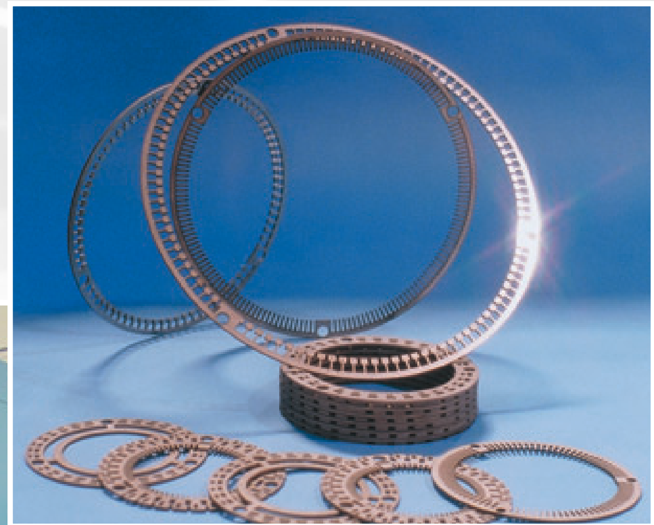
Bulletin OZ3000 10/05

Noise Control Manual



Masoneilan

Experience, Knowledge & Technology...In Control

DRESSER

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Foreword

This noise manual contains informative material regarding noise in general and control valve noise in particular.

Noise prediction methods used by Masoneilan for aerodynamic noise and hydrodynamic noise are based on the latest publications of the Instrument Society of America (ISA) and the International Electrotechnical Commission (IEC), see reference Section 6.0. The calculations required by these methods are quite complex, and the solution of the equations is best accomplished by computer. For this purpose, the Masoneilan valve sizing and selection computer program ValSpeQ® provides a convenient and efficient working tool to perform these calculations.

1. Control Valve Noise

1.1 Introduction

Noise pollution will soon become the third greatest menace to the human environment after air and water pollution. Since noise is a by-product of energy conversion, there will be increasing noise as the demand for energy for transportation, power, food, and chemicals increases.

In the field of control equipment, noise produced by valves has become a focal point of attention triggered in part by enforcement of the Occupational Safety and Health Act, which in most cases limits the duration of exposure to noise in industrial locations to the levels shown in Table 1.

Duration of Exposure (Hours)	Sound Level (dBA)
8	90
4	95
2	100
1	105
1/2	110
1/4 or less	115

Table 1

1.2 Acoustic Terminology

Noise
Noise is unwanted sound.

Sound
Sound is a form of vibration which propagates through elastic media such as air by alternately compressing and rarefying the media. Sound can be characterized by its frequency, spectral distribution, amplitude, and duration.

Sound Frequency
Sound frequency is the number of times that a particular sound is reproduced in one second, i.e., the number of times that the sound pressure varies through a complete cycle in one second. The human response analogous to frequency is pitch.

Spectral Distribution
The spectral distribution refers to the arrangement of energy in the frequency domain. Subjectively, the spectral distribution determines the quality of the sound.

Sound Amplitude
Sound amplitude is the displacement of a sound wave relative to its “at rest” position. This factor increases with loudness.

Sound Power
The sound power of a source is the total acoustic energy radiated by the source per unit of time.

Sound Power Level
The sound power level of a sound source, in decibels, is 10 times the logarithm to the base 10 of the ratio of the sound power radiated by the source to a reference power. The reference power is usually taken as 10^{-12} watt.

Sound Pressure Level: SPL
The sound pressure level, in decibels, of a sound is 20 times the logarithm to the base of 10 of the ratio of the pressure of the sound to the reference pressure. The reference pressure is usually taken as 2×10^{-5} N/M^2.

Decibel: dB
The decibel is a unit which denotes the ratio between two numerical quantities on a logarithmic scale. In acoustic terms, the decibel is generally used to express either a sound power level or a sound pressure level relative to a chosen reference level.

Sound Level
A sound level, in decibels A-scale (dBA) is a sound pressure level which has been adjusted according to the frequency response of the A-weighting filter network. When referring to valve noise, the sound level can imply standard conditions such as a position 1 m downstream of the valve and 1 m from the pipe surface.

1. Control Valve Noise (cont.)

1.3 Human Response to Noise

Frequency

Given a sound pressure, the response of the human ear will depend on the frequency of the sound. Numerous tests indicate that the human ear is most sensitive to sound in the frequency region between 500 and 6000 Hz and particularly between 3000 and 4000 Hz.

Sound Weighting Networks

A weighting network biases the measured sound to conform to a desired frequency response. The most widely used network for environmental noise studies, the A-weighting network, is designed to bias the frequency spectrum to correspond with the frequency response of the human ear, see Figure 1.

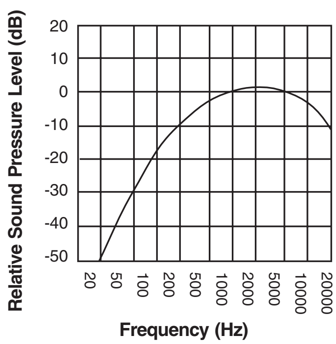


Figure 1

IEC Standard A-Weighting Curve for Sound Level Meters

Relative Energy	Decibels	Example
1×10^{14}	140	Proximity to jet aircraft
1×10^{13}	130	Threshold of pain
1×10^{12}	120	Large chipping hammer
1×10^{11}	110	Near elevated train
1×10^{10}	100	Outside auto on highway
1×10^{9}	90	Voice - shouting
1×10^{8}	80	Inside auto at high speed
1×10^{7}	70	Voice - conversational
1×10^{6}	60	Voice - face-to-face
1×10^{5}	50	Inside general office
1×10^{4}	40	Inside private office
1×10^{3}	30	Inside bedroom
1×10^{2}	20	Inside empty theater
1×10^{1}	10	Anechoic chamber
1	0	Threshold of hearing

Table 2

Comparison of Energy, Sound Pressure Level, and Common Sounds

Increase in Sound Level	Human Subjective Response
3 dBA	Just perceptible
5 dBA	Clearly noticeable
10 dBA	Twice as loud
20 dBA	Much louder

Table 3

Changes in Sound Level

1.4 Major Sources of Noise

Mechanical Vibration

Mechanical noise is caused by the response of internal components within a valve to turbulent flow through the valve. Vortex shedding and turbulent flow impinging on components of the valve can induce vibration against neighboring surfaces. Noise generated by this type of vibration has a tonal characteristic.

If this turbulence induced vibration of trim parts approaches a natural frequency of the plug-stem combination, a case of resonance will exist. A resonant condition is very harmful, since it can result in fatigue failure of trim parts. Noise from mechanical vibration does not occur often in control valves, especially since the introduction of top and cage guided valves. Should it occur, steps must be taken to eliminate that resonant condition, to reduce the noise but more importantly to preclude fatigue failure.

Possible cures for this type of noise include change in trim design or capacity, reduction of guide clearances, larger stem sizes, change in plug mass, and sometimes reversal of flow direction. These steps are intended to shift the natural frequency of parts and the excitation frequency away from each other. There is presently no reliable method for predicting noise generated by mechanical vibration in control valves.

However, fundamentally, the quantity of *acoustic energy* generated by the flow provides realistic estimates of the energy available to excite *component vibration,* because both aerodynamic noise and component vibration energy are generated by the same physical mechanism: turbulent pressure fluctuations in the fluid field. Thus, the IEC aerodynamic noise prediction method enables one to evaluate potential for trim and valve vibration. Our experiences confirm this result: We have found that operating a control valve below 95 dBA will prevent onset of vibration problems. Since most valve applications require sound pressure levels below 85 dBA, valves satisfying this aerodynamic noise requirement will avoid vibration problems.

Aerodynamic Noise

Aerodynamic noise is a direct result of the conversion of the mechanical energy of the flow into acoustic energy as the fluid passes through the valve restriction. The proportionality of conversion is called acoustical efficiency and is related to valve pressure ratio and design. See Sections 2, 3 and 4.

Hydrodynamic Noise

Liquid flow noise, cavitation noise, and flashing noise can be generated by the flow of a liquid through a valve and piping system. Of the three noise sources, cavitation is the most serious because noise produced in this manner can be a sign that damage is occurring at some point in the valve or piping. See Section 5.

2. Aerodynamic Noise Prediction

2.1 An Introduction to the Prediction Method

Aerodynamic noise prediction described in this section is based on the equations and nomenclature of the international standard for control valve noise prediction, IEC-534-8-3. Because of the extent and complexity of these calculations, only a general description of the calculation methods are included here.

The IEC control valve aerodynamic noise method consists of four basic processes. (1) The method determines the process conditions to calculate the trim outlet velocity and solves for the valve noise source strength at the valve. (2) This method estimates the portion of the sound generated at the valve that propagates into the downstream piping. (3) The third step of the method models how the pipe walls attenuate the noise as it passes from the inside to the outside of the pipe. (4) This method describes the radiation of the sound from the pipe wall to estimate the A-weighted sound-pressure level (SPL) at a distance of one meter from the piping wall. In addition, the method takes into account noise generated by flow expansion upon exiting the valve body and adds this expander noise to the valve noise, yielding the aerodynamic noise produced by the *valve system* one meter downstream of the valve exit and one meter from the piping wall.

2.2 Further Explanation of the Prediction Method

The problem of predicting control valve noise is two-fold. First, the sound power generated in the fluid inside the valve and piping due to the throttling process must be estimated. Secondly, the transmission loss due to the piping must be subtracted to determine the sound level at a predetermined location outside the piping.

Noise prediction for a freely expanding jet is based on multiplying the mechanical energy conversion in the jet by an efficiency factor. This theory is modified to take into account the confined jet expansion in a control valve, and the inherent pressure recovery.

In order to accommodate the complex nature of valve noise generation, the prediction method addresses the calculation of significant variables in five different flow regimes. Among the significant variables are an acoustic efficiency, sound power, and peak frequency. From these and other variables, the internal sound power is calculated.

The transmission loss model is a practical simplification of complex structural transmission loss behavior. The simplification is rationalized on the basis of allowable tolerances in wall thickness.

The downstream piping is considered to be the principal radiator of the generated noise. The transmission loss model defines three sound damping regions for a given pipe having their lowest transmission loss at the first coincidence frequency. The transmission loss is calculated at the first coincidence frequency and then modified in accordance with the relationship of the calculated peak frequency to the coincidence frequency.

A correction is then made for velocity in the downstream piping.

The predicted sound level is then based on the calculated internal sound pressure level, the transmission loss, velocity correction, and a factor to convert to dBA.

2.2.1 The flow regime for a particular valve is determined from inlet pressure, downstream pressure, fluid physical data, and valve pressure recovery factor.

Five flow regimes are defined as:

Regime I - Subsonic
Regime II - Sonic with turbulent flow mixing (recompression)
Regime III - No recompression but with flow shear mechanism
Regime IV - Shock cell turbulent flow interaction
Regime V - Constant acoustical efficiency (maximum noise)

The following explanation is based on Regime I equations, but will serve to illustrate the methodology employed.

2.2.2 The stream power of the mass flow is determined (for Regime I) as:

Noise Source Magnitude

- Magnitude: Proportional to Stream Power, W_m, at Vena Contracta

$$W_{mI} = \frac{\dot{m} \cdot U_{vc}^{2}}{2}$$

2.2.3 For the confined jet model, the acoustical efficiency is calculated as:

- Mixed Dipole – Quadruple Source Model

$$\eta_I = 0.0001 \cdot M_{vc}^{3.6}$$

2.2.4 In Regime I, the peak frequency of the generated noise is determined as:

Noise Frequency

- Peak frequency of noise generation, f_p
- Varies with flow regime
- Always scales with jet diameter and velocity at the throttling vena contracta

$$f_{PI} = \frac{0.2\, U_{vc}}{d_{vc}}$$

Jet vena contracta diameter is a function of jet pressure recovery and valve style modifier, F_d (throttling flow geometry).

2. Aerodynamic Noise Prediction (cont.)

2.2.5 Only a portion of the sound power propagates downstream. That portion is designated as a factor r_w. This factor varies with valve style.

Valve Noise Propagation
- A portion of valve noise propagates downstream
- This ratio, r_w, varies with valve style
- Reflects "line-of-sight" through valve

Valve Style	r_w
Globe (21000, 41000)	0.25
Rotary Globe (Varimax)	0.25
Eccentric Rotary Plug (Camflex)	0.25
Ball	0.5
Butterfly	0.5
Expander	1

2.2.6 The sound pressure level in the downstream piping is determined as:

Downstream Piping Internal Noise
- Average valve sound pressure level over cross-section of downstream piping

$$L_{pI} = 10\text{Log}_{10} \frac{3.2 \cdot 10^{9} W_a \rho_2 c_2}{d_p^{\,2}}$$

2.2.7 An increase in noise occurs with increased Mach number on the downstream piping.

Downstream Noise Propagation
- Higher Mach number in the downstream piping, M_2, increases noise by L_g
- Alters wave propagation (Quasi-Doppler)

$$L_g = 16\text{Log}_{10} \left(\frac{1}{1\text{-}M_2} \right)$$

Note: Moderate M_2 Controls Noise

2.2.8 The sound transmission loss due to the downstream piping is determined as:

Basic Sound Transmission Through Piping Wall

$$TL = 10\text{Log}_{10}\ 7.6 \cdot 10^{-7} \cdot \left\{ \left(\frac{c_2}{t_p f_p} \right)^2 \frac{G_x}{\left(\frac{\rho_2 c_2}{415 G_y} + 1 \right)} \left(\frac{P_a}{P_s} \right) \right\}$$

Note: Increasing Wall Thickness Increases Loss

2.2.9 The transmission loss is dependent upon frequency.

Frequency-Dependent Sound Transmission Through Piping (I)
- Pipe Ring Frequency, f_r

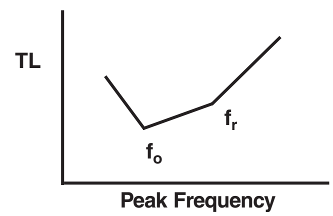
$$f_r = \frac{\text{Pipe Material Sound Speed}}{\text{Pipe Circumference}} = \frac{c_p}{\pi d_p}$$

- Pipe Coincidence Frequency (Minimum Transmission Losses), f_o

$$f_o = \frac{\text{Pipe Circumferential Bending Wave Speed}}{\text{Pipe Acoustic Wave Speed}} \approx \left(\frac{f_r}{4} \cdot \frac{c_2}{c_{air}} \right)$$

2.2.10 The transmission loss regimes can be illustrated graphically:

Transmission Loss Regimes
- $f_p < f_o$: Larger TL (non-resonant wall fluid coupling)
- Smallest TL, $f_p = f_o$ (~ circumferential bending & acoustic modes coincident)
- $f_p > f_r$: TL increases markedly (~ flat-plate radiation)



The slope to the transmission loss in the three regimes can be determined by the following relationships:

$$f_p < f_o \quad \Delta TL = 20\text{Log}_{10} \left(\frac{f_o}{f_p} \right) + 13\text{Log}_{10} \left(\frac{f_o}{f_r} \right) \quad \text{(Expanders, Std Valves)}$$

$$f_o < f_p < f_r \quad \Delta TL = 13\text{Log}_{10} \left(\frac{f_o}{f_p} \right) \quad \text{(Standard Valves)}$$

$$f_r < f_p \quad \Delta TL = 20\text{Log}_{10} \left(\frac{f_p}{f_r} \right) \quad \text{(Low Noise Trims)}$$

Based on the above, Frequency Factors Gx and Gy are applied per the IEC standard.

Note: Higher f_p (smaller d_{vc}) can increase piping damping and reduce control valve throttling noise.

2. Aerodynamic Noise Prediction (cont.)

2.2.11 The net sound level at the pipe wall converted to dBA is:

- Sum Noise, Add +5dB for A-Weighted SPL

$$L_{pAe} = 5 + L_{pi} + TL + L_g$$

2.2.12 At one meter from the pipe wall, the valve noise is:

- Cylindrical spreading model yields noise at 1m from pipe wall

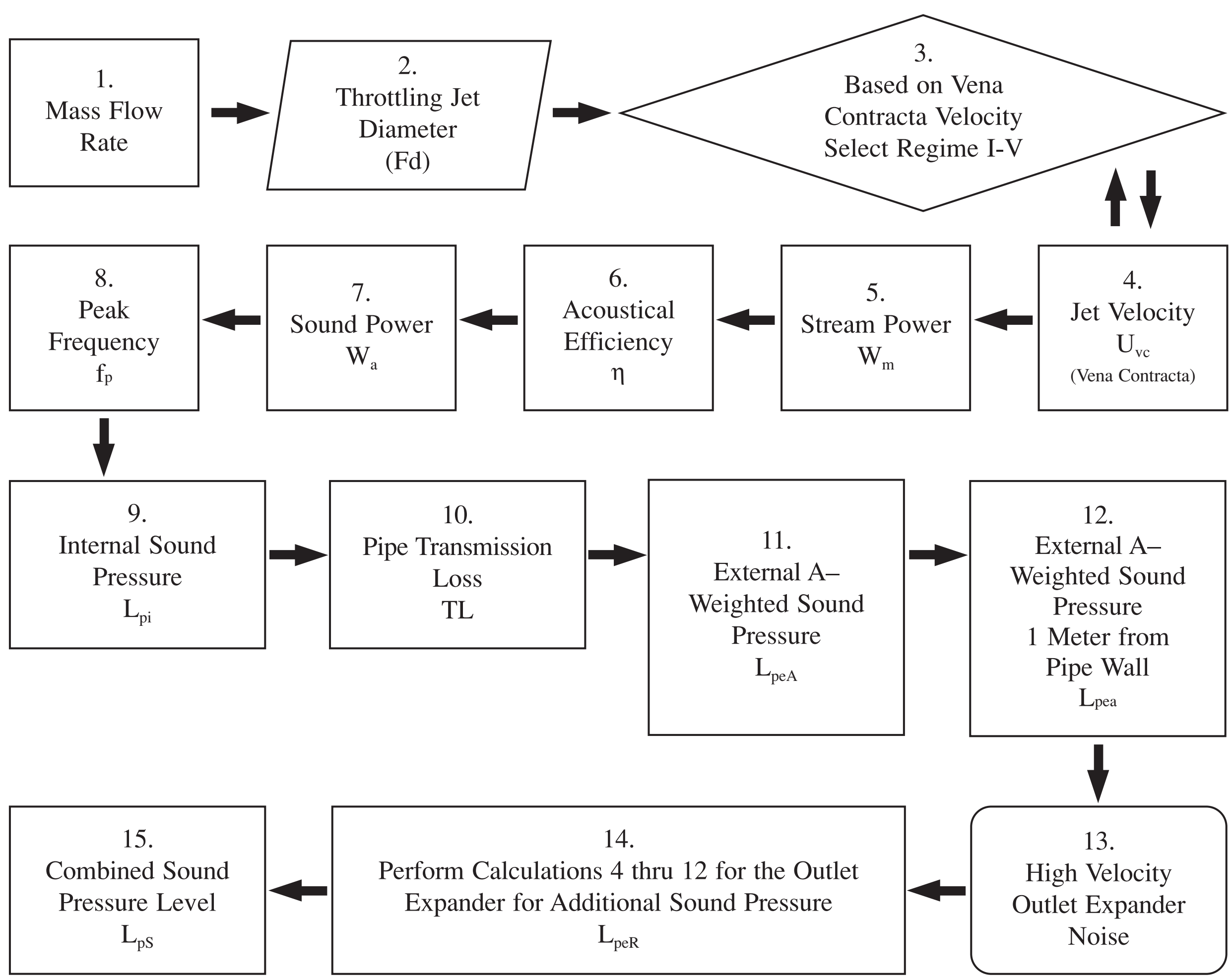
$$L_{pAe,1m} = L_{pAe} - 10\mathrm{Log}_{10}\left[\frac{d_p + 2t_p + 2}{d_p + 2t_p}\right]$$

2.2.13 In the case of an expander downstream of a valve, the noise generated in the expander is calculated in a manner similar to the Regime I method, and added logarithmically to the valve noise to determine an overall sound level (L_e).

High expander noise occurs when high Mach number exit flow jets into the larger downstream piping (> 0.3 Mach). This is very important as this noise source can readily overwhelm trim noise and result in damaging low frequency noise which can excite piping structures. This outlet expander noise is accounted for in steps 13 and 14 below.

2.2.14 A flow chart illustrating the aerodynamic noise prediction method is shown below.

Control Valve Aerodynamic Noise Prediction Flow Chart



2. Aerodynamic Noise Prediction (cont.)

2.3 Additional Comments

Examples of the use of the prediction methods are shown in detail in the respective standards. These examples may be used to verify a computer program.

The IEC standard also provides for prediction for proprietary low noise trim designs and other valve configurations not specifically covered by the standard. The manufacturer is required to incorporate additional changes in sound pressure level as a function of travel and/or pressure ratio, in addition to the sound pressure level obtained by using the appropriate clauses applying to valves with standard trim. Masoneilan has accomplished this requirement in our valve sizing and selection computer program ValSpeQ®.

Nomenclature

A = flow area

C_v = valve capacity

c = sound speed of gas

c_p = sound speed of piping

d_p = outlet pipe inner diameter

d_v = outlet valve inner diameter

d_{vc} = trim jet vena contracta diameter

F_d = valve style modifier

F_L = pressure recovery coefficient

f = sound frequency

f_o = acoustic-structural Coincident Frequency

f_p = flow peak frequency

f_r = pipe ring frequency

L_e = expander and pipe flow noise sound-pressure level, A-weighted and 1 meter from the pipe wall

L_g = pipe Mach number correction factor

L_{pAe} = A-weighted sound-pressure level

$\dot{m}$ = mass flow rate

$L_{pAe.1m}$ = A-weighted sound-pressure level, 1 meter from pipe wall

L_{pi} = pipe internal sound-pressure level

M = Mach number

Mw = molecular weight of gas

p = pressure

p_a = actual pressure outside pipe

p_s = standard pressure outside pipe (1 atmosphere)

R = universal gas constant

T = gas temperature

TL = pipe transmission loss

TL_{fr} = pipe transmission loss at ring frequency

t_p = pipe wall thickness

U = velocity

W_a = sound power

W_m = stream power of mass flow

Greek

γ = ratio of specific heats

η = acoustical efficiency factor

ρ = gas density

Subscripts

1 = upstream of valve or vena contracta

2 = downstream of valve or vena contracta

e = expander

v = valve outlet

vc = vena contracta

I = Regime I

3. Aerodynamic Control Valve Noise Reduction

3.1 Methods

Reduction of control valve aerodynamic noise can be achieved by either source treatments (preventing the noise generation) or path treatments (pipe insulation, silencers, or increasing pipe schedule). Source treatment often becomes the preferable method. Sound, once generated, propagates virtually unattenuated in downstream pipe. In addition, as discussed in the Appendix, very high sound levels inside piping systems can damage the pipe and mechanical components located downstream by inducing excessive vibration.

3.1.1 Source Treatment

The generation of noise can be controlled by using trim components specially designed for low noise production. There are basically two methods employed in reducing noise generated in the valve trim:

1. **Use of Small, Properly Spaced Fluid Jets**
   The size of the fluid jets affects noise generation in three ways. First, by reducing the size of the fluid jets (and consequently the size of the eddies), the efficiency of conversion between mechanical and acoustical power is reduced. Second, 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. Third, the higher frequency sound, if raised above 10000 Hz, is de-emphasized by both the A-weighting filter network and the human ear.

   The spacing of the fluid jets affects the location of the point downstream at which the fluid jets mutually interfere. The mutual interaction of the fluid jets at the proper location downstream thereby reduces the shock-eddy interaction that is largely responsible for valve noise under critical flow conditions. This factor further reduces acoustical efficiency.

2. **Adiabatic Flow with Friction**
   The principle of "Adiabatic Flow with Friction" is to reduce pressure much like the pressure loss which occurs in a long pipeline. This effect is produced by letting the fluid pass through a number of restrictions, providing a tortuous flow pattern dissipating energy through high headloss rather than through shock waves.

   The flow area of the valve trim is gradually increased toward the downstream section. This compensates for expansion of the gas with pressure loss and ensures a nearly constant fluid velocity throughout the complete throttling process.

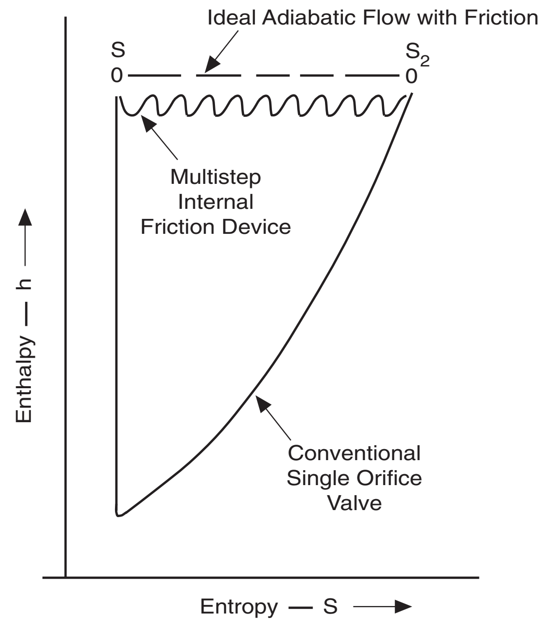


Figure 2

As shown on Figure 2, in conventional orifice type valves, internal energy is converted into velocity (kinetic energy). This results in a sharp decrease in enthalpy. Downstream turbulence accompanied by shock waves, reconverts this velocity into thermal energy with a permanent increase in entropy level (corresponding to the pressure change P_1-P_2). These same shock waves are the major source of undesirable throttling noise. In a Lo-dB valve, however, the velocity change is minimized and the enthalpy level remains nearly constant.

Most Masoneilan Lo-dB valves use both of the previously mentioned methods to limit noise generation to the minimum levels possible. When controlling noise using source treatments, such as Lo-dB valves, it is imperative that the fluid velocity at the valve outlet is limited to avoid regenerating noise at this potential source. Low noise valves are inherently quieter (less efficient noise generators), due to their special trim designs. The noise generated by the outlet, if not properly limited, can easily dominate over the noise generated by the trim, rendering the low noise trim virtually ineffective. There are two methods used to control outlet velocity. First, the downstream pressure can be increased by using Masoneilan Lo-dB cartridges and expansion plates. This method, from Bernoulli's principle, decreases the velocity at the valve outlet by increasing the pressure immediately downstream of the valve. The second method is simply to choose a valve size that is adequate to ensure the proper outlet velocity.

3. Aerodynamic Control Valve Noise Reduction (cont.)

3.1.2 Path Treatment

There are three basic methods of incorporating path treatment into control valve systems:

1. **Silencers**
   Silencers can be effective in reducing control valve noise provided they are installed directly downstream of the valve. However, there are several technical problems often encountered in their use. First, to be effective, they require low flow velocities which often make them impractical, especially for use in high capacity systems. Second, the acoustic elements are not always compatible with the flowing medium, and third, the operating conditions may be too severe.

2. **Increase in Pipe Schedule**
   An increase in the wall thickness of downstream piping can be an effective means to reduce control valve noise. However, since noise, once generated, does not dissipate rapidly with downstream pipe length, this method must normally be used throughout the downstream system.

3. **Pipe Insulation**
   This method, like that of increasing pipe thickness, can be an effective means to reduce radiated noise. However, three restraints must be noted. First, as with the pipe schedule method, insulation must be used throughout the downstream system. Second, the material must be carefully installed to prevent any "voids" in the material which could seriously reduce its effectiveness. Third, thermal insulation normally used on piping systems is limited in its effectiveness in reducing noise. Unfortunately, more suitable materials often are not acceptable at high temperature, since their binders may burn out, radically changing their acoustical and thermal qualities. In application, noise reduction of acoustical insulation reaches a practical limit of 11-12 dBA due to acoustical "leaks" from the valve bonnet and top works, see Figure 3.

3.2 Equipment

3.2.1 Historical Perspective

Masoneilan's innovative research and development has pioneered solutions to control valve application problems for years. Before OSHA was established Masoneilan developed the first high performance valves for reducing control valve noise and minimizing the effects of cavitation. Among these were the 77000 and 78000 Series valves, followed by the introduction of our first globe valves with special Lo-dB trim.

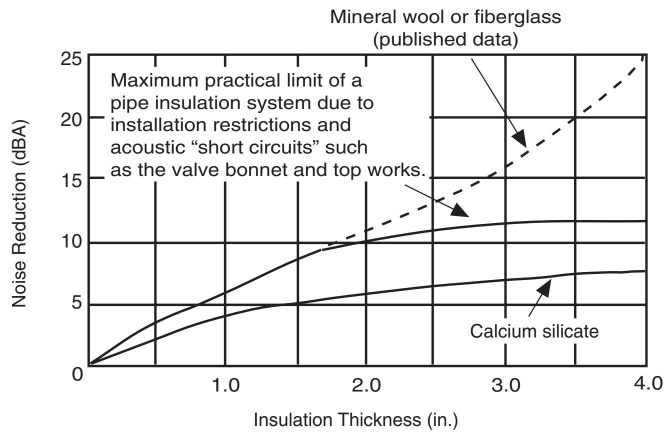


Figure 3
Additional Noise Reduction from Typical Pipe Insulation Systems

Since 1975, Masoneilan laboratory studies have led to a steady stream of innovative designs. Examples include new Lo-dB trim for popular cage guided and top guided globe valves. Masoneilan has led in the application of the static restrictor as an effective means to reduce noise control costs.

In addition to the development of new designs, Masoneilan has continued to conduct both pure and applied research at Masoneilan's corporate laboratories. The result has been numerous internationally published technical articles, and the first universal noise prediction method.

Masoneilan has contributed to the work of ISA and IEC standards organizations, whose efforts have resulted in the noise prediction methods now employed by Masoneilan.

3.2.2 Products and General Selection Criteria

Masoneilan offers a wide variety of low noise valves and valve systems. Some Lo-dB valves provide low cost solutions to relatively general purpose applications. Others, such as the V-LOG® technology can be custom-made for particular applications. This wide selection provides a cost effective solution to virtually any control valve problem. A brief description of each unit, its typical uses, and noise reduction performance is given below in order of increasing cost.

Lo-dB and V-LOG Static Restrictors - Cartridges and Plates

Lo-dB and V-LOG restrictors can be installed downstream of any valve (conventional or low noise) to create back-pressure and limit the pressure drop that is taken across any single device.

3. Aerodynamic Control Valve Noise Reduction (cont.)

Depending on the overall pressure drop ratio (P_1/P_2) of the system, either Lo-dB or multi-stage V-LOG technology can be applied to reduce the overall system noise to a much quieter level. These restrictor plates reduce the velocity in the valve outlet as well as reduce the pressure drop ratio (P_1/P_2) across the valve trim by creating back-pressure. Both of these factors are significant contributors to the overall system noise level and cost.

When used in series with a conventional valve (Camflex®, Varimax®, etc.), over 20 dBA of noise reduction can be achieved at modest cost. This level can be extended to over 30 dBA when used in series with a Lo-dB valve design. Improvements in installed life are also recognized as the reduction in noise and vibration allow for longer durations in between maintenance periods.

Camflex (35000 Series) Valve with DVD Plate
This eccentric rotary control valve with optional DVD® (Differential Velocity Device) low noise plate is a unique solution to moderate noise reduction applications. The patented DVD element is strategically located downstream of the eccentric throttling plug. It reduces noise by the principle of producing a lower velocity annulus (by pressure staging) surrounding the higher velocity core flow stream. Operated flow-to-open (FTO), this valve produces noise levels approximately 15-20 dBA lower than a standard globe valve. Because this element fits within the valve face to face dimensions it can be field retrofitted into any of our highly successful standardized Camflex valves. Accordingly this option can be a very cost effective noise solution in place of a more expensive globe valve with noise reducing trim.

21000 Series with Lo-dB Trim
This valve model fills the moderate cost, moderate noise reduction category of the product line. Operated flow-to-open (FTO), it produces noise levels approximately 16-19 dBA lower than conventional valves. The Lo-dB trim is based on Masoneilan's multiple-orifice cage concept. It is completely interchangeable with other 21000 Series parts. A two-stage noise reducing trim is now available for greater noise reduction.

The 21000 Lo-dB is the optimum choice for a broad range of process applications due to its simple construction, tight shutoff, and effective noise reduction.

2600 Series with Lo-dB Trim
This valve is ideally suited to chemical and other industries. Its key features include a modular approach to valve construction that results in angle, globe or other configurations, availability of numerous body materials, quick change trim and separable flanges. The Lo-dB trim based on Masoneilan's multiple-orifice cage concept, generates up to 12 dBA less noise than conventional valves.

41000 Series with Lo-dB Trim
The 41000 Series control valve can be equipped with five different efficient noise reduction packages which comply with process conditions. These trim packages are directly interchangeable with conventional construction. These packages include:

1. Standard Capacity Lo-dB
2. High Capacity Lo-dB
3. Reduced Capacity Lo-dB
4. Multi-Stage Lo-dB
5. Multi-Stage V-LOG

The cage, which is the Lo-dB element, has been designed using the latest in hole sizing and spacing technology from both Masoneilan research and NASA funded programs. Proper hole sizing and spacing prevents jet reconvergence and shock-induced effects, which reduce acoustic energy formation.

30000 Series Varimax® with Lo-dB Trim
The Varimax® Lo-dB valve provides control of high pressure compressible fluids without the erosion, vibration and high noise levels associated with conventionally designed rotary valves. Because the Varimax® has relatively large flow passages it is particularly well suited for applications involving gases. For high pressure ratios, Lo-dB cartridges in the globe adaptor are recommended.

The high rangeability 100:1 of this Varimax® Lo-dB valve allows wide variations in controlled flow. Operation is stable because the plug is equilibrated. This uniquely balanced plug has no secondary balancing seal and mounts with a standard seat.

72001 and 72002 Series Lo-dB
Using the proven 41000 Lo-dB trim design in an angle body configuration, the 72001 (single stage Lo-dB) and 72002 (double stage Lo-dB) Series fabricated, low noise, angle valve provides high flow capacity with high noise attenuation. Typical applications involve gas collection systems, compressor surge control, and gas-to-flare lines.

3. Aerodynamic Control Valve Noise Reduction (cont.)

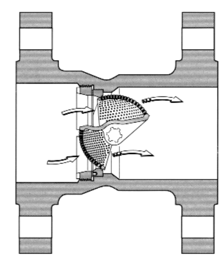
Noise prediction and attenuation are identical with the 41000 Lo-dB Series. Furthermore, an optimal second stage cage provides added noise reduction on high pressure drop service when required. The 72000 Series valves are available with outlet sizes up to 36" and capacities up to 13500 Cv, and with expanded outlet to reduce valve outlet velocity.

72009 Series V-LOG

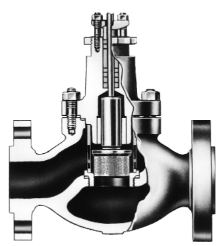
Used on very high pressure ratio applications (usually > 10 to 1), where the 2-stage drilled cage design cannot provide acceptable noise levels and/or some trim velocity limitations are required. The trim design is a brazed stack of overlapping discs which form individual tortuous flow paths. High path flow resistance is achieved by right angle turns with some contractions and expansions. Customized staging and flow characteristics can be achieved because each stack uses individual laser cut discs. The small flow paths shift sound frequency to increase transmission losses while area expansion and path resistance reduce trim velocities and decrease sound source strength. The V-LOG trim is available in the very large and versatile 72000 Series style angle bodies with expanded outlets to reduce valve outlet velocities. Like the 72001 and 72002 Series this product is typically used in large gas line applications, Vent-to-Flare, Soot Blower, and Compressor Recycle.

77000 Series Labyrinth Trim

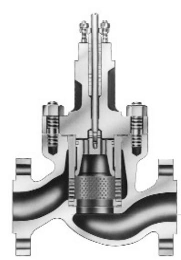
This is a specialized valve, of extremely tough construction fitted with an effective multiple-staged labyrinth trim. The multi-step labyrinth type plug and seat ring incorporate Stellite-faced seating surfaces which, when coupled with leveraged actuator force, provide tight shutoff. The labyrinth flow pattern, with a large number of steps, results in gradual pressure reduction and quiet operation - approximately 20 dBA quieter than conventional valves. Perhaps most importantly, the shape of the flow passages are designed to prevent deposits and entrapment of solids that may be entrained in the fluid stream. Combined with low fluid velocity, longer wear is ensured. These plus many other features make the 77000 Series valve ideal for high pressure drop applications, especially those involving solid-entrained fluids typical of drilling rig platforms, where it has achieved notable success.



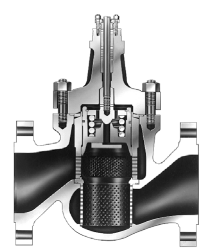
30000 Series Varimax® with Lo-dB Trim



21000 Series Valve with Lo-dB Trim

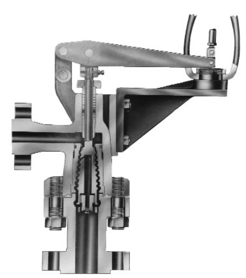


41000 Series Valve with Lo-dB Trim

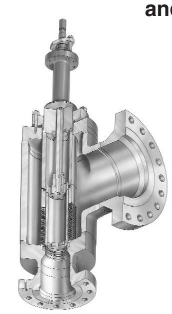


41000 Series Valve with Lo-dB Trim and Optional Diffuser

41000 Series Double Stage with Lo-dB Trim

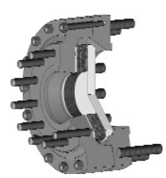


77000 Series Lo-dB Valve



72000 V-Log

Camflex DVD



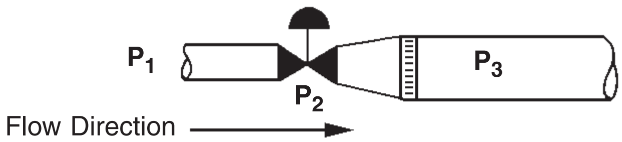
V-Log Restrictor

Lo-dB Cartridge

Lo-dB Expansion Plate

3. Aerodynamic Control Valve Noise Reduction (cont.)

3.3 Lo-dB Static Restrictor Selection



Used with either conventional or low noise valves, Lo-dB cartridges and expansion plates can be an extremely cost effective low noise system.

The static restrictor should be sized using valve sizing equations. Normally, the pressure ratio across the restrictor should be taken as 2 to 1 for initial sizing purposes. The addition of a restrictor holds a higher downstream pressure on the control valve, reducing the noise generation of the valve. A pressure drop of at least 20% of the total pressure drop should be taken across the valve to assure good control. If a conventional valve requires a pressure drop of less than 20% to meet the acceptable noise level, a Lo-dB valve must be considered.

For high system pressure ratios, two or more restrictors may be used. For sizing purposes, a pressure ratio of 2 to 1 should be taken across each restrictor.

3.3.1 Estimation of Sound Level

Aerodynamic noise generated by a low noise static restrictor (Lo-dB cartridge or expansion plate) can be calculated by using the same procedure employed to estimate low noise control valve noise level.

When a valve and a restrictor are in series, the method for calculating the overall noise level will vary somewhat depending upon how the valve and restrictor are connected (i.e., reducer or length of pipe). The following methods are used to calculate system noise.

Case I

Valve and downstream restrictor(s) are close coupled by reducer(s).

1. Calculate the aerodynamic valve noise for a conventional valve or a low noise valve, using the restrictor inlet pressure as valve downstream pressure, and pipe wall thickness and pipe diameter downstream of the restrictor(s).
2. Calculate the sound level of the restrictor(s) by using the methods for low noise valves.
3. Find the total sound level for the valve and restrictor(s) combination.
   a. From the sound levels calculated in Steps 1 and 2, subtract 6 dBA for each restrictor downstream of a noise source. The limit of 12 dBA applies with 2 or more downstream restrictors.
   b. Determine the final sound level by logarithmic addition. Logarithmically add the results above according to Figure 4 to obtain the estimated sound level downstream of the final restrictor.

Note that the close coupling of the valve and Lo-dB cartridges and expansion plates results in a lower predicted noise level than when separated by pipe.

Case II

Valve and downstream restrictor(s) are separated by a length of pipe (not close coupled).

1. Calculate the sound level downstream of the final restrictor as in Case I.
2. Calculate the sound level for the control valve using restrictor upstream pressure as valve downstream pressure, and pipe wall thickness and pipe diameter of the connecting pipe. This is the sound level radiated by the connecting pipe.
3. Compare sound levels of the connecting pipe and downstream of the final restrictor. The connecting pipe is an effective noise source that should be examined to determine overall system performance.

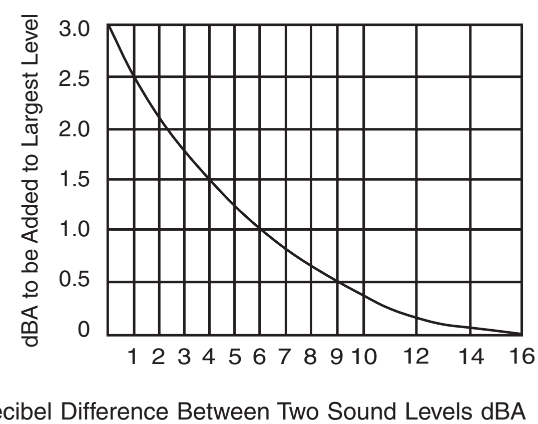


Figure 4

4. Atmospheric Vent Systems

4.1 Introduction

Noise emitted from atmospheric vents, using either conventional, low noise valves, or valve-restrictor systems can be calculated using the procedure below. Spherical radiation is assumed which reduces noise by 6 dBA for each doubling of distance. However, at long distances, much lower noise levels would be expected due to atmospheric absorption and attenuating effects of topography, wind, temperature gradients, and ground effects.

Lo-dB static resistors (cartridges and plates) used with either Lo-dB or conventional valves, can often provide the most cost effective solution to vent applications. If these systems are used, only the final system sound level is considered.

4.2 Noise Calculation Procedure

Step 1 Calculate the base sound level for a conventional valve, low noise valve, or static restrictor by the methods given in the previous sections. However, in each case, use transmission loss, TL, equal to zero. Correct for distance, r, by subtracting 20 log r/3 for distance in feet (or 20 log r for distance in meters) to obtain the corrected sound level.

Step 2 **Correct for Directivity**
The directivity index is important in vent applications because of the directional nature of high frequency noise typical of control valve signature. Figure 5 is based on typical average peak frequencies of 1000 to 4000 Hz. If a silencer is used, the directivity index will change appreciably. Silencers, by design, absorb the high frequency (directional) components from the valve spectral signature, leaving predominantly low frequency noise. Consequently, for silencer applications, use one half the directivity index at each angle shown. Add the directivity index to the sound level determined in Step 1.

Step 3 For large distances, make appropriate corrections for wind and temperature gradients, topography, and atmospheric absorption, for a specific application.

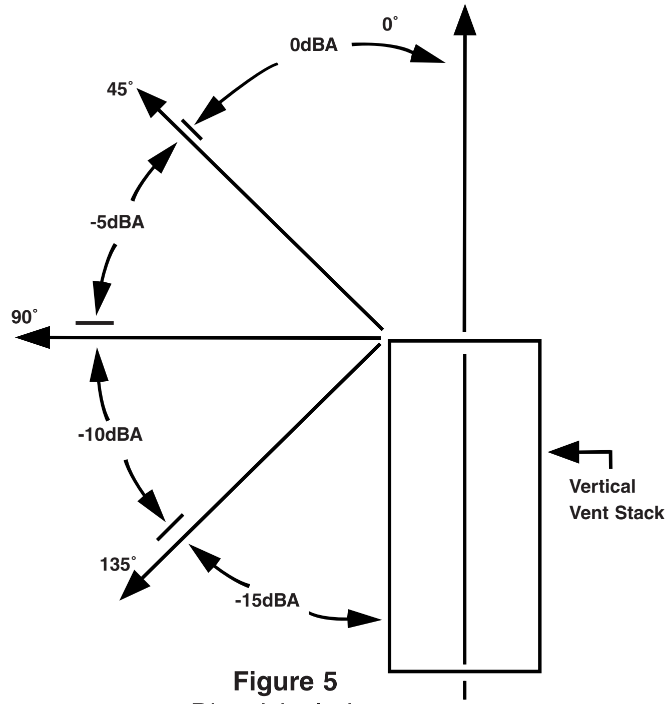


Figure 5
Directivity Index

5. Hydrodynamic Noise

5.1 Prediction

The basic sources of hydrodynamic noise include:

- Turbulent flow
- Flashing
- Cavitation
- Mechanical vibration resulting from turbulent flow, cavitation, and flashing

Problems resulting from high hydrodynamic noise levels are flashing erosion, cavitation erosion, and combined erosion/corrosion. Unlike aerodynamic noise, hydrodynamic noise can be destructive even at low levels, thus requiring additional limitations for good valve application practice.

The international standard for control valve hydrodynamic noise prediction is IEC-534-8-4. The method in this standard is based on physics principles, and can be applied to any valve style. Like the aerodynamic standard the intended accuracy is plus or minus five decibels.

5. Hydrodynamic Noise (cont.)

The factors used in the calculations are:

- **F_L** Pressure Recovery, Choked Flow Factor
- **X_F** Differential Pressure Ratio

$$X_F = \frac{\Delta P}{P_1 - P_v} = 1/\sigma$$

σ = Cavitation Index (See Sizing Manual, OZ1000)

- **X_{FZ}** Pressure Ratio at Which Cavitation Inception is Acoustically Detected
- **η_F** Acoustical Efficiency Factor (Ratio of Sound Power to Stream Power)

  $1x10^{-8}$ for Std. Globe Valve

- **ΔL_F** Valve Specific Correction Factor for Cavitating Flow

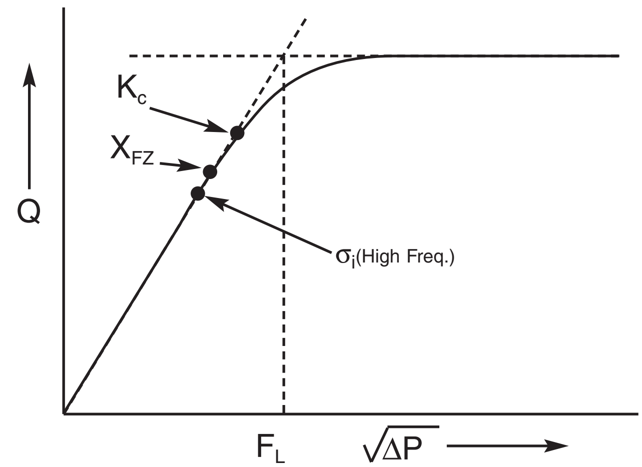
  L_{wi} Measured - L_{wi} Calculated

- **L_{wi}** Internal Sound Power
- **F_B** Factor to Account for Cavitation of Multi-Component Fluids Having a Range of Vapor Pressures

5.1.1 The graph below depicts a typical liquid flow curve for a control valve over a wide range of pressure drops, with constant inlet pressure. Flow rate is plotted on the vertical axis versus the square root of pressure drop. At relatively low to moderate pressure drop, in the range of fully turbulent and non-vaporizing flow, flow is proportional to the square root of pressure drop. At high-pressure drop, flow is choked; that is, further decrease of downstream pressure does not result in an increase in flow rate. Note that the pressure recovery factor, F_L, is determined by test at the intersection of the straight line representing non-vaporizing flow and the straight-line representing choked flow. The factor K_c is determined as the point of deviation of the straight-line flow curve. The newer factor X_{FZ} is determined acoustically as the point where an increasing noise level is detected. Although not required for use in this standard, the cavitation factor Sigma denotes the inception of vaporization determined by high frequency detection through the use of an accelerometer. Note that the X_{FZ} and Sigma may be very close, and for all practical purposes can be considered the same point, unless Sigma incipient is found by use of very high frequency detection.

Flow Curve

- X_{FZ} is approximately σ_i (not high frequency detection)



5.1.2 Non-Cavitating Flow

The basic equations for non-cavitating flow are shown below:

- **Stream Power**

$$W_m = \frac{\dot{m} \cdot \Delta P}{\rho_F}$$

- **Sound Power**

$$L_{wi} = 10\text{Log}_{10}\left(\frac{\eta_F \cdot \dot{m} \cdot \Delta P}{\rho_F \cdot w_o}\right) \qquad w_o = 1x10^{-12}$$

$$\boxed{L_{wi} = 120 + 10\text{Log}\eta_F + 10\text{Log}\dot{m} + 10\text{Log}\Delta P - 10\text{Log}\rho_F}$$

Key Factors: Mass Flow and Pressure Drop

5.1.3 Cavitating Flow

The equation for cavitating flow has additional quantities:

$$L_{wi} = 120 + 10\text{Log}\eta_F + 10\text{Log}\dot{m} + 10\text{Log}\Delta P - 10\text{Log}\rho_F + \Delta L_F + 180 \cdot \frac{(X_{FZ})^{0.0625}}{(X_F)^{X_{FZ}}} \cdot (1 - X_F)^{0.8} \cdot \text{Log}_{10}\left(\frac{1 - X_{FZ}}{1 - X_F}\right)$$

ΔP is Limited by Critical Pressure Drop

Key Factors: ΔL_F, Ratio X_{FZ} / X_F

NOTE: FB factor is included in the cavitation noise adder for multi-component fluids.

5. Hydrodynamic Noise (cont.)

5.1.4 Pipewall Transmission Loss

The internal frequency spectrum is first

$$L_{wi(f)} = L_{wi} - 10\mathrm{Log}_{10}\left(\frac{f_m}{500}\right) - 2.9$$

- Standardized Spectrum Based on Std. Single Seated Globe Valve Water Testing
- Noise Spectrum in the Single Octave Band Range of 500 Hz through 8000 Hz

The pipewall transmission loss is then calculated for each frequency band:

- Pipe Wall Transmission Loss

$$TL(f) = 10 + 10\mathrm{Log}_{10}\left(\frac{C_p \cdot \rho_p \cdot t}{C_f \cdot \rho_F \cdot d_0}\right) + 10\mathrm{Log}_{10}\left(\frac{f_r}{f} + \left(\frac{f}{f_r}\right)^{1.5}\right)^2$$

Key Factors: Pipe Diameter, Wall Thickness, Ratio of Center Frequency to Ring Frequency

5.1.5 External Sound Pressure Level

The external unweighted sound power level is next calculated in each frequency band:

$$L_{we(f)} = L_{wi(f)} - 17.37\left(\frac{l_p}{2d_o}\right) \cdot 10^{-(0.1TL(f))} - TL(f) + 10\mathrm{Log}_{10}\left(\frac{4 \cdot l_p}{d_o}\right)$$

l_p = 3 Meters Min.

Key Factors: Diameter and Length of Pipe and TL

Then the A-weighted external sound power level is determined:

- A-Weighting Sound Levels (L_{WA})

fm, Hz	500	1000	2000	4000	8000
Correction Values, dB	-3.2	0.00	+1.2	+1.0	-1.1

- A-Weighted Sound Power Level

$$L_{wAe} = 10\mathrm{Log}_{10} \sum_{n=1}^{5} 10^{0.1LwAn}$$

L_{wAn} is the External A-Weighted Sound Power Level of the n^{th} Octave Band

Finally, the external sound pressure level is calculated:

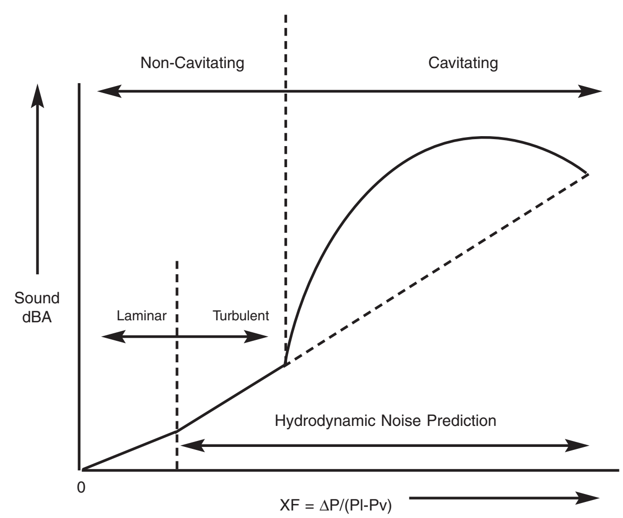
- Based on open field conditions and cylindrical radiation, the sound pressure level 1 meter downstream of the valve outlet flange and 1 meter lateral of the pipe is:

$$L_{pAe} = L_{wAe} - 10\mathrm{Log}_{10}\left(\frac{\pi \cdot l_p}{l_o}\left(\frac{d_i}{d_o} + 1\right)\right)$$

5.1.6 A generalized hydrodynamic noise curve is depicted in the graph shown below. Sound level in dBA is plotted on the vertical axis versus the pressure drop ratio (pressure drop divided by inlet pressure minus vapor pressure). The most interesting feature of this curve is that a rounded curve is superimposed on the intermediate straight line (shown in the turbulent region) and the dotted line projection. This illustrates the result of the use of the sound power equation that adds an additional quantity in the cavitating region.

The method predicts that all globe valves having equal pressure recovery will produce the same noise level in the non-cavitating region. However, by selecting a valve with higher X_{FZ} (or lower Sigma), the inception of increased noise due to vaporization and cavitation will be forestalled to higher-pressure drop. With the selection of anti-cavitation valve trim, the resulting noise levels will be dramatically reduced.

Hydrodynamic Noise

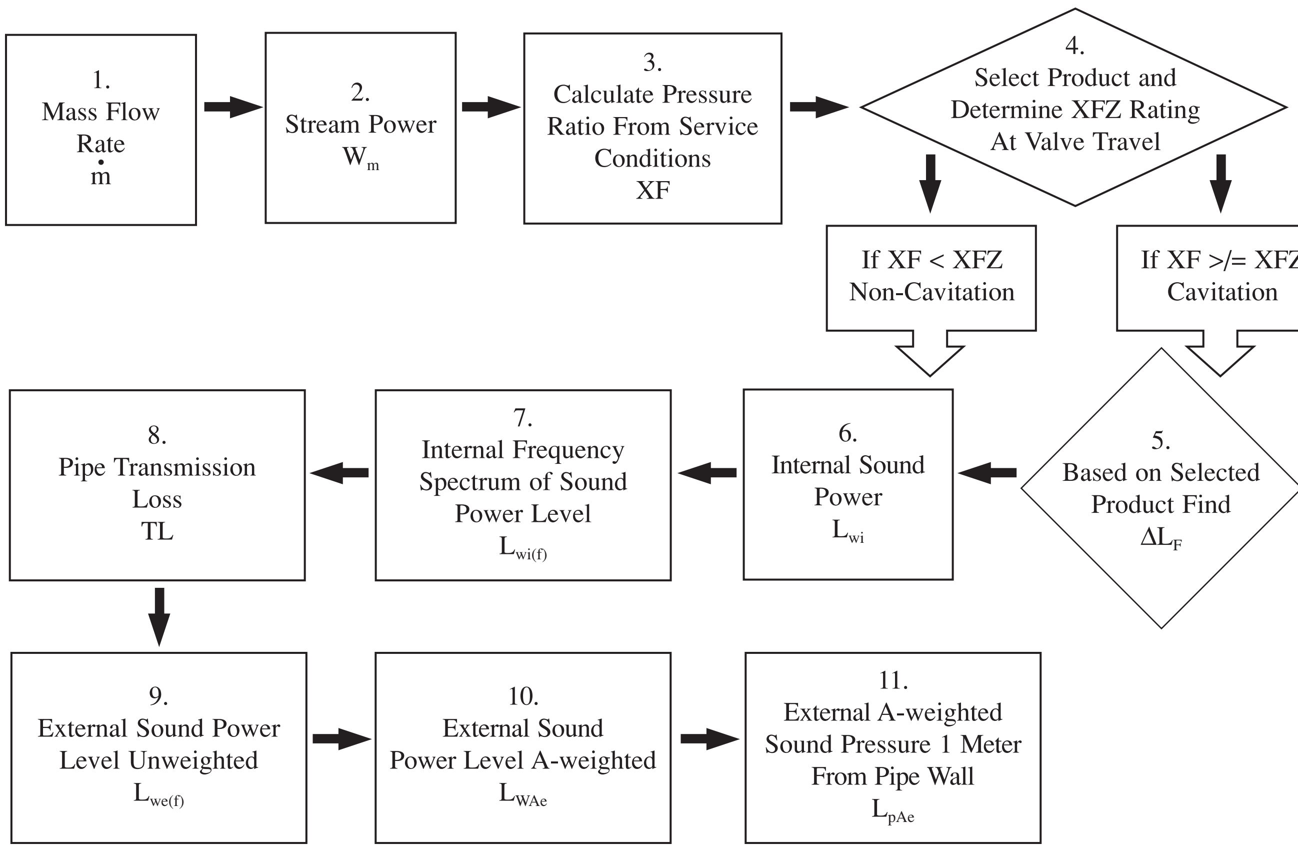


Acoustical Version of the Sigma (σ) Curve

5. Hydrodynamic Noise (cont.)

5.1.7 A flow chart illustrating the hydrodynamic noise prediction method is shown below.

Hydrodynamic Noise Prediction Flow Chart



Nomenclature

c_F	=	speed of sound in the fluid
c_p	=	speed of sound of the longitudinal waves in the pipe wall
C_v	=	flow coefficient
d_i	=	inside diameter of the downstream pipe
d_o	=	outside diameter of the downstream pipe
f	=	frequency
f_m	=	octave center frequency
f_r	=	ring frequency
F_B	=	Factor to account for cavitation of multi-component fluids having a range of vapor pressures.
F_L	=	liquid pressure recovery factor
l_o	=	reference length of pipe = 1
l_p	=	length of pipe
L_{pAe}	=	A-weighted sound-pressure level external of pipe
L_{WAn}	=	A-weighted sound power level of the n^{th} octave band
L_{We}	=	external sound power level (unweighted)
L_{WAe}	=	A-weighted sound power level external of pipe
L_{Wi}	=	internal sound power level
ΔL_F	=	valve specific correction value
$\dot{m}$	=	mass flow rate
P_o	=	reference sound pressure = 2 x 10^{-5}
p_v	=	absolute vapor pressure of fluid at inlet temperature
p_1	=	valve inlet absolute pressure
p_2	=	valve outlet absolute pressure
ΔP	=	differential pressure between upstream and downstream (p_1-p_2)
T_1	=	inlet absolute temperature
T_L	=	transmission loss (unweighted)
t	=	thickness of wall pipe
U_2	=	fluid velocity at outlet of valve
W_m	=	fluid power loss in the valve
W_o	=	reference sound power = 10^{-12}
x	=	ratio of differential pressure to inlet absolute pressure ($\Delta P/p_1$)
x_F	=	differential pressure ratio ($\Delta P/p_1$-p_v)
x_{Fz}	=	characteristic pressure ratio for cavitation
η_F	=	acoustical efficiency factor for liquid (at ϕ = 0.75)
ρ_F	=	density (specific mass) at p_1 and T_1
ρ_p	=	density (specific mass) of pipe material

5.2 Application Guidelines and Equipment Selection

5.2.1 Cavitating Fluid

Cavitating fluid, usually water, can be one of the most devastating forces found in control valve applications. Caused by high localized stresses incurred by vapor implosion, it can quickly destroy critical valve parts if not properly controlled or eliminated. Fortunately, because the imposed stresses are highly localized, the vapor implosion must occur at or very close to valve metal surfaces to cause damage. This attribute provides many methods of controlling these destructive forces, some of which are described below.

The damage potential of any cavitating fluid is directly proportional to:

1. **Inlet Pressure P_1:** The inlet pressure is directly related to the amount of energy available to cause damage. The greater the inlet pressure, the greater the potential energy applied to the cavitating fluid and the greater the damage potential.
2. **Degree of Cavitation:** This factor, related to the percentage of the fluid which cavitates, is proportional to the required vs. actual valve F_L and to the degree that the fluid vapor pressure is well-defined. For example, using a valve with a F_L of 0.9, a system with a required F_L of 0.98 will have a much greater percentage of fluid cavitating than a system with a required F_L of 0.92, both at the same P_1, and will, therefore, experience greater damage. Secondly, a fluid that does not have a well-defined vapor pressure, that will boil over a wide temperature range, will likely be self-buffering in a cavitating application. Consult Masoneilan Engineering.

5. Hydrodynamic Noise (cont.)

3. Fluid Surface Tension: Since fluid surface tension affects the amount of pressure recovery experienced before vapor implosion, it directly affects the amount of energy so released. Consequently, fluids with low surface tension will tend to cause less damage.

For a more detailed description of cavitation in control valves and the factors impacting cavitation damage see reference 6.8.

5.2.2 Equipment Application

Preventative Measures: There are several preventative measures that can eliminate cavitation damage. First, however, it cannot be overstressed that cavitation must be eliminated or controlled. Further use of hard materials is not a solution and will only delay ultimate valve failure. On all but the lowest pressure systems, this delay will be insignificant. Several steps which can be taken are as follows:

1. Use a valve with low pressure recovery (high F_L): Often on a moderately cavitating system, cavitation can be eliminated by using a low pressure recovery valve such as a cage guided globe. The goal is to increase the critical pressure drop, F_L^2 $(P_1\text{-}P_v)$, above the valve ΔP.
2. Reduce ΔP: If the ΔP can be reduced so that the vena contracta pressure does not drop below the vapor pressure, cavitation will be eliminated. Often this can be done by changing the physical location of the valve (elevation, etc.).
3. Use of back pressure plates: If system rangeability permits, use of back pressure plates to increase P_2, reducing ΔP below the critical ΔP_1 can be the most cost effective solution.

Cavitation Control: At low to moderately high pressures, cavitation can be controlled by use of specially designed trims. These trims function in two ways:

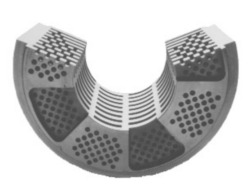
1. High F_L: Recall, cavitation damage is directly proportional to the percentage of fluid cavitating. Consequently, valves with low pressure recovery (high F_L) will experience less cavitation damage.
2. Containment: Because cavitation is a highly localized phenomenon which requires direct impingement on metal surfaces to cause damage, use of a design which diverts the bubble implosion away from metal surfaces can be effective.

Masoneilan's 21000 and 41000 Series control valves, when equipped with single or doubled stage anti-cavitation trims, are examples of "cavitation control" valves. This method of "cavitation control" is a very cost effective solution, however, there are limitations to the amount of energy that can be absorbed by these trim designs. For further details contact Masoneilan Application Engineering.

Cavitation Prevention: Where high potential energy exists (high P_1) in cavitating fluid, cavitation must be eliminated through use of good multiple-stage trim, designed specifically for anti-cavitation service. Ideally, the pressure staging should be such that the smallest pressure drop occurs at the last stage to minimize overall valve pressure recovery. To minimize plug damage, the flow should be axial, parallel to the plug; for good control, there should be no dead spots in the trim, providing a good smooth flow characteristic. Finally, since most valves of this type will be seated much of the time, extra-tight shutoff should be provided. For further information concerning control valves providing "cavitation containment" trim designs see Masoneilan's Specification Data Brochure on the following: 41000, 72000, and the 78400 LincolnLog®.

Flashing Fluid: When flashing exists in a control valve, potential physical damage to the valve must be considered. Flashing fluid vapor carries liquid droplets at high velocity, quickly eroding carbon steel. Use of higher alloys such as chrome-moly will result in acceptable performance. Flashing noise is determined by use of the IEC calculation method @ $X_F \approx 1.0$ and $\Delta L_F = 0$.

Axial Flow, LincolnLog 78400 Trim



Variable Resistance Trim Type S VRT® Sectioned to Show Flow Passages

Variable Resistance Trim Type C Sectioned to Show Flow Passages

6. References

6.1 CEI/IEC 60534-8-3, 2nd Edition, 2000
"Control Valve Aerodynamic Noise Prediction Method"

6.2 CEI/IEC 534-8-4, 1st Edition, 1994
"Prediction of Noise Generated by Hydrodynamic Flow"

6.3 CEI/IEC 534-8-1, 1st Edition, 1986
"Laboratory Measurement of Noise Generated by Aerodynamic Flow Through Control Valves"

6.4 CEI/IEC 534-8-2, 1st Edition, 1991
"Laboratory Measurement of Noise Generated by Hydrodynamic Flow Through Control Valves"

6.5 ISA Standard ISA S75.17, 1989
"Control Valve Aerodynamic Noise Prediction"

6.6 ISA Standard ISA S75.07, 1987
"Laboratory Measurement of Aerodynamic Noise Generated by Control Valves"

6.7 ISA Recommended Practice ISA RP75.23, 1995
"Consideration for Evaluating Control Valve Cavitation"

6.8 Bulletin OZ1000
"Masoneilan Control Valve Sizing Handbook"

Appendix: Installation Considerations

In closed systems, control valve noise generated by the throttling process is radiated to the atmosphere through downstream piping. Noise calculations are based on laboratory conditions, including an acoustic “free field” (an environment without acoustic reflections) and with piping systems designed so that they will not contribute to generated noise. Consequently, like any other equipment in a facility, these factors should be considered when developing expected installed control valve noise levels.

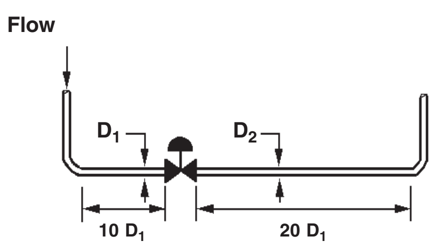
Acoustical Environment

The acoustical environment refers to the type of “field” in which the valve is installed. It is a measure of the sound build-up expected due to acoustic reflections from boundaries, other equipment, as well as the total size (volume) of the installed environment. These factors are explained in any basic acoustics text but cannot be anticipated by the control valve manufacturer.

Piping Design Guidelines

The following guidelines should be considered for optimum results.

1. **Straight run before and after valve**
   Straight pipe for at least 10 diameters upstream and 20 diameters downstream of the valve is recommended.



2. **Isolating Valves**
   Isolating block valves must be selected to ensure minimum resistance to fluid flow. Full bore type is preferred.

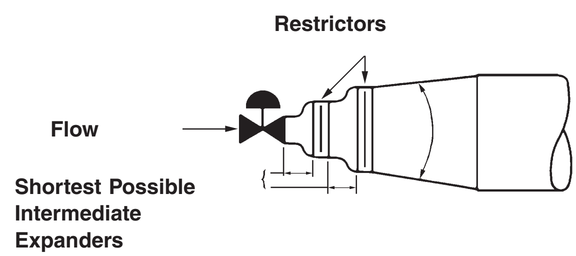
3. **Fluid Velocity**
   Depending on velocity, fluid flow may create noise levels higher than that produced by the control valve. Masoneilan provides a means for calculating the Mach number (M) at service pressure and temperature conditions.

$$M = \frac{\text{Average Velocity of Flowing Medium}}{\text{Sound Velocity in the Flowing Medium}}$$

   With Lo-dB trim and fluid velocities above $^1/_3$ Mach, fluid velocity noise must be calculated and total system sound level reevaluated.

4. **Expanders and Reducers**
   Like any other source of turbulence in a fluid stream, expanders and reducers may be the cause of additional system noise. Concentric expanders and reducers with included angles smaller than 30° upstream and 15° downstream of the valve are recommended.

   As an exception to the above, short reducers (large included angles) are recommended with Lo-dB restrictors because of their inherent stiffness and the fact that velocity is low upstream of the restrictors.



5. **Bends, T's and other Piping Connections**
   Drastic disruptions in the fluid stream, especially if high fluid velocity exists, are potential noise sources. Possible improvements to conventional design for piping connections are shown in Figure 6.

Piping Supports

A vibration free piping system is not always possible to obtain, especially when thin wall piping such as Schedules 5S and 10S are used. Supports in strategic locations, however, will alleviate a lot of the potential structural problems. At the same time, they reduce the possibility of structure borne noise. In some cases, piping may be buried to reduce noise and vibration problems.

Appendix: Installation Considerations (cont.)

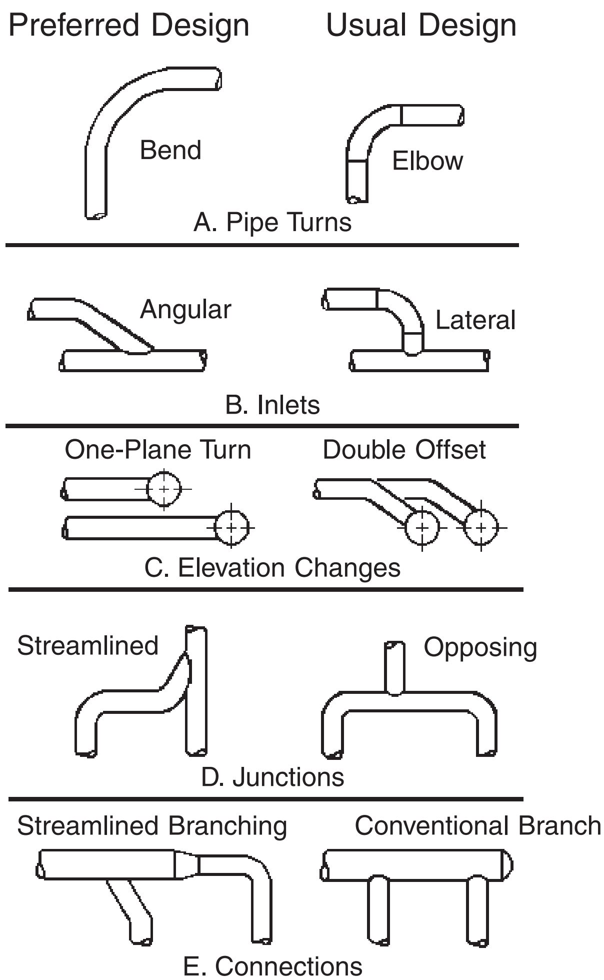


Figure 6

Extreme Sound Levels

Fluid borne valve generated noise induces mechanical vibration in the piping system which is radiated to the environment as valve noise. The valve sound level is indicative of this surface motion. Excessive vibration can cause failure or damage to valve and pipe mounted instruments, and accessories. Piping cracks, loose flange bolts, and other problems can develop. For this reason, valve noise should be limited to 115-120 dBA. If higher levels are expected, Lo-dB valves, Lo-dB static restrictors or other alternatives should be used to reduce noise below the recommended levels. Note that pipe insulation and certain other "add on" noise control treatments, which do not change the pipe wall surface motion, are ineffective. In most cases, such extreme sound levels are precluded by occupational and environmental noise requirements anyway.

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