The Effect of Pulsations On the Accuracy of Gas Metering

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The recent boom in natural gas production has led to an increased need for gas compression infrastructure, including meter stations that can transport larger volumes of gas at higher velocities and pressures. When improperly designed, these meter stations can create poor conditions for accurate flow measurement. Among the factors that should be considered when developing a meter station are the static pressure environment (for pulsation minimization) and pulsation distortions of the velocity profile. Measurement accuracy can be improved through the understanding of the influence of these factors on various meter types and by identifying and mitigating potential sources. This paper is a survey paper composed of previously published material by current and previous employees of Southwest Research Institute: S. Simons, T. Grimley, R. McKee, R. Durke, E. Bowles, and K. Brun as noted in the reference section. This paper will discuss the basics of pulsations in piping systems, various methods for attenuating pulsations, and the effect on orifice, turbine, ultrasonic, and Coriolis meter readings. Field case studies of different types of problems experienced at meter stations will be used to demonstrate these effects.

Pulsation Basics

Pulsations in metering applications can be generated by reciprocating compressors, centrifugal compressors, flow-induced vortex-shedding, pressure reducing valves, or turbulent flow. Of these sources, reciprocating compressors and flow-induced vortex-shedding produce the majority of pulsations that affect meter readings by creating widely fluctuating line pressures and distort the flow velocity profile.

Pulsation is a periodic fluctuation in local pressure imposed on a piping system by a machine or as a result of a flow phenomenon. The Bernoulli principle describes that a variation in pressure will produce a corresponding variation in velocity (see Equation 1 for a simplified form). Figure 1 shows how the pulsations created by a reciprocating compressor affect the molecular distribution in a gas by creating pressure (and velocity) waves that travel through a piping system.

$$\frac{V^2}{2} + \left(\frac{\gamma}{\gamma - 1}\right)\frac{P}{\rho} + gz = constant \qquad \text{(Equation 1)}$$

Where,

- V = nominal fluid velocity
- P = static line pressure
- ρ = fluid density
- g = gravitational acceleration
- z = elevation of a point above a reference plane
- γ = ratio of the specific heats of the fluid

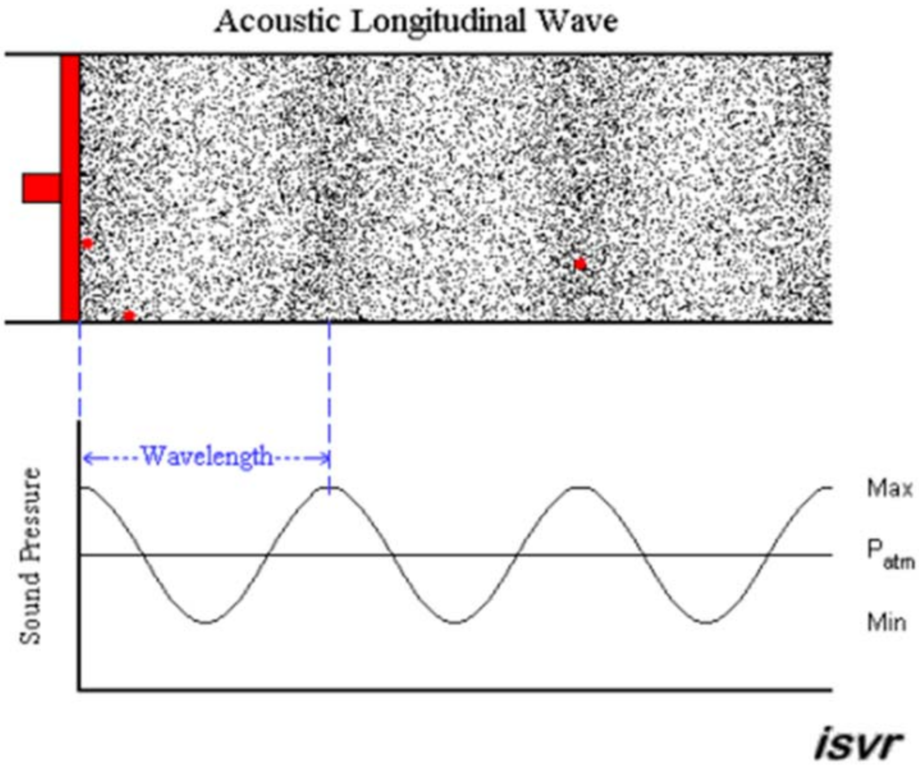


Figure 1. Illustration of an Ideal Generation of a Pulsation Wave in the Shape of a Sine Wave [1]

Pulsation Sources

Pulsations imposed by a piston on a fluid typically travel in the shape of a sine wave through the piping system. However, due to the distortions imposed by the compressor valves, the actual wave has significantly more high frequency content and obtains some characteristics of a square wave. The pulsation wave is composed of frequency multiples of the compressor running speed with more energy on the odd orders if the compressor cylinders are single-acting and more energy on the even orders if the compressor cylinders are double-acting. Pulsation excitations from reciprocating compressors are typically less than 100 Hz in metering areas. The acceptability of pulsation amplitudes from positive displacement machinery is typically evaluated using API standards, such as API 618 for reciprocating compressors.

The interaction of a gas flow stream with changes in the geometry of a piping system can also introduce pulsations at discrete frequencies into the system. The shear or boundary layer of fully developed turbulent flow will separate creating the formation of vortices. Vortex-shedding occurs at regular intervals creating an oscillating pressure field that can excite acoustic and mechanical natural frequencies in the piping system. The most common source of low-frequency (less than 100 Hz) vortex-shedding in a metering area is flow past a closed stub or dead leg [2]. Higher frequency vortex-shedding in metering areas can be caused by instrumentation (such as thermowells and sample probes), orifice plates, and control valve internals.

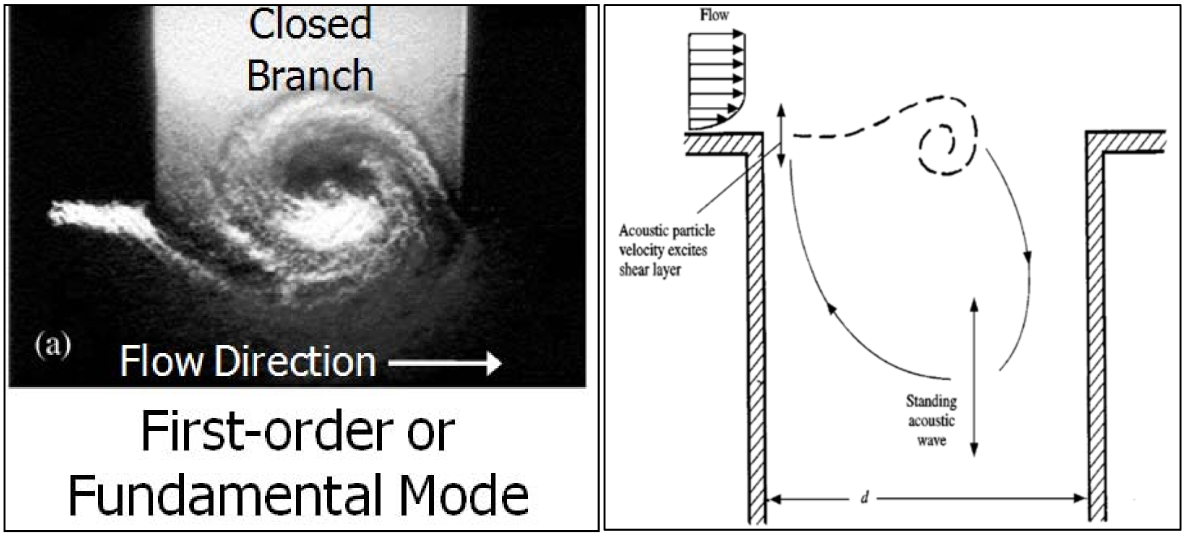


Figure 2. Vortex Formation in Water and Feedback Mechanism of Vortex-Shedding Excited Acoustic Resonances at Closed Side Branches [S. Dequand, et al.; Ziada and Shine, 1999]

The Strouhal number relates the vortex-shedding frequency to the flow velocity (U) and the characteristic dimension (d) of the piping geometry causing the flow disturbance as shown in Equation 2. It is typically determined through experimentation and used to calculate the range of excitation frequencies in a piping system. To avoid discrete-flow-induced pulsations and vibrations in the piping system, a vortex-shedding analysis can be performed to ensure separation exists between the excitation frequency and the natural frequencies of the piping. Several standards, such as ASME PTC 19.3 and the Energy Institute (EI) Guideline, as well as guidelines developed by engineering companies and consortiums, exist to evaluate piping systems for potential flow-induced problems.

$$S_t=f_s d/U \qquad \text{(Equation 2)}$$

Piping Acoustic Natural Frequencies

A piping system has two types of acoustic natural frequencies: longitudinal, 1-dimensional modes and circumferential (radial) 3-dimensional modes. Longitudinal acoustic natural frequencies are defined, similar to mechanical natural frequencies, by the end conditions of a piping segment. Closed ends may be the capped ends of headers, closed branch line valves, or terminations of gauge or drain lines. Open ends may large volumes such as scrubbers, headers with large diameter ratios to the main piping, or locations where a small branch line connects to a larger diameter pipe. When the pulsation excitation frequencies from the compressor or vortex-shedding align with the acoustic natural frequencies of the piping system, *acoustic resonance* occurs, amplifying the pulsations.

For 1-D longitudinal modes, the relationship between the speed of sound of the fluid, the pipe length, *L*, and pipe end conditions determine the acoustic natural frequency, *f*. The acoustic natural frequency equation for the modes of the response of a pipe segment with one open and one closed pipe end is that of multiples of a quarter wave as shown in Equation 3a and Figure 3. The acoustic natural frequency equation for the modes of the response of a pipe segment with both open, or both closed pipe ends is that of multiples of a half wave as shown in Equation 3b and Figure 4. These frequencies are typically less than 300 Hz for natural gas piping systems.

$$f = \text{n}\ \frac{c}{4L},\ \text{n}=1,3,5,\ldots \qquad \text{(Equation 3a)}$$

$$f = \text{n}\ \frac{c}{2L},\ \text{n}=1,2,3,\ldots \qquad \text{(Equation 3b)}$$

where:

L = Acoustic length of pipe span

c = speed of sound of the flowing gas

f = pulsation frequency

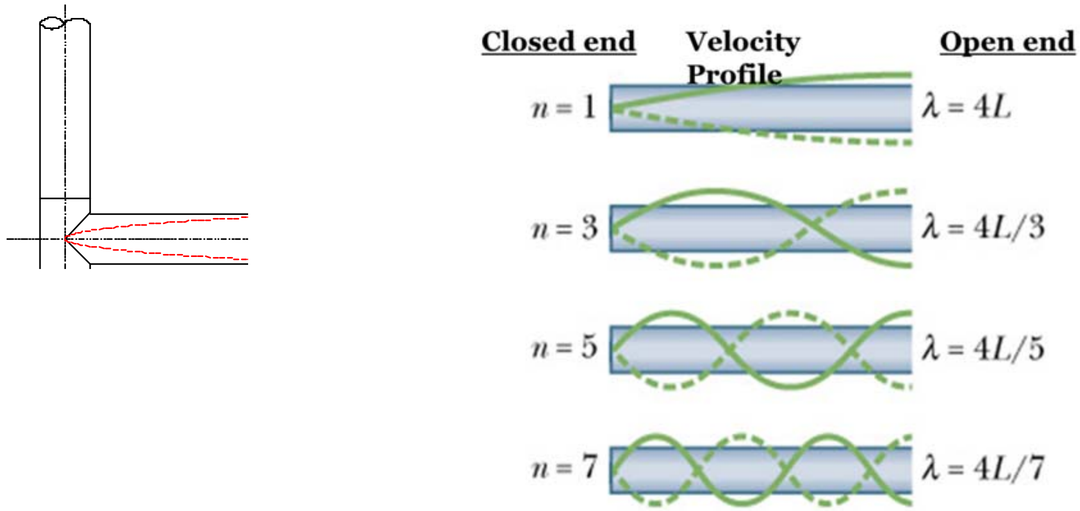


Figure 3. Quarter-Wave Mode Shapes for Open and Closed Ends

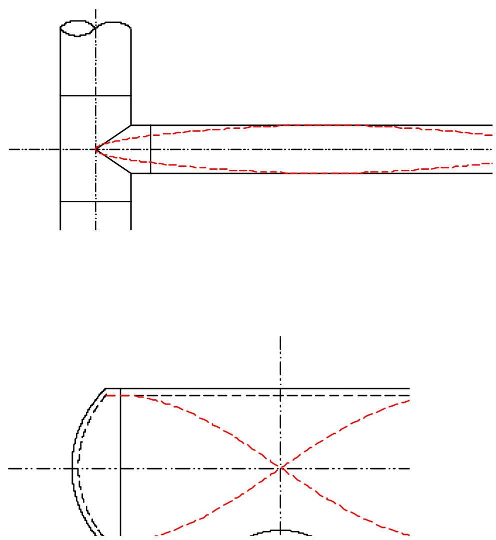
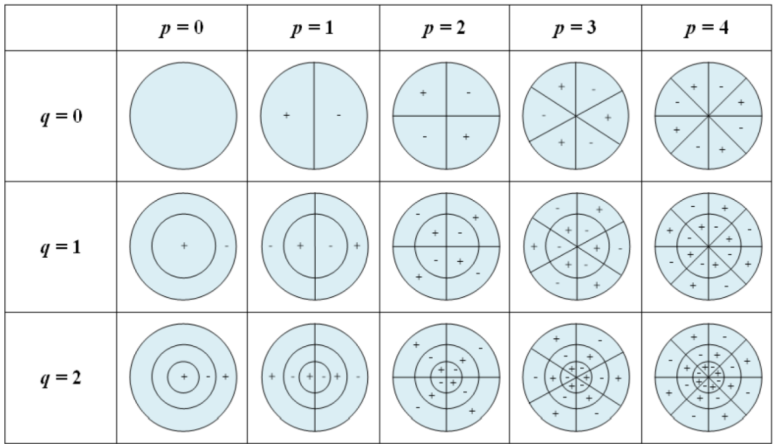


Figure 4. Half-Wave Mode Shape for Open-Open and Closed-Closed Ends

Wavelengths that are shorter than the diameter of the pipe propagate radially creating excitation of the acoustic cross-modes of the piping. The 3D wave equation used to calculate the radial modes can be simplified into a solution in the form of a Bessel Function. The acoustic radial and circumferential piping modes are described in the equation below, Equation 4, with the coefficient, α, determined by the circumferential and radial modes of the piping as defined by the qth zero of the first derivative of the pth order of the Bessel function. The first several acoustic 3D mode shapes are shown in Figure 5 below. Using equation 3 for a 6-inch standard pipe, the frequency of the first mode, with p=0, q=1, is at 600.7 Hz using a speed of sound (c) of 1,135 ft/s. For natural gas main process piping, 3D acoustic modes are typically greater than 300 Hz.

$$f = \frac{c\alpha_{pq}'}{2\pi r} \qquad \text{(Equation 4)}$$



α_{pq}	p = 0	1	2	3	4
q = 0	0.000	1.841	3.054	4.201	5.318
1	3.830	5.331	6.706	8.015	9.282
2	7.016	8.536	9.969	11.346	12.682

Figure 5. Acoustic Piping Radial Modes and Associated Bessel Constants

Pulsation Impact On Various Meter Types

Pulsations result in distorted velocity profiles and include cyclical variation in the bulk velocity that affect the flow measurement in varied ways depending on the sensing technology. The most well-known adverse effect of pulsations is commonly referred to as the square root error (SRE) for orifice meters. Since the orifice flow rate is proportional to the square root of the pressure differential (see a simplified orifice flow equation below), the presence of dynamic flow leads to sampling and averaging errors resulting from the difference between the average differential pressure and the average of the square root of the differential pressure.

$$\Delta P = K * Q^2 \qquad \text{(Equation 4)}$$

where:

ΔP = differential pressure measured across the orifice plate

K = empirical coefficient

Q = volumetric flow rate

SRE increases with increasing pulsation amplitude. If a pressure transmitter is installed with a frequency response range high enough to accurately measure the differential pressure across an orifice plate experiencing pulsating flow, the mathematical relationship shown in Equation 5 can be utilized to determine the magnitude of the SRE. This method for determining SRE is a patented process developed at Southwest Research Institute during research sponsored by the Gas Machinery Research Council[i] and is the basis for the commercially available Square Root Error Indicator (a.k.a., SREI) that measures the SRE.

$$SRE = \frac{\sqrt{avg\Delta P} - avg\sqrt{\Delta P}}{avg\sqrt{\Delta P}} * 100 \qquad \text{(Equation 5)}$$

where:

SRE = the square root error in percent of differential pressure transmitter reading

$\sqrt{avg\Delta P}$ = the square root of the time averaged value of the differential pressure across the orifice plate

$avg\sqrt{\Delta P}$ = the time averaged value of the square root of the instantaneous differential pressure across the orifice plate.

The interaction of pulsations with the pressure sensing (gauge) lines can also distort the values presented to the pressure sensor and lead to additional errors (Durke, et al., 2012). In addition to the differential pressure measurement error, velocity profile distortions (e.g, Figure 6) caused by pulsations cause a shift in the orifice discharge coefficient that results in additional error. Therefore, it is important to note that the measured amount of *SRE cannot* be used to perfectly correct for orifice measurement error associated with flow pulsation, but *SRE* can be used to indicate if pulsation is causing a significant problem at an orifice flow meter. The maximum allowable pulsation level specified in Section 2.6.4 of Part 2 of American Gas Association (AGA) Report No. 3[ii], is 10% root mean square (RMS) variation in the ΔP (RMS is a statistical measure of the magnitude of the variation in the ΔP), which corresponds to an *SRE* value of approximately 0.125% of reading. This applies to single frequency flow pulsations with or without several harmonics and to broadband flow pulsations/noise. Any *SRE* above this threshold indicates that the pulsation is adversely affecting the orifice meter accuracy. Section 2.6.4 further states that… "*Currently, no satisfactory theoretical or empirical adjustment for orifice measurement in pulsating flow applications exists that, when applied to custody transfer measurement, will maintain the measurement accuracy predicted by this standard. Arbitrary application of any correcting formula may even increase the flow measurement error under pulsating flow conditions. The user should make every practical effort to eliminate pulsations at the source to avoid increased uncertainty in measurements.*"

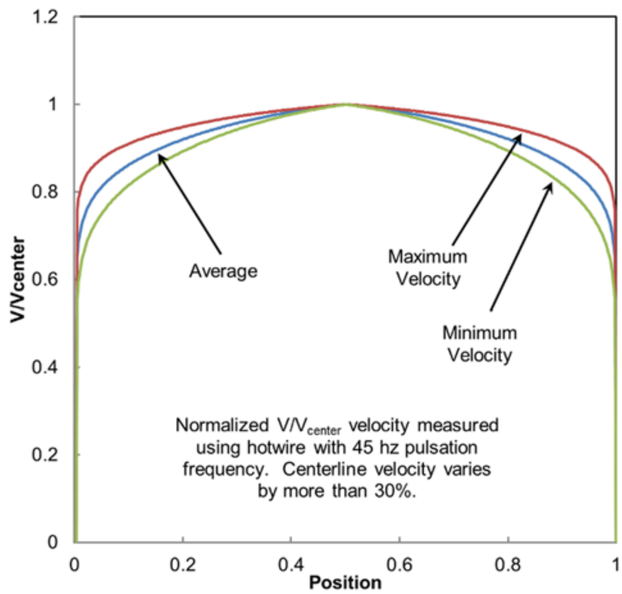


Figure 6. Pulsation Changes the Shape of the Velocity Profile (Durke et al., 2017)

Turbine meters utilize a rotor that is designed to turn in proportion to the flow through the meter. In the ideal situation, this proportionality is independent of the flow rate. However, when turbine meters are exposed to pulsating flow, the inertia of the rotor does not allow the meter to instantaneously respond to flow variations. The result is that the rotor lags the flow fluctuations and the meter flow rate indication is incorrect; typically, the meter over-registers as shown in Figure 7. AGA Report No. 7[iii] recommends that in order to avoid pulsation-related measurement errors, pulsation should eliminated or reduced.

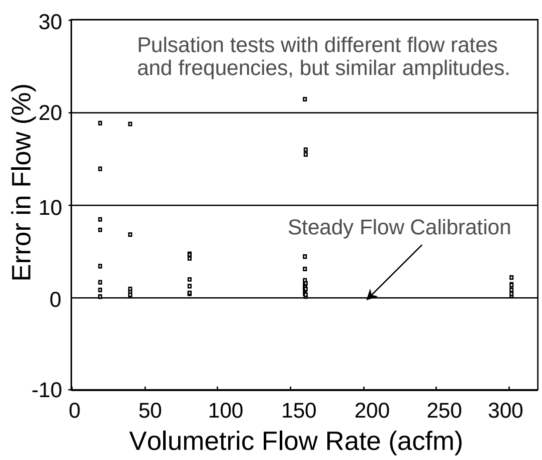


Figure 7. Example Pulsation Test Results for a Turbine Flow Meter (Durke, et. al. 2017)

Coriolis flow meters rely on the inertial force imparted on a flow tube that results from electro-mechanically oscillating the flow tube at its natural frequency. Sensors detect the phase shift that results from flowing fluid and the meter electronics convert this into a mass flow rate. Although Coriolis meters are largely immune to pulsations, the presence of pulsations or vibrations that interact with the flow tube motion can prevent the sensors from properly measuring the phase difference and result in measurement error. The frequency at which Coriolis meters operate varies depending on the meter size and design but, in general, the natural frequency values are greater than 100 hz.

Ultrasonic meters (USMs) measure the transit time of high-frequency (greater than 100-200 KHz) acoustic pulses traveling at angles through the gas flow. Each pair of transducers samples the velocity profile along a specific path and the velocity measurement results from multiple paths are typically combined to determine the overall flow rate. As shown in Figure 8, pulsations can change the shape of the velocity profile across a pipe thereby changing the measurement samples. Depending on the methods used to combine the path velocity measurements and the severity of the velocity profile distortion, significant error can result. A portion of the error is also a result of the meter's sampling frequency relative to the pulsation frequency, but even with perfect sampling, errors would still be expected from the profile distortion. Broadband pulsations and noise resulting from pressure control valves can interfere with the measurement of acoustic pulses and cause flow measurement errors (McKee, 2009). Some valves shift acoustic frequency noise into the ultrasonic frequency range to reduce environmental noise, but in doing so, may create a noise source for the transducers used in ultrasonic meters.

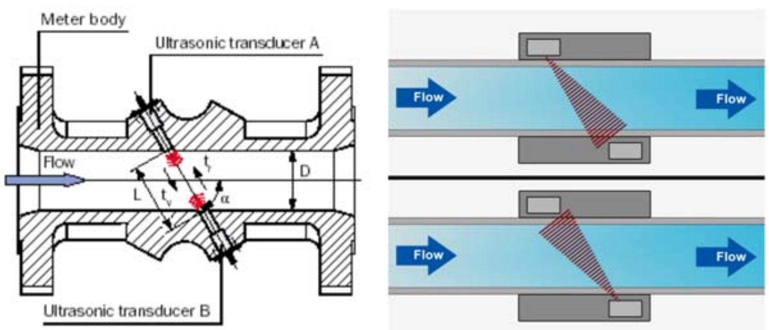


Figure 8. Schematic of an Ultrasonic Flow Meter
(Images courtesy of Bureau of Analytical Complexities & Systems and Alicat Scientific)

<u>Pulsation Control Methods</u>

Pulsations can propagate over long distances as shown in Figure 9 with lower frequencies taking longer to decay than high frequencies. Therefore, placing long lengths of piping between meter stations and reciprocating compressors often does not provide sufficient attenuation. Typical pipeline pulsation frequencies in the 5 to 45 Hz range can travel several miles from the source and higher frequencies can exist at significant levels up to a mile from the source.

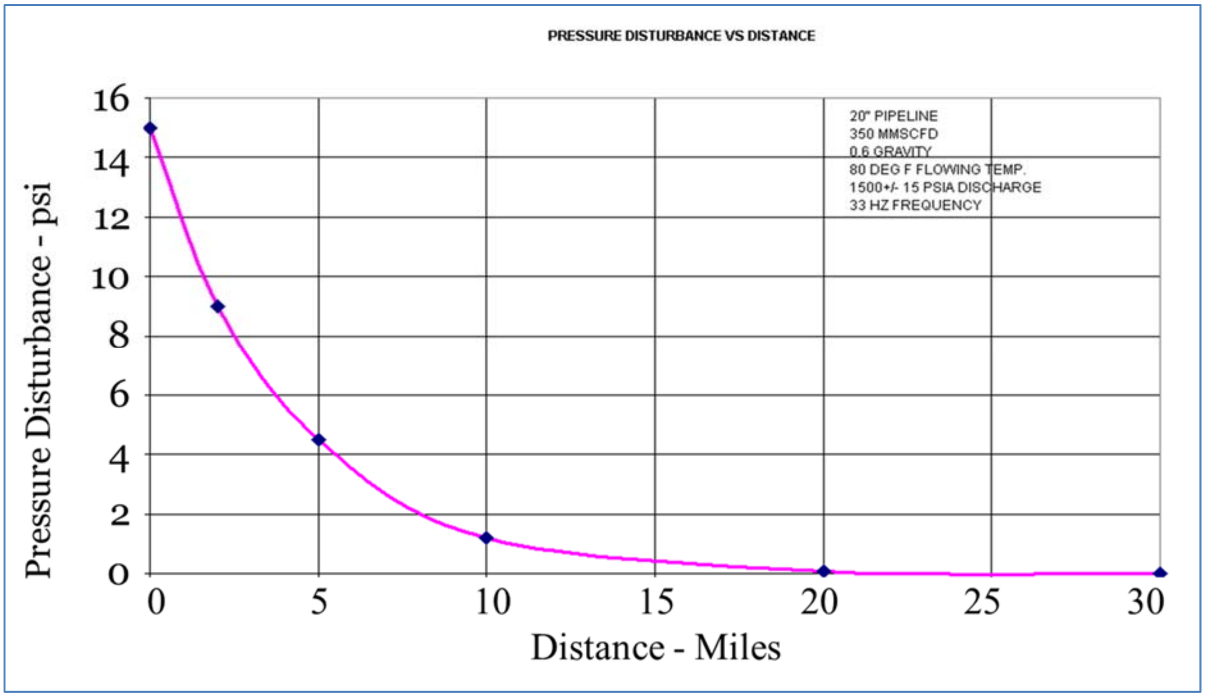


Figure 9. Decay of a 33Hz Pulsation in a Pipeline [Kurz et al., 2003]

Some simple changes that can be made to avoid localized high amplitude pulsation such as that in a branch connection line is to change the branch line length. For orifice flow meters, the pressure transducer gauge line length should be selected to avoid resonance at a reciprocating compressor running speed or other known excitation frequencies. One mitigation technique that reduces the pulsation induced error in orifice meter readings is to reduce the orifice beta ratio, which, in turn, increases the ΔP. As long as the pulsation amplitude remains constant, the *SRE* effect on the orifice will be reduced. However, these changes are typically not sufficient to mitigate the effect of pulsation amplitudes on meter readings. The most effective methods of attenuating pulsations rely on choking, shifting or damping the pulsations. This is accomplished through the use orifice plates, Helmholtz resonators, side-branch absorbers or volume-choke-volume filters. Physically most mitigation methods rely on three attenuation mechanisms: choking, shifting, and damping.

Attenuation through Orifice Plates

When a pulsating flow stream enters an area reduction, such as an orifice plate, pressure builds up on upstream side of the area restriction. The rise in pressure and corresponding dynamic velocity due to the flow pulse causes the flow through the area restriction to increase proportionally. However, often times the increase in local velocity in the flow particles within the pulse would exceed the speed of sound in the gas stream if this relationship remained proportional. When this occurs, the flow through the area restriction is limited or choked and held to a fixed value of Mach number equal to one.

In order to illustrate the choking effect, a typical example is presented in Figure 10. An orifice plate is inserted into a pulsating flow stream. Since this calculated dynamic velocity in the orifice bore would exceed the speed of sound in the gas, the pressure pulse will be momentarily choked by the area change. Pressure will build up on the upstream side of the orifice plate and elongate the pressure pulse on the downstream side of the place. This choking effect will also result in a lower amplitude pressure. However, there is a significant pressure loss penalty associated with the use of orifice plates; additionally, the orifice must be located in a node in the piping where it can effectively attenuate the pulsation amplitudes.

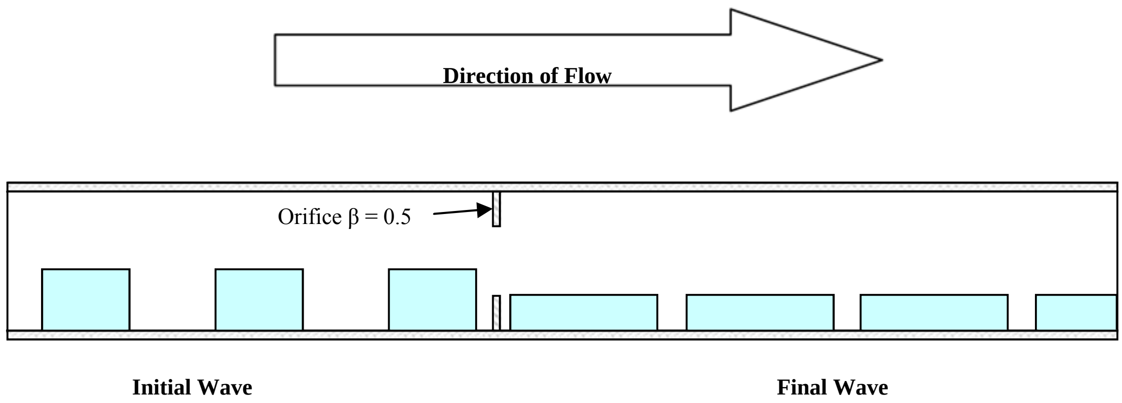


Figure 10. Pulsations through an Orifice [5]

Attenuation through Side Branch Absorbers or Pipe Splitting

The second physical means of reducing pulsations is through a shifting effect. This occurs when a portion of the pulsating flow stream is diverted to a closed end in the piping system, which can be either a short stub (side-branch) or a more elaborate closed end achieved through a side branch volume (Helmholtz resonator). Figure 11 below shows a set of pulses traveling down a piping system and splitting into two lower amplitude sets (assuming equal pipe flow areas at the tee intersection). The pulses in the stub branch travel down the length of the branch and then reflect back against the closed end. For the purposes of the example, the stub branch has a length equal to the one-fourth of the wavelength. When the reflected pulse rejoins the flow stream, its phase has effectively shifted over one complete cycle. This results in tuning out the frequency corresponding to $f = c/4L$. The advantage to using this method of attenuation is the low pressure loss associated with this device; however, it can only reduce one pulsation amplitude frequency and the device can become very large if the frequency is low.

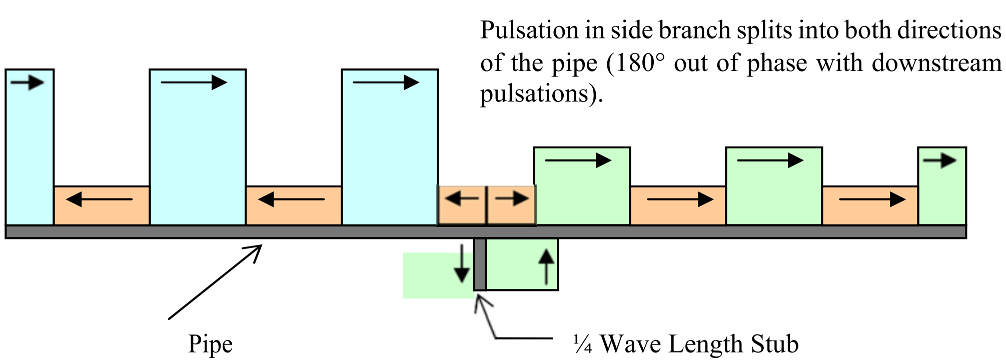


Figure 11. Pulsations through a Pipe with a Side Branch Attenuator [5]

Attenuation through Damping or Acoustic Filters

Damping is typically most effective when combined with choking such as in an acoustic filter discussed below. A large empty vessel can provide significant damping of pulsations; however, it typically is only used as a secondary device to bring pulsation amplitudes below API acceptable levels since most vessels (such as scrubbers) are not large enough to provide sufficient attenuation on their own.

One of the most effective method for reducing pulsations near metering areas is to install an acoustic filter between the pulsation source and the flow meter. An acoustic filter is analogous to a low-pass electrical filter and can be thought of as a spring mass spring mechanical system. An acoustic filter passes pulsation below its natural frequency and filters out (or, at least, significantly reduces) pulsation at frequencies above the natural frequency. One important characteristic of this type of filter is that it amplifies pulsation at and near its natural frequency. Therefore, pulsation energy that is to be controlled should never be coincident with the natural frequency of the filter. There are many approaches to designing acoustic filters. The two filter designs pictured in Figure 12 are a symmetric, in-line acoustic filter and a single bottle, two chamber filter that can be used to eliminate pulsation over a wide frequency range. Acoustic filters can be non-symmetric with various sizing and characteristics when designed for a piping system's specifications.

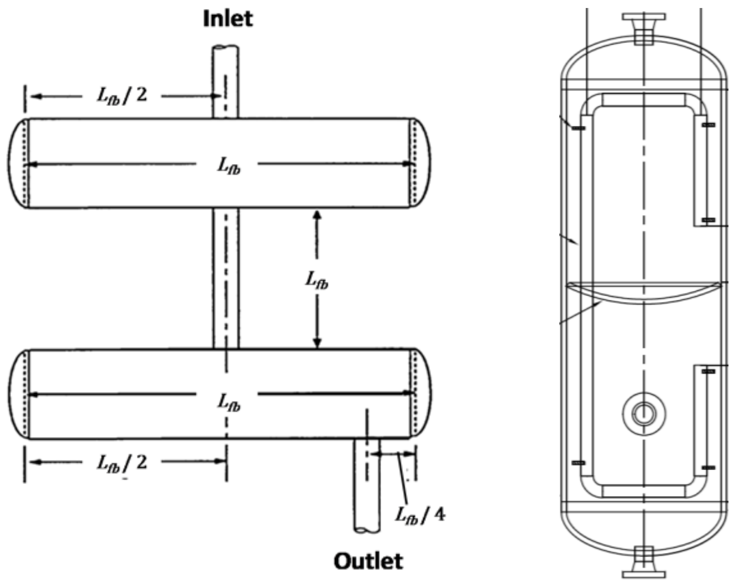


Figure 12. Example of a Two Bottle Symmetric [6] and One Bottle Two Chamber Acoustic Filter Design

Case Studies

Case Study of a Pulsation Problem in a Metering Station

A meter station experienced high vibrations causing shaking of the piping, ground, and instrument lines; high noise levels; and disabling of electronics used in meter readings. A field study was performed to obtain pressure and vibration data over a range of flows and operating conditions. High pulsation amplitude was clearly seen at 40 Hz. This corresponded to several mechanical piping response modes near that frequency creating vibration problems.

It was determined by pressure transducer data that the source of the pulsations was vortex-shedding excitation of bypass piping around a valve in the main line (see Figure 13). Vortex-shedding of the flow past the tandem tees connecting the bypass line to the main line excited the acoustic natural frequency of the two identical closed piping stubs at certain operating conditions. Although the gas flow velocity of excitation was considered normal to low at less than 60 ft/s, two identical acoustic standing waves located in close proximity to each other can create resonances that sharply increase in amplitude due to feedback with each other.

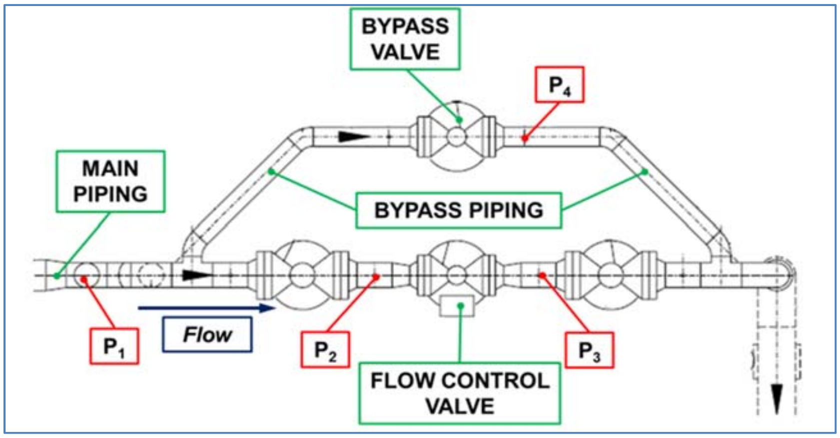


Figure 13. Piping Area with Pulsation Problems

Pulsation levels in the main line were amplified by an order of magnitude in the bypass line near the closed valve as shown in Figure 14. As a point of reference, the recorded pulsations were 170 times greater than API 618 allowable levels at 40 Hz.

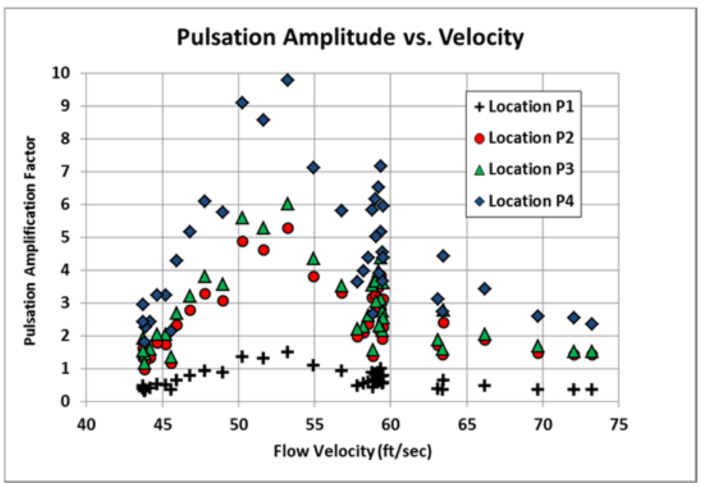


Figure 14. Pulsation Amplitude Factor vs. Frequency in Bypass Area Piping

To eliminate the pulsation problem, the bypass valve was partially opened thus changing the reflective end condition of the piping stubs and shifting the acoustic natural frequency of the line as well as disrupting the excitation source.

Flow Control Valves Problems: Noise Reducing Trim

The majority of installed meter stations contain a flow control valve skid to regulate the pressure or flow of gas going through the station. A reduction in pressure or velocity is ideally an isenthalpic (zero work) process, but some energy is lost due to viscous shearing in the flow. The energy loss taken at these valves is primarily converted to heat; however a significant portion is converted into noise. To reduce the noise associated with the energy loss, most modern flow control valves have an internal noise reducing trim which can be multi-stage and/or multi-path. This allows the pressure loss to be taken in steps and forces the flow through very small passageways. The reasoning for this is that the small passageways will shift the noise frequency of the energy well above the ring frequency of the pipe such that it does not transmit well through the pipe wall and can be at frequencies well above human hearing. Additionally, noise power level is proportional to the square root of the pressure drop. Therefore, adding the noise generated by a series of pressure drops result in a significant reduction of noise over that generated by one large pressure drop.

In one metering facility a loud, high-pitched noise was reported emanating from the flow control valve skid (see Figure 15). According to FERC regulations, 18CFR380.12(k)(2), all new and modified meter stations must conduct an ambient sound survey to quantify noise levels at noise sensitive areas (NSAs). Noise attributed to operation of the station must not exceed a day-night sound level (L_{dn}) of 55 dBA at pre-existing NSAs such as houses, farms or places of employment. A field study performed at the facility used Sound Pressure Level (SPL) meters at various locations to record the noise levels and dynamic piezoelectric pressure transducers were installed in the available flow control skid piping taps to record the pressures. Vibration data was collected using magnetically-mountable tri-axial accelerometer probe. The opening of the control valves in combination with other station valves was varied to record data for a range of pressures and flows as well as evaluate the likelihood of the control valves being the source of excitation.

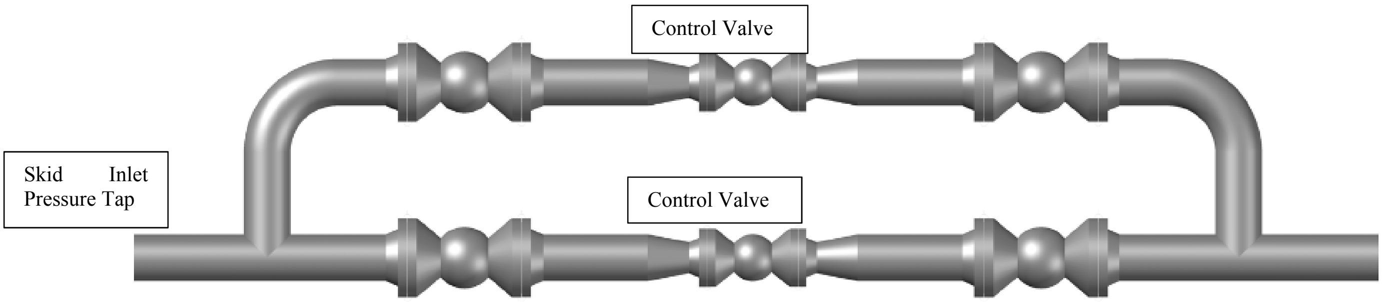


Figure 15. Control Valve Skid Configuration

When the control valves were fully open in the maximum flow conditions, no clear tonal noise was noted. The total dB levels recorded by the SPL meters were well under 55 dBA. However, there was a strong pulsation amplitude of approximately 17 psi peak-peak at 4400 Hz recorded by the pressure transducers located at the control valve inlet (see Figure 16). However, this frequency was above the ring frequency of the pipe (4,100 Hz), and while little noise external to the pipe was recorded by the SPL meters at this frequency there was concern that it was affecting the meter readings. It is typical to have high amplitude acoustic excitation near the valves created by the high velocity flows and pressure drop associated with the control valves; one of the purposes of the noise reducing trim is to shift the dominant excitation frequencies above the ring frequency to reduce the radiation efficiency of the acoustic waves.

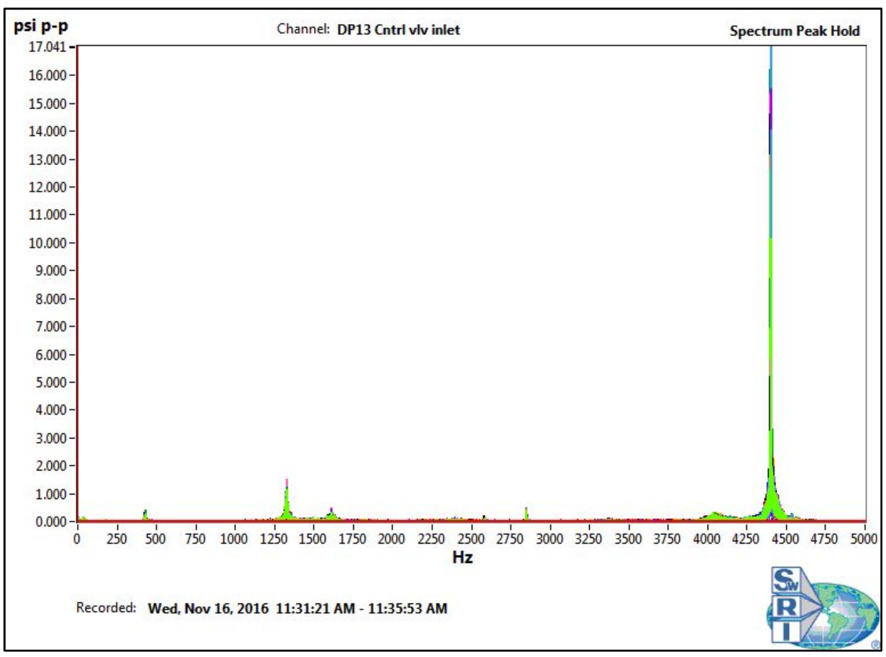


Figure 16. Control Valve Skid Inlet Maximum Pulsation, Low Noise Due to Frequency

When the control valves were operating in a partially closed positions (50-80% open), a high amplitude tonal noise occurred at 3300 Hz clearly radiating from the control valve skid (Figure 17). The recorded external decibel level was approximately 106 dBA at the valves. Although the recorded internal pulsation amplitudes were not as high as other test conditions (approximately 7.47 psi p-p in Figure 17), this excitation frequency was transmitting through the pipe wall with few losses, likely due to coupling with a mechanical shell mode of the 20-inch piping.

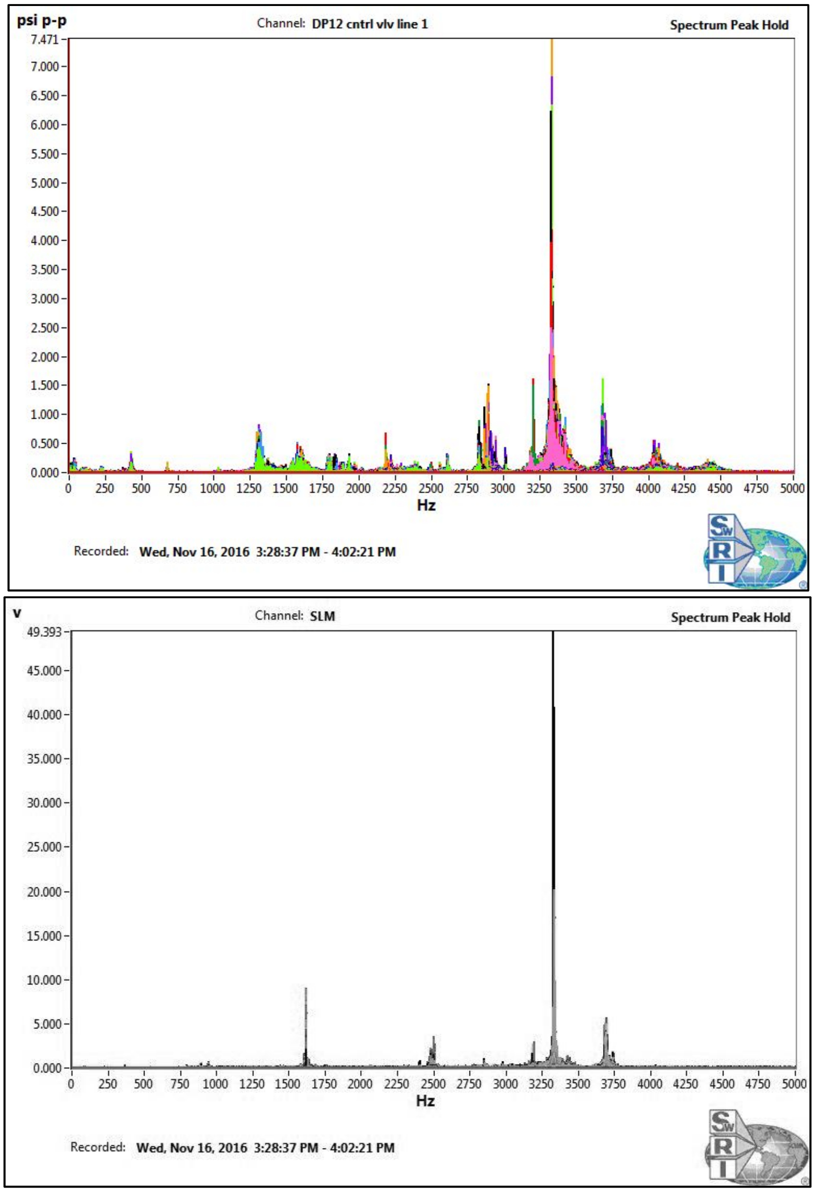


Figure 17. Control Valve Skid Inlet Pulsation Amplitude and SPL Maximum Recorded Measurement (Volts)

As the control valves were closed from 50% to 0% open, noise levels dropped to acceptable levels. It was clear from the testing that the noise was originating at the valves and was created by a frequency that was excited when partially closing the valves. The valve internals consisted of various flow areas partitioned by perforated plates with small diameter holes. As the plates shift position to partially close the valve, cavities of various depths were created which had acoustic modes near 3300 Hz at the station operating conditions. The excitation of an acoustic resonance internal to the valve coupled with a mechanical shell mode in a frequency range with low transmission loss typically creates high external noise levels.

After these findings, the station replaced the control valves with a different type of control valve. A follow-on field study was performed to record SPL data and dynamic pressure measurements over the same range of operating conditions and valve opening positions. The high frequency noise in the 1,500-4,000 Hz range was virtually eliminated as shown in Figure 18.

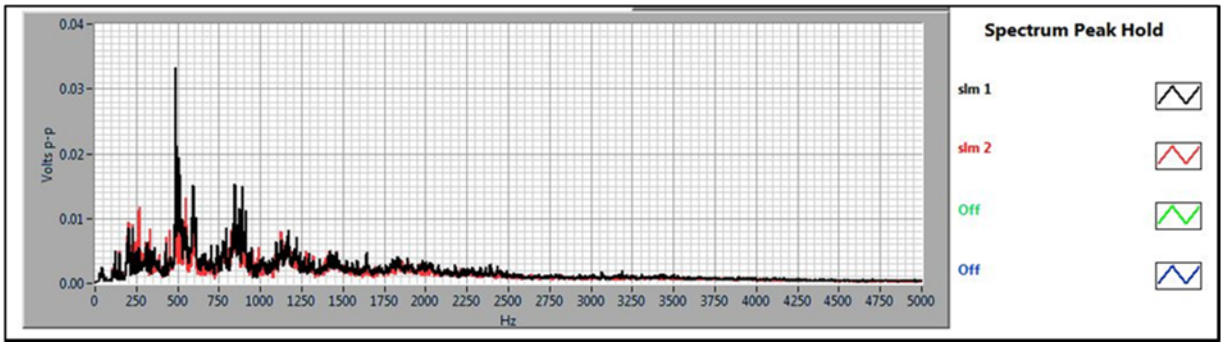


Figure 18. SPL Measurement (Volts)After Valves Were Replaced

Conclusions

Gas flow with high levels of pressure and velocity fluctuations can create significant errors in flow measurement readings. While the error mechanisms for each meter type will vary, it is best to avoid these errors by minimizing pulsations upstream of the metering devices. It is recommended that pulsation amplitudes be measured within approximately 100 feet upstream of a meter installation. Some general guidelines for acceptable levels are as follows: 3-5% velocity perturbations from discrete pulsations and 3-5% velocity perturbations from turbulence. If measured amplitudes exceed these levels, an attenuation device can be installed or the piping, instrumentation or valves can be modified to reduce pulsation amplitudes such that large errors in measurement are avoided.

References

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