Controlling Noise and Vibration from a Pipeline Pump Station

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The amount of shale oil extracted increases every year, and the amount transported has increased proportionally. This increase in demand on the transportation system has resulted in new pipeline installations and repurposing of existing lines to carry crude and refined products. Some of the first pipeline right-of-ways date back before WW II and were typically located in rural areas, which have now been suburbanized. This article will chronicle the problems encountered with one such repurposed pipeline, its surrounding residential homes and noise pollution, which provoked the ire of the residents resulting in a barrage of grievances.

Most pipeline pump stations are remote, with any noise generated typically by the drive motors and pumps, sufficiently attenuated before reaching populated areas. This particular station had been in and out of service for many years. Over those years, the population density around it increased. In 2011, the line and all the pumping stations were revamped to handle light petroleum products that required new variable speed motor/pumps and additional above-grade piping.

Since the upgrades, the surrounding residents have complained of excessive noise from the station, described as an "annoying high-pitched hum." Even though the overall noise amplitudes were within local code requirements of 85 dBA at ground level, complaints still abounded. The noise emitting from the station continued to vex residents to the point of threating legal action.

Vane-Pass and Pump Design

Typically, the most significant generators of noise during operation at any pump station are those emissions from drive motors and pumps, which in this instance are located inside a concrete block pump house (Figure 1).

All the pumps on this line are a typical multistage, dual-volute horizontal design (Figure 2), but with an atypical design feature. The suction (first stage) and fourth-stage impellers have six vanes (Figure 3) and the remaining stages have seven-vane impellers. Typically, vane-pass frequencies dominate the vibration signatures manifesting as structural vibration and acoustic emissions. At this station, it was the acoustics that troubled the neighbors.

Atypical Design

What makes this pump design atypical is the dual-volute design and impellers with even-numbered vane counts. Typically, impellers with an even number of vane counts are installed in single-volute pumps (Figure 4). The reason is to minimize the impeller-generated pressure pulsation energies referred to as vane pass frequencies.

In an even-numbered impeller-vane/dual-volute combination (Figure 5), the pulsations occur twice per revolution as opposed to odd-number impeller vane/dual-volute combinations, where the pressure pulsations occur once every revolution. For example, a six-vane impeller in a single-volute pump will generate six vane-pass pulsations per revolution (6×RPM). However, the same impeller in a dual-volute pump will still generate 6×RPM vane-pass pulsations but at twice the energy level for each revolution.

The vane-pass pulsation frequencies are intermixed with the general hydraulic and machinery noises. It is not uncommon for the vane-pass components to dominate the noise spectrum and manifest as structural vibration sources or acoustic emissions. As previously noted, in this particular case, it wasn't structural, but acoustics, that were the problem.

Investigation

The investigation began with determining the mechanical and hydraulic condition of the pump and current operating loads. Hydraulic noise and vane-pass pressure pulsation levels can be affected by the pump load. Therefore, it's important to know where the pump is operating with respect to its best-efficiency point (BEP) or head and flow, where the pump is most efficient.

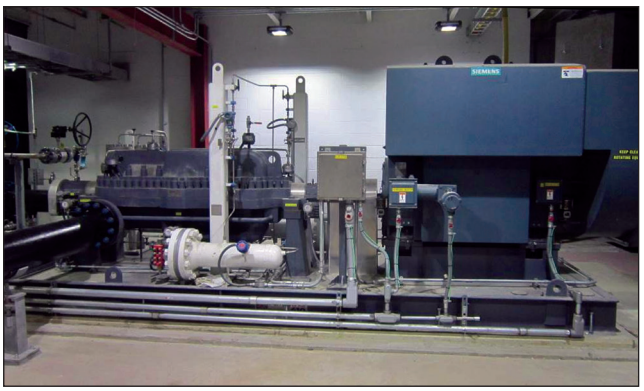

Figure 1. 1500-HP, horizontal, nine-stage centrifugal pipeline motor and pump.

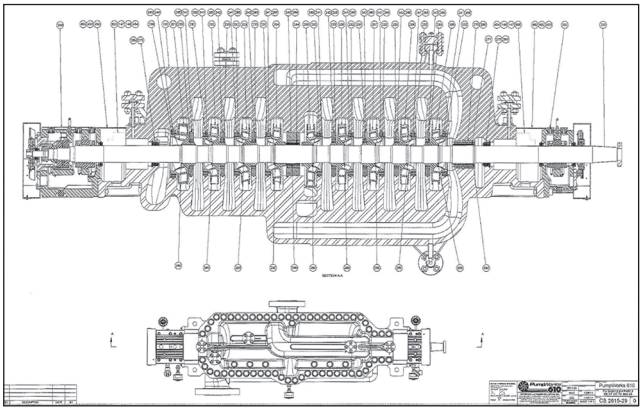

Figure 2. Cross section of subject pump.

The measured hydraulic data were reviewed and compared to the pump's performance curves indicated the pump was operating within 86% of the BEP at 2,500 RPM. The vane-pass amplitudes being generated were going to be the norm, since operators informed us that this was a nominal load point.

Seismic vibration signatures were obtained in three XYZ planes from the pump drive-end (DE) bearing housing (PIH, PIV, PIA) and non-drive end (NDE) bearing housing (POH, POV, POA) planes. The unfiltered overall vibration velocities (Figure 6) were higher than the two discrete frequencies of interest, 6×RPM and 7×RPM vane-pass velocities, as expected. However surprisingly, the relatively low velocities of the vane-pass frequencies, in particular, the 6×RPM vane-pass components, were unexpected.

As expected, the overall noise level generated by the drive motor and pump within the pump house was 110-120 dBA; warranting hearing protection. Outside the pump house, a survey of the piping yard with a sound level meter indicated a persistent average noise level of 103 dBA emanating from the area of the discharge piping near the pump house (Figure 7) at points D1, D2, and D3. With the pump operating at 2500 RPM, seismic vibration signatures were recorded from the piping to pinpoint the source; vibration signatures were recorded at five locations on the discharge piping (Figure 7) and four on the suction piping (Figure 8).

A comparative review of the vibration velocities recorded from

Figure 3. Centrifugal six-vane impeller.

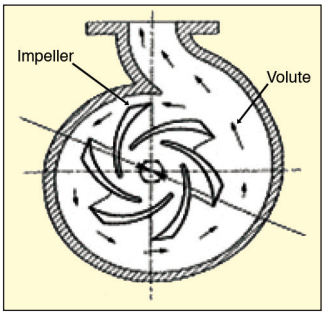

Figure 4. Centrifugal six-vane impeller in single-volute pump.

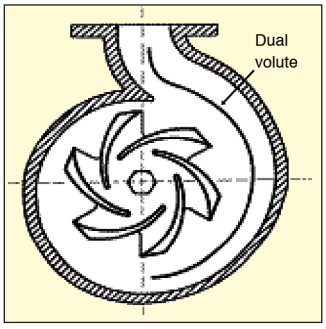

Figure 5. Centrifugal six-vane impeller in dual-volute pump.

the pump to those recorded from the discharge piping are presented in Figure 9 and show an unexpected amplification of the overall values. Even more interesting, the amplification of the 6×RPM component at the D1 and D3 locations. A review of the associated D1 and D3 spectra (Figure 10) confirms the dominance of the 6×RPM vane-pass component.

Similarly, a comparative review of the velocities recorded from the pump to those recorded from the suction piping are presented in Figure 11, again revealing amplification of the overall values at locations S1-S3 and of the 6×RPM component at locations S1 and S3. No amplification was indicated at S4. Similar to the discharge piping spectra, the associated S1 and S2 spectra (Figure 12) confirmed the dominance of the 6×RPM vane-pass component.

Acoustic Resonance

It is not uncommon in refined petroleum product pipeline systems for a section of piping to be of the proper dimension and containing the proper fluid density to result in the creation of a standing wave – typically at vane-pass frequencies. Based on experience, it was first thought that the amplification of the vane-pass energy at 6×RPM noted in the seismic spectra was the result of a resonant standing wave at 250 Hz in the discharge pipe section at point D3 (Figure 13). The general formula for calculating the resonant frequencies of sine waves that will resonate in a pipe closed at both ends is:

$$F_n = \frac{nc}{2L} \qquad (1)$$

where n is the resonant frequency harmonic multiple, Hz; c is the speed of sound in the medium, m/s; and L is the pipe length, m.

The medium was a pressurized (1466 psi) mixture of 60% ethylene and 40% propane. The speed of sound in the medium was measured using an ultrasonic flow meter at approximately 500 m/s. However, for the suspect 1.22-meter section of pipe (D3), the lowest

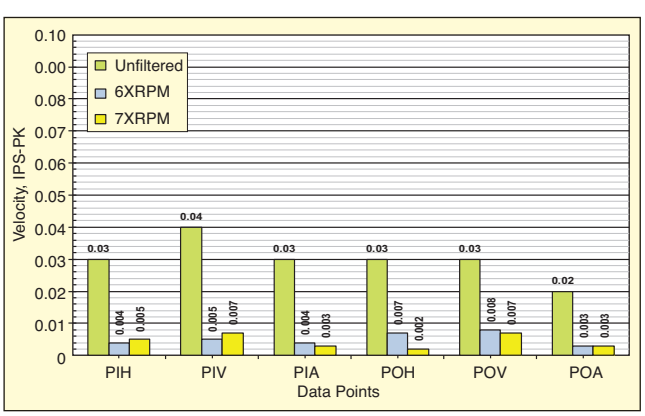

Figure 6. Vibration velocity measured from pipeline pump.

Figure 7. Discharge piping vibration data points.

Figure 8. Suction piping vibration data points.

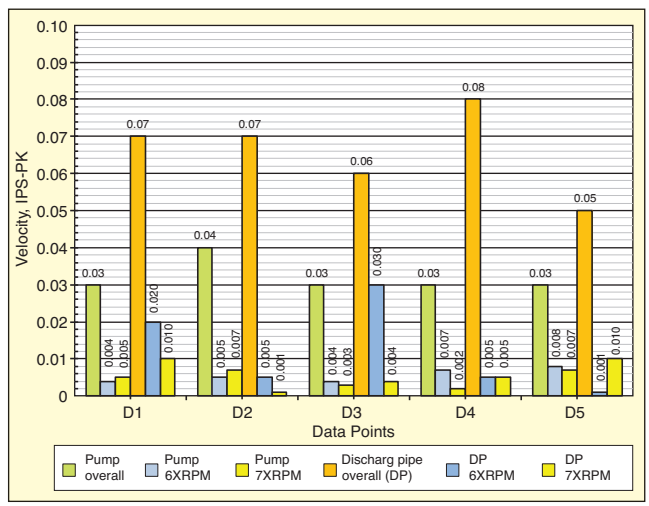

Figure 9. Discharge pipe vibration amplitudes compared to pump-generated amplitudes.

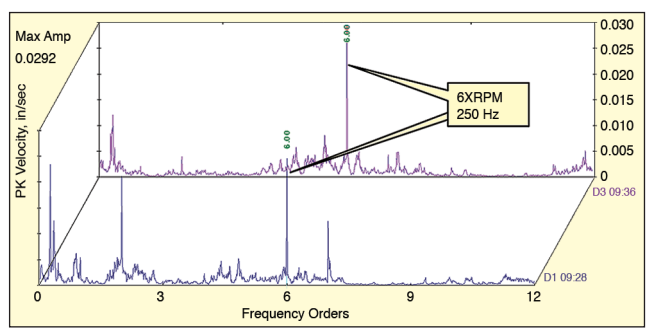

Figure 10. D1 and D3 spectra revealing dominance of 6×RPM vane-pass components

frequency standing wave that could be supported was 204 Hz, not the measured 250 Hz. The standing-wave theory became even more questionable, since there was also indication of amplification of the same frequency in the longer suction piping runs at S1 (2.44 meters) and S3 (3.65 meters).

Later that afternoon, to satisfy production requirements, the pump speed was increased from the 2513 RPM to 2802 RPM, increasing the 6×RPM vane-pass frequency from an average 250 Hz to 280 Hz. Additionally, the average noise levels increased from

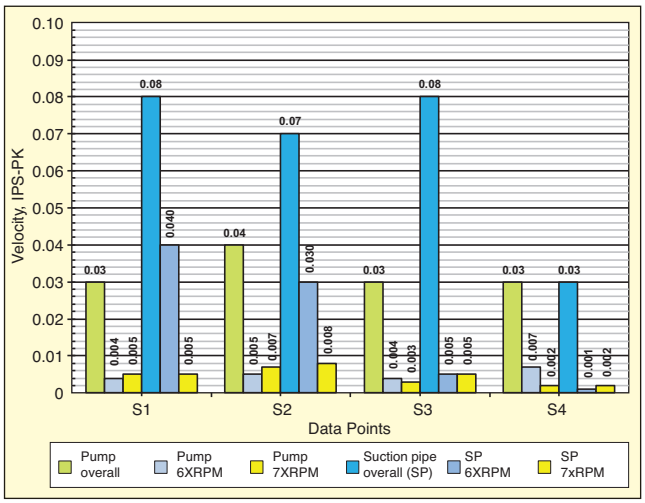

Figure 11. Suction pipe vibration amplitudes compared to pump-generated amplitudes.

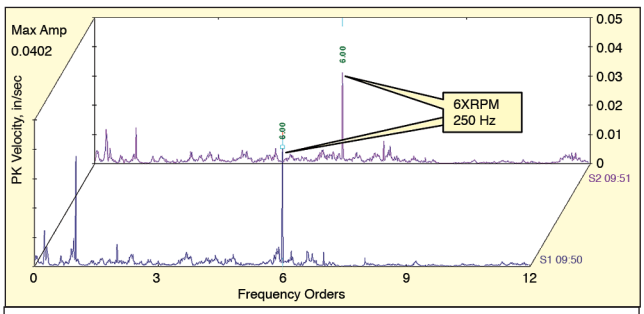

Figure 12. S1 and S2 spectra revealing dominance of 6×RPM vane-pass components.

Figure 13. Section of discharge piping where standing wave was suspected.

89 dBA to 102 dBA. The combination of location and these events made the tuned-resonance, acoustic, standing-wave theory moot.

Generated hydraulic noise pressure waves emanating from the pump consisted of a broad spectrum of frequency nodes (zero pressure) and antinodes (highest pressure). If at the end of the pipe there is an area of high impedance, such the control valve in this case, some of those waves will reflect back on themselves amplifying the sound pressure modulating the pipe walls. The seismic vibration signatures verified this mechanical movement of the piping walls.

Transmission of Noise

To conduct a more focused analysis of the noise emanations from the piping, a sound analyzer with a frequency response range of 100-4000 Hz was used to measure acoustic signatures in the pipe yard at the locations identified in Figure 14 at two pump operating speeds of 2513 and 2802 RPM.

As noted in Figure 15, an increase in the measured sound

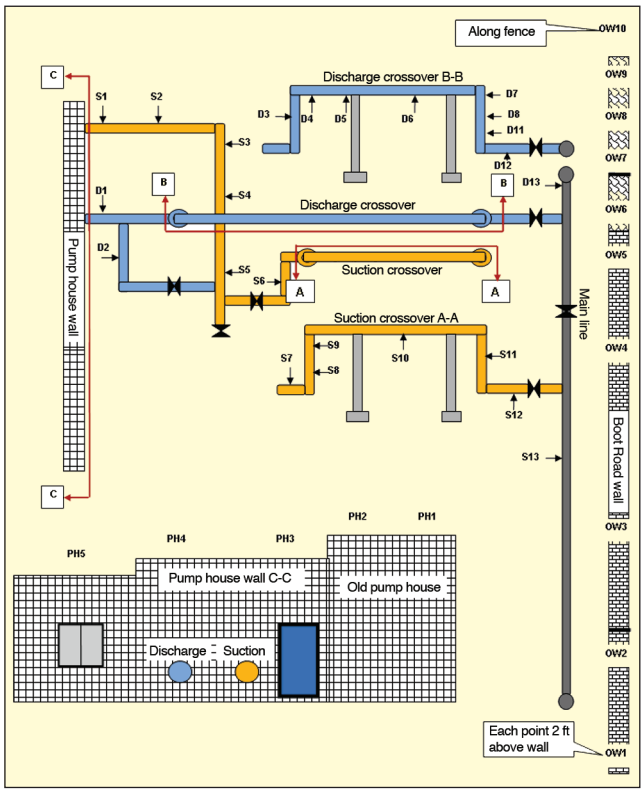

Figure 14. Sound signature collection data points.

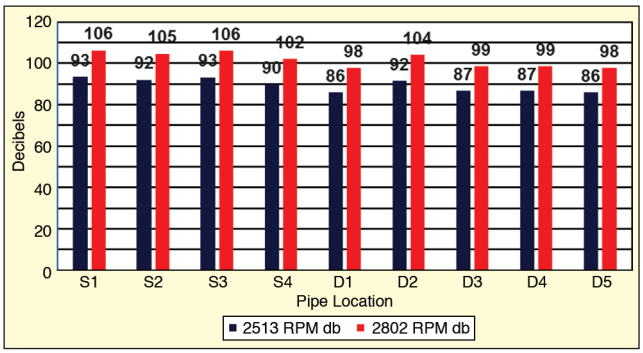

Figure 15. Comparison of noise amplitude changes due to increase in pump speed.

levels emanating from the suction and discharge piping (Figure 14) was due to an 11.5% increase in pump speed, suggesting that vane-pass frequency components in suction or discharge piping segments were not the sole source but a contributing one. A review of the noise spectra indicate that the noise bandwidth (125-1000 Hz) appeared to be amplified by the suction and discharge piping. The question becomes what mechanisms in the flow stream could account for the increase in noise emissions from the piping.

Velocity and Pressure Waves

For centrifugal or radial-flow-type pumps, the effects of speed variations can be described using affinity laws, where capacity (flow) varies directly with speed; head (pressure) varies with the square of the speed; and brake horsepower varies with a cube of the speed:

$$\frac{S_1}{S_2} = \frac{Q_1}{Q_2} = \frac{\sqrt{H_1}}{\sqrt{H_2}} = \frac{\sqrt[3]{BPH_1}}{\sqrt[3]{BPH_2}} \qquad (2)$$

where S is speed, rpm; Q is flow capacity, gpm; H is head (pressure), ft or psi; and BHP is brake horsepower.

In this instance, it's the relationship of flow and pressure with changes in speed that are of particular interest. The 11.5% increase in speed increased the flow by the same amount, but nearly double the percentage to an approximate 23% increase in the pressure head. This elevates the energy levels of the pressure waves with a

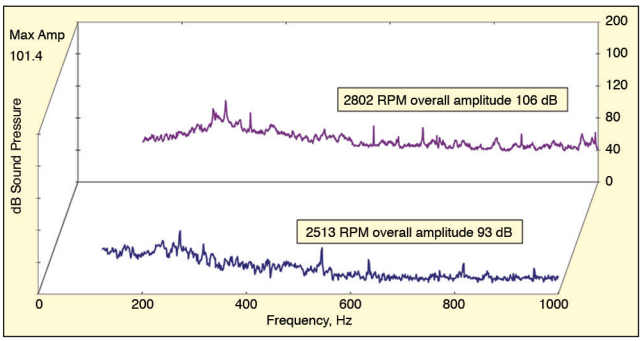

Figure 16. Representative spectral sample from suction and discharge piping measured at Figure 14 location.

Figure 17. Exposed yard piping with acoustic insulation installed.

corresponding increase in wave velocities due to increased flow. This subsequently increases the mechanical displacement of the pipe walls, resulting in the heightened volume of noise emission from the pipe. The described effect is illustrated in the representative spectra in Figure 16. As indicated at location S3, the 11.5% increase in speed resulted in an increase in the overall values of the spectra.

Observations

Local code restrictions required the maximum allowable environmental noise measured at the property boundaries not to exceed 85 dBA. The measurements in Table 1 indicate, with the average noise levels were within the code limits except for pf point PW1. However, the noise emissions from the crossover piping, 15 ft. above grade (Figure 14), averaged 94 dBA (Table 2). This was the noise the neighbors nearby were hearing, and depending on time of day and atmospheric conditions, the noise could be heard up to 500 feet from the pump station and affected numerous residences.

Solutions

Modifications could have been made to the pump to reduce the vane-pass noise; however, given the suspected waveguide amplification, there was no guarantee that costly and long-term modification alone would reduce the overall noise levels sufficiently.

Another method was discussed – damping the sound by enclosing the entire pipe yard in a structure with some type of sound-barrier wall. Again, this is a costly and longer-term solution. Additionally, a large imposing structure would no doubt raise significant objections by an already annoyed populous.

After some debate it was decided that the most cost effective and quickest solution was to apply lagging to the exposed piping in the yard. It was decided to first wrap the piping in a dense 2-inch layer of fiberglass matting, followed by layer of 0.11-inch-thick mass-loaded vinyl. On top of the vinyl a 1-inch-thick layer of sound absorption material was applied, followed by vibration-damping material, then two more layers of the 0.11-inch-thick mass-loaded vinyl. The entire package was then sheathed with aluminum cladding for weather protection (Figure 17).

The reason for so many layers is, for proper attenuation, the thickness of the lagging material should be equal to the wall thickness of the mass; the pipe in this case. For example, a pipe wall thickness of 0.38 inch will require the layered mass loaded vinyl to be approximately 0.38 inch thick or greater. Since each sheet is 0.110 inch thick, three layers were installed. The 0.08 inch sheet bonded to a backing with damping compound provides a total of 0.41 inch of lagging material in addition to the absorption matting.

After the pipe lagging was installed, the customer requested another sound survey, and it was conducted again with the same instrumentation at the points identified in Figure 14. As illustrated in Table 3, the average overall noise from the crossover piping was reduced from 94 to 78 dBA.

Table 1. Perimeter wall noise, dBA.

Location	Pump RPM	Overall dB
PW1	2745	95
PW2	2746	83
PW3	2743	82
PW4	2743	82
PW5	2737	81
PW6	2734	82
PW7	2724	82
PW8	2724	82
PW9	2729	75
PW10	2716	65
Averages	2734	81

Table 2. Crossover piping noise, dBA.

Location	Pump RPM	Overall dB
S9	2772	97
S10	2776	93
S11	2772	87
D4	2764	99
D5	2759	98
D6	2759	94
D7	2760	94
Averages	2766	94

Table 3. Before and after crossover piping noise, dBA.

Location	Before dB	After dB
S9	97	65
S10	93	79
S11	87	77
D4	99	88
D5	98	83
D6	94	83
D7	94	71
Averages	94	78

Conclusions

As stated in the beginning of this article, resonant standing waves typically excited by pump impeller vane-pass frequencies are common in petroleum pipeline systems, and modifications to the pump and/or piping will resolve the issue. What made this particular case unusual was the amplification by the system piping of the hydraulic noise. In this case, since a specific frequency was not being amplified, costly alteration to the pump such as changing the atypical six-vane impeller to a five-vane impeller may not have had a sufficient reduction in the noise energies. But more importantly, the customer informed us that the noise complaints have ceased and relations with the neighbors are on the mend.

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