Control-valve-induced Pipeline Vibration Corrected By Variable-speed Pumping

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During startup and commissioning of two stock pumps in a pulp mill, violent vibrations took place in the associated piping systems. Figure 1 shows one of the two pumps as installed. Over time, the piping and its supporting structures sustained major damages, 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. These events resulted in production delays and safety issues which were of great concern to the owner of the systems. Initial analysis indicated that the pumps were performing properly, so attention turned to the piping system. Several modifications were made to better anchor the pipe and stiffen its support structures, but no significant reduction in the vibrations resulted.

A field test and analysis of the nature of the vibrations was undertaken to determine their source. The pulsation frequencies measured were well below any of the normal pump-related frequencies so the pump was not considered to be the source of the problem. The pulsation at vane passing frequency was measured at 1 psi peak-peak (approximately one-third of one percent of developed head) which is normal for a pump of this type.

The most plausible hypothesis for the pressure pulsation is broad-band turbulence generated as the pulp passes through the restriction (V-notch) of the control valve. In a flow stream, any structure with a sufficiently broad trailing edge sheds vortices in a subsonic flow. Periodic forces on the structure are generated as the vortices are alternately shed from each side. These relationships hold true for restrictions in pipe flow. The dimensionless Strouhal number (S) is the name

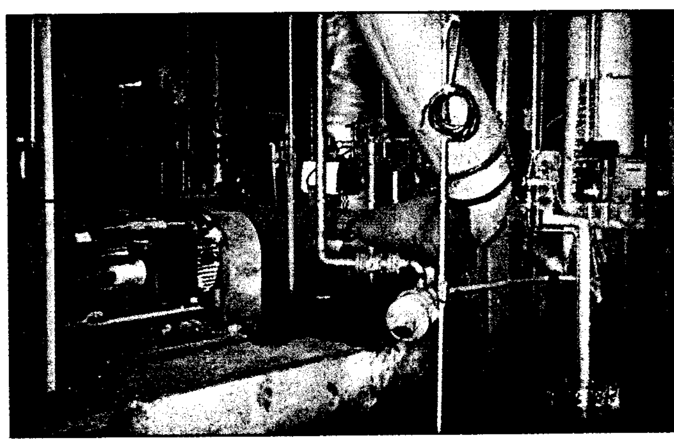

Figure 1. Stock pump, standpipe, control valve and discharge pipe.

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 relation:

$$f_s = S\left(\frac{U}{D}\right)$$

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 as well as for in-line 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 normal production rates and 12.5% consistency pulp stock, the control valve operated in the range of 30% to 40% open with 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 frequencies of the control valve excited several acoustic natural frequencies of the piping system, resulting in a large amplification of the valve turbulence to as much as 110 psi peak-peak pressure pulsations (Fig. 2). These pulsations generated forces as high as 30,000 lb peak-peak at the elbows, causing large vibrations at widely distributed structural natural frequencies.

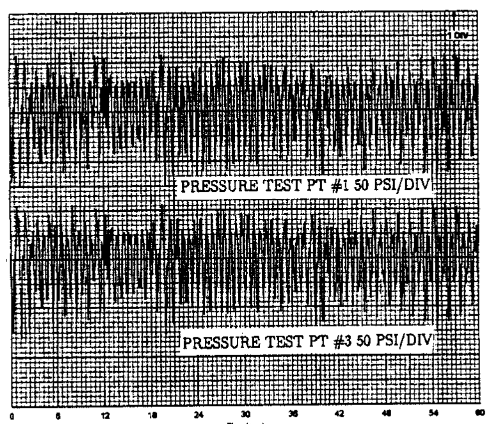

Figure 2. Pressure vs time at point 1, downstream of the control valve; and point 3, downstream at the first piping elbow.

A number of design changes to the system were considered:

1. Reduce the size of the control valve or the speed of the pump in order to reduce the energy level of the vortices and change the Strouhal frequency spectra from the control valve. These changes could reduce the level of pulsations 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. Using an acoustic analysis with a normalized forcing function and the attenuation (damping), a study of the effect of pressure drop at the outlet of the pipe system was performed. It showed that for a 2-psi drop, a reduction in the pressure pulsation from 50 psi zero-peak to 14 psi zero-peak could be expected. When the pressure drop was increased to 10 psi, the resonances were damped out.
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 inch 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 or change the 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 might change the frequency sufficiently to prevent the resonance with the piping acoustic frequency. An orifice plate mounted in the pipe immediately after the valve could 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 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 a system requirement (gas source) with reliability and maintenance issues.
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 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 1 and 3 were discounted as being either too invasive to the system operation or too costly. Number 2 (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-inch 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 characteristics of medium consistency pulp. This may be a viable solution to problems of this sort which do not have the physical constraints of medium consistency pulp. Number 4 was 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 pulsing at another 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 (number 5) by introducing a gas into the system, thus detuning the resonant acoustic frequencies of the piping from the forcing frequencies at the valve. This was partially successful, but was not considered to be a permanent solution because plant management did not want to introduce the requirement of maintaining a gas source.

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 was measured during a 2-hour test period, as pulp was pumped at consistencies ranging from 10% to 12% 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. With the VFD, the system experienced sudden, but less severe, pressure drops instead of pressure pulsations as with the control valve (Fig. 3). The period of the pressure drops was directly proportional to the ramp rate (the rate of change of motor speed).

The occurrence of these pressure drops resulted from 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. The elimination of the harmonic standing waves in the pipe significantly reduced the vibrations and the harmful damage that had been occurring. The net result was that the removal of the control valve with its 85 psi pressure drop eliminated 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.

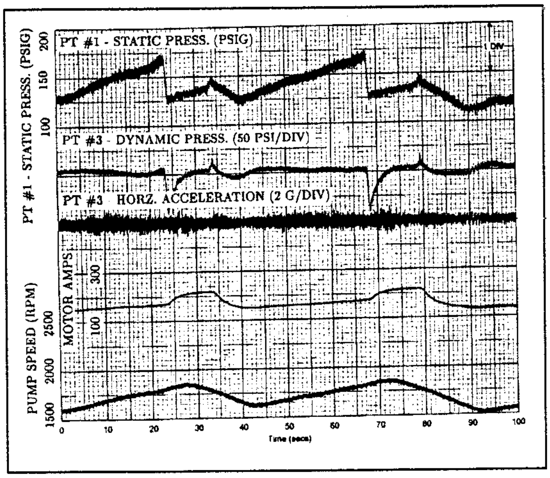

Figure 3. Effect of replacing control valve with variable-speed pump: static pressure at Point 1, dynamic pressure and acceleration at Point 3 vs time.

CONCLUSIONS

The following conclusions can be drawn from this case history:

1. 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.
2. 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.
3. Injection of air or other 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.
4. 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 use 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. **PT**

Copies of the complete paper are available – ITT Goulds Pumps, Seneca Falls, NY. **Circle 327**

■ For information about on-site diagnosis of vibration problems in pumping systems – Engineering Dynamics, San Antonio, TX. **Circle 328**

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