-frequency pressure disturbances downstream of the restriction. There is also a probability that acoustic resonance will occur and that high-amplitude periodic pressure surges can be present in the system.

The system supplier and the pipe system support designer must work together to minimize the effect of acoustic resonance so the final system design does not have pressure pulsations that could damage the piping and support structure. **TJ**

Lewis is technical product specialist and Roll is project manager at Goulds Pumps Inc., 240 Fall St., Seneca Falls, NY 13148.