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Fundamentals of Aerodynamic Noise in Control Valves

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This paper provides a basic understanding of the mechanisms by which aerodynamic noise is generated in a control valve. A fundamental approach is taken to the description of these noise mechanisms and the terminology associated with noise. This terminology is consistent with that used in the IEC valve noise prediction standard.

Fundamentals of Aerodynamic Noise in Control Valves

THE CONTROL VALVE AS A NOISE SOURCE

Control valves are a major source of noise in any industrial environment. A well designed control valve can reduce its noise to an acceptable level, but in order to ensure that an appropriate valve selection is made for any given application, one must understand how and why control valves produce and control noise, and how the amount of noise produced by a control valve can be predicted before purchasing the valve and placing it in service.

INTRODUCTION TO NOISE

Sound is the physiological phenomenon which occurs when fluctuations in air pressure register against our eardrums. Our brain then interprets these fluctuations as sound. Not all air pressure fluctuations, however, result in sound being interpreted by the brain. Only those pressure fluctuations in the general frequency range of 20 Hz to 20,000 Hz, for individuals with good hearing, are detected and interpreted by the brain as sound.

Air pressure fluctuations which are outside the range of human hearing (audible range) simply do not register in the brain as sound. For example, normal changes in atmospheric pressure occur so gradually that they are a much lower frequency than the audible range and therefore cannot be heard. For this, we can be truly thankful. Likewise, there are some sounds, such as a dog whistle, which are sufficiently high enough in frequency that they are above the range of human hearing. Most dogs, however, can hear frequencies well above the audible range of humans.

Since sound is defined as air pressure fluctuations against the eardrum, the implication seems to be that if there is no ear, there is no sound. Every beginning physics student sooner or later hears the question, “If a tree falls in the forest, and there is no one there to hear it; is there sound?” This type of philosophical question may be fun to debate at cocktail parties, but it sheds little useful light. For example, imagine a control valve in a pipeline which is passing a large amount of turbulent flow. The turbulent fluid downstream of the valve produces vibrations in the pipe wall which in turn disturb the surrounding air causing air pressure fluctuations to eventually impinge upon the ear drum of a person standing nearby the pipeline. The observer might state that there is noise coming from the valve, yet there is no observer inside the valve or the flowing fluid to hear the noise. On the other hand, it is illogical to believe that there is no noise inside the valve or flowing fluid, but somehow noise magically appears outside the pipeline. Therefore, this manuscript will take a much more pragmatic approach to this issue and talk about noise inside the valve, the fluid, the pipeline, etc..

The amount of energy contained in the pressure wave (i.e., its sound power) also affects the ability of the ear to register the pressure wave as sound. There are some pressure waves whose energy levels are low enough that it does not register as sound in the human ear. In other words, the pressure disturbance is below the human threshold of sound, or in common parlance, it is not loud enough for us to hear. In other words, how loud the noise sounds is a measure of the power of the sound wave.

Although the term “loudness” is one that would feel very comfortable to most of us, it is a term which will not be used in this manuscript. “Loudness” has a technical definition which is used in the field of architecture, but has no significance within the scope of this discussion. We may use relative terms indicating that one noise is more (or less) loud than another, but we will not attempt to quantify “loudness” as such.

The greater the power of the pressure wave disturbance, the louder the sound; however, this is not a linear relationship. We cannot double the power of the sound wave and produce a sound which is twice as loud. In order to quantify how loud one sound is compared to another, we will use either the Sound Power Level or Sound Pressure Level of the noise disturbance. Both of these sound measurement quantities follow a logarithmic relationship which we will define shortly. We measure both of these quantities in units called “decibels” or “dB.” Measuring things in units of decibels (dB) is outside our range of normal experience and may seem complex, or even magical, at first, but it is really quite straightforward once one learns the basic principles involved. Dealing with decibels is so fundamental to the field of valve noise it is essential that we have a good grasp of this type of measurement. If you feel uncomfortable about your understanding of decibels as a form of noise measurement, a review of Fisher Control’s Technical Monograph, TM-42, “Understanding Decibels, (dB or not dB)” will allow you to talk about decibels like an expert.

The human ear does not possess the same sensitivity to noise at every frequency in the audible range. Sound at some frequencies tend to seem louder than sounds at other frequencies, even though the power level of the sound waves are the same. For example, sounds at the high frequency end of the audible range typically do not seem as loud as sounds of the same power level at middle range frequencies. Research has shown that the human ear is most sensitive to sounds at a frequency of 1000 Hz.

In order to have an equitable way to compare the effect of two sounds on the human ear, scientists have developed a weighting scale (called “A-weighting”) which adjusts the dB sound level at other frequencies to a level which would be the perceived equivalent to a sound at 1000 Hz. To determine the amount of adjustment, the power level of a sound at 1000 Hz is adjusted until it sounds equally as loud as the test noise. The test noise is then assigned the value of the sound power level of the 1000 Hz sound, regardless of the actual power level of the test noise. These units of adjusted power level are then listed as dB(A) or dBA and are typically used when dealing with government regulations which limit the amount of sound to which human beings may be exposed. In other words, two sounds which appear to be equally loud would have the same dB(A) number regardless of their frequency or actual power level.

Naturally, we would expect that some of the things we have discussed so far, such as the A-weighting, frequency sensitivity of the human ear, threshold of hearing, and the audible range of sound, etc. will tend to be unique to a specific human being at a specific point in time. Since it is not possible to deal with noise prediction on this individualistic basis, scientists have developed a statistical representation of an average, 27 year old, American male. So when anyone refers to how noise affects the human ear, they are referencing this statistically derived model.

Noise is simply unwanted sound. As you might suspect, there are many psychological, cultural, and contextual factors which determine whether a particular series of air pressure waves are called sound or noise. Even the same sound experienced by the same individual may undergo different classification depending upon the context in which the sound is heard. For example, the sound emanating from an aircraft engine may be noise to some individuals, but it is sweet music to the ears of a pilot flying a single-engine airplane across a large body of water. Since the primary focus of this manuscript is the unwanted sound produced by control valves, nearly all future references will be to noise.

Sound waves cannot radiate in a vacuum. They must have some material medium to propagate the wave. In order for sound to get from its source of origin to the human ear, it must eventually pass through the air as a pressure wave which registers against the human eardrum. Before it can get to the ear, however, it may have to pass through one or more additional material media. Any reduction in the eventual sound pressure fluctuations at the ear which is caused by the noise passage through the different media, or the boundaries between these media is called “transmission loss.”

If we think of the valve as a point source of noise sending out spherical sound waves in all directions, we can recognize that only the sound radiated in a direction which can arrive at the observers ears is effective at producing perceived noise. Noise generated in the fluid at the valve must then be transmitted through the pipe due to vibration induced in the pipe wall. Due to reflections of the sound waves from the inner surface of the pipe, only the portion of the energy in the sound wave which produces a radial component against the pipe wall will be effectively transmitted through the pipe. The rest will be called “transmission loss.” Finally, as the pipe wall vibration induces pressure fluctuations in the air surrounding the pipe, these sound waves must radiate outward to the observer. The type of waves generated would be classified primarily as a “line” source which would radiate sound waves radially outward from the surface of the pipe. Additional transmission losses occur through the air as these waves expand and spread the noise energy over an ever-increasing surface area of the sound wave. Thus, additional transmission losses will occur which are dependent upon the observer’s distance from the pipe.

NOISE LEVEL

When someone asks us how loud a given noise is, the first response should be, “As compared to what?” Typically, when we measure anything, we usually measure it with respect to some reference level. For example, when we talk about a given pressure in a vessel, we are actually talking about the pressure level above or below atmospheric pressure (i.e., the reference level). Likewise, with noise or sound we also have a reference level. In this manuscript, we will use the letter (L) to represent noise "level."

The level of the noise is, of course, a function of the power generated by the noise producing disturbance. We call this the "acoustic power" or "sound power" and we designate it as (W_a). The standard noise reference level is the lowest sound power which can just barely be detected by the average person with average hearing. This statistically derived threshold of hearing is called the "reference sound power" and is designated as (W_o). Both W_o and W_a are typically measured in units of Watts of power. The statistically derived value for the reference sound power is $W_o = 10^{-12}$ Watts.

As we increase the acoustic power we increase how loud the noise sounds to us, but not in a linear fashion. For example, if we were to double the sound power, we would not get a noise that sounds twice as loud. Due to this nonlinear nature, as well as due to the wide range of sounds to which the human ear can respond, it is convenient to use logarithms when dealing with noise measurements. Likewise, it is typical to quantify how loud a noise is compared to a reference level which is chosen as the threshold of hearing. Thus, the Sound Power Level is defined as:

$$L_w = \mathrm{Log}_{10}(W_a/W_o) \qquad \text{(Units are "Bels")} \qquad (1)$$

The units of measurement for this noise level are "Bels;" named, of course, after Alexander Graham Bell, the grandfather of acoustic science. As it turns out, however, the units of Bels are normally too large for practical usage. Consequently, acoustic researchers quickly began using a smaller unit of noise measurement called a "decibel (dB)" which equals one-tenth of a Bel. We can easily convert from Bels to dB as illustrated in the example below. Imagine a noise level such that

$$L_w = 8 \text{ Bels} = (8 \text{ Bels})(10 \text{ dB/Bel}) = 80 \text{ dB} \qquad (2)$$

It is easy to conclude from this example, that if we wished to rewrite the equation (1) in terms of dB (decibel) units, it would become

$$L_w = 10\ \mathrm{Log}_{10}(W_a/W_o) \qquad \text{(Units are "dB")} \qquad (3)$$

You will often see this equation incorrectly written as $dB = 10\ \mathrm{Log}_{10}(W_a/W_o)$ which can be somewhat confusing since it is actually using the units of measurement as the name of the parameter being calculated. This is similar to saying that m^2 = (length)x(width) as the area (A) of a rectangle. We need to recognize that this incorrect practice exists, but we should refrain from perpetuating it.

COMPARING NOISE SOURCES

We don't always want to compare a particular noise source with the reference threshold of hearing. Sometimes we may want to compare one noise source (W_1) to a second noise source (W_2). We can use our noise level definition here as well; i.e., the following equation can tell us how the noise power level of source (W_2) compares to the noise power level of source (W_1).

$$\Delta L_w = 10\ \mathrm{Log}_{10}(W_2/W_1) \qquad \text{(Units are "dB")} \qquad (4)$$

If the answer is positive, it means that W_2 is so many dB louder than W_1, and if the answer is negative, it means that W_2 is not as loud as W_1.

There is another comparison which is often of special interest to us. Let's look at what would happen if we were to actually double the power produced by the noise source; i.e.,

$$\Delta L_w = 10\ \mathrm{Log}_{10}(W_2/W_1) = 10\ \mathrm{Log}_{10}(2W_1/W_1)$$
$$= 10\ \mathrm{Log}_{10}(2) = 10(0.301) = +3.01 \cong +3 \text{ dB} \qquad (5)$$

The usual custom, as inferred above, is to round the 3.01 value to an even 3. In almost all noise control problems it makes little sense to deal with small fractions of decibels. The precision of 0.1 dB is rarely required and noise levels are nearly always best stated only to the nearest decibel.

The above result means that if we started out with a noise level of 80 dB and we doubled the sound power, the noise level would increase only to 83 dB. By the same token, if we were to cut the sound power in half, the noise level would decrease only to 77 dB.

COMBINING NOISE SOURCES

In a typical industrial environment, a control valve is seldom the only noise source present in a particular location; i.e., there are usually motors, compressors, turbines, other valves, other machines, etc. which are contributing to the general noise level. Although control valves can generate a significant amount of noise, they often are not the major noise source in a given area. The noise level in a particular area is the result of combining the noise generated by each source in the vicinity. When doing this, however, it is important to recognize that **dB's don't add; the powers add**.

Imagine that you are standing halfway between two identical valves installed in two different pipelines. Consider further that you measure a noise level of 60 dB from each valve, when it is operating by itself. Now, you want to know what the noise level would be if you operated both valves simultaneously. Since the powers add (not the dB's), we know that the total resultant power would be twice as much as before. From the example above, we can see that the new noise level would only be 63 dB, not 120 dB.

What happens when we combine noise sources of two different levels? Consider the example of the two valves above, but this time while one valve produces 60 dB of noise by itself, the other produces 100 dB by itself. It turns out that 100 dB is so much louder than 60 dB that the combination is essentially still 100 dB. We know that the result is really larger than 100 dB, but the difference between the resultant overall noise level and the noise generated by the 100 dB valve by itself is negligible. This may seem strange at first, but the mystery should disappear when we consider the following.

Recalling that powers add rather than dB's, we can solve equation (3) above in reverse to discover that 100 dB represents a power of 0.01 Watt (Don't forget to take into account the reference power). On the other hand, 60 dB represents a power of only 0.000001 Watt. When we combine these two noise sources to get a total of 0.010001 Watt we can see that there is indeed negligible difference between 0.010001 and 0.01. These two examples allow us to develop two useful rules of thumb:

RULE NO. 1: When a secondary noise source is combined with a louder noise source, the overall resulting noise level cannot exceed 3 dB greater than the loudest source by itself. This maximum would occur only when the secondary noise source became equally as loud as the loudest primary source.

RULE NO. 2: When trying to reduce the overall noise, determine which noise sources are the loud, dominant ones and correct them. Negligible improvement will be attained by eliminating a minor noise source. Even eliminating one of two equal, dominant sources will only result in a 3 dB improvement.

Table 1 below provides a practical aid in helping to combine two different noise sources or to aid in determining how much improvement would be gained by quieting one of several different noise sources.

Table 1: Combining Two Point Noise Sources

dB Difference Between Two Sources	dB Difference Between Total Noise and Louder Source
0	3.01
1	2.54
2	2.12
3	1.76
4	1.46
5	1.20
6	0.97
7	0.79
8	0.64
9	0.52
10	0.42
11	0.33
12	0.27
13	0.22
14	0.17
15	0.14
16	0.11
17	0.09
18	0.07
19	0.06
20	0.05

You can use Table 1 to combine any number of noise sources. You can do this by combining any two sources, then combining the result with another source, etc. As an example of how this can be done, consider the following example:

EXAMPLE 1: Use table 1 to combine four point noise sources which generate noise levels of 102 dB, 96 dB, 108 dB, and 102 dB respectively. Determine your answer to the nearest dB.

ANSWER: First combine 102 dB and 102 dB to get 105 dB. Next, combine this 105 dB with the 108 dB. The difference is 3 dB. From Table 1 then, 1.76 dB must be added to 108 dB to get 109.76 dB. The difference between 109.76 dB and the remaining 96 dB source is 13.76 dB. We could interpolate between 0.17 dB and 0.22 dB in the table; however, to the nearest dB the combined noise level is 110 dB.

SOUND POWER VERSUS SOUND PRESSURE

Although the acoustic power (W_a) generated by a noise disturbance is related to how loud the noise sounds, it is not always easy to directly measure the sound power. On the other hand, it is relatively easy to measure the noise induced air pressure disturbances (p_d) that exist in the vicinity of our ears. From a pragmatic point of view, we are often more interested in noise at the point where the noise pressure fluctuations impact our ears than we are at the actual source of the noise. All of this leads us to the conclusion that it would be useful to understand how sound power and sound pressure are related to each other and to the noise level.

It is a fundamental fact of nature that "power" is always proportional to the square of the "potential." In electrical systems, for example, we all know from our high school physics that electrical power is proportional to the square of the electrical potential (E). Likewise, the acoustic power is proportional to the square of the sound pressure. We can express this as an equation where C_1 is a constant of proportionality involving the area (A), the density (ρ), and the speed of sound (c).

$$W_a = (A/\rho c)\,(p_d)^2 \qquad (6)$$
$$W_a = C_1(p_d)^2 \qquad (7)$$

If we combine this fact with another fundamental fact about logarithms [$\text{Log } X^a = a\text{Log } X$], we can write two different "level" equations which are simply two different ways of looking at what is essentially the same phenomenon.

$$L_{pwr} = 10 \text{ Log}_{10}(W_a/W_o) \qquad (3)$$

$$L_{pres} = 20 \text{ Log}_{10}(p_d/p_o) = 10\text{Log}_{10}(p_a/p_o)^2 \qquad (8)$$

NOTE: In both equations, the units are "dB".

The statistically derived value for the reference sound pressure is $p_o = 2\text{x}10^{-5}$ Pa.

NOTE: L_{pres} is often written as L_p and is called "Sound Pressure Level."

Equation (8) tells us that if we double the sound pressure we will get a 6 dB increase in noise level; whereas equation (3) tells us that if we double the sound power we will get only a 3 dB increase in noise level. At first, this sounds inconsistent to some individuals until they realize that we are not talking about the same thing in these two cases.

Equation (7) reminds us that when we double the sound pressure, we actually quadruple the sound power. Now, when we double the sound pressure in equation (8) we must quadruple the sound power in equation (3) in order to be consistent. Thus, in both cases, we will get a 6 dB increase in noise level. This leads us to another rule:

RULE NO. 3: The noise level of any given noise disturbance which exists in the environment will result in exactly the same number of dB's, regardless of whether we use the sound power or sound pressure to determine it, as long as we remember and account for the relationship described by equation (7).

CAUTION: Although equations (8) and (3) are closely related to each other as we have indicated above, we should be careful about concluding that these two equations are always equal to each other because as we shall shortly see, they are not.

The relationship between sound power and sound pressure is much like the relationship between mass flow and pressure in a pipeline. As we proceed down a long pipeline, the mass flow is the same at every point even though the pressures will be different due to friction losses, changes in pipe diameter, changes in temperature, etc. Likewise, sound power will be the same as it "flows" outward from the source, but the sound pressure will vary at different distances from the source. To illustrate, let's take the simplest case of a point source of noise.

If a sudden noise disturbance, such as an explosion, occurs at a point it will generate a spherically shaped sound pressure wave which emanates outward from the source in all directions. Even though the total power of the wave is the same at every point away from the source, this total power is being spread uniformly over an increasing surface area of the spherical wave. Thus, the Watts/m^2 will decrease dramatically at any point on the spherical surface as the surface area increases when we move further and further from the source.

Since the sound pressure at any point is proportional to the Watts/m^2 at that point, rather than the total power, it follows that the sound pressure, which is what we typically measure, will be different depending upon how far away we are from the noise source. This situation is analogous to the phenomenon we witnessed as children when we tossed a rock into a quiet pond. As the rock entered the water it would produce a disturbance which generated a circular wave which traveled outward from the source. As we watched, we would see the amplitude of these waves decrease as they moved further and further from the source. Even though the total power in each wave was staying the same, it was getting dispersed over the rapidly increasing circumference of the wave circle. Eventually this total wave power would get so dispersed that there would be no detectable amplitude of the wave left.

The dispersal of power and corresponding decrease in sound pressure as we move away from the source is the reason why it is important to have a standard reference point for measuring and comparing the sound pressure levels of noise sources. If we don't measure them at the same distance from the source each time, there would be no basis for comparison. The standard distance for measuring is typically one meter from the source.

We can see now that equation (3) may tell us that the noise power level at the source is perhaps 100 dB, while at the same time we may measure a sound pressure level at the one meter reference point of only 85 dB. Of course, the sound pressure level would be even less as we moved further from the source, even though the TOTAL sound power level would still be 100 dB.

In valve noise studies, we are often talking about looking at changes or reductions in noise levels. It is reassuring to note that a 10 dB reduction in noise is always a 10 dB reduction whether we are talking about sound power level or sound pressure level. For instance, in the hypothetical example above, a 10 dB reduction in noise would reduce the sound power level at the source from 100 to 90 dB, while the sound pressure level measurement at the standard reference point would be reduced from 85 dB to 75 dB.

One final point should be made about the differences between sound power level and sound pressure level. Some people get very confused about this issue and tend to make their distinctions between the two cases based on things such as the coefficient in front of the Log term (i.e., 10 or 20), or they make the distinction based on whether the equation contains a power term (W) or a pressure term (p). Neither of these distinctions by themselves can lead one to the correct conclusion. Equation no. (8) makes it clear that the coefficient method is unreliable. The following rule of thumb can assist you in making a definitive distinction between sound power level and sound pressure level:

RULE NO. 4: If the log equation contains a power term ratio such as (W_1/W_2), the expression is most certainly sound power level. If the log equation contains a pressure ratio and an area term; e.g., $(p_1^2A_1/p_2^2A_2)$, the expression is also sound power level, because as equation (6) reminds us, the square of the pressure acting over a specific area is actually a measure of power. Only when the log equation contains a pressure ratio, but is stripped of the area relationship can we consider the equation as sound pressure level. The following examples should help to illustrate this rule:

$L = 10Log_{10}(W_1/W_2)$	Sound power level
$L = 10Log_{10}(p_1^2/p_2^2)$	Sound pressure level
$L = 10Log_{10}(p_1^2A_1/p_2^2A_2)$	Sound power level
$L = 10Log_{10}(p_1/p_2)^2$	Sound pressure level
$L = 20Log_{10}(p_1/p_2)$	Sound pressure level
$L = 10Log_{10}[(p_1/p_2)^2(A_1/A_2)]$	Sound power level

NOISE FREQUENCY AND WAVELENGTH

Sound, from the point of view of control valve noise, consists of pressure disturbances produced and propagated by waves in solid or fluid media. For practical purposes, the most important type of wave is the simple harmonic or sinusoidal wave shown in Figure 1. This is a snap-shot picture of one cycle of a simple harmonic sound wave.

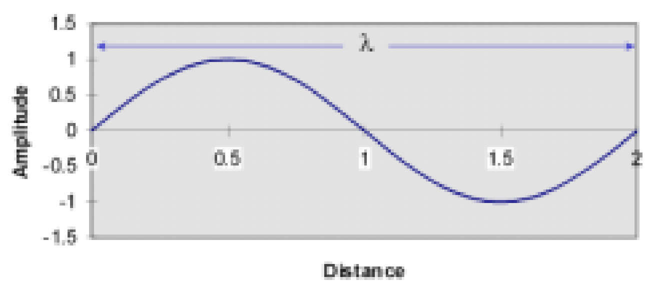

Figure 1: Simple harmonic sound wave

For illustration purposes, we can think of this as a single-frequency sound wave traveling through the air in the x-direction. The ordinate measures the amplitude of the air pressure disturbance above and below the ambient atmospheric pressure. It is this varying pressure against our ear drums that registers as sound.

Figure 1 only shows one complete cycle of the sound wave. The physical distance spanned by this one complete cycle is called the wavelength (λ). Actually, the distance between any two corresponding points on the wave is the wavelength. Typically, the wavelength is measured in meters (m).

The wave propagation corresponds to the motion of the whole sinusoidal figure to the right along the x-axis with a velocity (c) which is the speed of sound in the air (meters/second).

The number of complete wavelengths (cycles) which pass any given point on the x-axis per second is the frequency (f) of the wave in cycles/sec (Hertz).

By concentrating on the dimensions of these fundamental factors, we can logically combine them in such a way as to derive the fundamental relationship between wavelength (λ) and frequency (f). In other words,

(cycles/sec)(meters/cycle) = (meters/sec)

$(f)(\lambda) = (c)$ which is often written as

$$f = c/\lambda \qquad (9)$$

Thus, we arrive at the logical conclusion that the shorter the wavelength of the sound, the higher the frequency. As we shall see later, this is a fact which is of great importance in noise abatement using drilled-hole valve trim.

NOISE FREQUENCY SPECTRUM

Industrial noise, including control valve noise, is rarely a pure tone such as that shown in Figure 1. The noise is usually made up of a whole band of frequencies, with varying sound power at each frequency. In order to represent this type of noise, a noise frequency spectrum is often used. Webster's dictionary defines a spectrum as any collection of radiant energy arranged in order of their wavelengths. In noise studies, it is more conventional to deal with the frequency rather than the wavelength. Thus, the noise frequency spectrum is a band of frequencies displayed in order of their frequencies. The audio spectrum, as was stated earlier, is typically defined as the band of frequencies from 20 Hz to 20,000 Hz. These frequencies are normally plotted along the horizontal axis of a chart, while the corresponding sound power for each frequency is plotted along the vertical axis as shown in Figure 2.

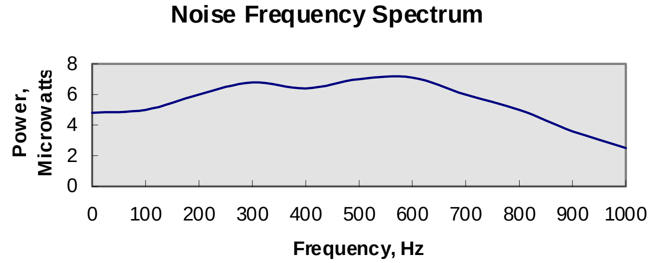

Figure 2: Noise Frequency Spectrum

For the type of noise generated in a control valve, the noise power spectrum tends to be more of a "hay stack" shaped curve, such as shown in Figure 3, rather than the type of irregularly shaped curve shown in Figure 2.

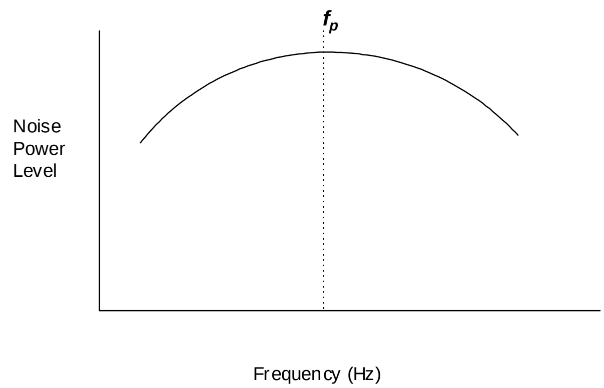

Figure 3: "Hay Stack" Noise Power Spectrum

Thus, as the noise being generated in the valve changes, the power spectrum curve rarely changes its basic shape, but it may move up or down as the peak noise power level changes, and it may move right or left as the frequency (f_p) changes at which the peak power occurs.

Strouhal, an early experimenter in turbulent flow, discovered that in general, the tone frequency of noise due to turbulent flow was proportional to flow velocity and inversely proportional to some characteristic dimension. To turn this proportionality into an equation, he invented what is known as the Strouhal number (S_T). Thus, the frequency equation becomes

$$f = \frac{S_T U_{vc}}{D_j} \qquad (10)$$

Since our noise studies typically deal with valve flow, the velocity we need becomes the velocity at the vena contracta (U_{vc}), and the characteristic dimension associated with that velocity becomes the diameter of the jet at the vena contracta (D_j).

When experimenters plotted noise power level versus frequency as shown in Figure 3, it was discovered that the peak frequency of the "hay stack" noise spectrum typically occurred around a Strouhal number of two tenths (i.e., $S_T = 0.2$).

A-WEIGHTING OF NOISE LEVEL (dB(A))

In literature concerning noise, we often encounter the terms dBA or dB(A). These terms both refer to what is known as A-weighted noise, but dB(A) is the preferred designation. Weighting a noise measurement consists of modifying a measured sound pressure level by some weighted factor which is a function of frequency. The A-weighting factors are intended to adjust actual, measured sound pressure levels at each frequency band to the sensitivity of the human ear.

The numerical value of the A-weighting factor at any frequency is determined by how loud a noise sounds compared to how loud a 1000 Hz tone appears to be. In other words, at 1000 Hz, the A-weighting factor is unity (1.0). The sound pressure level of a 1000 Hz tone is adjusted until it appears to be equally as loud as the unknown noise source. If the sound pressure level of the 1000 Hz tone measures 105 dB when that match occurs, we say the unknown source "sounds like" 105 dB, regardless of what its sound pressure level would measure. For example, if we listen to a sound at 50 Hz, which to us appears to be just as loud as the 105 dB was at 1000 Hz, we say the 50 Hz tone sounds like 105 dB regardless of the measured sound pressure level. In this case we would say that the 50 Hz A-weighted noise level is 105 dB(A).

If two or more sounds at different frequencies sound equally loud, they are the same dB(A), regardless of what their individual sound pressure levels may be. Government standards are written in terms of dB(A) since they are really more concerned about how loud the noise actually sounds, rather than with what the sound pressure level actually is.

A-weighting of noise level is a much more significant factor for hydrodynamic noise (noisy liquid flow) than for aerodynamic noise (noisy gas flow). The reason for this has to do with the sensitivity of the human ear. The response of the human ear to noise is fairly flat in the frequency range of 600 Hz to 10,000 Hz. This means that in this frequency range there is negligible difference between the actual measured sound pressure level and the A-weighted noise level.

Since aerodynamic noise in a valve is generated primarily in this same frequency range of 600 Hz to 10,000 Hz, the A-weighting factor is essentially unity. Thus, whether we predict the noise level in dB or in dB(A), we will arrive at approximately the same number when we are dealing with aerodynamic noise.

On the other hand, hydrodynamic noise in a valve can have appreciable energy at frequencies below 600 Hz. Because the human ear is much less sensitive to noise at these lower frequencies than it is at 1000 Hz, the A-Weighting factor is going to be considerably less than unity. Therefore, when dealing with hydrodynamic noise, it is extremely important to take into account the A-weighting factor.

GOVERNMENT STANDARDS FOR NOISE EXPOSURE

Society values people and their health and welfare. In our democracy, it is considered to be the proper role of government to ensure each citizen's health and welfare through various regulations and programs. When it became apparent to various governments in the 1960's that industrial noise pollution was becoming a serious problem, federal and state governments began developing legislation to establish standards and penalties which are intended to protect people from noise pollution.

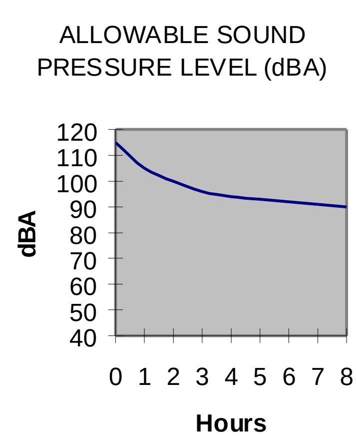

Figure 4: Federal Noise Regulation

Figure 4 shows the allowable sound pressure level which the USA government will allow an individual to be exposed to for any given duration. The allowable noise level can be louder if the individual is only going to be exposed briefly than if he or she must work a full 8-hour shift in the noisy environment. Notice also that the standard uses A-weighted sound levels (dB(A)) since the actual sound pressure level is not as important as how loud it actually sounds to the human ear.

There are three elements of noise standards which are essential considerations when dealing with industrial noise problems. These are people, noise level, and exposure time. If there are no people involved, there is no problem as far as the government standards are concerned. So, one way to address the issue is to fence off and isolate noisy equipment from people. In airports, for example, the majority of people are isolated from the jet noise either by fences, noise barriers, or by the insulated walls of the terminal building. Employees who can't be completely isolated wear ear protectors to insulate their ears from damaging noise. Since the amount of hearing damage done by noise is a function of how long an individual is exposed to the noise, various governments have developed regulations which relate noise level to exposure time.

THE CONTROL VALVE AS A NOISE GENERATOR

Noise is the result of energy dissipation in the control valve. The major sources of control valve noise are mechanical vibration of components, hydrodynamic noise, and aerodynamic noise.

MECHANICAL NOISE

Vibration of valve components is a result of random pressure fluctuations within the valve body and/or fluid impingement upon the movable or flexible parts. Noise that is a by-product of vibration of valve components is usually of a secondary concern and may even be beneficial since it warns that conditions exist which could produce valve failure. Mechanical vibration has for the most part been eliminated by improved valve design and is generally considered a structural problem rather than a noise problem. Accordingly, mechanical noise is not addressed by the IEC noise standard.

HYDRODYNAMIC VALVE NOISE

The major source of hydrodynamic noise (i.e., noise resulting from liquid flow) is cavitation which is caused by implosion of vapor bubbles formed in the cavitation process. Cavitation occurs in valves controlling liquids when the service conditions are such that at some point within the valve the pressure drops below the vapor pressure causing bubbles to form, while the static pressure downstream of the valve is greater than the vapor pressure causing the vapor bubbles to implode and release energy.

As the fluid velocity increases due to the restriction formed by the valve trim parts, vapor bubbles are formed in the region of minimum static pressure (highest velocity) and are subsequently collapsed or imploded as they pass downstream into the pressure recovery region. Noise is produced by the energy dissipation of the imploding bubbles. Noise produced by cavitation in a valve has a broad frequency range, however, it can have appreciable energy at frequencies below 600 Hz. Cavitation noise is often described as a rattling sound similar to that which would be anticipated if gravel were in the fluid stream.

Cavitation may produce severe damage to the solid boundary surfaces that confine the cavitating fluid. Generally speaking, noise produced by cavitation is of secondary concern. In addition, test results and field experience indicate that noise levels from non-cavitating liquid applications are quite low and generally would not be considered a noise problem.

AERODYNAMIC VALVE NOISE

The major source of aerodynamic valve noise (i.e., noise resulting from gas flow) is a by-product of a turbulent gas stream. A control valve controls gas flow by converting potential (pressure) energy into turbulence. Most of the energy is converted into heat; however, a small portion of this energy is converted into sound. It is possible to determine an acoustical efficiency factor (η) which indicates how much of the initial energy in the flowing medium is converted into sound. This acoustical efficiency factor varies as the valve service conditions change, and this changes the flow patterns through the valve. The IEC valve noise prediction standard defines five different flow regimes which affect the acoustical efficiency factor. Since the conditions which exist in each of these flow regimes result in slightly different noise generation mechanisms, it is important that we understand the flow regimes that have been defined by the standard.

IEC FLOW REGIMES

As the pressure drop across the valve increases, the flow energy intensifies and the flow patterns change and the noise generation mechanisms change accordingly. For purposes of the IEC noise standard, five different flow regimes are defined. The boundaries of these regimes are defined by the relationship of the valve outlet absolute pressure (p_2) to the following four pressures; i.e., the valve outlet absolute pressure at critical flow conditions (p_{2C}), the absolute vena contracta pressure at critical flow conditions (p_{VCC}), the valve outlet absolute pressure at a break point (p_{2B}), and the valve outlet absolute pressure where the region of constant acoustical efficiency begins (p_{2CE}). The break point pressure (p_{2B}) occurs at the point where the shock cell-turbulent interaction mechanism begins to dominate the noise spectrum over the turbulent-shear mechanism. Both p_{2B} and p_{2CE} will be discussed in more detail later.

The concept of flow regimes can best be understood using a graphical illustration of the flow through a control valve. For purposes of simplification, a control valve at any flow opening can be represented by a simple restriction in the line as shown in Figure 5.

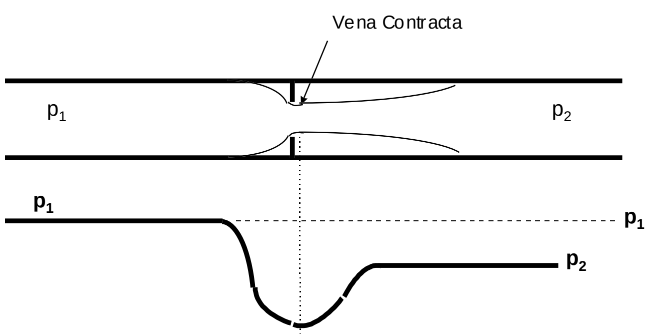

Figure 5: Valve Pressure Profile

As the flow passes through the physical restriction, there is a necking down, or contraction, of the flow stream. The minimum cross-sectional area of the flow stream occurs at a point called the *vena contracta,* which is just a short distance downstream of the physical restriction. It is important that we understand the interchange between the kinetic energy (energy of motion) and the potential energy (pressure energy in this case) of a fluid that is flowing through a valve or other restriction. To maintain a steady flow of gas through the valve, the velocity must be greatest at the vena contracta where the cross-sectional area is the least. This increase in velocity, or kinetic energy, comes about at the expense of the pressure, or potential energy as illustrated in Figure 5.

The pressure profile along the valve shows a sharp decrease in the pressure as the velocity increases. The pressure will decrease to a minimum at the vena contracta where the velocity is the greatest. What happens downstream of the vena contracta is a function of several things, including the general flow efficiency of the valve style and other external process conditions. Typically, however, there will be a decrease in velocity and a corresponding increase in pressure as the fluid stream expands into a larger area. Of course, the pressure downstream of the valve never recovers completely to the pressure that existed upstream. The pressure recovery downstream of the valve is often called “recompression.”

The pressure differential that exists across the valve is called the Δp (delta p) of the valve. This Δp is a measure of the amount of energy that was dissipated in the valve. Useful energy is lost in the valve because of turbulence and friction. The energy is dissipated primarily as heat and some noise. The greater the Δp for a given flow area, the greater the energy dissipated in the valve.

The amount of energy dissipated in the valve is influenced greatly by the design of the valve and its flow efficiency. If the valve design minimizes the amount of energy dissipated in turbulence and friction, there will be more energy left over for recovery in the form of downstream pressure. Such a valve would be relatively streamlined and would be classified as a high recovery valve. In contrast, a low recovery valve dissipates more energy due to turbulence and friction and consequently has a greater Δp for the same flow.

Two important points can be made here. First of all, the recovery properties described above are an inherent characteristic of the valve design and can thus be assigned a fixed index number (F_L) called the “recovery factor” of the valve. This will be discussed in more detail later. The second point is that it would be a mistake to believe that there is always a relationship between the Δp and the flow through the valve.

Regardless of the recovery characteristics of the valve, the amount of gas flow is determined primarily by the density of the gas, the flow area at the vena contracta, and the flow velocity at the vena contracta. Therefore, assuming the case of constant inlet pressure, if the flow area is constant, such as would be the case when the valve is wide open, any increase in flow must come from an increase in velocity at the vena contracta.

Due to the interchange of energy from one form to another, an increase in velocity at the vena contracta results in a lower vena contracta pressure (p_{vc}). This train of logic leads to the conclusion that the pressure differential (p_1 - p_{vc}) between the inlet and the vena contracta is directly related to the flow rate. The larger this pressure differential, the greater the flow. While this statement remains true, there is a limit to the amount of flow that can be achieved.

In a normal control valve design, it is impossible for the compressible gaseous fluid to achieve a velocity greater than the speed of sound at the vena contracta. Thus, when sonic velocity is reached at the vena contracta, there will be no further increase in velocity, there will be no further increase in flow, and there will be no further decrease in the vena contracta pressure. This condition is referred to as either “choked flow” or “critical flow.”

At the point where critical flow is first reached, the pressure at the vena contracta is designated as (p_{VCC}); i.e., the absolute vena contracta pressure at critical flow conditions, and the downstream pressure where this occurs is designated as (p_{2C}); i.e., the valve outlet absolute pressure at critical flow conditions.

External process conditions may force the valve outlet pressure (p_2) to drop below the valve outlet critical pressure (p_{2C}). This increase in Δp across the valve will result in additional energy dissipation (and therefore produce more noise), but it will not increase the flow through the valve nor the velocity at the vena contracta. Regardless of what happens to p_2, the pressure differential (p_1 - p_{vc}) will always be directly related to the flow.

Still assuming a constant absolute inlet pressure (p_1), we can look now at how the flow regimes are defined as a function of valve outlet absolute pressure (p_2).

FLOW REGIME I: ($p_2 \geq p_{2C}$)

Figure 6 illustrates the definition of flow Regime I (i.e., for the operating condition where $p_2 \geq p_{2C}$). In Regime I, the flow is subsonic. Since p_2 has not yet reached the critical pressure, the velocity at the vena contracta is less than the speed of sound. In this regime, the gas is partially recompressed and the amount of pressure recovery depends, of course, on the design of the valve. Therefore, it is only to be expected that the valve recovery factor (F_L) will play a role in determining the amount of sound power generated by the valve.

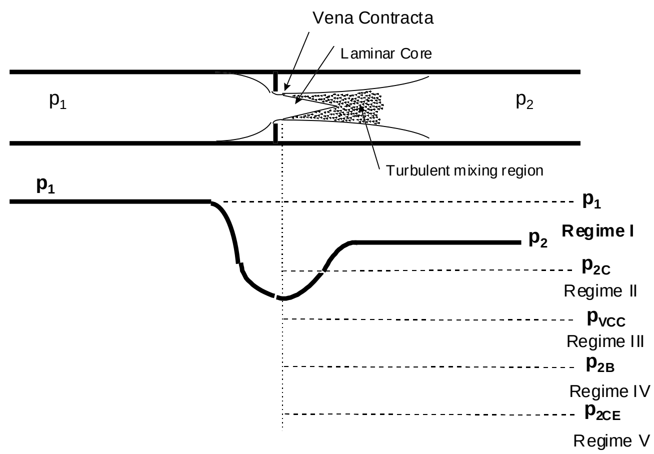

Figure 6: Flow Regime I

Note in Figure 6, that under this non-choked flow condition, there is a jet core which is essentially laminar flow in nature. Although, the jet core itself can be quite uniform, it is surrounded by an area of intense turbulence. Under non-choked flow conditions, aerodynamic noise is primarily a result of the Reynolds stresses or shear forces created in the flow stream as a result of rapid deceleration and expansion of the fluid. The principal area of noise generation is in this shear/mixing region where the flow field is characterized by extreme turbulence and mixing. This noise generation mechanism is known as “turbulent shear flow.”

FLOW REGIME II: ($p_{2C} > p_2 \geq p_{VCC}$)

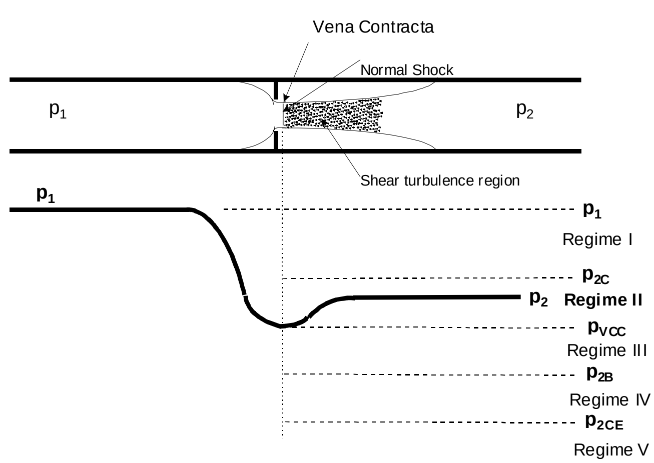

Figure 7: Flow Regime II

Figure 7 illustrates the definition of flow Regime II (i.e., for the operating condition where $p_{2C} > p_2 \geq p_{VCC}$). In Regime II the flow is sonic. Since p_2 has dropped below the critical pressure, the velocity at the vena contracta has reached the speed of sound. In this regime, some recompression still exists; however, it decreases as p_2 drops lower and lower in this regime.

Note in Figure 7, that under this choked flow condition, the laminar jet core has disappeared and a normal shock is formed at the vena contracta. This is essentially a stationary shock region that the flow must pass through. As the flow passes through the normal shock, it rapidly dissipates energy as it goes from sonic velocity to a subsonic velocity in a very short distance. Much of this dissipation of energy goes into noise. This noise generation mechanism is known as, "shock-turbulence interaction."

Downstream from this normal shock, there is still a region of even more intense turbulence than in Regime I. This, of course, is due to the greater pressure drop. Because some recompression does exist, the pressure drop across the valve will still have some effect on the amount of noise generated due to increased turbulence, even though it does nothing to increase the flow since choked flow already exists at the vena contracta. As p_2 drops lower and lower in this flow regime, the amount of noise power produced increases proportionately until it reaches a maximum at the boundary between Regimes II and III. At this boundary, the Δp across the valve will no longer have an effect on the amount of energy available in the fluid stream; however, it will have an effect upon the acoustic efficiency (η) which continues to increase the amount of acoustic power converted from the stream power.

Under these choked flow conditions in Regime II, aerodynamic noise is still primarily a result of the Reynolds stresses or shear forces created in the shear-turbulence region downstream of the normal shock, but there is some contribution from the shock-turbulence interaction. As p_2 drops lower and lower into Regime II, the normal shock begins to "protrude" further downstream. As it does so, more and more reflected waves are spawned from the shock and there is an increase in interaction between these reflected waves and the turbulent mixing region which increases the acoustic efficiency (η_2) somewhat; however, the flow-shear turbulence mechanism continues to dominate.

<u>FLOW REGIME III</u>: ($p_{VCC} > p_2 \geq p_{2B}$)

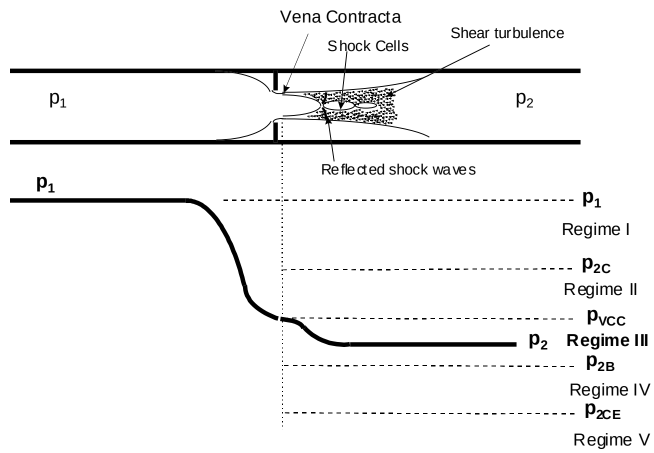

Figure 8: Flow Regime III

Figure 8 illustrates the definition of flow Regime III (i.e., for the operating condition where $p_{VCC} > p_2 \geq p_{2B}$). In Regime III the flow is still basically sonic in a macro sense; however, there will be regions of localized supersonic flow. Since p_2 has now dropped below the vena contracta critical pressure, there is no further pressure recovery, or recompression.

Note in Figure 8, that multiple shock cells have formed. Surrounding these shock cells, there is still a region of intense turbulence where some noise is generated as before, but now the shock-turbulence interaction begins to come more into play. Notice the presence of the reflected shock waves which have broken away from the shock cells and pass through the region of turbulent mixing. As these reflected shock waves pass through and interact with the turbulence, additional energy is dissipated and more noise is produced. This is an expansion of the mechanism known as "shock-turbulence interaction." Only a single shock wave is shown here, but in reality there will be whole families of these reflected shock waves formed as p_2 continues to decrease.

In flow Regime III, there is no difference in the acoustic efficiency from that in Regime II. The only major difference in noise generation is the effects of Δp which was prevalent in Regime II, but has now disappeared in Regime III. Noise is produced by both flow-shear turbulence and shock-turbulence interaction, with the latter increasing in importance as the boundary (called the "break point") between Regimes III and IV is approached.

<u>FLOW REGIME IV</u>: ($p_{2B} > p_2 \geq p_{2CE}$)

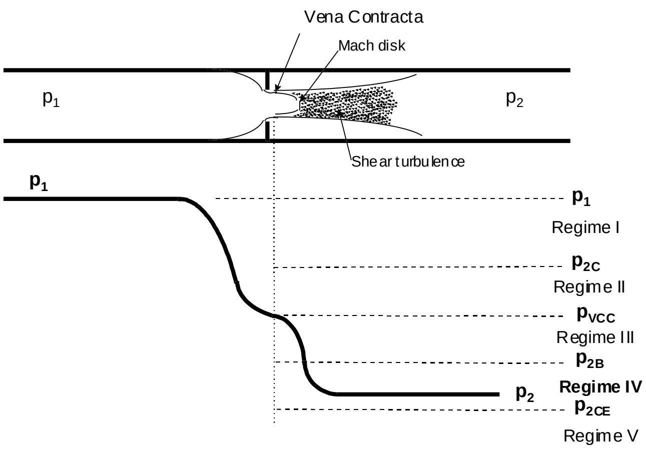

Figure 9: Flow Regime IV

Figure 9 illustrates the definition of flow Regime IV (i.e., for the operating condition where $p_{2B} > p_2 \geq p_{2CE}$). In Regime IV, p_2 has now dropped below the break point pressure, which by definition is that pressure below which the shock-turbulence interaction mechanism begins to dominate the noise spectrum over the turbulent-shear mechanism. Turbulent-shear noise generation is still present, of course, but it pales in significance compared to the shock cell-turbulence interaction noise generation.

Note in Figure 9, that the multiple shock cells have disappeared, and have been replaced by what is called the "Mach disk." You can visualize the Mach disk as a stationary shock wave with a much more intense energy gradient through which the flow must pass, dissipating energy into noise as it does so. The flow at the vena contracta is still choked sonic flow, but the flow in the Mach cone between the vena contracta and the Mach disk is locally supersonic.

Flow Regime IV is basically a transition region. There is a slight decrease in the acoustic efficiency as it transitions from the high Mach number dependence in Regime III to the constant value in Regime V.

<u>FLOW REGIME V</u>: ($p_{2CE} > p_2$)

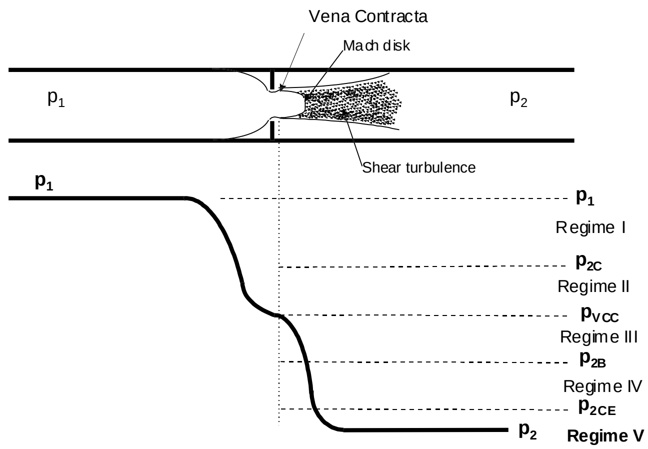

Figure 10: Flow Regime V

Figure 10 illustrates the definition of flow Regime V (i.e., for the operating condition where $p_{2CE} > p_2$). In flow Regime V, p_2 has now dropped below a pressure, beyond which there will essentially be no further increase in acoustical efficiency. Shock-turbulence interaction mechanism continues to dominate the noise spectrum, even though the turbulent-shear mechanism is still present.

Note in Figure 10, that there is little difference in the flow structure except that the Mach disk has grown slightly in diameter over that shown in Figure 9; however, there is little significance that can be attached to this from a noise generation point of view. The growth in the Mach disk is due to the somewhat higher local pressure which exists in the Mach disk area as it exits into the lower pressure downstream region. As before, the flow at the vena contracta is still choked sonic flow, but the flow in the Mach cone between the vena contracta and the Mach disk is locally supersonic.

Once the flow has entered Regime V where the acoustic efficiency remains constant, there will be no further increase in valve noise generation, regardless of how low we drop p_2.

LOW NOISE VALVE DESIGNS

The IEC noise standard basically recognizes three different categories of noise reducing trim. These are
- o Single stage, multiple flow passage trim
- o Single flow path, multistage pressure reduction trim

o Multi-path, multistage trim

The IEC noise standard develops a noise calculation procedure and equations which are intended to apply to any standard valve construction. When special low noise trims are used in valves, the general calculation procedure and most of the equations still apply, however, these special trims need some special consideration. Here we will not discuss the special techniques used in the standard. This paper will only deal with the theory behind why these trims provide noise reduction.

SINGLE STAGE, MULTIPLE FLOW PASSAGE TRIM

"Single stage" means that the flowing fluid goes from the upstream pressure condition at the valve inlet (p_1) to the downstream pressure condition at the valve outlet (p_2) in one step, or stage. This is the typical arrangement in most conventional control valves.

"Multiple flow passage" means that the flowing fluid, in going from the valve inlet to the valve outlet, passes through several flow openings rather than just one orifice. There are a couple of restrictive conditions on this definition, however, which are important to remember.

First of all, the flow passages must be sufficiently separated in distance so that there can be no interaction between the jets emanating from each flow opening. Secondly, the calculation procedures of the standard require that all of the multiple flow passages have the same hydraulic diameter (d_H). "Hydraulic diameter" is just a term used to account for the fact that each flow opening might have some unusual or irregular shape other than circular. Hydraulic diameter then simply becomes the diameter of a circular hole that has the same area as the irregularly shaped flow passage. In the case of a drilled-hole cage, the hydraulic diameter would simply be the diameter of each identical hole.

The question we will attempt to answer here is, "How do these single-stage, multiple flow passage designs reduce valve noise?" The answer may surprise you. At one time, it was widely believed that passing the flow through several holes instead of one would reduce the noise power level. The following quote is from an early article on noise reduction which attempted to explain this phenomenon. "The acoustic power of a single flow restriction increases as a function of $(C_g)^2$. Changing the area by a factor of 2 results in a corresponding 6 dB change of power level, whereas, the power level is changed only 3 dB when the number of equal noise sources is changed by a factor of two. Thus noise reduction to be derived from utilization of many small restrictions rather than a single or few large restrictions is self-evident."

The reader should clearly understand that we now know that the explanation in the previous quote IS NOT CORRECT! In fact, the amount of acoustical power generated by several small holes is precisely the same as that generated by one equivalent large hole!

The considerable success of drilled-hole cages in noise reduction is not due to a reduction of the internal noise power level, but due to the fact that the noise generated by the smaller holes has been shifted to a higher frequency where it is no longer as serious a problem to human hearing or the pipework.

Reviewing the shape of the typical "hay stack" noise power spectrum in Figure 3 will help us understand how this works. The peak power frequency (f_p) for a single hole flow opening often falls into a lower audible range (e.g., 200 - 6000 Hz) where the ear is more sensitive to sounds than it is to the upper end of the audible spectrum. By using several smaller holes to pass the same flow, the peak power frequency is moved to a much higher frequency. Thus, it is apparent from the "hay stack" curve that by moving the peak frequency higher, the power level is decreased in the lower frequency range. Therefore, even though the maximum power level is still the same, the dB(A) level is reduced because of the reduced sensitivity of the human ear at the higher frequencies.

This phenomenon produces an additional benefit as well with regard to pipe transmission loss. For a typical pipeline, the greatest coupling between the dynamic characteristics of the pipe and the internal noise field will occur in the 1000 Hz to 6000 Hz frequency range. Thus, by moving the peak frequency of the internal power higher, we reduce the amount of energy that is effective in exciting pipe vibrations and therefore reduce the radiated noise.

Let's now investigate how the use of a drilled-hole cage accomplishes this shift in peak power frequency. When a pressure wave exits from a flowing jet, the wavelength of that pressure wave is primarily determined by the diameter of the jet orifice. The wave frequency, of course, is just the inverse of the wavelength. Thus, we can see that a small hole will produce a shorter wavelength (and therefore a higher frequency) than will a larger hole. It's as simple as that! In fact, theory tells us that for a simple drilled hole cage noise trim, the amount of noise reduction is related directly to the number of holes; i.e., the more holes, the greater the noise reduction capability.

In addition, experience in the gas production, petrochemical and other industries has demonstrated that acoustic energy in high capacity, gas pressure reducing systems can cause severe piping vibrations, and in extreme cases have led to piping fatigue failures. By shifting the peak frequency of the internal power higher, we reduce the amount of energy that is effective in exciting pipe vibrations, and since stress is directly proportional to the level of vibration, there is a reduction in the potential for fatigue damage in the system. Thus, the use of many smaller holes in the drilled hole cage allows us to win two ways; less noise and less pipe fatigue.

Fisher Control's research performed on both internal and external large-scale piping systems lead to recommended guidelines for maximum valve noise levels designed to ensure safe levels of pipe stress due to acoustic vibrations. These guidelines, which are summarized in Figure 11, were published in an ISA paper in 1986[1]. Figure 11 gives a recommended Sound Pressure Level (for standard weight pipe) measured at the standard location of 1.0 meter from the pipe wall as a function of the nominal pipe diameter in inches.

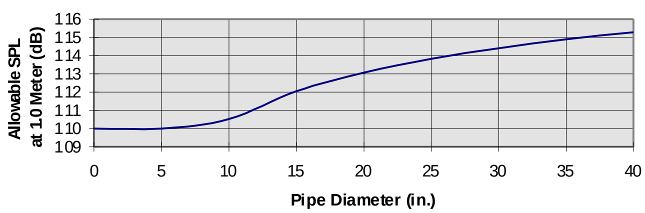

Figure 11: Recommended Maximum Valve Noise Levels for Structural Integrity of the Piping System

SINGLE FLOW PATH, MULTISTAGE PRESSURE REDUCTION TRIM

"Single flow path, multistage" means that the same flow stream passes through each of the multiple pressure reduction stages sequentially in series. This means that the outlet pressure of the first stage becomes the inlet pressure to the second stage, and so on down the line. In its simplest sense, this is equivalent to placing two or more control valves in series to reduce the pressure drop across each valve, rather than taking the entire pressure drop (p_1 - p_2) across one valve. It would help to keep that perspective when reviewing the following explanation of how multistaging helps reduce the noise.

A review of the acoustic efficiency factors (η_n) in the IEC standard shows that the acoustic efficiency increases proportionately to some higher power of the velocity (i.e., the Mach number), ranging from 3.6 to approximately 6.6 depending upon the flow regime. The point to be made here is that noise generation is very sensitive to the flow velocity. For purposes of simplification and to illustrate the basic principle, let's assume that the acoustic efficiency is equal to the 4th power of the velocity. Actually, this is not too far from the case in flow Regime I where the exponent is actually 3.6. Based upon this simplifying assumption, we can continue.

We already know that the flow (and therefore the velocity) is proportional to the square root of the pressure drop across the flow restriction. Thus, if we reduce the pressure drop by a factor of two, we can reduce the noise power level by a factor of four (-6 dB). This means that if we take the total pressure drop equally across two valves instead of one, the noise generated by each valve will be 6 dB less than if all the pressure drop was taken across one valve. We know, however, that when we combine two equal noise sources, the total noise power level is only 3 dB greater than either of the noise sources alone. This means that we have a net gain in noise reduction of 3 dB.

Dividing the pressure drop equally across more than two stages will, of course, reduce the noise even more. For example, if we divide the flow equally across 4 valves in series, we will reduce the noise power level of each valve by a factor of sixteen (-12 dB). When we combine valves 1 and 2 we will get a 3 dB increase for a total of -9 dB. Likewise, we will get the same - 9 dB when we combine valves 3 and 4 together. Now if we combine these two groups of valves together, we will get another 3 dB increase for a total noise level

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[1] Dr. Allen C. Fagerlund, "Recommended Maximum Valve Noise Levels," Advances in Instrumentation, Vol. 41-Part 3, Proceedings of the ISA/86 Internal Conference and Exhibit, Houston, Texas, October 13-16, 1986.

reduction of - 6 dB for all four valves.

By following this same logic, we can show that each time we double the number of equal stages, we will get an additional 3 dB reduction in noise. We are rapidly approaching the point of diminishing returns if we have to double the number of restrictions in series for each 3 dB improvement. When we place all of the multiple restriction stages inside a single valve, however, we get a dramatic improvement in the picture.

Each restriction still produces its share of noise, just as it would if it were a separate valve; however, the noise generated by all the interior stages will have a more difficult time radiating to the environment since the noise must pass through the rigid metal structure of the valve body in order to reach the observer. Only the jet from the orifice of the final stage will be able to effectively radiate sound waves directly into the flow stream at the outlet of the valve, where it can then be radiated through the pipe wall to the observer. Thus, if we have a valve with an 8 stage trim, the noise power level produced by the final stage would only be 1/64 ($1/8^2$) of the noise from a single valve. This would provide a noise reduction of 18 dB ($10\log_{10}1/64$) instead of only the 9 dB we would get with 8 separate valves.

In reality, however, the picture may not always be this rosy, since a small amount of the noise generated by some of the interior stages may get introduced into the flow stream and carried downstream into the pipe at the valve outlet, but this is likely to be relatively minor and the general principle holds. That is why the noise calculations in the standard only consider the pressure drop across the final stage of the multistage trim.

One final note should be made. The multistage concept discussed so far is not the same thing as what has been called the "tortuous path" method of noise reduction. The true multistage design has a series of restrictions, each followed by its own pressure recovery chamber; whereas, the "tortuous path" design is simply one long restriction which depends upon friction and flow-direction changes to dissipate the flow energy. It could be thought of as the equivalent of placing a series of elbows in a pipeline to absorb the pressure drop rather than placing a simple restriction in the line. The "tortuous path" method is an attempt to absorb the pressure drop over the flow path without dramatically increasing the velocity and therefore the noise producing turbulence. While this method can provide some noise reduction, it is not as effective as the true multistage method.

<u>MULTI-PATH, MULTISTAGE TRIM</u>

This configuration is simply a combination of the two previous methods and the noise reduction is likewise a combination of the two phenomena.

PIPE REDUCERS AND EXPANDERS (SWAGES)

Pipe reducers and expanders, which are often called "swages" are often lumped together under the general term of "fittings." It should be understood that the phenomena of valve noise generation discussed so far assumes that there are no swages attached to the valve.

Due to the fluid velocity changes which take place in the swage fittings, these swages are sources of noise generation in themselves. Thus, when a valve is fitted with swages at the inlet and outlet, there are really three noise sources; i.e., the inlet reducer, the valve, and the outlet expander. Each of these sources must be treated as a separate noise source and the results combined using Table 1 to determine the total noise coming from the combination of valve and fittings. The IEC noise standard offers a fittings compensation factor to account for the attached fittings; however, this technique places severe and unreasonable velocity limitations on the outlet of the assembly. It is better to treat these fittings as separate noise generating entities.

FLUID VELOCITY CONSIDERATIONS

Fluid velocity at the outlet of the valve has a significant effect upon the noise produced by the downstream turbulence. At the time this manuscript was written, the IEC standard had no provision for predicting noise levels for valve installations where the outlet velocity exceeds Mach 0.3. In order to maintain realistic installation costs, however, it is not always possible to limit valve outlet velocities to this range. For example, a control valve may be selected in a size smaller than the adjacent piping for economic reasons; however, the piping size is still subject to the normal selection process involving gas density and mass flow. With pressure reducing valves, for example, this invariably leads to a downstream pipe that is larger than the valve size, thus dictating the use of a pipe expander with resulting higher velocities in the valve outlet and expander.

Intense turbulence, caused primarily by the difference between the gas velocity in the valve outlet and expander and the larger downstream pipe, creates its own noise source which can often exceed the noise level of the valve itself.

There are techniques available to accurately predict control valve noise at high exit velocities, and work is being done to incorporate these techniques into a later revision of the standard.

NOISE TRANSMISSION LOSSES

Even though the control valve may be producing noise power in the fluid stream at the valve outlet, it is of little concern to us until it actually reaches the ears of a human observer in the vicinity of the valve. The first hurdle that the noise must overcome is to somehow get from pressure disturbances in the fluid stream to pressure disturbances in the air surrounding the pipeline. To do this, of course, the noise energy in the fluid stream must cause the pipe wall to vibrate in some manner so as to disturb the surrounding air and produce sound waves that will impact on the observer.

How effectively the noise power in the fluid stream can establish vibration of the pipe wall depends upon many factors. It will be most effective when the fluid noise power at any frequency coincides with a resonant frequency of the pipe. This implies that the noise power spectrum, and in particular the peak frequency, is an important consideration. Likewise, the geometry and material properties of the pipe will also have an effect on its resonant frequencies and whether they will match up with the fluid noise frequencies.

There are basically three frequencies that we need to discuss in order to better understand the acoustic coupling between the fluid in the pipe and the pipe wall. These three frequencies are the Acoustic Cutoff Frequency (f_C), the Ring Frequency (f_r), and the First Coincidence Pipe Frequency (f_O). These three frequencies always obey the following relationship:

$$f_C \leq f_O \leq f_r \qquad (11)$$

<u>CUTOFF FREQUENCY</u>

The cutoff frequency (f_C) is basically a property of the flowing fluid. It is the lower limit of energy transmission to the pipe wall since below this frequency there is very little radial displacement of the pipe. When the wavelength of the acoustic wave in the fluid is roughly equal to or longer than twice the diameter of the pipe, the wave moves down the pipeline essentially as a normal, plane wave at the speed of sound (c_2) in the fluid, as illustrated in Figure 12.

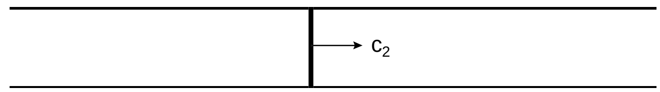

Figure 12: Normal Plane Wave

Since this normal sound wave is perpendicular to the pipe wall, it has little ability to transmit any energy to the pipe wall. Therefore, below the cutoff frequency (f_C), where the wavelength is somewhat greater than the pipe diameter, there will be essentially no acoustic coupling with the pipe and no sound will be transmitted to the outside observer. We would say that the transmission “loss” is very high.

<u>FIRST COINCIDENCE PIPE FREQUENCY</u>

The first coincidence pipe frequency (f_O) is actually a property of both the flowing fluid and the pipe. As the noise frequencies become greater than the cutoff frequency, the wavelengths become shorter and the acoustic wave patterns in the fluid become much more complex. The shorter wavelengths attempt to radiate in all directions and so begin to propagate outward toward the pipe wall at some angle from the centerline. These waves then strike the pipe wall at an angle, reflecting off the wall and passing across the pipe to the opposite wall. As these acoustic waves propagate by reflection down the pipeline, they tend to proceed in a spiral wave pattern.

As the acoustic wave in the fluid goes spiraling down the pipeline, the radial component of this wave tends to induce a corresponding wave in the pipeline which also spirals down the pipe at the same speed as the acoustic wave spiral. Aside from the spiraling nature, this wave in the pipe is really a bending or flexure wave similar to what would occur in a flat sheet of metal if we grabbed it by the end and shook it rapidly up and down. Imagine for a moment, that we have removed a narrow section of the pipe wall for some distance along the pipe as shown in Figure 13.

As we imagine this strip to be narrower and narrower, the curvature becomes less predominant and we can further stretch our imagination to think of this as simply a thin, narrow, relatively long strip of “flat” metal.

If we were to grab the left end of this long strip and shake it rapidly up and down one time, we would expect to see something similar to a sine wave that would travel at some speed down the length of the strip, just exactly analogous to the kind of waves we used to make with jump ropes when we were kids.

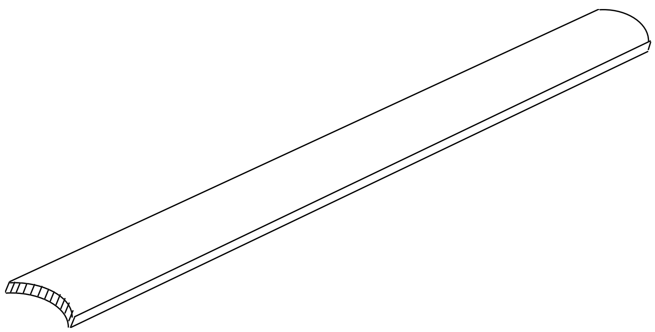

Figure 13: Strip section of the pipe wall along the length of the pipe

What we have excited in the strip is the fundamental (lowest) frequency mode which results basically in a second-order, spring-mass oscillation of the strip. If we were to shake the strip more vigorously in a random manner, we could probably excite some of the higher frequency modes which would result in even more complex wave shapes in the strip. For right now, however, we are interested only in this lowest frequency, fundamental mode of the pipe strip. We recognize, of course, that this fundamental mode frequency is a function of the pipe material and geometry.

Let's stretch our imaginations slightly further to think of this strip not as a long straight strip down the pipe, but as a spiral strip down the pipe which spirals at the same rate down the pipe as the acoustic wave inside the pipe. Thus, the radial component of the acoustic wave continuously excites this strip causing it to want to vibrate. The frequency, however, with which the acoustic wave excites the strip will determine just how effective the vibration response will be. We know from experience that, because of the wide band of acoustic frequencies, there will most likely always be a frequency component of the acoustic wave that precisely matches the lowest frequency mode of the strip. As the spiraling wave continuously excites the lowest frequency mode of this strip, the amplitude of pipe wall motion will be the greatest. This means that the noise transmission through the pipe will be the greatest, or the transmission "loss" will be at a minimum. The frequency of the acoustic wave where this occurs is called the "first coincidence pipe frequency, (f_O)." Webster's dictionary defines "coincidence" as, "Condition, fact, or instance of coinciding; correspondence." In our case, the acoustic wave speed as it spirals down the pipe exactly coincides with the bending wave speed down the pipe.

RING FREQUENCY

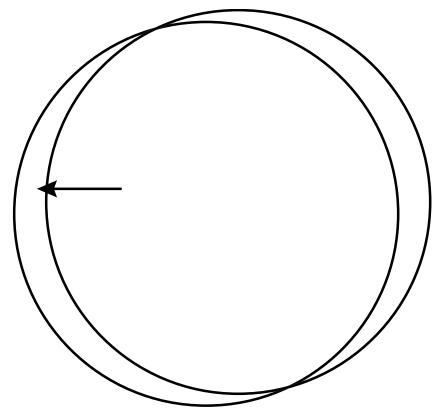

Figure 14: Ring section of pipe

Let's now imagine a very short section along the length of the pipe. What we would have would resemble a "ring" as shown in Figure 14. If we were to apply a static transverse force to the pipe as shown in Figure 14, the deformation will be predicted by treating the pipe as a hollow beam with a particular bending stiffness determined by the radius and wall thickness. When the stresses in the pipe are analyzed, we will find that there will be nearly uniform compressive stresses through the wall thickness on the side of the pipe where the force is applied, and nearly uniform tensile stresses through the wall thickness on the opposite side of the pipe. The tensile stress on the opposite side of the pipe occurs due to the compression wave which transmits the effect of the force around the circumference of the pipe ring.

If the force is now changed to an oscillating force, such as we would see from the radial component of the acoustic wave, we would find that the stresses will vary with time since it takes the compression wave around the circumference of the ring a finite amount of time to travel to the other side of the ring. If the frequency of excitation is very low, then the picture at any instant will closely resemble the static case, where the stresses are opposite in sense on opposite sides of the pipe.

As the frequency of excitation increases, the speed of

the compression wave around the circumference of the
pipe ring will begin to come into play. For a particular frequency of excitation, the wave will arrive at the opposite side of the pipe out of phase with the driving force. At this frequency the traveling wave will arrive in phase with the driving force back at the point of excitation, due to a delay of exactly one period of the wave cycle. This “in phase” reinforcement feedback causes an amplification effect which greatly magnifies the movement of the pipe wall, thus amplifying the amount of sound transmitted; i.e., a minimum transmission “loss.”

The frequency where this resonant amplification occurs is call the “ring frequency (f_r)” and is defined by the condition where the wavelength (λ) of the compression wave is exactly equal to the circumference (πD_i) of the pipe ring.

TRANSMISSION LOSS SPECTRUM

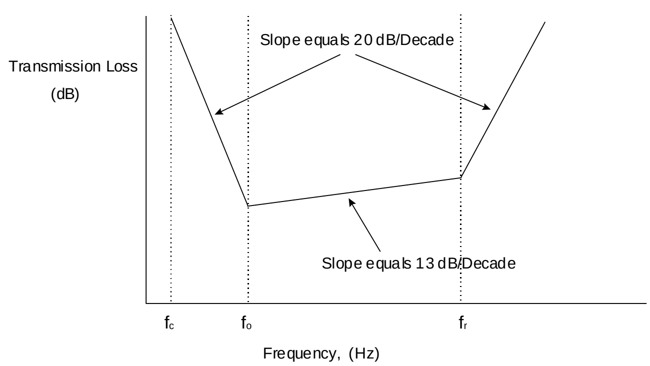

Figure 15: Pipe Transmission Loss Spectrum

Since our perspective is noise reduction, we usually talk in terms of transmission losses instead of transmission efficiency. As we have just implied, this transmission loss for the pipe will vary with frequency as shown in Figure 15. Below the cutoff frequency (f_C), the transmission loss is very large (essentially infinite) since there is no radial component of the normal shock wave to excite the pipe wall.

As the frequency approaches the first coincidence pipe frequency (f_O), the acoustic waves begin to develop some radial component to stimulate the pipe wall and noise transmission becomes more and more efficient; i.e., the transmission loss decreases at a rate of 20 dB per decade of frequency to a minimum value precisely at the first coincidence pipe frequency.

As the frequency increases from the first coincidence pipe frequency (f_O) toward the ring frequency (f_r), the transmission loss increases slightly due to the complex propagation path of the spiral acoustic waves. The rate of increase in this region is 13 dB per decade of frequency.

At frequencies above the ring frequency (f_r), the wavelengths become significantly shorter compared to the pipe dimensions and the structure begins to respond more and more like a flat plate. In this region, the transmission losses begin to increase more rapidly (20 dB per decade) with frequency.

SUMMARY

Obviously, this has been a rather simplified approach to an extremely complex topic; however, an elementary understanding and appreciation of the theory behind aerodynamic noise generation and transmission allows one to make a fully informed decision regarding the selection of a control valve for any given noise application. There are a number of common errors that can be avoided through application of this basic knowledge.

For example, when comparing noise quotations for a valve assembly, we recognize that it is important to make certain that we are comparing apples to apples. One quotation may be limited to just the valve, while the other may include the attached swages. Knowing that the swages may be the dominant noise source, and that the lower noise generated by the valve will have minimal effect on the overall noise level, we can avoid paying too much for an expensive noise control trim that will have neglible effect on the overall noise level.

Likewise, we might want to seriously consider the wisdom of buying an expensive noise control trim for a valve that is going to be installed next to a much larger noise source, such as a compressor, etc. Remember, even if the noise of the valve is equal to the other noise source, the valve will only increase the overall noise level by 3 dB. A valve vendor might take advantage of this fact by promising a lower valve noise level than can actually be achieved, knowing that the likelihood of verification is slim. The best defense against this tactic is to verify the stated valve noise prediction via the IEC noise prediction standard.

The general rule of thumb for controlling the noise level

in a given space is to identify the dominant noise sources and eliminate or reduce them. Only then is it logical to put precious resources into reducing valve noise.
Finally, an understanding of pipe transmission loss theory helps us realize that controlling the frequency of the noise being generated is perhaps even more important than the noise power. There are three important benefits to the frequency shifting method of noise control.

First, much of the high frequency noise will fall outside the audible range where it has neglible effect on humans. Secondly, the high frequency noise is concentrated mainly in the region of the frequency spectrum where the transmission loss is very high (see the right side of Figure 15). In this region, very little of the noise that is generated will get into the outside environment where humans will be located. Finally, the reduced coupling between the internal sound field and the pipewall at these high frequencies means that there are reduced levels of stress in the piping structure which will help to prevent fatigue damage.

As in many other areas, knowledge is the best defense against waste and inefficiency.

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