# MANAGING NOISE IN NATURAL GAS FACILITIES

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## Introduction

Webster defines noise as "a sound, especially one that lacks agreeable musical quality or is noticeably unpleasant." In the gas industry, objectionable sounds can come from relief valve discharges, gas blowdowns, compression equipment running, control valves throttling, and normal (and abnormal) pipeline flows.

Noise emanating from pipelines can be caused by reasons other than gas flowing. Debris left in a pipeline after construction can cause metallic noises. Noise can also be generated by liquids moving in the gas flow stream.

Noise can travel long distances, as was evidenced by a problem a gas transmission company experienced many years ago in Georgia. A woman started hearing a "popcorn popping" sound in her front yard where a gas transmission line was buried. The transmission company spent months trying to determine the source of the noise without success, until a routine maintenance check of their mainline valves uncovered a vandalized valve which had been partially closed. When this valve (which was located almost ten miles upstream) was opened, the noise disappeared in the lady's front yard.

This paper will primarily focus on noise coming from the throttling of gas in regulating stations. Control valve generated noise, resulting from gas pressure reduction (regulation), can exceed OSHA or local noise limits or can cause destructive damage to regulating equipment and pipe components. Too often, potential regulation generated noise is an "afterthought" of station design. Its importance is realized only after noise complaints or noise generated damage to regulating equipment is brought forcefully to the designer's attention.

Including a look at regulation noise as an integral part of station design can avoid excessive noise problems and expensive noise abating post installation modifications. Following are brief descriptions of the causes of regulation noise and the tools available to keep them within acceptable levels.

## The Sound Measurement Scale

Sound is measured in decibels, which is defined as 20 times the logarithm of the ratio of the root mean square sound pressure level to a reference level. Mathematically it is expressed as:

$$dB = 20\mathrm{Log}_{10}(P/Po)$$

where P is the measured root mean square sound level and Po is the reference level.

Noise typically consists of a broadband of different sound frequencies. However, the human ear is most sensitive to sound in the 500 Hz to 6,000 Hz frequency range. The A-weighting scale (dBA) biases the measured sound to conform to the frequency response of the human ear. This is the most widely used scale for environmental studies. For further information, refer to any of the major control valve manufacturer's technical bulletins on

noise for a detailed description of the dBA scale.

**Noise Limits**

OSHA has established a sound criterion level of 90 dBA as the maximum 8-hour time weighted average (TWA) for employee hearing conservation. The American National Standards Institute (ANSI) S12.19 (1996), "Measurement of Occupational Noise Exposure" defines the criterion sound level as a "constant sound level in decibels (dBA), which, if it continues for the criterion duration, would provide 100% of an employee's allowable noise exposure." Therefore, for design purposes, 90 dBA should be considered the maximum permissible limit for control valve generated noise. Typically, local ordinances are much more restrictive, and are associated with the property lines. It is not unusual for urban areas to require noise levels to not exceed 70 dBA at the station boundaries.

In addition to environmental limits, a noise level, as stated in the Fisher technical bulletin on noise, "above **110 dBA** is generally not recommended since high vibration levels can result, which can lead to severe damage to the control valve, actuator, instrumentation, and downstream piping." Other manufacturers have the same nominal recommended limitation. Noise in excess of this level can result in catastrophic failure. However, compliance with environmental limits ensure that destructive noise levels are avoided.

For standardization purposes, control valve produced noise is referenced one meter (3 feet) downstream of the valve and one meter (3 feet) from the pipe. Regulation generated pipeline noise falls 3 dBA for each doubling of the distance from the source, commonly called a line source, per the following formula:

$$\Delta dBA = -10 Log_{10}(r_2/r_1)$$

where $r_1$ is the distance from the source to the first point in feet and $r_2$ is the distance from the source to the second point in feet

Table I, a hypothetical regulation noise source, illustrates this principle.

Table I. Noise related to distance from source

| Feet From Source | Noise Read - dBA |
|---|---|
| 3 | 90 |
| 6 | 87 |
| 12 | 84 |
| 24 | 81 |

If the noise source is truly a point source, such as a relief valve discharge or a blowdown stack exit, then the equation is:

$$\Delta dBA = -20 Log_{10}(r_2/r_1)$$

**Two Sources of Noise**

**Mechanical** noise is caused by turbulent flow induced vibration of valve trim parts. Noise thus generated has a tonal quality. If the induced vibration of trim parts approaches a natural frequency, a harmful resonant condition will occur that can result in fatigue failure.

Fortunately, contemporary valves seldom experience noise due to mechanical vibration, especially since the advent of top and cage guided valves. However, should this condition occur, immediate action must be taken to eliminate the resonant condition to avoid potential fatigue failure.

**Aerodynamic** noise results from the conversion of gas pressure energy to velocity energy inside

the vena-contracta of the valve. For subcritical conditions where the jet velocity at the vena-contracta is below sonic, control valve noise is generated by the intense turbulence created in the shear layer downstream of the vena-contracta. For critical conditions, additional noise is induced by the interaction between the turbulence and the shock waves developed by the sonic flow velocity.

**Physics of Aerodynamic Noise**

The regulation or throttling process is not isenthalpic within the control valve. Within the confines of the valve body the gas stream is dramatically accelerated to high velocities. It is here that the gas shoots through the smallest passage reaching its highest velocity at the vena-contracta (the minimal stream restriction a small distance downstream of the smallest passage). If sonic conditions occur at the vena-contracta, speeds around 1400 fps for natural gas result.

The dramatic increase in velocity results from a conversion of pressure energy to kinetic energy. The high velocity causes the turbulence that generates the sound energy. The velocity energy, through turbulence and sonic shocks, is dissipated at the exit of the control valve. Some energy is lost in the form of sound.

**Factors Determining Regulation Noise**

There is no standard industry recognized formula for predicting regulation noise levels such as there is for valve sizing (ISA). It seems that all the major valve manufacturers have developed their own empirical noise formulae based on extensive laboratory testing of their unique valves. An examination of the manufacturer's technical noise bulletins shows no two identical techniques or noise formulae. However, all use the same independent variables to some degree.

These factors include the flow rate (or Cv factor), $\Delta P$, inlet and outlet pressure, pressure ratio ($\Delta P/P_1$), pipe size, and pipe wall thickness. For typical valves without special noise attenuating trim, the two dominate factors that determine regulation noise are flow rate **Q** (in SCFH) and the pressure ratio **($\Delta P/P_1$),** where $\Delta P$ is the difference between the valve inlet and outlet pressure and $P_1$ is the inlet pressure in psia. Taking some liberty with the manufacturer's formulas and greatly simplifying them, the following relationship describes the impact of these two independent variables on regulation noise levels:

$$\text{Noise dBA} \propto 20\text{Log}_{10}(Q\,R^n)$$

where, $Q =$ SCFH

$R = \Delta P / P_1$

$n \approx 1.5$

The value n varies somewhat, depending on the R value, indicating a more complex relationship. Also, other independent variables are used by the manufacturers to more precisely predict the noise level such as $P_1/P_2$, $\Delta P$, and other factors previously noted.

Caution must be exercised with this formula. It is not intended to calculate absolute noise levels. Further, it is most representative of standard valves without special noise attenuating features. Focusing on these two variables illustrate their approximate relationship in predicting noise levels. The absolute noise determination is more complicated, requiring the factoring in of other independent variables. Furthermore, the pipe diameter and schedule, the physical location of the regulator station, and most importantly, the valve style have significant impact on the noise

level. Calculation of the noise level is best accomplished with the valve manufacturer's formulae or noise program software.

The reason these two variables are significant in determining noise levels can be seen after some exploration of the physics. It is reasonable to expect noise intensity to be proportional to flow rate, all else being equal; that is valve size, pressure ratio, and other factors previously noted. For instance, if the mass flow rate through a valve is doubled, but the velocity remains constant, the noise intensity should double. In fact the calculated noise level consistently increases 6 dBA for each doubling of flow assuming all else is constant. The math is left to the reader. The table below shows this relationship for a hypothetical regulation station assuming the only factor that changes is the flow rate.

Table II. Noise related to flow rate

| **Flow Rate MSCFH** | **Noise dBA** |
|---|---|
| 250 | 80 |
| 500 | 86 |
| 1,000 | 92 |
| 2,000 | 98 |

Concerning the pressure ratio, it is known that the predominate control valve generated noise is related to the cube of the mach number or the gas velocity through the valve. The velocity is generally proportional to the square root of the pressure ratio R (see ISA flow equation). Thus, $(\sqrt{R})^3 = R^{1.5}$.

What does this second factor tell us? Among other things, that pressure ratio, unless there is noise mitigating features, is the primary culprit creating regulation noise. More importantly, it is a misconception to think of pressure reduction alone as a determining factor. The pressure ratio is often the key independent variable determining regulation noise. When considering the noise potential of a station, it is common to think in terms of pressure reduction as the predominate factor. Often this is not the case. At least for standard globe valves it is the pressure ratio that is the dominate factor. Table III illustrates this principal.

Table III

Mean dBA for two typical single stage globe valves calculated using valve manufacturer's noise programs. Flow 500 MSCFH. Six-inch valve with six-inch standard wall outlet pipe.

| **$P_1$ - psia** | **$\Delta P$ - psi** | **R** | **dBA** |
|---|---|---|---|
| 200 | 20 | 0.1 | 77 |
| →200 | 100 | 0.5 | 94 |
| 200 | 180 | 0.9 | 102 |
| →1000 | 100 | 0.1 | 75 |
| 1000 | 500 | 0.5 | 93 |
| 1000 | 900 | 0.9 | 101 |

Note that a large 500 psi drop at 1,000 psia inlet results in no more noise than a moderate 100 psi drop at 200 psia inlet. The key is that both have the same pressure ratio.

What happens if pressure reduction is considered the fundamental variable? Observe from the table the two 100 psi pressure reductions (marked with an →), one with a 200 psia inlet and the other a 1,000 psia inlet pressure respectively. Although the pressure reduction is identical for 200 psia and 1,000 psia, the noise varies dramatically, 94 and 75 dBA respectively. The 19 dBA difference represents close to 80 times the sound intensity. The higher pressure ratio (0.5 for the former) accounts for the dramatic noise level difference.

In many cases the pressure ratio **R** is the dominate variable for determining regulation

noise. Flow rate **Q** is the second most significant variable.

**Worst Case Noise**

Adequately sizing a regulator requires determining worst case conditions. Typically, worst case conditions for sizing a district regulator station entail maximum station flow rate at minimum expected inlet pressure. This method logically expects minimum inlet pressure to correspond with maximum flow.

What conditions determine worst case noise potential for a district regulator station? It may be tempting to apply the worst case sizing criteria. However, worst case sizing conditions involve the minimum inlet pressure, a condition less conducive to noise generation since the most important variable, pressure ratio, is at its minimum. A number of pressure and flow conditions may have to be evaluated to determine worst case noise conditions. This requires more homework on the part of the designer who must gather much more data and run that data through the manufacturers software to determine worst case noise. The upfront additional homework will save many times over the cost of retrofitting a too noisy station. The highest potential noise will often occur close to when the product of flow rate times the pressure ratio raised to the 1.5 power ($QR^{1.5}$) is at its maximum value.

After the worst case noise scenario is determined, utilize the manufacturer's programs to accurately predict noise potential with specific valves and noise attenuating equipment if applicable.

**Frequency Distribution**

Typical control valves create the bulk of their acoustic noise in the 500 to 10,000 Hz frequency range. Pipe adjacent to the control valve are excited according to their harmonic ring frequencies. Environmental noise is further determined by the efficiency with which the pipe surface radiates sound to the surrounding air. Unfortunately, pipe effectively transmits the acoustic sound frequencies most commonly generated by regulators. Peak noise at regulating stations is typically in the 500 to 3,000 Hz frequency band dependent primarily on the pipe diameter.

Typically the most energetic frequencies (f) generated by a control valve are related to the "Strouhal number" (Ns), velocity (V), and valve slot or hole width (D) as follows:

$$f = (V)(Ns)/(D)$$

Ns is nominally 0.22

It can be seen that the frequencies generated can be affected by the hole or slot size of the control valve trim. More on this later.

**Velocity Affects**

Manufacturer's noise predictions typically do not take into account the additional noise that can be generated because of high gas velocity in the valve exit throat and/or adjacent pipe. An outlet velocity of mach 1 will exceed the regulation (throttling) noise and become the determining noise source. A good rule of thumb is to limit the valve body outlet velocity to mach 0.5 (about 700 fps for natural gas) so that it does not contribute in a measurable way to noise generation. Valves with noise mitigating trim often specify outlet mach numbers as low as 0.3 to be effective.

## Two Noise Attenuation Approaches

The two basic approaches used to attenuate noise are source treatment and path treatment. Source treatment reduces the noise that would otherwise be generated within the control valve, while path treatment reduces the noise after it has been generated by the control valve.

### Source Attenuation

A common source treatment method of noise attenuation is use of the "quiet trim" style control valve. The "quiet" term is intended as a generic description of this type of valve, not a specific brand. All the major control valve manufacturers produce quiet trim type control valves with noise attenuating trim. This is one of the most effective and least cost methods of attenuating regulator noise. Globe style control valves are particularly amenable to quiet trim modifications. They come in a variety of trim packages depending on the severity of noise and pressure ratio.

The most common form of quiet trim uses small, strategically placed holes or slots producing small fluid jets and eddy currents. As noted previously, the noise generation frequency is shifted higher since it is inversely proportional to the hole diameter. The higher frequency is less conductive to pipe transmission or to that perceived by the human ear. The following from the Masoneilan "Noise Control Manual" describes how these valves achieve noise attenuation:

> The size of the fluid jets affects noise generation in three ways. First, by reducing the size of the fluid jets, the efficiency of conversion between mechanical and acoustical power is reduced. Secondly, the smaller eddies shift the acoustic energy generated by the flow to the higher frequency regions where transmission through the pipe walls is sharply reduced. Thirdly, the higher frequency sound, if raised above 10,000 Hz, is de-emphasized by both the A-weighting filter network and the human ear.

This form of quiet trim is common to all the major control valve manufacturers. Proper placement of the holes encourages mutual interference downstream at a location calculated to minimize shock-eddy interaction.

For a minimum cost over standard trim, this basic form of quiet trim achieves around 10 to 18 dBA environmental noise attenuation as long as the pressure ratio stays below 0.65. This is a significant reduction achieving the most "bang for the buck" related to noise attenuation methods. Unfortunately, its effectiveness rapidly diminishes with pressure ratio, effectively vanishing when the pressure ratio exceeds 0.9. Furthermore, the noise attenuation effectiveness is also dependent on an exit mach number less than 0.5, otherwise noise generated in the valve outlet and pipe will overwhelm the noise generated due to throttling.

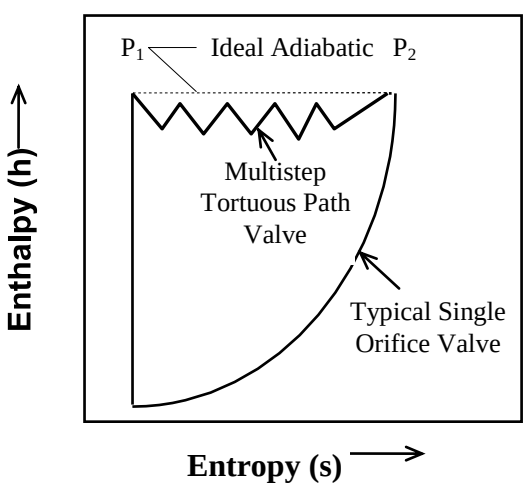
Manufacturers have developed quiet trim using the same principal of smaller slots for quarter turn valves. For example, the v-notch style ball valve and the ball valve with quite trim layers. Similar noise attenuation is achieved over standard trim.

An interesting and recently observed nuisance caused by quiet trim valves is their greater interference with ultrasonic meter measurement.

The ultrasonic meter determines flow rate by measuring acoustic transit times through a flow stream. Ironically, shifting the regulation noise into a higher frequency band less sensitive to the human ear, is more likely to cause interference with the acoustic pulses emitted by ultrasonic meters. The meter has difficulty distinguishing between the acoustic pulse and ultrasonic noise generated from the quiet trim valve, that is, quiet for the ear, noisy for the ultrasonic meter. In contrast, standard "noisy" valves with their large ports, particularly quarter turn valves, cause less observed interference with the acoustic pulses.

Further enhancement of the quiet trim valve involve layers of high flow restrictive plates to provide a tortuous flow path dissipating energy through high head loss rather than through turbulence shear and shock waves. This style valve also uses the principal of shifting the frequency to higher bands. However, it adds significant noise attenuation capability by achieving the pressure reduction in a series of small incremental steps.

The noise attenuation principle is illustrated below.




Solid curved line represents the thermodynamics for a typical single stage control valve, globe or quarter turn. Darker saw-tooth line represents the thermodynamics for a multilayered tortuous path valve.

The conventional single orifice valve creates one large pressure reduction at almost constant entropy, converting pressure energy (enthalpy) into velocity (kinetic energy). Downstream turbulence reconverts this velocity into thermal energy with a permanent increase in entropy and the desired pressure reduction. With the tortuous path valve, the pressure is reduced in a series of small incremental changes thus minimizing velocity changes that create valve noise.

This type of valve consists of a series of plates with small holes on the upstream side which are gradually increased in size toward the downstream section thus achieving constant velocity throughout the pressure reduction process. This design is for the severest applications where large flow and pressure cuts are experienced for example withdrawal control valves at underground storage fields. They are capable of achieving noise attenuation exceeding 30 dBA.

Unfortunately, tortuous path valves have five major disadvantages that must also be considered.
First, they are much more expensive than standard valves.
Second, their small restrictive holes and layers of plates significantly reduce the valve capacity. For instance an eight-inch valve may have no more capacity than a three-inch standard trim valve.
Third, their small passages are more susceptible to plugging from debris in the line or from hydrate formation resulting from the throttling process.

Fourth, they limit the valve characterization available to accomplish the most effective station control -- the valves ultimate purpose. Most stations operate best with equal percentage trim, the tortuous path quiet trim valves are primarily linear in nature.
Fifth, their rangeability (or minimum to maximum stable operation zone) is less than that of standard trim valves.

## Path Attenuation

Path attenuation techniques commonly used are; heavier wall pipe, larger diameter pipe (to reduce flow stream velocity), acoustical insulation, in line diffusers or silencers, and restriction plates. For extremely noisy stations, both path and source treatment can be used.

It is important to recognize that some forms of path treatment do not attenuate the noise in the pipe, but rather block its emission. The noise reappears with very little attenuation where the path treatment ends. For example, acoustic insulation may reduce the noise where it is wrapped by 15 dBA. However, where the wrap stops, the noise will be at the same level as an unwrapped pipe. Unfortunately, acoustic noise energy inside a pipe attenuates very slowly along the length of the pipe. Furthermore, this form of path treatment does not eliminate the destructive features of high intensity noise.

**Heavier wall pipe** reduces the noise level roughly 3 dBA for each increase in pipe schedule. As noted, to be effective it must be utilized the length of the above ground piping.

**Larger diameter pipe** reduces the noise level about 2 dBA for each nominal pipe size increase. This is based on the decrease in velocity for the same flow rate.

**Acoustic insulation** can attenuate noise over 20 dBA with several inches of acoustic blanket. Some acoustic wraps achieve some degree of sound absorption in addition to sound blockage. Usually, a mineral wool is wrapped next to the pipe and metal sheeting wrapped over the wool. The seams of the metal wrap must be carefully sealed to prevent cracks that allow noise emission or moisture entrance points. All exposed piping downstream and sometimes part way upstream must be wrapped for this method to be successful. Furthermore, the regulators and valves must also be wrapped with a noise blanket to prevent noise emission.

A serious concern with acoustic insulation is the possibility of moisture entrapment under the wool blanket adjacent to the pipe. A porous wrap that permits condensation to accumulate is a sure recipe for corrosion pitting. It is critical that the acoustic wrap be installed to prevent condensation accumulation. All holes, cracks, and seams must be completely filled with a water repellent sealant.

**Line diffusers or silencers** are installed immediately downstream from the noise producing regulator. They achieve their pressure attenuating features using some of the same techniques common to quiet trim valves. They have the further benefit of absorbing the sound energy so that it does not propagate downstream. They are capable of noise attenuation up to 25 dBA. Even greater noise attenuation is achieved in conjunction with quiet trim valves. In addition to added cost, they reduce the station capacity due to their restrictive design.

**Restrictive plates** achieve the same purpose as a quiet trim valve by shifting the frequency higher and making a series of gradual pressure reductions. The restrictive plate or plates are

placed immediately downstream of the regulator and cause another pressure reduction across the plate thus lowering the severity of the pressure cut and hence noise produced by the regulator. They can achieve noise attenuation approaching 20 dBA when used with standard trim valves.
They have two drawbacks. First, they reduce the capacity of the regulator. Second, they are not as effective if the flow rate drops significantly below design resulting in a slight pressure reduction across the plate so that the regulator takes the bulk of the pressure cut with the attendant high noise levels. Restrictive plates are effective if properly matched to a given pressure and flow rate.

**Burying the Station**

A very effective noise attenuation method is to bury the control valves or place the regulators in underground vaults. Quarter turn control valves buried at least four-feet below surface achieve better than 30 dBA noise attenuation. Regulators in vaults are attenuated roughly 20 dBA from their above ground counterpart. Higher attenuation can be achieved by placing acoustic absorbent material under the vault lids. More importantly, the buried pipe is completely supported so that destructive vibrations arising from pipe resonance modes for above ground pipe are detuned in the buried condition.

Above ground piping should be supported at all piping turns (elbows, tees, etc) and heavy masses to minimize the destructive potential of noise induced vibrations.

At times, extreme measures have to be taken to blend regulation facilities into the environment, and to make them “noiseless”. The authors know of one utility that had an existing aboveground station end up on a lot adjacent to a new cul-de-sac. In order to satisfy the developer, a complete residential style home was built over the station, with sidewalks, driveway, and landscaping. To service the station, you opened the garage door and drove in. The structure was wood frame with stucco outside and normal wall insulation inside (which provided outstanding noise attenuation). The roof was equipped with additional vents and the structure had explosionproof wiring.

**PG&E Experiences**

In typical district regulator stations, PG&E has experienced a limited number of instances where dramatically higher noise occured when the regulator was switched by field personnel to being upstream of the monitor. Switching the regulator control to the downstream valve dropped the noise level dramatically. The reason for this is PG&E installs the regulator and monitor the same size which causes a high velocity through a downstream monitor, contributing to the high noise. It is also possible that resonance modes occur between the upstream regulator and downstream monitor. Should you experience an extremely noisy station and the regulator is upstream, try switching it to the downstream side. It may solve a noise problem at no cost. If downstream monitoring is preferred, then the monitor needs to be larger than the regulator.

When a fully buried major control station was rebuilt in 1985, it was thought the neighbors about 50 feet away would be unaffected. The new facilities were also fully buried or in covered pits; but due to different equipment, more congested piping, and higher flows, the noise complaints started. A concrete block sound wall was built, but was not completely satisfactory. Ultimately, fiberglass covers were placed over the control valve high heads and the underside of the pit lids were soundproofed.

Some years ago, an existing buried station in the flight path of an international airport was rebuilt to handle higher flows at a different downstream setpoint. Although the noise generated by the new regulation was below the 75dBA local noise ordinance level at the station fence line, neighbors complained. It was strictly a case of a different sound they were hearing and not the magnitude of the noise.

Several years ago, a large transmission regulator station was constructed which had two metered and regulated feeds. The ball valve regulators and monitors were buried and the downstream meters were aboveground (because they were ultrasonic). When placed in service, the regulation noise emanating from the above ground meters bothered neighbors more than 500 feet away, so a building costing nearly $200,000 had to be placed over all the above ground piping. Quiet trim control valves costing $112,000 each could have been used but would have probably detrimentally affected ultrasonic flow measurement.

At a power plant primary regulator station, a Grove Flexflo regulator (vintage 1955) was replaced with a conventional style globe valve for maintenance convenience. The existing piping configuration just downstream of the valve consisted of a flange bolted to the valve and welded to a 45 degree elbow. When the valve was placed in service, the noise generated by the new valve cracked the elbow in 5 days. Thinking the elbow was faulty, the maintenance crew replaced the ell. The new ell failed in three days, prompting contact with engineering who replaced the internals in the valve with quiet trim which has worked perfectly to this date.

Finally, and although not a control valve noise issue, the blowing of gas to atmosphere can create noise concerns. PG&E's most noted gas discharge problem was at an existing station that became surrounded by a subdivision. An automated blowdown at this station was used to evacuate a pipeline on an Interstate Highway bridge should an emergency occur. This blowdown, which had measured levels of 130 dBA, had to be relocated to the other side of the bridge and placed in a remote area more than 5 miles from its original location to satisfy all concerned.

## Conclusion

Forethought in designing for regulator station noise can ensure that the noise level does not exceed code or destructive limits. The two most important variables determining regulation noise are the flow rate through the control valve and the pressure ratio (defined as the pressure cut divided by the absolute station inlet pressure).

Numerous techniques are available to attenuate regulation noise. Manufacturer supplied quiet trim valves provide a cost effective means of source treatment. Additional attenuation can be achieved by wrapping pipe and components with acoustic blankets or installing in-line silencers. Heavier wall pipe and/or larger diameter pipe will also provide some minimal additional noise attenuation.

With the tools available for noise attenuation, even the most severe pressure cut at the highest flows can be attenuated to acceptable noise levels. However, always consider that "One man's music is another man's noise."

## References

The authors recommend the manufacturer's technical bulletins or manuals for additional

help in dealing with control valve noise issues. Three excellent sources used for this paper are the *Fisher Noise Catalog*, the *Masoneilan Noise Control Manual*, and the *Valtek Understanding Noise bulletin*.