UNIVERSAL™

SILENCER APPLICATION HANDBOOK

1993 EDITION

By Jim R. Cummins jr., P.E.
and Bill G. Golden

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FORWARD

As president of UNIVERSAL SILENCER, for some years I have wanted this company to produce a practical handbook on noise fundamentals and the application of noise control products — specifically industrial silencers. I felt that the experience and resources of UNIVERSAL SILENCER uniquely qualified it to undertake such a project.

A couple years prior to the publication of the 1993 edition, the assignment to write the handbook was turned over to Jim Cummins and Bill Golden, whose credentials are summarized elsewhere in this book. It was agreed that we wanted the end result to be a handbook that improved upon any similar resource material currently available; it should be a useful tool for our customers and our employees, as well as consultants, instructors and generally anyone with an interest in basic noise or its control. While the book obviously would be written to a relatively high technical level, we wanted it to be understandable and useful for non-engineers as well.

We are pleased with the finished product, and we extend thanks and congratulations to Jim and Bill for the professionalism and diligence they displayed in making this handbook a reality. We hope it will help readers better understand noise and how to control it.

Roy McDaniel, President
Universal Silencer
September, 1993

PROFESSIONAL PROFILE

Jim R. Cummins jr., P.E., Mem. INCE

Jim Cummins is Manager of Engineering with UNIVERSAL SILENCER, A Division of Nelson Industries, Inc. He has extensive experience in noise control, structural and mechanical design, fluid flow and computers. His professional experience includes twelve years as a Senior Engineer with the Inlet & Exhaust Systems and Power Plant Engineering units of the General Electric Gas Turbine Products Department in Schenectady, New York, and seven years as an independent consultant with J. R. Cummins and Associates. In addition, Mr. Cummins developed software for Enable Software of Ballston Lake, New York, as a Senior Software Engineer.

Mr. Cummins has both BS and MS degrees from Southern Methodist University with majors in physics and mathematics. He has done extensive post-masters work in Applied Mechanics at the University of Connecticut and in Computer Science at Rensselaer Polytechnic Institute. He is a registered Professional Engineer in New York and Wisconsin and is Board Certified by and a member of the Institute of Noise Control Engineering. He is also a member of the Acoustical Society of America, the American Society of Mechanical Engineers, the American Welding Society and is a Professional Member of the American Institute of Steel Construction. Mr. Cummins is a Certified Welding Inspector per ANSI/AWS QC1-88. He has a patent on heat recovery anti-icing systems, assigned to General Electric Company, and has published a technical paper on gas turbine exhaust noise modeling.

PROFESSIONAL PROFILE

Bill G. Golden

Bill Golden is a semi-retired acoustical consultant to UNIVERSAL SILENCER, A Division of Nelson Industries, Inc. He has more than 35 years experience with industrial noise control equipment as both a design engineer and researcher and as manager of the research, development and sales departments of several companies. He has also been an independent acoustical consultant for the last five years.

Mr. Golden has a BS degree in mathematics from East Texas State University and has done additional graduate study at both Southern Methodist University and Texas Tech University. He is a retired member of both the American Society of Mechanical Engineers, where he has over 25 years of membership, and the Acoustical Society of America. Mr. Golden has a patent on liquid separators and has written numerous papers relating to noise control for the ASME and other technical publications.

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CHAPTER ONE
INTRODUCTION

Excessive noise can be both objectionable and hazardous. Even low amplitude noise can cause extreme distress to either workers or neighbors. Any noise problem can be handled in one of three ways: control the source, control the path or control the destination/listener. For our purposes, noise control is defined as controlling the source, or preventing or lessening of noise before it is generated, while noise reduction refers to controlling the path (i.e., reducing noise after it is produced). With tighter legal requirements now in effect and more in sight, it is best to reduce noise at its source. Unfortunately, many of our modern conveniences have noise as a natural side effect, so to accomplish our goal of a quieter environment we need to proceed in a logical and systematic manner.

This handbook introduces the application engineer to industrial noise control and reduction as it applies to engines, turbines, blowers, compressors, vacuum pumps, vents, blowdowns and fans. These are but a few of the major sources of plant and process noise caused by aerodynamic sources, gaseous fluids such as air, steam and natural gas. Plant and area noise of this type depend on many things, including the type and number of sources, equipment speed (rpm), process pressures and the economy of layout and construction. Noise administration — controlling the listener's access to the noise source — is not addressed in this handbook.

A noise analysis, however complex, should be included in the initial planning and engineering of a new plant or facility. Failure to understand or recognize the complications that may be caused by legal requirements or neighborhood response to excessive noise can cost money and time at a later date.

There are many reasons for noise control and reduction, including:

- Risk of hearing loss,
- Adverse community response,
- Area criteria,
- Annoyance,
- Speech interference, and
- Safety and economics.

Aside from the environmental and human consequences of excessive noise, the application engineer must also avoid potential mechanical problems by recommending the proper silencer or silencers for a specific application.

The development of silencers that reduce industrial noise dates back to the mid-thirties. By 1960, there were numerous silencer designs (many of which were patented) being marketed. More recently, the emphasis has been on improving materials and manufacturing techniques of silencers, such as UNIVERSAL SILENCER's enhanced computer operation, for optimum quality control and increased product life. The newest noise reduction technique is called "active" noise control since it depends on a dynamic control process for its implementation. This will be briefly covered in Chapter 4. The computer, especially the personal computer, has eliminated most of the time consuming details and drudgery involved in the design and application of silencers and other noise abatement products. Even so, the application engineer must fully understand the logic of the computer program, that is, the basis of the calculations and all of the input data. An engineer must also be able to override the actual computer printout when necessary since no single current program covers all of the many variables encountered in plant noise analysis. Accordingly, there is no substitute for theoretical knowledge and/or experience.

C H A P T E R   T W O

FUNDAMENTALS

2

INTRODUCTION

SOUND is both a physical phenomenon and the sensation of hearing. One hears sound but the pressure waves still exist if there is no listener. By definition, **NOISE** is unwanted sound. One person's sound, rock music, for example, can be another person's noise. Unlike sound, noise can only exist in the presence of a listener. Noise may be intermittent, erratic or continuous depending upon its source and may be within certain discrete frequencies or broadband across the entire frequency spectrum.

The **BEL** is a logarithmic unit by which noise is measured and evaluated. A more convenient unit is the **DECIBEL** (dB), which is one-tenth the size of the **Bel** and is in common usage. The decibel is a dimensionless unit used to express the relationship of one sound to another, either in terms of pressure (Figure 2-1) or in terms of power (Figure 2-2). The **SOUND PRESSURE LEVEL** is the logarithm of the ratio of the actual pressure to a common reference value of 0.0002 microbar (20 μ pascals) and is often abbreviated as L_p. The **SOUND POWER LEVEL** is the logarithm of the ratio of the actual power of the sound to 10^{-12} watt and is often abbreviated as L_W. See Figure 2-3 for a concise summary of the definitions of L_W and L_p. Since the decibel is a logarithmic function, it allows us to represent a very wide range of sound levels.

L_W has been described as resembling the power or wattage rating of a light bulb, while L_p is compared to the amount or intensity of light that is produced at a given distance and in a given environment. Just as more light intensity is produced in a light-colored room than in a dark-colored room, L_p is higher in a reverberant area than in a highly absorptive area.

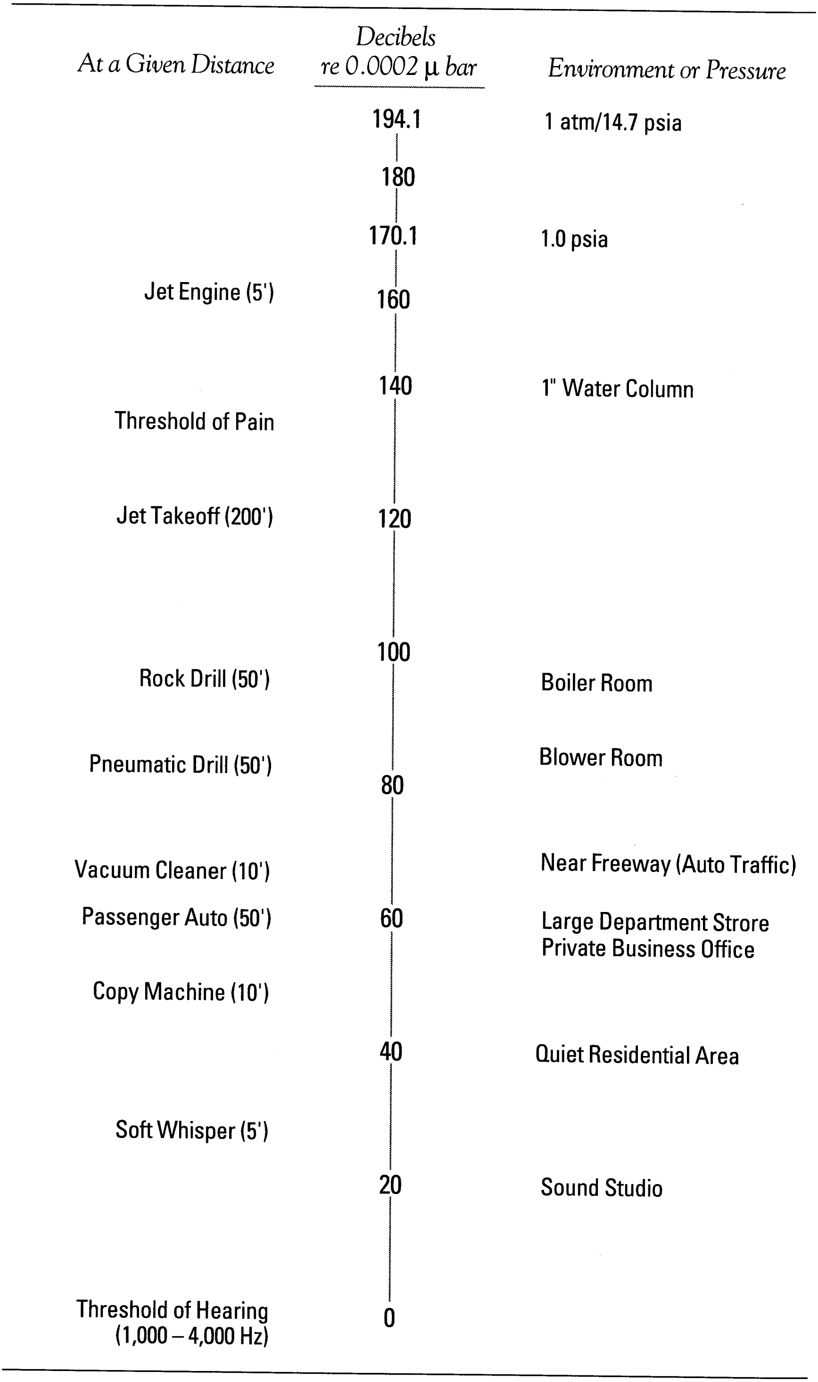

Figure 2-1. Example Sound Pressure Levels of Common Sounds

Power (watts)	Decibels re 10^{-12} watt	Typical Sources
100,000,000.	200	
		Saturn Rocket
1,000,000.	180	
		Jet Engine/Afterburner
10,000.	160	
100.	140	Large Centrifugal Fan
		Large Orchestra
1.0	120	Large Chipping Hammer
		Loud Radio
0.01	100	
		Shouting
0.0001	80	
		Normal Conversation
0.000001	60	
		Small Electric Clock
0.00000001	40	
		Soft Whisper
0.0000000001	20	
0.000000000001	0	

Figure 2-2. Example Sound Power Levels of Common Sounds

Generally speaking, L_W levels cannot be measured directly, but must be calculated in part. L_p levels, which are dependent on distance (divergence) and to some extent on the acoustic environment, are easily measured. L_p values without reference to distance are meaningless.

The lowest sound level that an average person can hear is about 20 dB. The human ear can tolerate sound levels to 100 dB, and in some instances even higher for short durations without lasting hearing damage. Sound levels in excess of 90 dB over an extended period can cause permanent loss of hearing.

SOUND POWER LEVEL (L_W) is the logarithm of the ratio of the sound power (W) to a reference power of 10^{-12} watt and is the total sound energy of a given source, as shown in the formula:

$$L_w = 10 \log \frac{W}{10^{-12}}, dB$$

SOUND PRESSURE LEVEL (L_p) is the logarithm of the ratio of a sound pressure (P) to a reference pressure which is usually chosen as 0.0002 microbar (20μ pascals), as shown in the formula:

$$L_p = 20 \log \frac{P}{0.0002}, dB$$

Figure 2-3. Definition of Sound Power and Pressure Levels

Decibels are combined (i.e., added) logarithmically, on an energy basis as given in Figure 2-4. The procedure for subtracting decibels is essentially that of logarithmic addition in reverse. Most modern scientific calculators, spreadsheets and computer programs can be set up to use the equation in Figure 2-4.

$$\text{Decibel (dB)} = 10 \log\left(\text{antilog}\ \frac{dB_1}{10} + \text{antilog}\ \frac{dB_2}{10}\right)$$

Figure 2-4. Adding Decibel Levels

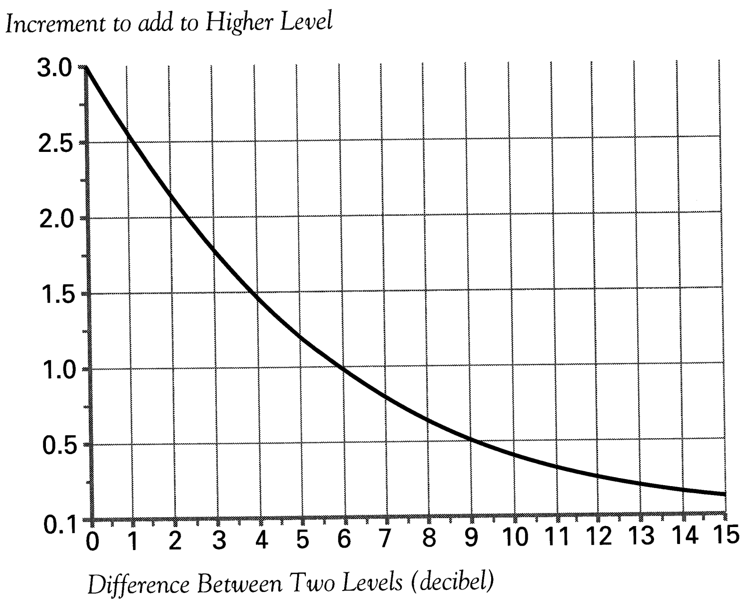

Figure 2-5. Adding Decibel Levels Graphically

Figure 2-5 gives a graphic method of combining two decibel levels. Figure 2-6 demonstrates the use of the equation in Figure 2-4 or Figure 2-5 to combine frequency bands by pairs to obtain the overall sound level. A similar combination by pairs could be done with distinct sources. The normal practice is to combine the smallest sources together and then add the larger ones.

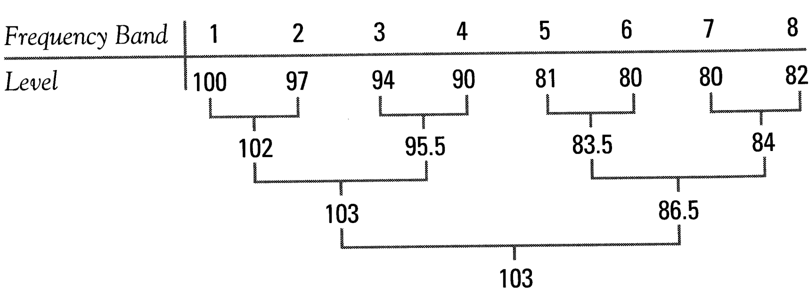

Figure 2-6. Examples of Combining Decibels

If several equal sound sources are present, then the total increase in sound level is dependent only on the number of sources, as shown in Figure 2-7. When combining equal sources in this manner, one must assume close proximity of sources, or that the measurement point is distant enough that the sources may be considered close.

Number of Equal Sources	Add to Single Source Level (dB)
2	3
3	5
4	6
5	7
6 – 7	8
8	9
9 – 10	10
N	10 log (N)

Figure 2-7. Combination of Equal Sources

FREQUENCIES AND WEIGHTINGS

The audible frequency range is roughly between 50 and 8,000 Hz. Most speech information is contained in the frequencies between 500 and 3,000 Hz; however, in qualitative analysis, it is often necessary to consider frequencies from 20 to 11,000 Hz and sometimes higher. The frequency spectrum can be divided into groups or bands of frequencies. One way to group frequencies is by establishing **OCTAVE BANDS.** An octave band covers a 2:1 range of frequencies (meaning that the upper limit is twice the lower limit) and is identified by its center frequency. Acoustic experts have agreed on a set of center frequencies and standardized them in ANSI S1.6-1984 (ASA 53-1984). The preferred series of octave bands, as they are called and shown in Table 2-1, are based on a decimal scale and have replaced the older series based on powers of two. Most specifications are written using octave bands 1-8 (63 Hz to 8k Hz).

Band Number	Octave Bands		
	Center Frequency (Hz)	Range (Hz)	
		Low	High
0	31.5	22	44
1	63	44	88
2	125	88	177
3	250	177	354
4	500	354	707
5	1,000	707	1,414
6	2,000	1,414	2,828
7	4,000	2,828	5,656
8	8,000	5,656	11,312
9	16,000	11,312	22,624

Table 2-1. Preferred Octave Bands

One-third octave band levels are used in the standard (ANSI S1.6) as the definition for the preferred center frequencies and are used for a more detailed analysis as each octave is split into three parts as shown in Table 2-2.

Other bands, such as the one-half octave band, have in most instances been replaced with that of one-third or narrow band frequency levels. Other fixed frequency band width spectrum, usually gotten from an FFT analyzer, are also available. These are most useful for vibration or other applications in which predominantly pure tones are found.

Any number of octave band combinations will produce the same overall sound level. Evaluation and analysis then must be based on octave bands (when available) and not merely upon overall levels.

When octave or other band levels are given, they may be combined either with or without weighting factors to obtain the overall sound level. This is shown diagrammatically in Figure 2-6, where octave band levels are combined in pairs using the equation in Figure 2-4 to finally obtain the so-called linear or unweighted sound level. When noise criteria are discussed in Chapter 3, the notion of weighting and its value will be clearer.

One-Third Octave Band Number	Center Frequency (Hz)	Range (Hz)		Corresponding Preferred Octave Band
		Low	High	
14	25	22	28	0
15	**31.5**	28	36	**22 – 44**
16	40	36	44	
17	50	44	56	1
18	**63**	56	71	**44 – 88**
19	80	71	88	
20	100	88	112	2
21	**125**	112	141	**88 – 177**
22	160	141	177	
23	200	177	224	3
24	**250**	224	282	**177 – 354**
25	315	282	354	
26	400	354	447	4
27	**500**	447	562	**354 – 707**
28	630	562	707	
29	800	707	891	5
30	**1,000**	891	1,122	**707 – 1,414**
31	1,250	1,122	1,414	
32	1,600	1,414	1,778	6
33	**2,000**	1,778	2,239	**1,414 – 2,828**
34	2,500	2,239	2,828	
35	3,150	2,828	3,546	7
36	**4,000**	3,546	4,466	**2,828 – 5,656**
37	5,000	4,466	5,656	
38	6,300	5,656	7,079	8
39	**8,000**	7,079	8,913	**5,656 – 11,312**
40	10,000	8,913	11,312	
41	12,500	11,312	14,130	9
42	**16,000**	14,130	17,780	**11,312 – 22,624**
43	20,000	17,780	22,624	

Table 2-2. One-Third Octave Band and Octave Band Definition

The human ear responds to amplitude (loudness) and frequency (pitch) but is not equally sensitive to all frequencies. At 3,000 Hz 90 dB sounds much louder than at 500 Hz.

Subjective reaction to noise is difficult to predict. A 10 dB increase doubles the perceived loudness, while a 3 dB increase is barely noticeable. Loudness is usually stated in sones and if a noise is judged twice as loud as another, it has a sone rating of twice the other source. Sones are calculated by adjusting each octave band level according to the ear's sensitivity in that band and then by adding the effect of all of the bands. Additional discussion of ways to account for the ear's sensitivity is included in Chapter 3.

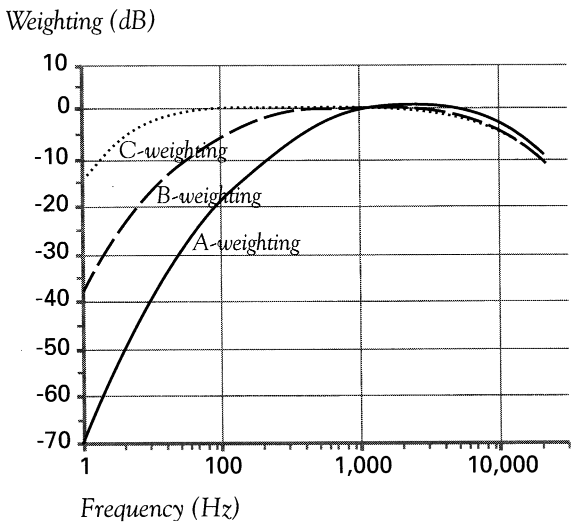

Figure 2-8. Standard SLM Weighting Networks

The standard sound level meter (SLM) has three basic frequency weighting networks (A, B and C) with the "A" scale being the more heavily weighted against lower frequency. The "A" scale response (dBA) to sound is approximately that of the human ear at the threshold of hearing. The "C" scale is only slightly weighted at low and high frequencies and for many measurements is used interchangeably with the linear or unweighted sound level. Figure 2-8 shows the standard A-, B- and C-weighting from ANSI S1.4-1983 (ASA 47-

1983). The B-weighting is seldom used now. To indicate the weighting of a sound spectra the units are often given with a suffix corresponding to the network, such as dBA, dB(A), dBC or dB(C). Purists are shocked by this usage but it is very common.

A-weighted sound levels may be measured or calculated from octave band sound pressure levels (L_p) by adding the weighting factors as in Table 2-3 and combining them into a single value.

Octave Band Center Frequency	31.5	63	125	250	500	1k	2k	4k	8k	16k
A-weighting Factor	-39	-25	-16	-9	-3	0	+1	+1	-1	-7

Table 2-3. Factors for A-weighting Octave Bands

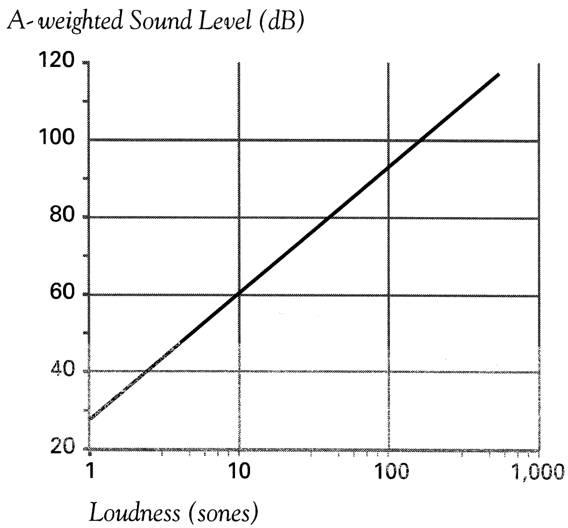

Figure 2-9. Sones vs. A-weighted Sound Levels

Loudness (sones) and A-weighted levels are similar as shown in Figure 2-9. While loudness is technically the more precise estimate of listener response, the A-weighted level is more often used because it is easier to measure.

PROPAGATION AND ABSORPTION

As one gets farther away from a source, the intensity of noise becomes less due to the spreading of the sound waves; this is called divergence and it follows an inverse square relationship with distance.

SPHERICAL DIVERGENCE of noise occurs when the source is relatively small and a far field condition exists (see near field on following page); where the area is free of obstructions and reflective surfaces, L_W and L_p are related by the equations below.

$$L_P = L_W - 20 \log r - 0.5 \text{ dB}$$

or

$$L_P = L_W - 10 \log (4\pi r^2) + 10 \text{ dB}$$

HEMISPHERICAL DIVERGENCE usually occurs when the source is relatively large and near the ground as are most industrial applications. In this case L_W and L_p are related by the equations below.

$$L_P = L_W - 20 \log r + 2.5 \text{ dB}$$

or

$$L_P = L_W - 10 \log (2\pi r^2) + 10 \text{ dB}$$

Where

r = *distance from source in feet.*

Table 2-4 is a tabular solution for the conversion value added to L_W to obtain L_p using the above equations.

Distance from Source (ft)	L_W to L_P Conversion (dB)
3	-7
5	-11
10	-17
15	-21
20	-23
25	-25
50	-31
75	-35
100	-37
200	-43
300	-47
400	-49
500	-51
600	-53
800	-55
1,000	-57

Table 2-4. Hemispherical Divergence

To calculate L_p values at a distance other than that specified or measured, then

$$\underset{\left(\substack{\text{at}\\\text{distance}\\\text{desired}}\right)}{L_P} = \underset{\left(\substack{\text{at}\\\text{known}\\\text{distance}}\right)}{L_P} - 20\log\left(\frac{\text{distance desired}}{\text{known distance}}\right)$$

Note that a smaller (closer) distance desired will yield an increase in L_p, whereas a larger distance desired will yield lower L_p. When the distance from the source is either halved or doubled, under far field conditions, the change is 6 dB. Measured L_p is generally lower than that calculated due to atmospheric conditions, natural or man-made obstacles and absorptive surroundings.

Divergence inside a room or building does not follow the inverse square law discussed above. Close to the source, the direct noise predominates. Away from the source, noise is reduced by a combination of distance, directivity and the surface treatment of the walls, ceiling and floors. This room divergence equation is given below and shown in Figure 2-10 for several room constants.

$$L_P = L_W + 10 \log\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) + 10 \text{ dB}$$

Where

Directivity Factor Q	=	*2 for a small source in the center of a room* *4 for a small source near the wall* *8 for a small source in the corner* *Large source Q may vary with direction*
r	=	*distance from source in feet*
R	=	*room constant* = $\frac{\alpha A}{1-\alpha}$
A	=	*total surface area of room in square feet, and*
α	=	*sound absorption coefficient of room surfaces at a given frequency.*

As an example, Figure 2-11 shows the computation of divergence for a blower or fan installed in the center of a hard surface room.

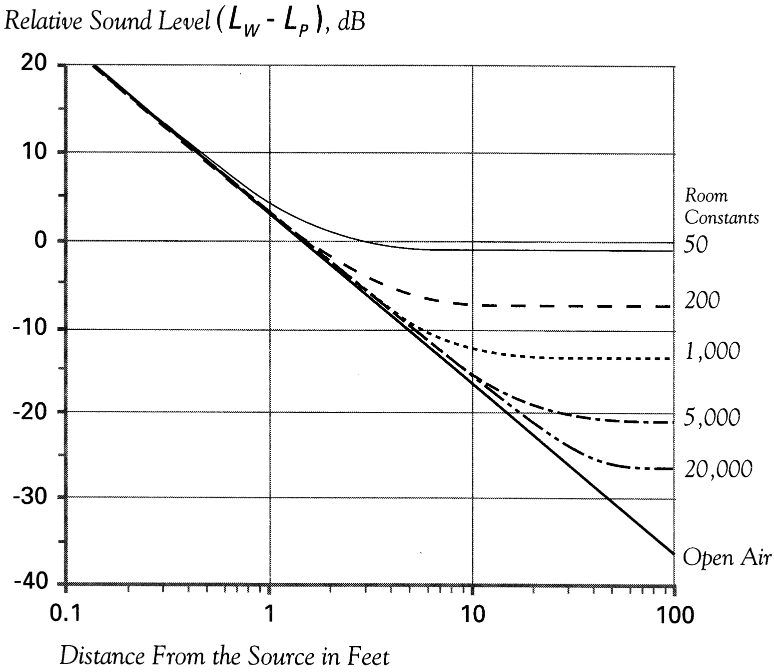

Figure 2-10. Graphic Solution of Room Divergence (Q = 2)

Q = 2

r = 5 feet

α = 0.15 (average)

A = 1,600 square feet, and

Calculating R, one gets

$$R = \frac{(0.15)(1{,}600)}{(1-0.15)} = 282 \text{ square feet}$$

From Figure 2-10 one gets $L_W - L_P = 6\,dB$,

while normal divergence would have given approximately $13dB$.

Figure 2-11. Example of Room Divergence

Atmospheric molecular absorption (often called **EXCESS AIR ATTENUATION**) also reduces the sound level in the upper frequencies for outdoor sources as indicated in Figure 2-12 and as more completely defined in ANSI S1.26-1978 (ASA 23-1978).

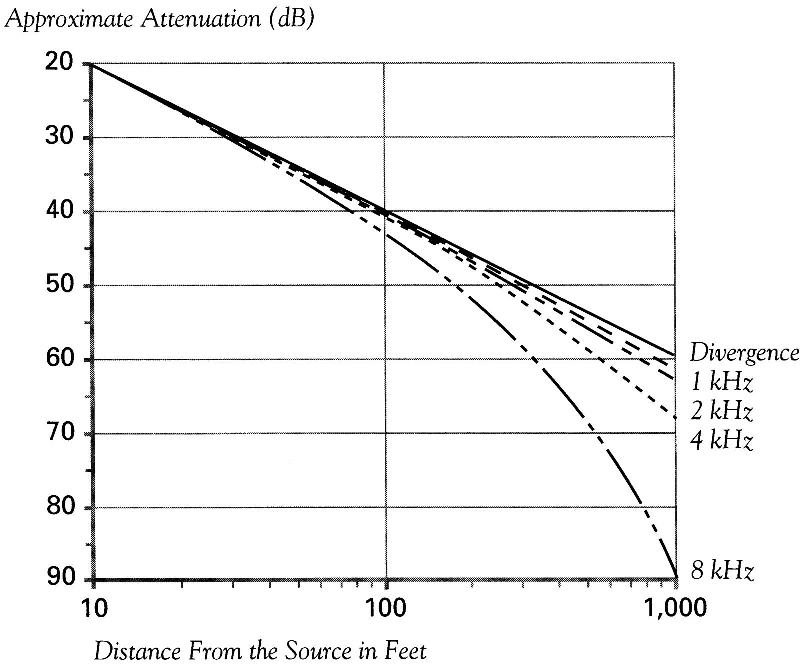

Figure 2-12. Excess Air Attenuation

NEAR FIELD MEASUREMENTS (that is, measurements close to the source) should be taken at a distance of not less than one wavelength of the lowest frequency of concern to avoid the presence of "pseudo-sound," or pressure fluctuations that do not propagate. To determine the wavelength of a sound, divide the speed of sound by the frequency.

Wavelength (λ) is defined as

$$\lambda = \left(\frac{c}{f}\right)$$

where λ has units of length in feet, and f is frequency with units of Hertz (Hz).

The speed of sound (c) in air is approximately

$$c = 49.03\sqrt{T}$$

with units of feet per second, and T is temperature with units of °R (460 + °F).

Another correction to spherical divergence is **DIRECTIVITY** which was mentioned briefly in the discussion of room divergence. Directivity may have positive or negative effects, depending upon the measurement position (angle) and the size of the intake or discharge opening. Approximate directivity corrections are tabulated in Table 2-5 and are accurate enough for most application purposes. Once the sound level (based on spherical or hemispherical divergence) is found, the correction for directivity is added to that number.

BACKGROUND NOISE or ambient noise must be considered if the difference between the source in question and the noise present without it is less than 10 dB. The background sound level should be subtracted logarithmically or using Figure 2-13.

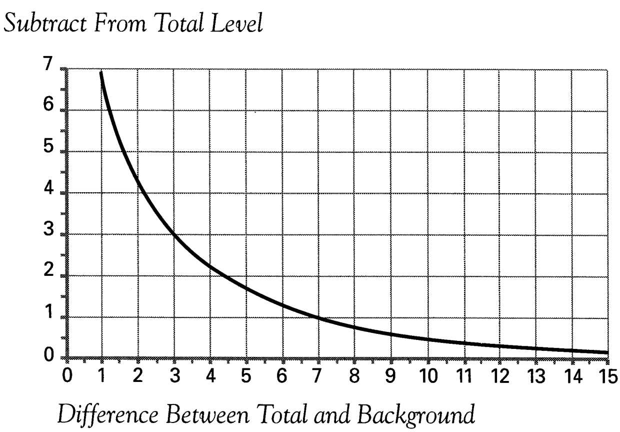

Figure 2-13. Background Noise or Ambient Correction

As an example, suppose the background noise in an industrial area is 65 dB and with the start-up of a continuous vent the level is increased to 70 dB. Here the difference is 5 dB. The correction from Figure 2-13 is 2 dB, which is subtracted from the total noise of 70 dB. The vent noise itself then is 68 dB, not 70 dB as measured since the already existing noise is 65 dB.

If the difference is less than 2 dB, then an accurate determination is not possible. If 10 dB or more, the background noise will not add to the total noise and can be ignored.

The **SOUND ABSORPTION** coefficient (α) is defined as the ratio of sound energy absorbed by a given surface to that incident upon the surface. Absorptive materials or acoustic panels may be added to the walls and ceilings to reduce noise within a room to acceptable levels. Noise reduction by added absorption follows Sabine's theory using the following equation:

$$L_p\,(Reduction) = 10\log\left(\frac{S_1}{S_2}\right) = 10\log\left(\frac{A\,(\alpha)_{(after)}}{A\,(\alpha)_{(before)}}\right)$$

Where

S = *total absorption in Sabines,*
A = *surface area in square feet, and*
α = *absorption coefficient of material.*

When absorption is doubled, L_p is decreased by 3 dB. Double the absorption again, and the total decrease is 6 dB. The practical limit is a reduction of 6 to 8 dB. The absorption coefficient (α) varies with both depth and density of the material as shown in Appendix XIII. The actual coefficient of a specific material or panel may be determined by test or obtained from the manufacturer.

The total absorption of a surface is the product of the absorption coefficient of that surface and its surface area.

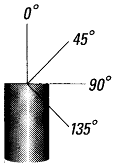 Angle to Direction of Flow	Source Diameter (in)	Octave Band Center Frequency (Hz)							
		63	125	250	500	1k	2k	4k	8k
0°	72 – 96	+4	+5	+5	+6	+6	+7	+7	+7
	54 – 66	+3	+4	+4	+5	+5	+5	+5	+5
	36 – 48	+2	+3	+3	+4	+4	+4	+4	+4
	26 – 32	+1	+1	+2	+2	+2	+2	+2	+2
	16 – 24	0	0	+1	+1	+1	+1	+1	+1
	8 – 14	0	0	0	0	0	0	0	0
	6	0	0	0	0	0	0	0	0
45°	72 – 96	+2	+3	+3	+4	+4	+5	+5	-5
	54 – 66	+1	+2	+2	+3	+3	+3	+3	+3
	36 – 48	0	+1	+1	+2	+2	+2	+2	+2
	26 – 32	0	0	0	+1	+1	+1	+1	+1
	16 – 24	0	0	0	0	0	0	0	0
	8 – 14	0	0	0	0	0	0	0	0
	6	0	0	0	0	0	0	0	0
90° and 135°	72 – 96	-1	-2	-5	-7	-10	-12	-15	-17
	54 – 66	0	-1	-2	-5	-8	-10	-13	-16
	36 – 48	0	0	-1	-3	-6	-7	-11	-15
	26 – 32	0	0	0	-1	-3	-5	-9	-14
	16 – 24	0	0	0	0	-1	-3	-7	-13
	8 – 14	0	0	0	0	-1	-2	-5	-11
	5 – 6	0	0	0	0	0	-1	-3	-6
	4	0	0	0	0	0	0	-1	-3

Table 2-5. Approximate Directivity Corrections (dB)

SOUND TRANSMISSION LOSS (TL) is defined as the logarithmic ratio of the incident sound power on one side of a barrier (or partition) to the sound power transmitted to the other side as calculated using the following equation:

$$TL\ (dB) = 10 \log \frac{1}{\tau} = 10 \log \frac{W_i}{W_t}$$

Where

τ = *sound transmission coefficient*
W_i = *incident sound power, and*
W_t = *transmitted sound power.*

Stated another way, TL is the difference between the incident sound power level (L_{W1}) and the transmitted or radiated sound power level (L_{W2}). TL (dB) is usually stated in octave bands. TL is a function of mass, stiffness and dampening and is frequency dependent as graphically shown in Figure 2-14. TL increases 6 dB per octave and 6 dB for each doubling of the mass above the plateau region shown in the center of Figure 2-14.

For the other regions, an empirical procedure for determining the approximate TL of various materials is often applied, where for frequencies below the plateau, the following equation should be used:

$$TL = 20 \log W + k$$

Where

W = *surface weight of the material per square foot, and*
k = *frequency constant from Table 2-6 which is equal to (20 log (f)-33).*

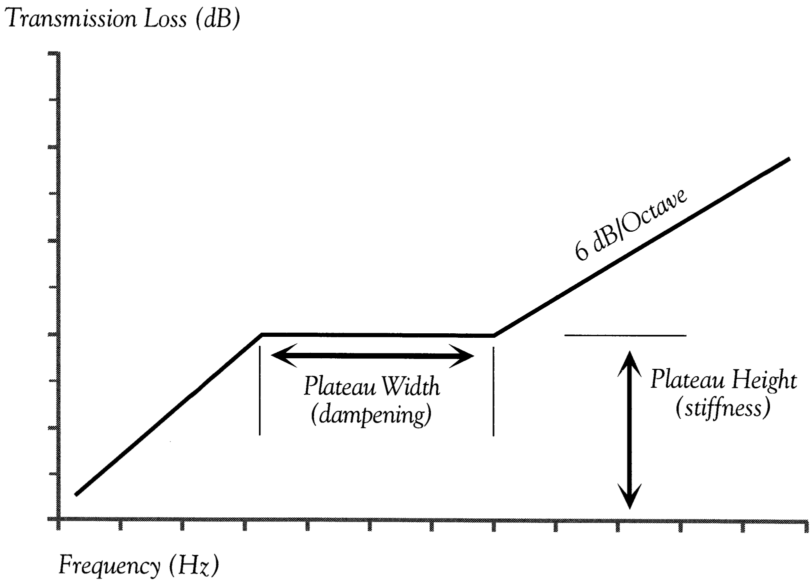

Figure 2-14. Generalized Sound Transmission Loss Curve

	Octave Band Center (Hz)							
	63	125	250	500	1k	2k	4k	8k
k (dB)	3	9	15	21	27	33	39	45

Table 2-6. Constants for TL Below Plateau

When TL, thus obtained, reaches the plateau height, it remains equal to the plateau height for the next three octave bands. The plateau height for various materials is provided in Table 2-7.

Type Material	Surface Density (lbs / ft^2 per inch)	Plateau Height (dB)
Aluminum	14	29
Steel	40	40
Glass	13	27
Lead	59	56
Concrete	12	38
Masonry Block	33	32

Table 2-7. Surface Density vs. Plateau Values

As stated previously, the TL increases 6 dB per octave above the plateau level.

To estimate the TL of 1/4" sheet steel using this procedure —

1. *Obtain the surface weight (W) and the plateau height from Table 2-7.*
2. *Using the equation on Page 21 and Table 2-6, calculate the TL for each octave band until the plateau height is reached. The TL is the same for the next three octave bands.*
3. *Finally, the TL for each of the remaining octave bands is increased by 6 dB.*

Figure 2-15. Summary of TL Calculation Method

Hz	63	125	250	500	1k	2k	4k	8k
TL ¼" Steel	23	29	35	40	40	40	40	46

Table 2-8. Sample Calculation for 1/4" Steel Barrier

Published TL values for various materials are tabulated and provided in Appendix XIV.

The approximate TL for pipe with various steel wall thicknesses can be estimated using the equation:

$$TL(dB) = 17 \log(tf) - 9dB$$

Where

t = *wall thickness in inches, and*
f = *center frequency (Hz).*

The TL of most acoustical materials has been tested by independent testing laboratories and is available from the manufacturer.

EFFECT ON TL OF OPENINGS

Even a small opening can have a significant effect on the TL of a wall or panel. The L_W of a sound that will pass through an opening into or out of a room is determined using the equation:

$$L_W = L_P + 10 \log A - 10\, dB$$

Where

L_P *is measured at or near the opening, and*

A *is the cross-sectional area of opening in square feet.*

High-frequency noise is particularly noticeable when it travels through small openings such as cut-outs for piping or conduits. These may require acoustical treatment. Large openings for ventilation or process air require absorptive-type silencers.

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NOISE CRITERIA

3

NOISE CRITERIA is defined as the maximum allowable sound pressure level (L_P) permitted at a specific distance and location from a noise source. This criteria is applied to areas where noise levels must be low enough to prevent employee discomfort and hearing loss and/or to prevent neighborhood complaints about noise. In either case, they form the basis on which the engineer decides how much noise reduction is required in any particular application. Figure 3-1 lists typical applications and the possible criteria that could be used.

Application	Criteria
Personnel Exposure	*dBA and sone levels*
Neighborhood Annoyance	L_P*, sones and equivalent dBA levels*

Figure 3-1. Noise Criteria Selection

There are numerous noise criteria now in use throughout industry and by local, state and federal agencies. Only the more common industrially related and binding criteria will be summarized and included in this chapter.

U.S. OCCUPATIONAL SAFETY AND HEALTH ADMINISTRATION (OSHA)

This noise regulation attempts to protect workers' hearing unrelated to the sound level. The maximum exposure levels for occupational noise identified by OSHA under the Federal Safety Act are shown in Table 3-1. The "slow response" on the sound level meter is used to obtain an average level when the levels fluctuate more than a few dB. In addition, exposure to impact noise must not exceed 140 dB. Some authorities have extended Table 3-1 to include a 16-hour exposure at 85 dBA.

Duration per Day (Hrs)	Sound Level dBA (Slow Response)
8	90
6	92
4	95
3	97
2	100
1 1/2	102
1	105
1/2	110
1/4 or less	115

Table 3-1. Permissible Noise Exposure Levels (OSHA)

Feasible engineering and/or administrative noise control measures are required when exposure levels exceed 90 dBA for any length of time. Engineering controls include reducing equipment and process noise through redesign, replacement and/or change in operation, or by various forms of acoustical treatment and/or installation of silencers. Administrative controls include reducing noise exposure by limiting the time an employee is exposed to excessive noise levels and by annual audiometric testing and providing for hearing protectors. When daily exposure consists of two or more periods of excessive noise at different levels, the combined effect must be considered if the sum of the following fractions exceeds unity:

$$\frac{C_1}{T_1} + \frac{C_2}{T_2} + \frac{C_3}{T_3} \cdots + \frac{C_n}{T_n} > 1$$

Where

C = *actual exposure time in hours at a given level, and*
T = *permissible exposure time from Table 3-1 for that level.*

As an example, if a worker is exposed to 90 dBA for 4 hours, 95 dBA for 3 hours and 100 dBA for 1 hour, the combined effect is:

$$\frac{4}{8} + \frac{3}{4} + \frac{1}{2} = 1.75 > 1.0$$

Since the total is greater than unity, engineering or administrative controls are required in this case.

Employers must administer continuing and effective hearing conservation programs when employee noise exposures equal or exceed an 8-hour, time-weighted average of 85 dBA. Regardless of the measured sound level, if an employee is found to have a hearing loss, the employer may be cited for non-compliance.

Any number of octave band combinations will produce the same overall dBA level. Evaluation and analysis must be based on octave bands (when available) and not merely upon overall noise levels.

COMMUNITY NOISE EQUIVALENT SOUND LEVELS (dBA)

Many community and state requirements use the concept of different acceptable sound levels for day and night. This concept determines the energy average equivalent A-weighted sound level and then applies a 10 dBA penalty for nighttime operation. Figure 3-2 defines the various quantities used in community noise evaluation. The equations in Figure 3-3 define the method of calculating L_{dn} and L_{eq} (24) from measured sound levels.

L_d	=	*The equivalent dBA level between 7a.m. and 10 p.m.*
L_{dn}	=	*The day/night equivalent dBA level (equivalent level for a 24-hour period, with an additional 10 dB imposed on levels between 10 p.m. and 7a.m.).*
L_{eq}	=	*The equivalent sound level (the dBA level which has the same energy level as that computed from the actual time varying sound measured over a specific time period).*
$L_{eq}(x)$	=	*L_{eq} over a period of (x) hours.*
L_n	=	*Equivalent dBA level (L_{eq}) between 10 p.m. and 7a.m. (Also known as nighttime equivalent sound level.)*
$L_x(L_{10}, L_{50}, L_{90})$	=	*That time varying dBA level that is expected x percent of the time.*

Figure 3-2. Common Quantities Used in Community Noise

$$L_{dn} = 10 \log \frac{1}{24}\left[15 \times 10^{\left(\frac{L_d}{10}\right)} + 9 \times 10^{\left(\frac{L_n+10}{10}\right)}\right]$$

$$L_{eq}(24) = 10 \log \frac{1}{24}\left[15 \times 10^{\left(\frac{L_d}{10}\right)} + 9 \times 10^{\left(\frac{L_n}{10}\right)}\right]$$

Figure 3-3. Calculation of L_{dn} and L_{eq}

$L_d - L_n$	Add to L_d for L_{dn}	Add to L_d for L_{eq} *(24)*
-4	10	2
-2	8	1
0	6.5	0
2	5	-0.7
4	3.5	-1
6	2	-1.5
8	1	-1.7
10	0	-1.8

If L_d = 80 and L_n = 82, then L_{dn} = 88 and L_{eq} (24) = 81 dBA.
If L_d = 75 and L_n = 75, then L_{dn} = 81.5 and L_{eq} (24) = 75 dBA.
If L_d = 70 and L_n = 64, then L_{dn} = 72 and L_{eq} (24) = 68.5 dBA.

Table 3-2. Relationships Between L_d-L_n, L_{dn} and L_{eq} (24) Sound Levels

Typical L_{dn} sound pressure levels at generalized types of locations are provided in Table 3-3.

Location	Typical L_{dn} (dBA)
Wilderness Ambient	35
Rural Residential	40
Agri Crop Land	44
Wooded Residential	51
Old Urban Residential	60
Urban Housing on Major Street	68
Urban High Density Apartment	78
Downtown with some Area Construction	79
¾ Mile from Major Airport	86
Apartment next to Freeway	88

Table 3-3. Typical L_{dn} Sound Levels at Various Locations

NR (ISO) Noise rating curves were developed by the International Standards Organization (ISO) to rate noise using the 1,000 Hz octave band as a common reference (Table 3-4). The equivalent overall dBA level has been added for reference. NC (Noise Criteria) curves are another common criteria, and these rate noise within an indoor space (Table 3-5). Compared to NC curves, NR curves permit higher noise levels at lower frequencies and lower levels at higher frequencies. For both of these curves the sound to be rated is plotted by octave bands on standardized paper and the highest rating penetrated is the simple number rating.

NR	Octave Band Center Frequency (Hz)									dBA Equivalent
	31.5	63	125	250	500	1k	2k	4k	8k	
20	69.0	51.3	39.4	30.6	24.3	20.0	16.8	14.4	12.6	30
25	72.4	55.2	43.7	35.2	29.2	25.0	21.9	19.5	17.7	35
30	75.8	59.2	48.1	39.9	34.0	30.0	26.9	24.7	22.9	39
35	79.2	63.1	52.4	44.5	38.9	35.0	32.0	29.8	28.0	44
40	82.6	67.1	56.8	49.2	43.8	40.0	37.1	34.9	33.2	48
45	86.0	71.0	61.1	53.6	48.6	45.0	42.2	40.0	38.3	53
50	89.4	75.0	65.5	58.5	53.5	50.0	47.2	45.2	43.5	58
55	92.9	78.9	69.8	63.1	58.4	55.0	52.3	50.3	48.6	62
60	96.3	82.9	74.2	67.8	63.2	60.0	57.4	55.4	53.8	67
65	99.7	86.8	78.5	72.4	68.1	65.0	62.5	60.5	58.9	72
70	103.1	90.8	82.9	77.1	73.0	70.0	67.5	65.7	64.1	77
75	106.5	94.7	87.2	81.7	77.9	75.0	72.6	70.8	69.2	82
80	109.9	98.7	91.6	86.4	82.7	80.0	77.7	75.9	74.4	87
85	113.3	102.6	95.9	91.0	87.6	85.0	82.8	81.0	79.5	91
90	116.7	106.6	100.3	95.7	92.5	90.0	87.8	86.2	84.7	96
95	120.1	110.5	104.6	100.3	97.3	95.0	92.9	91.3	89.8	101
100	123.5	114.5	109.0	105.0	102.2	100.0	98.0	96.4	95.0	106
105	126.9	118.4	113.3	109.6	107.1	105.0	103.1	101.5	100.1	111
110	130.3	122.4	117.7	114.3	111.9	110.0	108.1	106.7	105.3	116
115	133.7	126.3	122.0	118.9	116.8	115.0	113.2	111.8	110.4	121
120	127.1	130.3	126.4	123.6	121.7	120.0	118.3	116.9	115.6	126
125	140.5	134.2	130.7	128.2	126.6	125.0	123.4	122.0	120.7	131

Table 3-4. NR (ISO) Noise Rating Curves

Noise Criteria	Octave Band Center Frequency (Hz)							
	63	125	250	500	1k	2k	4k	8k
NC 20	51	41	33	26	22	19	17	16
25	54	45	38	31	27	24	22	21
30	57	48	41	35	31	29	28	27
35	60	53	46	40	36	34	33	32
40	64	57	51	45	41	39	38	37
45	67	60	54	49	46	44	43	42
50	71	64	59	54	51	49	48	47
55	74	67	62	58	56	54	53	52
60	77	71	67	63	61	59	58	57
65	80	75	71	68	66	64	63	62
70	83	79	75	72	71	70	69	68

Table 3-5. NC (Noise Criteria) for Occupied Spaces

Speech Interference Level (SIL) is the average L_p levels within the speech range of 350 to 3,000 Hz under conditions where speech is possible (Tables 3-6 and 3-7). These criteria are used to rate the difficulty of speech communication in an area.

Distance (ft)	Normal	Raised	Very Loud	Shouting
1	77	83	89	95
3	67	73	79	85
6	61	67	73	79
12	55	61	67	73

Table 3-6. Approximate Speech Interference Levels

Table 3-7 compares the usual indoor criteria, NC and SIL, with A-weighted dB levels.

NC	20	30	40	50	60	70
dBA	31	40	49	58	68	77
SIL	22	32	42	51	61	71

Table 3-7. Comparison of NC, dBA and SIL Levels

U.S. ENVIRONMENTAL PROTECTION AGENCY (EPA)

The Environmental Protection Agency (EPA) established the Office of Noise Abatement and Control (ONAC) in response to the Noise Control Act of 1972. In 1974, EPA published the so-called "levels document" titled, "*Information on Levels of Environmental Noise Requisite to Protect Public Health and Welfare with an Adequate Margin of Safety*." The purpose of this document was to establish sound level criteria that would protect against noise-induced hearing loss in the general population. After review and analysis of the existing evidence and consultation with experts, it was concluded that an L_{eq} of 70 dBA over a 24-hour day and an L_{eq} (8) limit of 75 dBA was appropriate. Additionally, the EPA has identified L_{dn} levels of 45 dBA for indoor activity interference and 55 dBA outdoor activity interference. These levels were considered long-term goals rather than standards. During the early 1980s, the EPA lost its funding for noise abatement and closed ONAC even though many of the regulations are still in place. No additional effort in this regard has been made since that time. This criteria is considered very restrictive for most applications and has never been incorporated into any regulations.

TYPICAL REGIONAL OR LOCAL REGULATIONS

This section outlines several regional or local sets of regulations which have been copied widely and enjoy some degree of acceptance in industrial applications.

1. State of Minnesota Noise Control Regulation

NAC	Day (7 a.m. - 10 p.m.)		Night (10 p.m. - 7 a.m.)	
	L_{50}	L_{10}	L_{50}	L_{10}
1	60	65	50	55
2	65	70	65	70
3	75	80	75	80

NAC is the area noise classification as given below:

1. Residential areas, hotels, hospitals, schools, etc.
2. Urban shopping areas, rapid transit terminals, finance, insurance and similar trade areas.
3. Manufacturing areas.

2. New York City Noise Control Code*

Ambient Noise Quality Zone	Standards in L_{eq} (1) dBA	
	7 a.m. - 10 p.m.	10 p.m. - 7 a.m.
Low Density Residential	60	50
High Density Residential	65	55
Commercial and Manufacturing	70	70

* *Measured for any one-hour period.*

All noise measurements are made at the property line or as close thereto as is reasonable. However, measurements are not to be at a distance of less than 25 feet from a noise source. The New York City Noise Control Code also contains allowable noise limits for various noise sources.

3. Chicago Environmental Control Ordinance Maximum Allowable Sound Pressure Levels (L_p)

Manufacturing Zoning Districts	Octave Band Center Frequency (Hz)									dBA
	31.5	63	125	250	500	1k	2k	4k	8k	
1. Residential Boundaries										
Restricted	72	71	65	57	51	45	34	34	32	55
General	72	71	66	60	54	49	44	40	37	58
Heavy	75	74	69	64	58	52	47	43	40	61
2. Business and Commercial Boundaries										
Restricted	79	78	72	64	58	52	46	41	39	62
General	79	78	73	67	61	55	50	46	43	64
Heavy	80	79	74	68	63	57	52	48	45	56

A. Districts as defined in the City of Chicago Zoning Ordinance.

B. Maximum levels in "Restricted" manufacturing zoning districts apply at property boundaries.

C. Maximum levels in "General or Heavy" manufacturing zoning districts apply at the boundary of the residence, business or commercial districts, or at 125 feet from the nearest property line of plant or operation.

The Chicago Environmental Control Ordinance also contains allowable noise limits for various noise sources in addition to those shown in the previous table.

4. General Specifications for Ships of U.S. Navy (Section 073)

The accompanying table and following notes summarize this Navy specification.

Maximum Allowable Sound Pressure Levels (L_p) in dB

Ship Space Category	Octave Band Center Frequency (Hz) 31.5	63	125	250	500	1k	2k	4k	8k	SIL Value
A	115	110	105	100	SIL Value Requirement			85	85	64
B	90	84	79	76	73	71	70	69	68	—
C	85	78	72	68	65	62	60	58	57	—
D	115	110	105	100	90	85	85	85	95	—
E	115	110	105	100	SIL Value Requirement			85	85	72
F	115	110	105	100	SIL Value Requirement			85	85	65

Category A. Spaces other than category E where intelligible speech communication is necessary.

Category B. Spaces where comfort of personnel in their quarters is normally considered to be an important factor.

Category C. Spaces where it is essential to maintain especially quiet conditions.

Category D. Spaces or areas where a higher noise level is expected and where deafness avoidance is a greater consideration than intelligible speech communication.

Category E. High noise levels where intelligible speech communication is necessary.

Category F. Topside operating stations on weather decks where intelligible speech communication is necessary.

5. U.S. Air Force Specifications (MIL-N-83155B)

These standards are applicable to noise suppressor systems used for ground run-up of jet aircraft engines that are mounted on test stands, in aircraft or on power check pads.

Far Field measurements of L_p are to be taken at 250 feet in 10 degree circular increments from the engine exhaust. The level shown in each octave band must not exceed that shown for the applicable grade of performance specified below.

Far Field Maximum Allowable Sound Pressure Levels (L_p) dB

Grade	Octave Band Center Frequency (Hz)								dBA
	63	125	250	500	1k	2k	4k	8k	
I	85	78	74	71	70	70	68	67	76
II	94	91	88	84	83	83	79	73	89
III	97	97	96	95	92	91	91	92	99

Near Field levels must not exceed the values specified below at any of the positions where run-up and maintenance crew members are normally stationed during engine operation.

Near Field Maximum Allowable Sound Pressure Levels (L_p), dB

Octave Band Center Frequency (Hz)								dBA
63	125	250	500	1k	2k	4k	8k	
114	114	114	114	117	117	117	120	124

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CHAPTER FOUR

TYPES OF NOISE REDUCTION

An urgent need for noise reduction (NR) is found in many segments of today's society, including

- Product research,
- Medical and diagnostic services,
- Commercial and industrial facilities, and
- Suburban and residential communities.

As defined in the introduction, noise reduction is the attenuation (or abatement) of noise after it is produced. Noise reduction is generally limited to the application of

- Silencers,
- Acoustic Enclosures,
- Partial Barriers, and
- Isolation and Damping Components.

We will discuss each of these types of noise reduction.

SILENCERS

The three basic types of silencers are

1. Reactive Passive (Chamber-Type),
2. Absorptive Passive (Dissipative-Type), and
3. Active or Dynamic (Electronic).

These types will be discussed separately and in combination. The two basic types of reactive passive silencers are

- Chamber-Type/Choke Tube (Low-Pass Filter), and
- Chamber-Type/Perforated Tubes.

CHAMBER SILENCERS (LOW-PASS FILTER)

A low-pass, filter-style silencer usually consists of two expansion chambers connected by a single choke tube. This type of device provides insertion loss (IL) by taking advantage of the reflection of noise at any junction of area change, where part of the noise continues down the pipe and part of it is reflected back toward the source.

The sizing of a classic low-pass filter–silencer is based upon the required cut-off frequency (f_c) for a given application using the equation,

$$f_c = \frac{c}{\pi}\sqrt{\frac{A}{LV}}$$

Where

c = *sonic velocity of the medium in feet per second,*
A = *area of choke tube in square feet,*
L = *length of tube in feet, and*
V = *total volume in cubic feet.*

Maximum performance occurs at the frequency where the length of the choke tube is equal to a quarter wavelength of the noise. Below this cut-off frequency, the low-pass filter is relatively ineffective. Larger unit diameters (added volume), as well as longer and smaller diameter choke tubes, can both be used to lower the cut-off frequency. Above the cut-off frequency, noise reduction can be very large, although limitations on size for shipping and pressure drop restrictions usually compromise the performance of the silencer.

Although in principle, the low-pass filter is also a pulsation dampener, its application is generally limited to the suction and/or discharge of low-speed reciprocating compressors. Even then, to ensure satisfactory performance, an acoustical analysis for the entire piping system is normally required to prevent possible troublesome noise pass-bands from occurring.

MULTIPLE-CHAMBER SILENCERS (WITH PERFORATED TUBES)

The multiple-chamber reactive silencer is primarily used for both low-frequency noise reduction and pulsation control. This type of silencer employs two or more volume chambers, usually connected in a labyrinth-like arrangement of perforated or slotted tubes. When used to silence engine exhaust noise, it is usually called a "muffler." In most other applications, the reactive-type silencer is commonly called a chamber-type silencer (or pulsation dampener).

UNIVERSAL SILENCER Chamber-Type Silencers (Figures 4-1 and 4-2) are essentially combinations of low-pass filters, but with two major deviations—that of the perforated tubes and the change in chamber volume ratios which was developed and patented in the early 1950s. Since then, there have been numerous modifications and changes in both concept and design. The perforated tube prevents system resonance and ensures predictable broadband silencer IL.

UC1	UCD	URB/URD	—
EN2	EN3	EN4	EN5
ET2	—	ET4	ET5

Table 4-1. UNIVERSAL SILENCER Multiple Chamber Silencer Series

Many physical arrangements are possible with this type of silencer. Double-ended tube lengths (preferably multiple tubes) extended into each chamber will lower the cut-off frequency and provide added low-frequency IL. However, for improved IL, there is no substitute for increased volume.

Individual chambers do not have to be located or positioned in the order shown, but the larger chamber is usually located next to the source. There is no precise method or equation for determining the design and size of chamber-type silencers. The design is based in large part on various empirical equations proven under actual operating conditions and extensive laboratory and field testing. The actual size is determined by the application requirements.

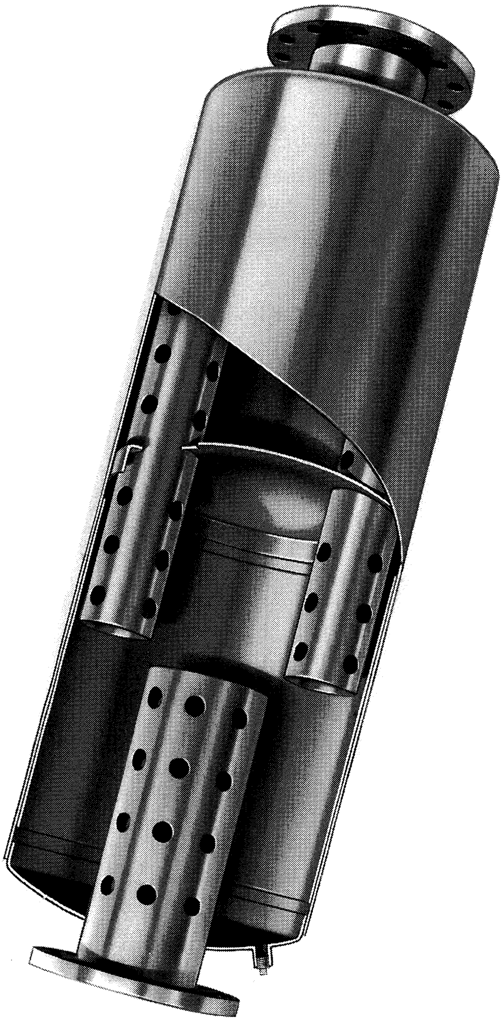

Figure 4-1. Basic UNIVERSAL SILENCER Chamber-Type Silencer (Standard Design)

ABSORPTIVE SILENCERS

The absorptive (dissipative)-type silencer is essentially a high-frequency attenuator that consists of various absorptive chambers and/or baffle arrangements. The acoustic fill is typically either fiberglass, polyester or ceramic fibers.

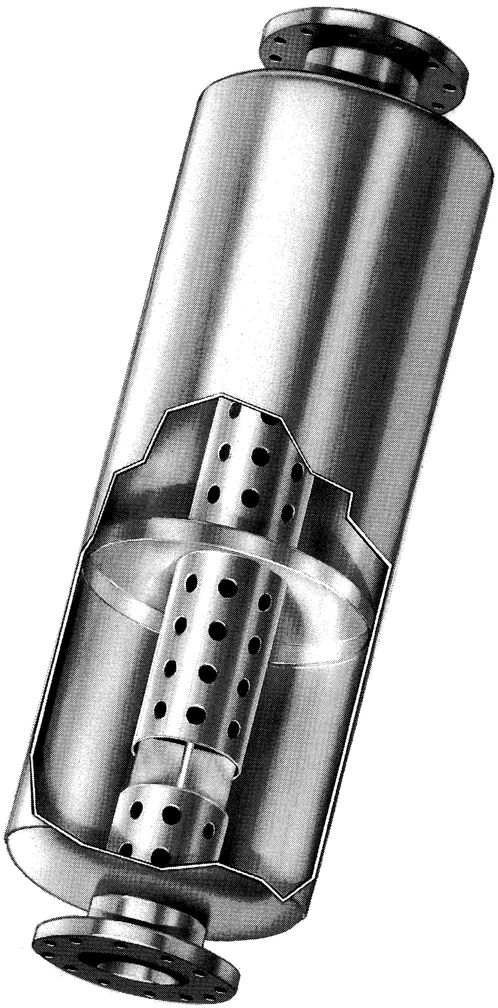

Figure 4-2. Basic UNIVERSAL SILENCER Chamber-Type Silencer (Straight-Through Low ΔP Design)

The absorptive silencer depends on sound absorption to dissipate the sound energy and may be either cylindrical or rectangular. Performance depends on design components such as pack thickness, air gap size and length and type of absorptive material used. Even the simplest cylindrical absorptive-type silencer can be considered a combination dissipative/volume resonator due to its geometry and construction, but it owes its main sound attenuation to the absorption properties of the pack material. All acoustic material that is used must be inert, durable, moisture-resistant, non-combustible and packed under not less than 10% compression to prevent voids and excessive settlement.

UNIVERSAL SILENCER Cylindrical Absorptive-Type Silencers Series U2, SU3, SU4 and SU5 are generally offered for reduction of high-frequency noise. There is a slight shift in performance toward the low-frequency end of the spectrum in the larger sizes, but in general the IL is quoted without regard to size.

UNIVERSAL SILENCER currently offers rectangular absorptive-type silencers for special applications such as the intake of forced draft fans and centrifugal compressors, the intake and exhaust of gas turbines and similar applications. A rectangular configuration is generally more effective than the cylindrical design where the piping or ducting permits. This is especially true in the larger sizes for air flows above 50,000 cfm.

ACTIVE NOISE CONTROL SILENCERS

The active cancellation of noise, or anti-noise, is not a new concept. The first patent for a "process of silencing sound oscillations" was granted to Paul Lueg in 1936. In this process, a microphone detects the undesired noise and provides an input signal to an electronic system that drives a loudspeaker. The output from the loudspeaker is adjusted so that the sound wave generated will destructively interfere with the incident noise, thereby canceling it. The entire process depends on the relatively slow propagation speed of the sound wave compared to the rapid processing of the electrical signal. Although many similar applications of this concept were suggested and studied over the years, it was not until twenty years later that electronics were sophisticated enough to produce an "electronic sound absorber." All of these early systems depended on analog electronic systems that were manually adjusted to obtain optimal results. In the late 1960s, work was published on active noise control systems that were automatically controlled to maintain optimal performance, but it was not until recently that both digital microprocessor systems and adaptive signal processing theory had advanced to the point where commercial applications were feasible.

It is unlikely that active noise control techniques will replace more conventional passive methods. Passive techniques are applied to multiple, incoherent sounds while active methods exploit interference between coherent sounds. The concept of energy addition of incoherent sources, where two sounds combine together to increase the noise as a whole, does not apply to active control where two precisely matched sounds cancel each other. Active noise control is most advantageous at low frequencies because it is not as dependent on large structures or large lengths to obtain sufficient noise reductions as are passive silencers. In addition, the dual problems of high sample rates in a digital system (i.e., twice the highest frequency of concern) and higher order modes of propagation are avoided by restricting the active control to lower frequencies. Attenuation of both low- and high-frequency sound will most likely be accomplished through use of hybrid or dynamic silencers that consist of an active low-frequency section and a passive high-frequency section.

The actual selection process for a dynamic silencer system is beyond the scope of this book, but several considerations in its application will be reviewed.

An effective active noise reduction system requires proper selection and application of the secondary loudspeaker source. The loudspeaker must be able to produce enough acoustical output to properly cancel the unwanted noise. Due to the high level of low-frequency noise produced by sources such as industrial fans and blowers, there is a need for very high power output speakers. This need is being addressed, but additional development is required.

Microphone placement is also a critical factor. Current active noise control silencers use two microphones; one is the input and one is the error microphone. Considerations include ensuring that the input microphone is not placed at a node of any standing wave and that it be placed far enough upstream from the secondary source to allow sufficient time for proper electronic processing of the input signal. For applications involving high-velocity gas flows, both turbulence fluctuations at the microphones and path variations due to turbulence may be serious problems. In addition, the error microphone must be carefully placed to avoid nodal locations before cancellation.

A final consideration is the operation of the system when the noise source is turned off. The active control system must be stable, and under low noise conditions, power itself down to avoid increasing the minimum sound levels.

Active noise control technology is finally beginning to achieve significant breakthroughs that will lead to practical applications. Dynamic silencing using a combination of active and passive silencing is an exciting development that will allow designers more flexibility in designing systems and optimizing operating parameters.

COMBINATION CHAMBER AND ABSORPTIVE SILENCERS

The combination reactive/absorptive silencer is used for broadband noise reduction with only a nominal increase in pressure drop. Combination silencers, such as UNIVERSAL SILENCER's series RIS, SD and RD, are similar to the chamber-type, except for the acoustic fill (pack material) that has been added in the first chamber (Figure 4-3). This chamber's performance is a result of both absorption and reflection of noise.

The more common acoustic fills used in both the absorptive and combination silencers are mineral wool (thermafiber), fiberglass and polyester. However, fiberglass cloth, fiberglass and mineral wool pack normally are not recommended for rotary-positive displacement blower service. Any service which produces high amplitude noise and pulsation tends to break down and pulverize the fibers, which may then cause a noise and/or a contamination problem. All pack material should be applied under at least 10% compression to prevent voids and excessive settling.

Occasionally a customer will specify a protective wrap of mylar or other plastic film. This substantially reduces the silencer IL, mostly in the upper frequency bands. The thickness of the film is generally designated by the customer. Where possible, its use should be discouraged altogether.

Figure 4-3. Basic UNIVERSAL SILENCER Combination-Type Silencer (Other Arrangements also Available)

When metallic pack is required, stainless steel mesh is recommended. In some instances, copper or stainless steel scrubble turnings may be used. Do not use stainless steel turnings except when added silencer TL is needed to prevent silencer shell ring.

Finally, a standard series design may not be the best design for a specific application, but in most instances, is offered for simplicity and cost effectiveness. When performance is critical, the application engineer should be contacted to ensure that the most effective and competitive design available is proposed.

ABSORPTIVE VENT AND BLOWDOWN SILENCERS

One special class of absorptive silencer is the cylindrical Vent and Blowdown Silencer. Silencers in the UNIVERSAL SILENCER HV series are of modular design, consisting of a capped inlet diffuser and plenum arrangement followed by a series of modular panels (absorptive-type) assemblies. The design of these silencers is dependent on the application requirements (Figure 4-4), and they are most often classified as a combination reactive/absorptive-type silencer.

Silencer velocities may run as high as 10,000 fpm or higher, depending on the application and service. Typical diffuser velocities range from 25,000 to 30,000 fpm, assuming atmospheric flow conditions. (Under actual conditions the velocity will be less.) The standard diffuser open area is 23%. The diffuser and inlet nozzle are both quality material with full penetration welding in accord with ASME Section IX welding procedures.

In applications with high upstream pressure, the maximum allowable working pressure at the inlet nozzle and across the diffuser must be calculated to assure a safe operation.

Pack material for oxygen (O_2) service must be free of oil and contaminants. All stainless steel construction is usually mandatory in O_2 service. However, where the application permits, long strand fiberglass which is certified at the factory as being oil-free and suitable for use in O_2 service is most often recommended. See Chapter 6G for more information on vent silencer applications.

Figure 4-4. Basic UNIVERSAL SILENCER Vent and Blowdown Silencer

ACOUSTIC PANELS, ENCLOSURES, PARTIAL BARRIERS AND PIPE LAGGING

Standard acoustical panels are perforated on one side and solid on the other and are used in almost all acoustical enclosures.

For best performance, the resonant frequency (f_o) of a panel arrangement should match that of the fundamental frequency of the source.

The approximate resonant frequency (f_o) can be calculated. First, the equivalent neck length (t') must be determined, and then used to calculate the f_o for a particular application. To find t', use

$$t' = \left(t + 0.8\sqrt{\frac{A_o}{n}}\right)$$

then calculate f_o using

$$f_o = \frac{c}{2\pi}\sqrt{\frac{A_o}{Vt'}}$$

Where

f_o = *resonant frequency in Hertz,*
V = *resonator volume in cubic feet,*
n = *number of holes,*
A_o = *total open area in square feet,*
t = *neck length in feet,*
t' = *equivalent neck length, and*
c = *speed of sound in feet per second.*

Splitter panels are perforated on both sides and are most often used in parallel baffle-type silencers.

Septum panels are essentially a splitter panel with a solid divider placed at its center to provide added TL and are generally used as an interior plenum or partition.

Hard panels are solid on both sides and are used for added TL, not absorption.

The perforated face sheet may be between 11% and 30% open, depending on the intended service. The standard is 23% open. A lower percentage of open area tends to increase the low-frequency performance, but with a corresponding decrease in performance in the higher frequencies. Thus, the more sparsely perforated the face sheet, the lower the resonant frequency. The more open the area, the better the high-frequency performance. The maximum open area that still has a performance benefit is 40% since the face sheet becomes acoustically transparent at this point.

Standard acoustical panels for commercial and industrial applications are usually constructed of

- 14 to 12 ga. CS or galvanized solid back and internal framing,
- 16 to 14 ga. CS or galvanized perforated face sheet (23% open), and
- 4 to 8 lb. density non-combustible acoustical fill.

Lighter or heavier panels may be required, depending on the overall size and rigidity required. The standard acoustical panel thickness is 2" to 8".

Most standard, commercially available acoustical panels are tested by independent laboratories when certified performance is required, with results similar to Table 4-2.

	Octave Band Center Frequency (Hz)							
	63	125	250	500	1k	2k	4k	8k
Absorption Coefficient (α)	—	0.5	1.01	1.11	1.06	1.02	0.95	—
Transmission Loss (dB)	18	22	29	40	50	55	57	58 >

Table 4-2. Typical Standard Panel (4" Thick)

ENCLOSURES

Commercially available acoustical enclosures are generally modular in concept and design and vary in size and arrangement, depending on specific needs.

Equipment enclosures are used to contain the noise and can range from a relatively small box placed around a blower or gear unit to a larger, more complex enclosure for a gas or steam turbine system. The equipment enclosure is usually shipped unassembled.

Personnel enclosures are used to protect workers from excessive noise. The smaller sizes may be shipped from the factory fully assembled.

Large enclosures may become increasingly complex in design and application and are not currently within the scope of this handbook.

BARRIERS

The modular acoustic panel itself is ideally suited for use in both indoor and outdoor barriers, partial enclosures, acoustical louvers and equipment plenums. It also provides a versatile and economical solution to many in-plant noise problems, when properly applied.

A sound barrier such as a wall, building, hill or some type of obstruction or solid structure, if large enough, will provide a significant amount of noise reduction within the "shadow" of the barrier itself.

The transmission loss (TL) of a barrier must exceed the noise reduction required of the barrier by at least 10 dB to be fully effective.

For a barrier to be effective, its lateral width must extend beyond the line-of-sight (between the source and barrier) by at least the height that the barrier extends above the line-of-sight. (See Figure 4-5 and Table 4-3.)

Barriers, unlike total enclosures, afford unrestricted accessibility to the equipment. Total enclosures are more effective for reducing noise, but forced ventilation is usually required when they are used.

An indoor barrier is usually less effective than an outdoor barrier, since noise will reverberate off the walls of a room.

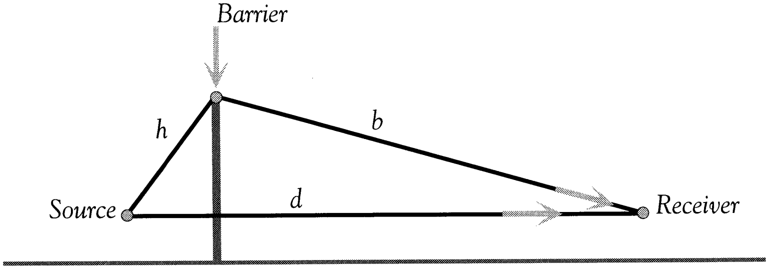

Figure 4-5. Barrier Wall Layout

Path Length Difference (ft) (h + b -d)	Octave Band Center Frequency (Hz)								
	31.5	63	125	250	500	1k	2k	4k	8k
0.01	5	5	5	5	5	6	7	8	9
0.02	5	5	5	5	5	6	8	9	10
0.05	5	5	5	5	6	7	9	10	12
0.10	5	5	5	6	7	9	11	13	16
0.20	5	5	6	8	9	11	13	16	19
0.50	6	7	9	10	12	15	18	20	22
1.00	7	8	10	12	14	17	20	22	23
2.00	8	10	12	14	17	20	22	23	24
5.00	10	12	14	17	20	22	23	24	24
10.00	12	15	17	20	22	23	24	24	24
20.00	15	18	20	22	23	24	24	24	24
50.00	18	20	23	24	24	24	24	24	24

Table 4-3. Approximate Noise Reduction (dB) Provided by a Solid Outdoor Barrier

A 2- or 3-sided barrier (with or without a top) will provide additional noise reduction opposite the closed walls of the barrier. When the barrier is located indoors, L_p at the receiver is equal to the source L_p plus the reverberant L_p. If the reverberant contribution is higher than the desired criteria, then the use of a partial barrier is not considered practical. Figure 4-6 shows a typical indoor barrier wall.

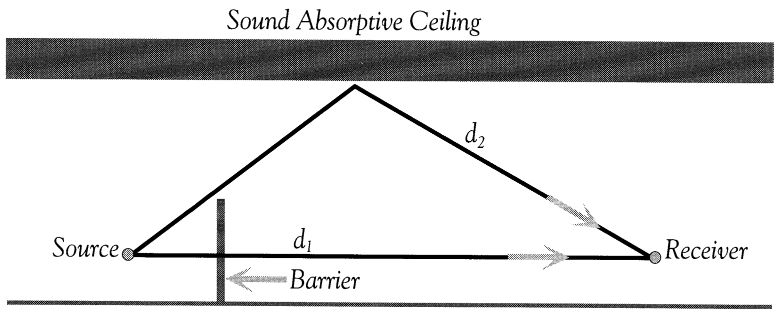

Figure 4-6. Barrier Wall with Absorptive Ceiling

The path length difference and the use of sound absorbent material on all adjacent reflecting surfaces provide a means of approximating the effectiveness of a barrier. Refer to Figure 4-6.

$$\text{Barrier } IL = L_p\,(\text{direct path}) - L_p\,(\text{ceiling path})$$

$$IL = 10 \log \left(\frac{d_2}{d_1}\right)^2 - 10 \log (1-\alpha)$$

Where

d_1 = *direct sound path in feet,*
d_2 = *ceiling reflected path in feet, and*
α = *sound absorption coefficient of the ceiling material.*

This simplified equation does not take into account the height of the barrier relative to the ceiling height. Tall barriers are obviously more effective due to the smaller opening at the top of the barrier.

Typical barrier noise reduction in a gas compressor station using two, 12'-high sound barriers to isolate a 4,000 hp gas engine from one which is down for overhaul is generally 10 to 13 dBA, depending upon the building size and acoustical characteristics of the building and barriers.

In many instances, barriers alone will reduce the noise to acceptable levels. In other instances, barriers provide only a partial solution.

Acoustic pipe lagging is another important form of noise reduction and is used primarily to prevent the radiation of pipe noise. When thermal lagging is provided, as it is in most exhaust systems, it may frequently reduce noise as well and no further treatment may be required.

Pipe lagging is normally done at the job site by the customer or an outside contractor, as opposed to the silencer manufacturer. On the other hand, external lagging of the silencer shell at the factory is very common.

The most common acoustic lagging material is fiberglass, which may be applied in various thicknesses and densities. An outer jacket is recommended for added TL and to protect the lagging material itself (Table 4-4).

The external jacket or cover is usually either

- 16 to 28 ga. galvanized carbon steel, or
- 16 to 20 ga. aluminum, or
- 16 to 28 ga. stainless steel.

The cover should be overlapped by 1" to 2" and then bonded on 12" to 18" centers. All irregular areas and shapes to be lagged are packed with loose material and then externally covered. Bands, screws and rivets are used as needed.

TL (dBA)	Effective Frequency Range (Hz)	Recommended Treatment
10 – 15	1,000 to 8,000	2" thick 4 lb density with external cover
10 – 15	250 to 2,000	4" thick 4 lb density with external cover
15 – 20	All Bands	Two 2" layers of 6 lb density with a lead-vinyl septum between the two layers and external cover

Table 4-4. Typical Acoustical Pipe Lagging

Resilient or spring-type pipe supports and equipment mounts, various coatings and spray compounds are also sometimes used for noise and vibration control.

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SILENCER PERFORMANCE

Overall silencer performance is usually determined on the basis of four criteria—three acoustic and one aerodynamic. They are

- Insertion Loss (IL),
- Shell Transmission Loss (TL),
- Self Noise (SN), and
- Pressure Drop (ΔP).

In most instances silencer performance is stated only in terms of insertion loss and pressure drop, but both self noise and transmission loss must be included in any acoustic analysis of a system.

ACOUSTIC

Insertion loss (IL), when measured under actual flow conditions, is often referred to as dynamic insertion loss (DIL) and is a function of the input noise amplitude and frequency, velocity, gas type, temperature and the area environment. While IL is relatively easy to measure under actual flow conditions, it is difficult to calculate or predict theoretically. Silencer manufacturers will give the IL of their equipment in either a catalog or technical data sheet. Determining the amount of IL in a given application is discussed later in this chapter and in Chapter 6 for specific applications.

Both self noise (SN) and shell transmission loss (TL) can limit the effectiveness of a silencer. Neither of these attributes are easy to calculate, so they should be determined by the manufacturer or a consultant. It is important that the design engineer be aware of them and consider them in an overall system design.

Shell TL is a function of the unit size and design. It must be compatible with the silencer IL so that the total radiated noise from the silencer itself will not become a problem. Standard UNIVERSAL SILENCER silencers are provided with double shells of varying thicknesses, depending upon actual IL requirements. When the IL of a silencer is above 40 dB, then a special design and/or external acoustical lagging may be required.

Self Noise (SN), sometimes called regenerated noise, is the noise created by the airflow within the silencer and is primarily a function of silencer exit velocity. Temperature and silencer size and design are other considerations. Most often excessive SN is caused by undersizing the silencer. SN should be at least 5 to 10 dB **less** than the required silenced level so as not to compromise the silencer's effectiveness.

PRESSURE DROP

When air (or gas) is moved through a pipe, a certain pressure (or head) is necessary to start and maintain the flow. The energy required is proportional to the total pressure. The total pressure consists of two components — static pressure and velocity pressure, sometimes called dynamic pressure.

Velocity pressure is the pressure required to move air through the pipe and represents the kinetic energy in the flow. If the pipe is closed and there is no flow, then only static pressure is present, since

---

Total pressure = static pressure + velocity pressure

Where

$\frac{1}{2}\rho V^2$ = *velocity pressure,*

ρ = *density, and*

V = *velocity.*

Static pressure, velocity pressure and total pressure are interrelated. When the velocity is decreased, such as where the area is increased in a duct, a portion of the velocity pressure is converted to static pressure. Conversely, when the velocity is increased, static pressure is partially converted to velocity pressure. These conversions are always accompanied by a certain amount of energy loss due to turbulence, shock waves and the like, depending on the change in area and shape. This loss is called pressure drop.

Silencer pressure drop (ΔP) is primarily a function of silencer design, velocity and gas density, but is usually represented by a constant times the velocity pressure, as shown in the equation below.

$$\Delta P = CH_V$$

Where

C = *loss coefficient, and*
H_V = *dynamic head or pressure.*

Adjusting for units the two equations below can be used to determine H_V or velocity.

$$H_V = \rho\left(\frac{V}{1097}\right)^2$$

and

$$V = 1097\sqrt{\left(\frac{H_V}{\rho}\right)}$$

For standard air (14.7 psia and 70°F) density of 0.075 lbs per cubic foot, this becomes

$$H_V = \left(\frac{V}{4005}\right)^2$$

and

$$V = 4005\sqrt{(H_V)}$$

The most widely used equation for calculating silencer pressure drop (ΔP) follows and can be used for various conditions and gases. Also given is the equation to determine ΔP for atmospheric air service by substituting for molecular weight, compressibility and standard conditions.

$$\Delta P = C\left(\frac{V}{4005}\right)^2\left(\frac{MW_a}{MW_s}\right)\left(\frac{P_a}{P_s}\right)\left(\frac{T_s}{T_a}\right)\frac{1}{Z}$$

ΔP *for Atmospheric Air Service*

$$\Delta P = C\left(\frac{V}{4005}\right)^2\left(\frac{530}{T_a}\right)$$

V *for Atmospheric Air Service*

$$V = 174\sqrt{\left(\frac{\Delta P T_a}{C}\right)}$$

ACFM for Atmospheric Air Service

$$ACFM = 174A\sqrt{\left(\frac{\Delta P T_a}{C}\right)}$$

The variables for these formulas are

ΔP = *silencer pressure drop in inches of* H_2O
ρ = *density of gas in pounds per cubic foot*
V = *silencer velocity in feet per minute*
H_V = *velocity pressure in inches of* H_2O
C = *silencer* ΔP *coefficient*
MW_a = *molecular weight of the gas*
MW_s = *molecular weight of standard air (28.97)*
P_a = *operating pressure psia*
P_s = *standard pressure (14.7 psia)*
T_a = *operating temperature, °R (460 + °F)*
T_s = *standard temperature, °F (460 + 70°F)*
A = *silencer flow area in square feet*
Z = *gas compressibility factor (where applicable) and*
$ACFM$ = *actual cubic feet per minute*

Silencer loss coefficients (C) are usually determined by laboratory or field tests and within a given silencer series do not vary with the size of the unit. A typical graphic solution for atmospheric air service is shown in Figure 5-1. A tabulation of the C values for the various standard UNIVERSAL SILENCER series is provided in Table 5-1.

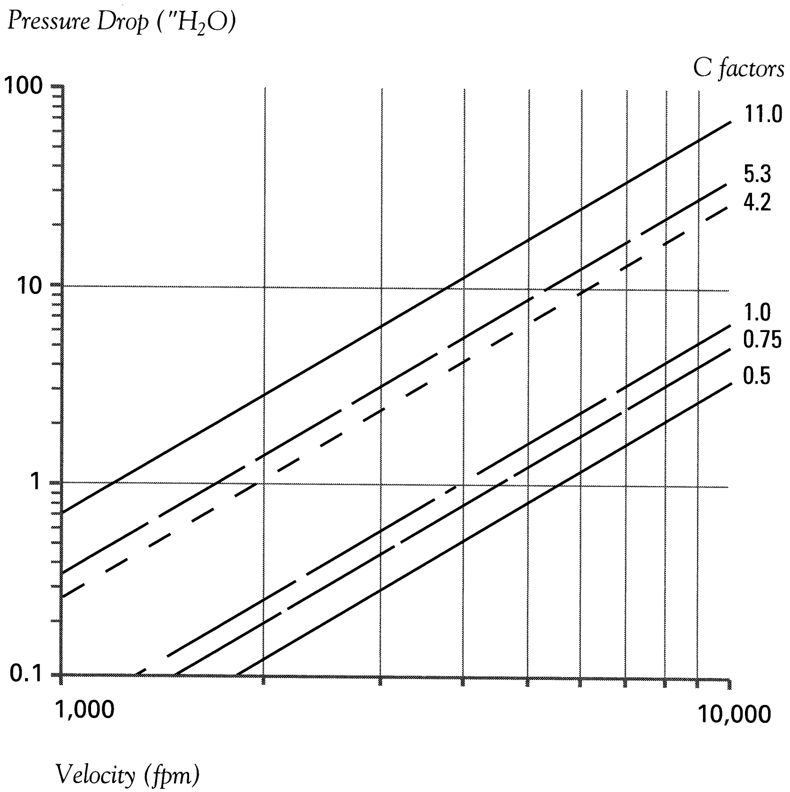

Figure 5-1. Silencer Pressure Drop for Various C Coefficients

In addition to the silencer pressure losses, the losses caused by other components must also be considered. Straight runs of ductwork or piping have losses due to friction and while not large, they should be considered if the length is very long or the diameter very small. An estimate of piping friction loss is given in Figure 5-2.

Silencer Model	C	Silencer Model	C
EN1, 2, 3, 4	4.00	URD	4.20
EN5 < 12	4.00	SD	4.20
EN5 > 10	5.30	SURS	5.30
ET2	0.50	RD	4.20
ET4, 5	1.00	—	—
ES2, 3, 4	4.20	GTE3	0.60
U2, 5	0.25	GTE4	0.75
SU3, 4	0.85	GTE6	0.90
SU5	0.75	HV5	11.25
URB	4.20	HV10	11.50
UCI	4.20	HV15	11.75
RIS	4.20	HV20	12.00
UCD	4.20	HV25	12.25

Table 5-1. Tabulation of UNIVERSAL SILENCER Pressure Drop Coefficients (C)

Smooth Pipe	Rough Pipe
$C = 0.0148\left(\frac{L}{D}\right)$	$C = 0.032\left(\frac{L}{D}\right)$

Where

L = *length of pipe* D = *diameter of pipe*

Figure 5-2. Piping Friction Losses

Turns and elbows are often expressed in an equivalent length of straight pipe. Because of their relatively high pressure loss, they should be avoided. The pressure loss of other types of components, such as entrances, exits, expansions, etc., can be determined using charts such as those shown in the Appendix or from ASHRAE.

ΔP must always be stated as either being a) across the silencer only, b) including entrance or exit losses or c) total system which includes all piping, elbows and fittings.

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6 SILENCER APPLICATIONS AND NOISE SOURCES

In this chapter we consider specific industrial noise sources and the application of silencers or mufflers to them. The following noise sources are considered:

- Internal combustion reciprocating engines
- Gas turbines
- Rotary positive displacement blowers
- Vacuum pumps, both dry and liquid seal
- Centrifugal blowers and compressors
- Reciprocating air compressors
- Vents and blowdowns to atmosphere
- Pressure regulators
- Centrifugal fans (industrial)

Each of these topics is covered along with specific guidelines and recommendations. Some of the topics covered overlap, but an effort has been made to make each discussion as complete as possible.

SECTION A
INTERNAL COMBUSTION RECIPROCATING ENGINES

All internal combustion reciprocating engines are noisy, some more so than others. Two-cycle and four-cycle engines of equivalent horsepower and speed produce about the same overall noise level due to their inherent combustion process and similar mechanical design and construction. However, there is a difference in the frequency distribution due to the higher firing frequency of the 2-cycle engine. Even so, the exhaust of 2-cycle engines (depending upon the scavenging means) is normally quieter than the 4-cycle engine.

It is difficult to predict the low-frequency noise output of large engines, due to operational variations and the effect of the intake and exhaust piping. The predominant frequency of a specific engine and that which is calculated may vary considerably. In actual practice, intake and exhaust silencers must be designed for broadband performance within the expected frequency range of operation. Tuned resonators and expansion chambers alone seldom provide acceptable performance.

BASIC UNSILENCED POWER LEVEL CALCULATION

In general, engine noise increases with horsepower. Added horsepower, by means of larger engines or by multi-engine installation of smaller engines, results in a logarithmic increase of 1½ to 3 dB when the horsepower is doubled. However, increased noise is only part of the problem. Noise levels that were acceptable a few years ago may no longer be tolerated.

The predominant sources of engine noise (listed in order of magnitude) are that of exhaust, intake and casing. The cooling fan may also contribute some noise.

1. Exhaust noise includes the various noise sources of the exhaust system (expansion joint, piping and the exhaust).
2. Intake noise includes all noise sources within the intake system (air filter, ducting or piping and the air intake itself).
3. Casing noise is the result of mechanical and structural propagation of radiated noise. Fuel contribution, engine

timing and the extent of component wear also contribute to the casing noise.

The unsilenced noise levels of specific engines should be obtained from the engine manufacturer or from actual tests (where possible), since no two engines produce identical noise levels. When such data are not available, the following empirical procedure for estimating unsilenced engine power levels may be used. The equations were derived from data published by the American Gas Association (1969), and are based on a study by Bolt, Beranek & Newman of 75 gas and diesel engines ranging from 10 to 6,000 hp.

Figure 6-1 shows the calculation method of the basic overall power level for the three noise sources outlined above as well as correction factors for tailpipe, fuel and speed. These levels are then distributed to octave bands using Table 6-1.

Exhaust: $$L_w = 10 \log hp + 117 \text{ dB}$$

Where

hp = *Engine horsepower*

Exhaust pipe length (L), ft	-L/4 dB
With turbocharger	-6 dB

Intake: $$L_w = 5 \log hp + 93 \text{ dB}$$

Intake pipe length (L), ft	-L/6 dB

Casing: $$L_w = 10 \log hp + 92 \text{ dB}$$

less than 600 rpm	-5 dB
600 – 1,500 rpm	-2 dB
greater than 1,500 rpm	0
Natural gas fuel	-3 dB
Liquid fuel	0

Figure 6-1. Basic Power Levels

Spectrum Corrections to be Subtracted from L_w Levels								
	Frequency							
Source	63	125	250	500	1k	2k	4k	8k
Exhaust	-8	-2	-6	-14	-18	-24	-34	-42
Intake	-10	-12	-12	-11	-8	-7	-8	-16
Casing	-11	-7	-7	-8	-8	-9	-15	-22

Table 6-1. Basic Spectrum Shape

VELOCITY AND PRESSURE DROP

Experience shows that optimum silencer performance is obtained within a relatively narrow range of velocities within the silencer. These values are summarized in Table 6-2 for intake systems and in Table 6-3 for exhaust systems. To determine silencer velocity, one must first know the engine displacement and flow rate. Figure 6-2 gives equations that determine the theoretical or displacement flow rate for 2- and 4-cycle engines. As can be seen from the equations, this consists of the volume of one cylinder times the number of cylinders times the engine speed. To know the silencer flow rate one must adjust this theoretical value by the volumetric efficiency as shown in Figure 6-3. This yields the actual flow rate in cubic feet per minute (CFM). Note that the difference between intake flow rate and exhaust flow rate is simply the ratio of absolute temperatures; normally the fuel flow is less than 1% of the exhaust flow and so can be ignored. If an exceedingly low Btu fuel is being used, then fuel flow may need to be considered in the exhaust flow. With this information, one can determine the required silencer velocity as shown in Figure 6-4, as well as the pressure drop as shown in Figure 6-5. If a maximum pressure drop requirement is given, then the second part of Figure 6-4 may be used to determine the maximum allowable velocity permitted.

SPECIAL CONSIDERATIONS

Special silencer designs with modified tubes and other internal changes are often required to meet operational and dimensional requirements. Engines (especially those over 1,000 hp) with low pressure drop (ΔP) and critical silencing requirements normally require velocities of no more than 4,000-5,000 fpm.

Where ΔP is not critical, silencer velocity is limited only by the silencing requirement and where silencer self noise (SN) is a major consideration. Reducing velocity (by increasing silencer size) provides added silencer performance (DIL) and reduced self noise (SN).

Engine Type	Intake Velocity (fpm)
4-cycle (4 or more cylinders) 2-cycle (2 or more cylinders)	4,000 – 6,000
4-cycle (2 to 3 cylinders) 2-cycle (1 cylinder)	2,000 – 3,000
4-cycle (1 cylinder)	1,000 – 1,500

Table 6-2. Average Intake Silencer Velocities

Engine	Speed (rpm)	Exhaust Velocity (fpm)
2-and 4-Cycle	< 350	4,000 – 7,000
	350 – 1,200	6,000 – 8,000
	1,200 >	8,000 – 10,000

Table 6-3. Average Exhaust Silencer Velocities

For 2-cycle engines

$$Displacement\ CFM = \frac{\pi B^2 \text{x}\ S \text{x}\ N \text{x}\ rpm}{4 \text{x}\ 1728}$$

For 4-cycle engines

$$Displacement\ CFM = \frac{\pi B^2 \text{x}\ S \text{x}\ N \text{x}\ rpm}{4 \text{x}\ 1728 \text{x}\ 2}$$

Where

CFM = *gas flow in cubic feet per minute,*
B = *cylinder bore in inches,*
S = *cylinder stroke in inches,*
N = *number of cylinders, and*
rpm = *revolutions per minute*

Figure 6-2. Engine Displacement Equation

$$Intake\ ACFM = Displacement\ CFM \times \frac{VE(\%)}{100}$$

Where

$ACFM$ = actual cubic feet per minute, and

VE = engine volumetric efficiency (%).

When VE is not known, use the following percentages for VE:

Natural aspirated	85%
Blower scavenged	120%
Turbocharged	145%

$$Exhaust\ Displacement\ CFM = Displacement\ CFM \times \frac{T_e}{T_i}$$

$$Exhaust\ ACFM = Intake\ ACFM \times \frac{T_e}{T_i}$$

Where

T_e = Exhaust temperature, °R(460 + °F), and

T_i = Intake temperature, °R(460 + °F).

Figure 6-3. Engine Flow Rates

$$Silencer\ Velocity = \frac{ACFM}{A}$$

Also,

$$V_{MAX} = 4005\sqrt{\frac{\Delta P}{\left(C\frac{530}{T_0}\right)}} = 174\sqrt{\frac{\Delta P(T_0)}{C}}$$

Where

A = silencer flow area in square feet,

ΔP = pressure drop ("H_2O),

C = silencer ΔP coefficient,

V_{MAX} = maximum silencer velocity in feet per minute, and

T_0 = Operating temperature, °R(450 + °F).

Figure 6-4. Silencer Velocities

$$\text{Silencer} \quad \Delta P = C\left(\frac{V}{4005}\right)^2\left(\frac{530}{T_0}\right)$$

Where

P = *pressure drop ("H_2O),*
C = *silencer ΔP coefficient,*
V = *silencer velocity in feet per minute, and*
T_0 = *operating temperature, °R (450 + °F).*

Figure 6-5. Silencer Pressure Drop (Atmospheric Service)

Large bore and stroke, low-speed engines (such as a 14" bore x 17" stroke at 257 rpm) often require a larger than normal silencer because of the adverse effects of pulsation produced in both the intake and exhaust piping. Conversely, in small, high-speed engine applications (such as a 3 3/4" bore x 5" stroke at 1,500 rpm), the silencer size may be reduced.

When an engine is installed inside a building, the exhaust must always be ducted to the outside. In addition, the intake of large engines is usually ducted from the outside.

Turbocharged engines may require acoustical treatment of both the turbocharger and the piping to reduce radiated and reverberant noise. In addition, the interior walls of the building may require some form of treatment.

An exhaust tailpipe, when properly applied, will provide added silencing in the lower frequencies. Acoustically, the tailpipe is considered open at both ends. Thus, the optimum tailpipe length is that which is equal to or less than the quarter wavelength, but no more than two-thirds of the half wavelength, based on the fundamental firing frequency of the engine. Theoretically, the quarter wavelength cannot support a standing wave. The optimum tailpipe length can be determined by using the equation below.

$$\textit{Optimum TP length} = \left(\frac{c}{4\,f_1} - \frac{D}{2}\right)$$

*or odd numbered multiples thereof, where f_1 and **D** are given in Figure 6-6.*

In order to avoid the half wavelength resonance, the maximum length for a particular system can be calculated using the equation given below.

$$\textit{Maximum TP Length} = \frac{2}{3}\left(\frac{c}{2f_1} - \frac{D}{2}\right)$$

or odd numbered multiples thereof.

$$f_1 = \frac{\textit{Number of Cycles} \times \textit{rpm}}{60} \qquad \textit{(2-cycle)}$$

$$f_1 = \frac{\textit{Number of Cycles} \times \textit{rpm}}{2 \times 60} \qquad \textit{(4-cycle)}$$

Where

TP = *tailpipe length in feet,*
c = *speed of sound in feet per second,*
D = *TP diameter in feet, and*
f_1 = *fundamental firing frequency of engine in cycles per second.*

Figure 6-6. Engine Firing Frequencies

There is little to be gained by beveling (angle cutting) the end of the tailpipe. Actually, it is more expensive than a square cut pipe and is more vulnerable to vibration and stress.

Often it is necessary to qualify the basic silencer recommendation to insure that the performance as quoted both is understood by and is acceptable to the customer. In some instances, it may be appropriate to point out the considerations listed below.

1. Thermal and acoustic lagging of the silencer and piping may be needed to insure that radiated noise will not become a problem.
2. Piping between the engine and silencer should be of adequate size and minimum length, otherwise, a transition and/or enlarged silencer inlet may be needed.
3. If the piping and installation arrangement is such that the silencer inlet tube is eliminated, overall silencer performance may be reduced by 3 to 6 dB or more.
4. When an analysis and recommendation is based on estimated noise levels, confirmation of the levels is required before a system can be built.

Application Data

Engine intake and exhaust silencers required

SERVICE: *Continuous (day and night)*

SILENCER CRITERIA: *Max. allowable L_p is NR 50 at 300 ft*

ENGINE DATA: *2,500 hp, 2-cycle turbocharged, 6 cylinder, 300 rpm*

INTAKE	EXHAUST
20" Diameter Connection	24" Diameter Connection
9,500 cfm at 14.7 psia and 60°F $\Delta P = 3.0$ "H_2O max. (silencer only)	22,105 cfm at 14.7 psia and 750°F $\Delta P = 3.0$ "H_2O max. (silencer only)

Silencer Recommendation:

FSH-20-8 (filter-type) – $V = \frac{9{,}500}{2.2} = 4{,}318 \text{ fpm}$	ET4-24 (reactive-type) – $V = \frac{22{,}105}{3.1} = 7{,}130 \text{ fpm}$
Silencer $\Delta P = 2.7$ "H_2O (re: Bulletin 241)	Silencer $\Delta P =$ $1.00\left(\frac{7{,}139}{4{,}005}\right)^2\left(\frac{530}{1{,}210}\right) = 1.39$ "H_2O (re: Bulletin 246)

Figure 6-7. Example Engine Application

			Acoustic Analysis Octave Band Center Frequency (Hz)							
Line	**Description**	**Reference**	**63**	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
1	Unsilenced Intake L_p at 300 ft	Engine Manufacturer	72	68	60	53	56	55	52	45
2	Filter Silencer DIL~dB	FSH – 20 – 8	-2	-3	-4	-5	-8	-13	-14	-13
3	Silenced Intake L_p at 300 ft	—	70	65	56	48	48	42	38	32
4	Unsilenced Exhaust L_p at 300 ft	Engine Manufacturer	82	80	72	63	63	60	55	50
5	Silencer DIL~dB	ET4 – 24	-26	-33	-30	-24	-19	-16	-16	-17
6	Silenced Exhaust L_p at 300 ft	—	56	47	42	39	44	44	39	33
7	Total Combined Silenced L_p at 300 ft	Line 3 Combined w/Line 6	70	65	56	48	50	46	42	36
8	Allowable L_p Max	NR 50	75	65	56	48	50	47	45	43

Figure 6-8. Acoustic Analysis for Example

N O T E S

SECTION B
GAS TURBINES

Industrial gas turbines are used to power such things as pumps, compressors and generators, and generally range in size from approximately 900 hp to more than 200,000 hp. Gas turbines are also used to generate electric power.

Gas turbines require exhaust silencers for all simple cycle and bypass operations where the turbine exhausts directly to the atmosphere. Silencers are also needed when there are stringent noise requirements on combined cycle or heat recovery applications. Generally, inlet silencers are required for all turbine systems. The total system may be located outdoors or inside a building and may, therefore, be either partially or totally enclosed.

Turbine inlet noise is predominantly a high-frequency "whine" corresponding to the blade passing frequency of the turbine's air inlet and in most cases is dominant in the near field.

The turbine inlet may be directly open to the atmosphere, but generally it has an air filter and an absorptive-type silencer. Most inlet silencers require a transition and/or plenum for unrestricted air flow and proper fit-up. The inlet may be super-charged by large forced draft fans which also produce noise. Provision for anti-icing and/or evaporative cooling is optional.

Turbine exhaust noise is predominantly low-frequency resulting from the mass flow and the high temperature combustion process. The exhaust spectrum also contains discrete frequency noise corresponding to the blade passing frequencies of the various stages of the power turbine.

The turbine exhaust may be directed to atmosphere or directed through either a heat exchanger or extended ducting to atmosphere. The exhaust is generally provided with an absorptive-type silencer. Most exhaust silencers require a transition and/or plenum and provision must be made for thermal expansion.

Pumps, compressors, generators and the like that are powered by a gas turbine are not normally a major noise source. However, when the turbine itself is enclosed, extension of the housing to include this equipment should be considered. In such a situation, cooling air must be provided.

Estimated unsilenced sound power levels (L_W) of a 15 MW gas turbine are show in Figure 6-9.

Gas turbine manufacturers provide L_W for the inlet, exhaust and casing of their units. These data should be used where possible. UNIVERSAL SILENCER has an extensive library of these data that cover most turbines and operating conditions.

The normal operating life of inlet and exhaust silencers for large gas turbines is between 24,000 and 160,000 hrs depending upon the design, installation and materials of construction. The goal of any gas turbine silencing system is to reduce the distance needed to lower the sound energy prior to its reaching a noise sensitive area without measurable loss of turbine efficiency.

The three major aspects of silencer design to be considered in all gas turbine silencing systems are

- Aerodynamics — Air flow distribution and pressure drop (ΔP),
- Acoustics — DIL, TL and SN, and
- Structural — Including corrosion, thermal expansion, seismic and wind load requirements.

The silencing system must be properly designed and installed so as not to restrict the air inlet and exhaust flow of the gas turbine. Typical components and allowable ΔP limitations are shown in Table 6-4.

Turbine System Pressure Drop

Inlet System (2 - 5 "H_2O)	Exhaust System (3 - 5 "H_2O)
• Air filter and inlet loss	• Expansion joint
• Silencer	• Plenum
• Transition	• Transition
• Plenum	• Silencer
	• Stack and exit loss to atmosphere

Table 6-4. Typical Component Pressure Drop (ΔP) Limitations

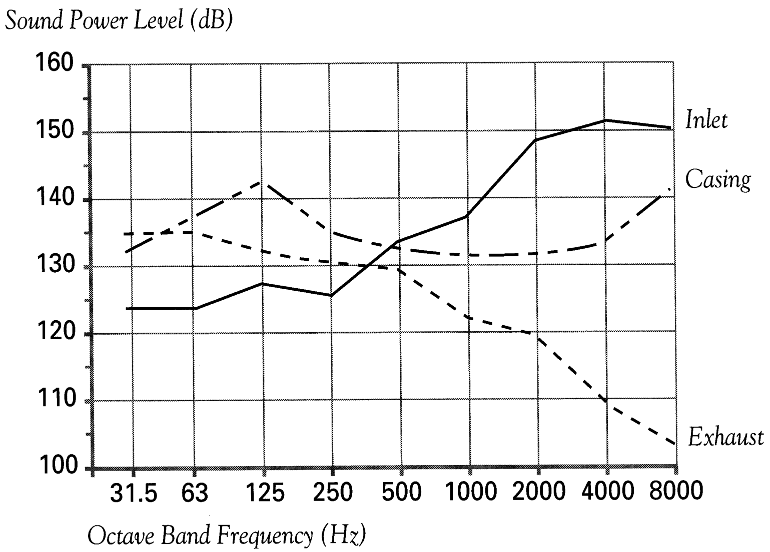

Figure 6-9. Estimated Unsilenced Octave Band Sound Power Levels (L_W) of a 15 MW (20,000 hp) Gas Turbine

Performance of absorptive-type silencers varies primarily with the depth, density and type of sound absorptive material used and the open area of the perforated face sheets.

The inlet silencer is basically designed to reduce the predominantly high-frequency inlet noise. The best overall performance is achieved by using relatively thin, straight-through, parallel baffles that are either cylindrical or rectangular.

The exhaust silencer is designed to reduce the predominantly low- to mid-range frequency spectrum of the exhaust, and this requires thicker baffles. Occasionally a reactive section must be used to effectively silence the low frequencies, but usually at a higher pressure loss. Specific details regarding the design and construction of these silencers are available on request.

Currently all rectangular-type silencers for gas turbine service are custom designed. Octave band IL obtained purely by acoustical absorption may be calculated or determined by laboratory or field tests, and is measurably affected by both velocity and temperature.

The primary mass flow of gas turbines is usually stated in lbs/sec, SCFM (standard cu ft/min at 14.7 psia and 70°F) or ACFM (actual cu ft/min). The relationship between SCFM and ACFM is expressed in the equations given below.

$$\text{Inlet ACFM} = 22.2(Q)\left(\frac{T_e}{P_a}\right)$$

or

$$\text{Inlet ACFM} = \frac{SCFM}{36.054}\left(\frac{T_e}{P_a}\right)$$

The relationship between exhaust and inlet **ACFM** *is given by the equation below.*

$$\text{Exhaust ACFM} = \text{Inlet ACFM}\left(\frac{T_e}{530}\right)$$

Where

T_e = *operating temperature, °R(460 + °F),*
P_a = *operating pressure, psia,*
Q = *mass flowrate, in pounds per second,*
$SCFM$ = *standard cubic feet per minute (14.7 psia, 70°F), and*
$ACFM$ = *actual cubic feet per minute.*

The ACFM is needed to determine the required silencer velocity, as shown below.

$$\text{Silencer Velocity (fpm)} = \frac{ACFM}{A}$$

Where

A = *silencer open area in square feet.*

Knowing both the allowable pressure drop (ΔP) across the silencer and the gas temperature, the following equation may be used to approximate the maximum velocity that can be used for silencer sizing. However, the self-generated noise of the silencer itself may require that the velocity be reduced.

$$\textit{Maximum Velocity (fpm)} = 4005\sqrt{\frac{\Delta P}{C\left(\frac{530}{T_e}\right)}}$$

Where

ΔP = *allowable drop in inches of* H_2O, *and*

C = *silencer pressure drop coefficient.*

In most atmospheric gas turbine applications, the basic silencer pressure drop equation is reduced to

$$\Delta P(''H_2O) = C\left(\frac{V}{4005}\right)^2\left(\frac{530}{T_e}\right)$$

In summary, silencer sizing is based upon velocity, allowable pressure drop, silencing criteria, size of the turbine connection or size of the transition connection, and any dimensional, space and weight limitations.

UNIVERSAL™

N O T E S

SECTION C

ROTARY POSITIVE DISPLACEMENT BLOWERS

The three basic types of rotary positive displacement (RPD) blowers are:

- Lobe (roots-type)
- Helical (screw)
- Vane (rotary)

The most common lobe blower is the figure-eight, 2-lobe impeller-type that produces 4 compression cycles per revolution. A newer, 3-lobe design produces 6 compression cycles per revolution. The lobe-type RPD blower includes the various models manufactured by Roots, Cooper, M-D Pneumatics and others.

The helical (screw)-type blower consists of 2 screw-like rotors that operate at a much higher rpm, again producing 4 compression cycles per revolution.

Helical (screw)-type blowers may be further divided into the two following categories:

- Spiraxial (and low-speed axial types) manufactured by Roots, Ingersoll-Rand, Gardner-Denver and others, and
- High speed axial types made by Ingersoll-Rand, AC Compressor and others.

The vane-type blower consists of a rotor and a series of vanes for compression. The compression cycle (number of impulses per revolution) is a function of the number of vanes. Various models of rotary vane-type blowers are manufactured by AC Compressor and Spencer.

Standard UNIVERSAL SILENCER silencers are recommended for most types of RPD blowers to 15 psig (Bulletins 244 and 245) with the exception of the high-speed axial-type. Silencers with acoustic fill frequently cannot be used on either the inlet or discharge of these blowers since the pack material tends to fatigue (break up) and then is carried into the gas stream. In these cases, a chamber-type silencer with external lagging is required. An ASME code design is recommended for the inlet and generally mandatory on the discharge of high-speed axial blowers due to the pressure requirements.

BASIC UNSILENCED POWER LEVELS

RPD blower noise is a function of

- Blower type and size,
- Speed (rpm) and timing gear diameter (TGD),
- Operating pressure and temperature, and
- Type of gas being compressed.

Unsilenced blower noise may exceed 135 dBA and because designs vary, it is wise to obtain this information from the blower manufacturer. Typically, the inlet noise is not as high as that produced by the discharge. However, when open to atmosphere, inlet noise may become predominant since much of the discharge noise is contained within the piping system itself.

Unsilenced blower operation may result in

- Excessive airborne radiated noise, and
- Troublesome pulsation-induced vibration.

Excessive noise may limit personnel exposure time under current OSHA regulations or in extreme situations cause the units to be shut down altogether. Pulsation-induced piping vibration may result in costly repair and loss of production.

The major components of RPD blower noise are

- Inlet (when open to atmosphere),
- Discharge piping,
- Blower casing (housing), and
- Drive unit.

Silencers for small blowers that do not produce high-amplitude, low-frequency noise and pulsation are sized on the basis of velocity up to 5,500 fpm and allowable ΔP, typically 5" to 12 "H_2O.

Silencers for large blowers that inherently produce high-amplitude, low-frequency noise and pulsations are sized on the basis of velocity, ΔP and in critical applications, the blower silencer volume ratios. For pressures above 15 psig, ASME code-type silencers are normally sized to match the blower or compressor connection size.

Blower capacity (volume) is usually expressed in CFM at inlet pressure and temperature conditions using the equation below.

$$ACFM = SCFM\left(\frac{T_a}{530}\right) = \frac{Q(v_q)}{60}$$

Where

$SCFM$ = *standard cubic feet per minute at 14.7 psia and 70°F,*
T_a = *operating temperature, °R (460+70°F),*
Q = *flow by weight in pounds per hour, and*
v_q = *specific volume of gas in cubic feet per pound.*

Silencer sizing, based on air blower capacities, pressure and ΔP, is provided in Table 6-5 and Figure 5-1. Velocity may be obtained by dividing the ACFM by the area of the silencer, and the pressure drop is given for the proper C factor.

Silencer Size	Capacity (Inlet CFM 14.7 psia & 70°F)						
		Discharge					
	Inlet	2 psig	4 psig	6 psig	8 psig	10 psig	15 psig
1	30	33	35	40	40	40	45
$1\frac{1}{2}$	70	70	80	85	90	95	105
2	120	130	140	150	160	165	185
$2\frac{1}{2}$	190	205	220	235	245	255	285
3	270	295	320	335	355	370	415
$3\frac{1}{2}$	370	405	430	455	480	505	560
4	480	525	560	600	630	660	735
5	750	820	880	935	985	1,030	1,150
6	1,080	1,180	1,260	1,340	1,410	1,480	1,650
8	1,920	2,100	2,250	2,390	2,510	2,630	2,940
10	3,000	3,310	3,520	3,730	3,930	4,110	4,590
12	4,320	4,760	5,070	5,370	5,660	5,920	6,600
14	5,880	6,450	6,890	7,310	7,700	8,060	8,990
16	7,680	8,450	9,000	9,550	10,000	10,500	11,800
18	9,720	10,800	11,400	12,100	12,700	13,300	14,900
20	12,000	13,200	14,000	14,900	15,700	16,400	18,400
22	14,500	15,700	17,000	18,100	19,000	19,900	22,200
24	17,300	18,700	20,200	21,500	22,600	23,700	26,400
26	20,300	22,300	23,800	25,200	26,600	27,800	31,000
28	23,500	25,900	27,600	29,300	30,800	32,200	36,000
30	27,000	29,500	31,700	33,600	35,400	37,000	41,300
Est. Temp.	70°F	90°F	115°F	140°F	165°F	190°F	240°F
ΔP ("H_2O)	7.9	8.7	9.3	9.9	10.4	10.8	12.1

Table 6-5. Silencer Capacity Based on Standard Air Conditions and Silencer Velocity of 5,500 fpm (Silencer C Factor of 4.2)

SPECIAL CONSIDERATIONS

The amplitude and spectrum shape of blower noise is a function of the blower type, size, rpm and pressure. Increased rpm in most instances becomes the major predictor of blower noise. RPD blower noise inherently reaches "problem" levels when the peripheral velocity of the timing gear exceeds the transition speed.

Once the blower transition speed (that is, the rpm at which the higher frequencies become predominant) is reached, a combination reactive/dissipative-type silencer is needed (Tables 6-6 and 6-7). The blower transition speed depends on the critical pitch-line velocity (PLV) which has been found by experiment and is expressed in the equation below.

$$\text{Blower Transition Speed (rpm)} = \frac{\text{Critical PLV}}{0.262\,(\text{TGD})}$$

Where

$$\text{PLV (fpm)} = \frac{\text{TGD (rpm)}}{3.820}$$

and

$$\text{TGD} = \text{Timing Gear Diameter in inches}$$

The critical PLV (2-lobe and other RPD blowers which produce 4 compression cycles per revolution) based on the TGD is typically

- Inlet—3,300 fpm
- Discharge—2,700 fpm

At timing gear pitch-line velocities above those shown, the noise energy produced by these blowers and the pulsation energy developed by the larger blowers becomes severe enough that the silencer requirements become more rigid than those for lower pitch-line velocities.

Operation at or above critical PLV can cause "pipe ring" in the downstream piping and "shell ring" in the standard silencer. For this reason, it is necessary that the absorptive "pack section" of the silencer inlet be installed at the blower discharge.

Blower Gear Diameter (in)	Inlet Silencer			Discharge Silencer		
	Trans. Speed (rpm)	Below Trans.	Above Trans.	Trans. Speed (rpm)	Below Trans.	Above Trans.
2	6,300	U5	U5	5,155	URB, UCD, URD	SD, RD
$2^1/_2$	5,040	U5	U5	4,125	URB, UCD, URD	SD, RD
3	4,200	U5	U5	3,435	URB, UCD, URD	SD, RD
$3^1/_2$	3,600	U5	U5	2,945	URB, UCD, URD	SD, RD
4	3,150	U5	U5	2,575	URB, UCD, URD	SD, RD
5	2,520	U5	U5	2,060	URB, UCD, URD	SD, RD
6	2,100	U5	U5	1,720	URB, UCD, URD	SD, RD
7	1,800	UCI, SU5, URB	RIS	1,470	URB, UCD, URD	SD, RD
8	1,575	UCI, SU5, URB	RIS	1,290	URB, UCD, URD	SD, RD
10	1,260	UCI, SU5, URB	RIS	1,030	URB, UCD, URD	SD, RD
12	1,050	UCI, SU5, URB	RIS	860	URB, UCD, URD	SD, RD
14	900	UCI, SU5, URB	RIS	735	URB, UCD, URD	SD, RD
16	785	UCI, SU5, URB	RIS	645	URB, UCD, URD	SD, RD
18	700	UCI, SU5, URB	RIS	570	URB, UCD, URD	SD, RD
20	630	UCI, SU5, URB	RIS	515	URB, UCD, URD	SD, RD
22	570	UCI, SU5, URB	RIS	470	URB, UCD, URD	SD, RD
24	525	UCI, SU5, URB	RIS	430	URB, UCD, URD	SD, RD
30	420	UCI, SU5, URB	RIS	345	URB, UCD, URD	SD, RD
36	350	UCI, SU5, URB	RIS	286	URB, UCD, URD	SD, RD

Table 6-6. Blower Transition Speed vs. Silencer Model Recommendations (Atmospheric or Vacuum Inlet and Pressure Discharge)

Transition speeds shown in Table 6-6 are for 2-lobe RPD blowers. For 3-lobe blowers, multiply the rpm shown by 0.67.

Blower Type (Lobe, Helical and Vane)	Inlet	Discharge
Below Transition	U5, SU5, UCI, URB	URB, UCD, URD
Above Transition	U5, RIS	SD, RD

Table 6-7. Silencer Model Recommendations

Silencers for the inlet and discharge of RPD blowers are typically classified by blower type and applied as shown in Figure 6-10.

Multi-Chamber (Reactive-Type)

- Lobe-type (operating < critical PLV)
- Helical (axial-type) requiring external lagging
- Vane-type (operating < critical PLV)

Absorptive (Dissipative-Type)

- Lobe-type (small blowers only)
- Helical (screw-type)
- Vane-type (small blowers only)

Combination (Reactive/Dissipative-Type)

- Lobe-type (operating > critical PLV)
- Helical (screw type)
- Vane-type (operating > critical PLV)

Figure 6-10. Application of Silencers to RPD Blowers

To summarize Figure 6-10, lobe-type RPD blowers require reactive (chamber-type) silencers to effectively reduce both low-frequency noise and pulsation. Operation "above transition speed" normally requires reactive/dissipative (absorptive-type) silencers to prevent added shell and piping noise.

The exception to the above is the absorptive-type silencer which may be offered for the atmospheric inlet and discharge of small blowers (regardless of blower rpm) and where low-frequency silencer DIL is not of major concern.

Silencers for helical (screw)-type blowers, which typically operate at higher speeds, are essentially the same as those offered for high-speed, lobe-type blowers.

Silencers for the rotary vane-type blower are the same as that offered for the lobe-type, except that at "above transition speeds" the carry-over of lubrication oil may prohibit the use of any form of acoustic fill in the discharge silencer. In such cases, a reactive (chamber-type) silencer with external acoustic lagging is normally applied. Special packless designed silencers are available for the higher speeds and pressures of axial-type blowers (compressors).

Hazardous gas services require that the silencer be of ASME Code Design and Construction (Section VIII).

Blower noise inside a room or enclosure is essentially that radiated from the blower casing, expansion joints and piping. Near the blower, casing noise usually predominates when the inlet is a closed system or the atmospheric inlet is located outdoors.

Since multiple sources are involved, silencer shell TL should be at least 6 to 10 dB more than the silencer DIL requirements.

The blower casing and discharge expansion joints frequently require some form of acoustical treatment (lagging or insulation) to protect workers from excessive radiated noise. When discharge pressure exceeds 8 psig, acoustical lagging of the discharge silencer is most often required to meet the 90 dBA criteria, especially in the larger pipe sizes.

Inlet (Atmospheric)	**Silencer Model**
• Below transition speed	UCI
• Above transition speed	RIS
90 dBA max. 3' from silencer inside of building 90 dBA max. 10' from atmospheric inlet outside of building 5% max. peak to peak pulsation at silencer inlet nozzle	
Discharge (15 psig max.)	**Silencer Model**
• Below transition speed	URD
• Above transition speed	RD
90 dBA max. 3' from discharge silencer 3% max. peak to peak pulsation at silencer outlet nozzle (based on a 3-chamber design)	

Table 6-8. Typical 90 dBA RPD Blower Silencer Recommendations

Large, high-speed RPD blowers may also require partial or total isolation (containment) of both the blower casing and the drive unit noise. Isolation is generally achieved by partially, or totally enclosing the blowers. When totally enclosed, the blower drive unit may require forced ventilation.

Experience shows that lobe-type blowers with TGD of 18" and larger, operating above critical PLV, may develop severe noise and pulsation induced vibration. In addition, when two or more blowers discharge into a common header, individual silencers upstream of the header are required to subdue the blower pulsation and to prevent unfavorable resonance from developing in the laterals and header. The intake and discharge silencer should be installed as close to the blower as possible to avoid piping resonance and radiated pipe noise.

Atmospheric inlet piping is especially susceptible to adverse lengths that can cause resonance and increase pulsation, vibration and noise. Adverse pipe lengths (lengths to avoid) are directly related to the actual piping arrangement.

Where the piping is considered **closed at the blower and open at the filter or silencer,** it is necessary to avoid the quarter wavelength ($\lambda/4$), defined as

$$\lambda/4 = \frac{c}{4f} = L$$

or odd multiples thereof.

When the piping is considered **closed at the blower and closed at filter or silencer** (as may be the case where small silencers may reflect a high percentage of the noise back to the source), it is necessary to avoid the half wavelength ($\lambda/2$), defined as

$$\lambda/2 = \frac{c}{2f} = L$$

or multiples thereof.

Where

λ = *wavelength in feet,*
c = *speed of sound in feet per second* = $49.03\sqrt{(T)}$ *for air,*
f = *blower fundamental in Hertz,*
T = *operation temperature,°R (460 + °F),*
L = *length of pipe in feet,*
and where the blower fundamental is

$$f = \frac{\text{RPM x Number of Impulses per Revolution}}{60}$$

Straight runs of piping should be used where possible, avoiding excessive use of elbows, reducers and other restrictive in-line components. When the predominant frequency or frequencies of the blower coincide with the natural frequency of the piping system, adverse noise and vibration conditions can develop.

Inlet air for atmospheric air blowers should be drawn from outdoors. Air taken from within the blower room itself will invariably create a noise problem. Most atmospheric air blowers require inlet filters with weatherhoods for installation outdoors (UNIVERSAL SILENCER Bulletins 241 and 242).

In conclusion, it must be remembered that multiple sources of noise, including background noise, combine to produce composite levels higher than that from any one individual source.

For example, the major components of blower noise are the inlet, discharge, casing, drive unit and piping. Therefore, installation of inlet and discharge silencers alone may not reduce the overall area noise to acceptable levels. *It follows then that a silencer performance (IL) guarantee is not the same as a system guarantee.*

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SECTION D
VACUUM PUMPS

The two most common positive-displacement vacuum pumps are

- Rotary positive-type (lobe, helical and axial, and rotary vane), and
- Reciprocating-type.

Unlike other applications, vacuum pumps often require a liquid separator on the intake and/or the discharge. We will consider the unique aspects of this application since base unsilenced levels are similar to other RPD systems.

ROTARY POSITIVE DISPLACEMENT (RPD)-TYPE

Water-Sealed

UNIVERSAL SILENCER separator silencers are used both to remove liquid from the air flow of water-sealed RPD vacuum pumps and to reduce the discharge noise to atmosphere to acceptable levels.

The inlet separator removes process water from the inlet system before it enters the pump. The discharge separator removes any carry-over from the inlet that remains, and more importantly, it removes the seal water being discharged to the atmosphere.

Two separate phases of the pump operation must be considered:

- Start-up, and
- Normal operations at vacuum conditions.

Both liquid removal and pressure drop (ΔP) must be considered during each phase of the operation to insure the proper selection and sizing of the separators (Table 6-9).

Vacuum measurements alone may not be sufficient to establish the system operating pressure. A simultaneous barometric reference is often required, as shown in Figure 6-11.

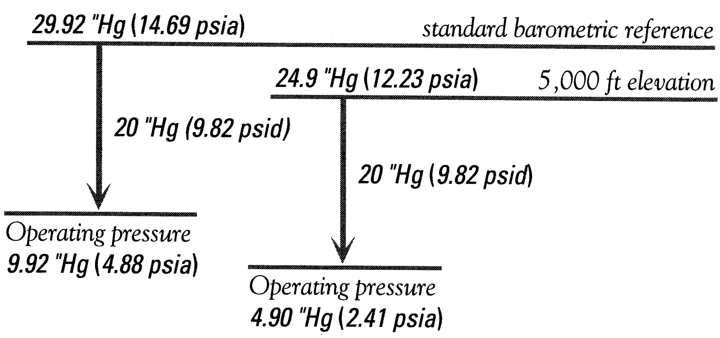
Model	C Factor	Efficiency (%)
Inlet (Vacuum Separator)		
VI	2.5	95
UV-S	4.0	99+
Discharge (Separator-Silencer)		
UDY/USVY	4.2	95
UCS	4.0	99+
USS	5.5	99+
URS	5.5	99+

Table 6-9. UNIVERSAL SILENCER Separator-Silencer Reference Guide (RPD Water-Sealed Blowers)

Maximum ΔP (and maximum power consumption) occurs at start-up. The inlet pressure decreases as the pump continues to evacuate the system. The system ΔP and power requirements also decrease.

The inlet air flow (ACFM) remains essentially constant throughout the operation. However, due to the decreasing inlet pressure, the discharge flow decreases until normal operation is reached.

Maximum capacity (inlet ACFM) for various operating conditions is given in Table 6-10.



Reference: 14.69 psia	=	29.92 "Hg	=	406.9 "H_2O
1 psi	=	2.04 "Hg	=	27.68 "H_2O
1 "Hg	=	0.49 psi	=	13.6 "H_2O

Figure 6-11. Effect of Barometric Pressure

Nominal Size	Operating Vacuum "Hg						
	0*	5	10	15	18	20	25**
1	30	36	45	60	75	90	98
$1^1/_2$	70	81	101	135	169	204	221
2	120	144	180	241	301	362	393
$2^1/_2$	190	225	282	376	471	565	614
3	270	324	406	541	678	814	884
$3^1/_2$	370	441	552	737	922	1,110	1,200
4	480	576	721	963	1,200	1,450	1,570
5	750	900	1,130	1,500	1,880	2,260	2,450
6	1,080	1,300	1,620	2,170	2,710	3,260	3,530
8	1,920	2,310	2,880	3,850	4,820	5,790	6,280
10	3,000	3,600	4,510	6,020	7,530	9,050	9,800
12	4,300	5,190	6,490	8,660	10,800	13,000	14,100
14	5,900	7,060	8,830	11,800	14,800	17,700	19,200
16	7,700	9,220	11,500	15,400	19,300	23,200	25,100
18	9,700	11,670	14,600	19,500	24,400	29,300	31,800
20	12,000	14,410	18,000	24,100	30,100	36,200	39,300
22	14,500	17,430	21,800	29,100	36,400	43,800	47,500
24	17,300	20,750	26,000	34,700	43,400	52,100	56,500
26	20,300	24,350	30,500	40,700	50,900	61,200	66,400
28	23,500	28,240	35,300	47,200	59,000	70,900	77,000
30	27,000	32,420	40,600	54,100	67,800	81,400	88,400

**This column is used for inlet vacuum separator silencers (5,500 fpm).*
***Operating vacuums greater than 20 "Hg are limited by start-up conditions.*

Table 6-10. UNIVERSAL SILENCER Separator-Silencer Capacities (all Series) for RPD Water-Sealed Vacuum Pumps

Separator-silencer ΔP is, for the most part, a function of the velocity pressure, the unit C factor and the pressure. See Chapter 5 for additional equations and charts to determine ΔP.

Inlet (Vacuum)

Velocity is determined by dividing the actual flow (ACFM) by the silencer flow area, and Figure 6-12 may be used to get start-up and separating ΔP.

$$\text{Start-Up } \Delta P = \frac{C}{6490}\left(\frac{ACFM}{P^2}\right)^2$$

$$\Delta P = \frac{C}{13.6}\left(\frac{V}{4005}\right)^2$$

$$\text{Operating } \Delta P = \frac{C}{6490}\left(\frac{ACFM}{P^2}\right)^2\left(\frac{29.92 - \text{"Hg. Vac.}}{29.92}\right)$$

or

$$\Delta P = \frac{C}{13.6}\left(\frac{V}{4005}\right)^2\left(\frac{29.92 - \text{"Hg. Vac.}}{29.92}\right)$$

See Figure 6-13 for variable definition.

Figure 6-12. Inlet Pressure Drop

As there is no well-defined and proven technique for predicting vacuum pump noise, such data must be obtained from the customer or pump manufacturer in most instances.

Discharge (Atmospheric)

As for the inlet, velocity for the discharge is determined by taking into account the difference in pressure and temperature, or Figure 6-13 may be used to determine ΔP.

Start-up ΔP is the same as inlet ΔP

$$\text{Operating } \Delta P = \frac{C}{6490}\left[\frac{ACFM\left(\frac{29.92 - "Hg. Vac.}{29.92}\right)}{P^2}\right]^2$$

Where

$ACFM$ = *inlet $ACFM$ at vacuum,*
P = *separator size in inches,*
ΔP = *pressure drop ("Hg),*
C = *silencer ΔP coefficient,*
A = *separator flow area in square feet,*
V = *velocity in feet per minute, and*
$"Hg. Vac.$ = *operating vacuum ("Hg).*

Figure 6-13. Discharge Pressure Drop

If the start-up ΔP is excessive, use the flow in Table 6-10 for a lower stated vacuum or calculate the separator sizing (P), as given in the following equation.

$$P = 0.1114\sqrt{ACFM\sqrt{\frac{C}{\Delta P}}}$$

Separators are required on both the inlet (vacuum) and atmospheric discharge. Only the more critical applications require inlet silencers. The discharge, when open to atmosphere, is extremely noisy and must be silenced.

The inlet separator provides corrosion protection for the pump by removing most, if not all, of the process liquid before it enters the pump.

When there is no inlet separator, both the inlet water and seal water is released to atmosphere. In this case, the discharge separator may have to be oversized if the liquid capacity of the separator is exceeded (Table 6-11).

If the over-capacity is caused by the inlet water, an inlet separator is recommended because excess water through the pump not only increases the power consumption, but can reduce the operational life of the pump and motor.

Both inlet and discharge separators require drain systems that provide sufficient head to both offset the vacuum on the inlet and the ΔP of the discharge (Figure 6-14). While the open drain system is not commonly used, it does highlight the need to seal the liquid lines.

Nominal Size	Liquid Capacity (Maximum gpm)						
	Inlet		Discharge				
	V1	UVS	VDY	USVY	UCS	USS	URS
2	—	15	15	—	15	15	15
$2^1/_2$	—	20	—	—	20	20	20
3	—	25	20	—	25	25	25
$3^1/_2$	—	35	—	—	35	35	35
4	25	50	25	—	40	40	40
5	40	75	30	—	50	50	50
6	60	120	40	—	60	60	60
8	100	120	50	—	80	80	80
10	140	200	100	—	100	100	100
12	180	200	120	180	120	120	120
14	210	300	—	210	140	140	140
16	250	300	—	250	160	160	160
18	300	—	—	300	—	—	—
20	350	—	—	340	—	—	—
22	400	—	—	370	—	—	—
24	450	—	—	400	—	—	—

Table 6-11. Separator Liquid Capacities

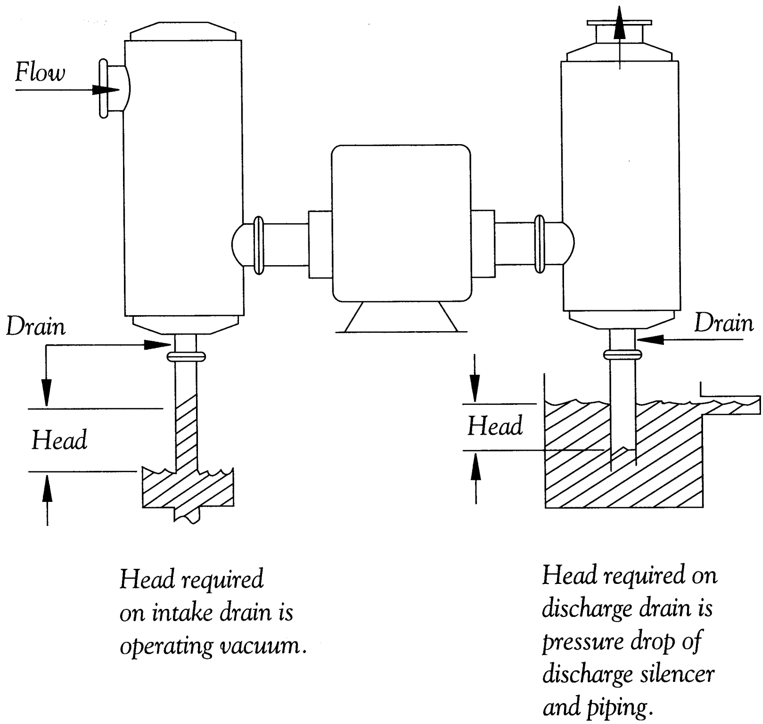


Figure 6-14. Open Drain System

Pressure required on the inlet drain is operating vacuum; a pump or trap is normally used due to the length of the required barometric leg. Pressure required on the discharge drain is the ΔP of the silencer and piping.

Dry-Type (Including Rotary Vane)

An inlet separator is required when process liquid is to be removed from the vacuum system; maximum inlet velocity is again 5,500 fpm. Inlet silencers are generally not required in these systems.

The discharge to atmosphere is typically extremely noisy, requiring a high-performance silencer. In critical applications, it may also require external acoustical lagging of the silencer shell and piping.

But, since there is no measurable liquid carry-over and the release to atmosphere is similar to that of a low-pressure (non-critical flow) vent, both the start-up and operating velocities may be increased. Even so, the operating velocity should not exceed 7,000 fpm (3,000 fpm for optimum performance). Otherwise:

- The selection and sizing parameters in Chapter 6, Section C (RPD blowers) apply.
- Silencer type is based upon the blower transition speed.
- Silencer size is based upon velocity and pressure drop.
- Critical applications may require increased silencer volume.

The ΔP is the same as that for water-sealed RPD blowers using Table 6-12.

Model	C Factor
Inlet (Vacuum Separator)	
VI	2.5
UVS	4.0
Discharge Silencer	
URB/URD	4.2
SD	4.2
RD	4.2

Table 6-12. UNIVERSAL SILENCER Separator-Silencer Reference Guide (Dry-Type and Rotary Vane RPD Blowers)

RECIPROCATING PISTON-TYPE

An inlet separator is required when process liquid is to be removed from the vacuum inlet and when the inlet piping might starve the pump; maximum inlet velocity is 5,500 fpm. Inlet silencers are generally not required.

The discharge is predominately low-frequency requiring a 2- or 3-chamber UNIVERSAL SILENCER silencer UCD, URB/URD or SURS series, depending upon the silencing criteria.

The discharge silencer is usually sized on the basis of ACFM and ΔP. Critical applications or those which have larger, lower-speed pistons may require added silencer volume.

The displacement CFM and pump efficiency may be obtained from the manufacturer or may be calculated as follows.

$$Displacement\ CFM = \left(\frac{\pi\, B^2 x\, S\, x\, N x RPM}{4 x\ 1728}\right)$$

Where

B = *cylinder bore in inches,*
S = *cylinder stroke in inches,*
N = *number of cylinders,*
RPM = *revolutions per minute*

$$Discharge\ ACFM = (Displacement\ CFM)\left(\frac{29.92 - \text{"Hg. Vac.}}{29.92}\right)\left(\frac{VE}{100}\right)$$

When compressor efficiency (VE) is not known, use 85%.

The maximum discharge start-up velocity, based on discharge ACFM at atmospheric inlet conditions, should not exceed 7,000 fpm. The maximum operating velocity should not exceed 3,000 fpm. The ΔP is determined as it is for other type vacuum pumps (See Table 6-13 for recommended silencer models and C factors).

Model	C Factor
Inlet (Vacuum Separator)	
VI	2.5
UVS	4.0
Discharge Silencer	
URB/URD	4.2
SURS	5.3*

*Three chamber design

Table 6-13. UNIVERSAL SILENCER Separator-Silencer Reference Guide (Reciprocating Piston-Type Vacuum Pumps)

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N O T E S

SECTION E

CENTRIFUGAL AIR BLOWERS AND COMPRESSORS

Centrifugal blowers and compressors used in plant and process air service range in size from about 25 up to 5,000 horsepower (hp). Compressor speed varies from approximately 5,000 to 20,000 rpm, depending upon the compressor size and service. The compressor may be powered by a steam or gas turbine, motor or engine. A gear unit may be required to match the speed of the driver to the compressor.

The major sources of centrifugal compressor noise are

- Air inlet (from atmosphere),
- Discharge piping,
- Compressor casing, and
- Drive unit.

Centrifugal blower and compressor noise is inherently high-amplitude, broadband (1,000 Hz and higher) and is a function of horsepower, blade tip-speed, compression ratio, piping arrangement and type of gas being compressed.

BASIC UNSILENCED POWER LEVEL

The compression impeller is the predominant source of the high-frequency noise produced at the blade-passing frequency, which is the number of blades times impeller revolutions per second. Secondly, a random, lower intensity broadband noise spectrum is generated by flow turbulence and other sources within the compressor and from the attached piping.

There is no well-defined and proven technique for predicting centrifugal compressor noise. Until a fully reliable method is developed, the application engineer should rely on the customer or the compressor manufacturer to provide such data.

However, the overall sound power level (L_W) of the inlet and discharge of centrifugal compressors may be approximated for estimating purposes using the equation:

$$L_w = 20 \log Hp + 50 \log \left(\frac{U}{800}\right) + 81\ dB$$

Where

Hp = *compressor horsepower, and*
U = *blade tip-velocity in feet per second.*

The frequency spectrum is relatively broadband, with maximum noise being produced at the compressor fundamental (f_o) and second harmonic of the blade-passing frequency; f_o is defined as

$$f_0 = \frac{N(RPM)}{60}$$

Where

N = *number of blades, and*
RPM = *compressor speed.*

The second (and higher) harmonies are integer multiples of f_o.

The spectrum shape will vary with each application but may be approximated as shown in Figure 6-15.

When estimating the L_W using this method, allowance must be made for the TL across the casing or pipe wall (an exception being the atmospheric inlet). Silencers are normally required for both the inlet and discharge of all centrifugal compressors.

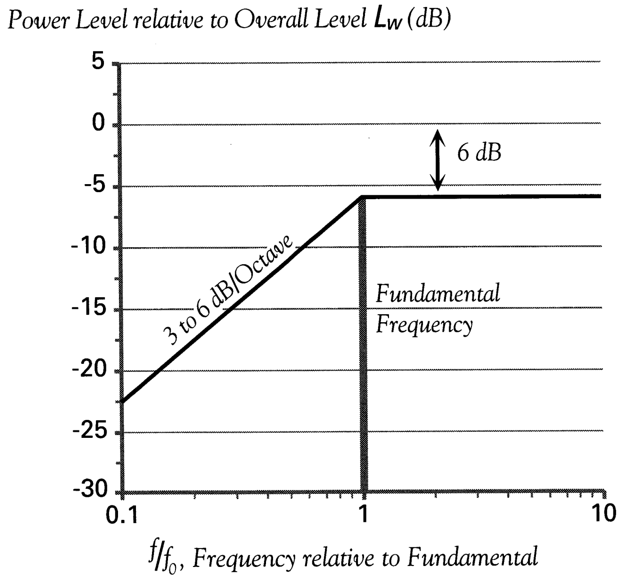

Figure 6-15. Generalized Spectrum Shape

When operating under ambient inlet conditions air blowers and compressors almost always are provided with an inlet filter and an inlet silencer or a combination filter-silencer, when space permits. Silencers for both the inlet and discharge usually are of a low-ΔP dissipative ("absorptive") design.

UNIVERSAL SILENCER's Bulletin 245 lists the U2, SU3, SU4 and SU5 series which meet this criteria. The acoustical material used in the silencers absorbs the sound by converting "acoustical" energy into heat using friction in the open cells (or passages) of the material.

The selection and sizing of inlet and discharge silencers for centrifugal blowers and compressors is typically based on

1. Acoustical criteria (silencer grade),
2. Velocity and ΔP limitation (sizing), and
3. Dimensional, weight and budget considerations.

The compressor volume flow rate may be stated in either actual or standard conditions or by weight flow, as shown in the equations below.

$$ACFM = SCFM\left(\frac{P_s}{P_a}\right)\left(\frac{T_a}{T_s}\right)$$

In most atmospheric air applications, this may be reduced to

$$ACFM = SCFM\left(\frac{T_a}{530}\right) = \frac{Q(V_g)}{60}$$

Silencer velocity as in all applications is

$$Velocity\,(V) = \frac{ACFM}{A}$$

Silencer pressure drop (ΔP) is then obtained by using the equation below.

$$\Delta P = C\left(\frac{V}{4005}\right)^2\left(\frac{P_a}{P_s}\right)\left(\frac{T_s}{T_a}\right)\left(\frac{MW_a}{MW_s}\right)$$

Where

$ACFM$ = *actual cubic feet per minute,*
$SCFM$ = *standard cubic feet per minute (at 14.7 psia and 70°F),*
P_a = *operating pressure psia,*
P_s = *standard pressure (14.7 psia),*
T_a = *operating temperature °R(460 + °F),*
T_s = *standard temperature (530°R),*
MW_a = *molecular weight of gas,*
MW_s = *molecular weight of standard air (28.97),*
Q = *flow by weight in pounds per hour,*
V_g = *specific volume of gas at operating pressure and temperature in cubic feet per pound,*
V = *silencer velocity in feet per minute,*
C = *silencer ΔP coefficient, and*
A = *silencer flow area in square feet.*

Silencers for gas compressors (other than air) usually require construction to meet ASME Code.

Near the compressor, inside a building for example, radiated noise from the compressor casing and piping usually predominates and may reach levels of 110 to 115 dB or more, when the inlet is a closed system or the atmospheric inlet is located outdoors. Otherwise the noise level may be even higher.

For optimum performance, the silencer should be installed as close to the compressor as possible. In addition, large units may require acoustical treatment of the compressor housing (casing) and the piping close to the source. Fixed or movable barriers, and even total enclosures are often needed in the more critical applications.

Considering the number of potential noise sources, the silencer TL should be at least 6 to 10 dB more than the required silencer IL.

Silencer velocity normally ranges from 3,000 to about 6,000 fpm. Maximum velocity should not exceed 7,500 fpm. Reduced velocities provide added silencer IL and in some instances added TL. Recommended silencer capacities, velocities and ΔP are tabulated in Tables 6-15 and 6-16.

Silencer Velocity (fpm)	3,000	3,500	4,000	4,500	5,000	5,500	6,000	6,500
Silencer Model	**Pressure Drop ("H_2O at 70°F)***							
U2	0.14	0.19	0.25	0.31	0.39	0.47	0.56	0.66
SU3, SU4	0.31	0.42	0.55	0.69	0.86	1.04	1.23	1.45
SU5	0.42	0.57	0.75	0.95	1.17	1.41	1.68	1.97
Silencer Size	**Flow Rate (ACFM)**							
8	1,047	1,222	1,396	1,570	1,745	1,920	2,094	2,268
10	1,650	1,925	2,200	2,475	2,750	3,025	3,300	3,575
12	2,370	2,765	3,160	3,555	3,950	4,345	4,740	5,135
14	3,210	3,745	4,280	4,815	5,350	5,885	6,420	6,955
16	4,200	4,900	5,600	6,300	7,000	7,700	8,400	9,100
18	5,400	6,300	7,200	8,100	9,000	9,900	10,800	11,700
20	6,600	7,700	8,800	9,900	11,000	12,100	13,200	14,300
22	7,800	9,100	10,400	11,700	13,000	14,300	15,600	16,900
24	9,300	10,850	12,400	13,950	15,500	17,050	18,600	20,150
26	11,100	12,950	14,800	16,650	18,500	20,350	22,200	24,050
28	12,900	15,050	17,200	19,350	21,500	23,650	25,800	27,950
30	14,700	17,150	19,600	22,050	24,500	26,950	29,400	31,850
32	16,800	19,600	22,400	25,200	28,000	30,800	33,600	36,400
36	21,300	24,850	28,400	31,950	35,500	39,050	42,600	46,150
42	28,800	33,600	38,400	43,200	48,000	52,800	57,600	62,400
48	37,800	44,100	50,400	56,700	63,000	69,300	75,600	81,900
54	47,700	55,650	63,600	71,550	79,500	87,450	95,400	103,350
60	58,800	68,600	78,400	88,200	98,000	107,800	117,600	127,400

**Entrance loss from atmosphere not included.*

Table 6-14. Silencer Capacity vs. Pressure Drop (ΔP) — Inlet of High-Speed Centrifugal Air Compressors

Silencer Size	Capacity (Inlet CFM at 14.7 psia and 70°F)						
		Discharge Silencer					
	Inlet Silencer	4 psig	6 psig	8 psig	10 psig	12 psig	15 psig
4	522	612	649	683	715	750	798
5	816	957	1,015	1,069	1,118	1,172	1,248
6	1,176	1,379	1,463	1,540	1,611	1,690	1,799
8	2,094	2,455	2,605	2,742	2,807	3,009	3,134
10	3,300	3,869	4,105	4,321	4,521	4,741	5,048
12	4,740	5,558	5,871	6,207	6,494	6,810	7,250
14	6,420	7,528	7,986	8,406	8,796	9,224	9,821
16	8,400	9,849	10,448	11,000	11,509	12,069	12,850
18	10,800	12,663	13,434	14,172	14,797	15,517	16,521
20	13,200	15,478	16,419	17,285	18,085	18,966	20,193
22	15,600	18,292	19,404	20,428	21,373	22,414	23,863
24	18,600	21,809	23,136	24,357	25,483	26,724	28,453
26	22,200	26,030	27,614	29,071	30,415	31,897	33,960
28	25,800	30,252	32,092	33,785	35,347	37,069	39,467
30	29,400	34,473	36,570	38,500	40,280	42,242	44,974
Temp (°F)	**70**	**115**	**140**	**165**	**190**	**210**	**240**
Silencer Model	**Pressure Drop ("H_2O)* — See Figure 5 - 1**						
U2	0.56	0.66	0.70	0.73	0.77	0.80	0.86
SU3, SU4	1.23	1.45	1.53	1.61	1.69	1.77	1.89
SU5	1.68	1.97	2.09	2.20	2.30	2.41	2.57

**Entrance loss from atmosphere not included.*

Table 6-15. Silencer Capacity vs. Discharge Pressure Drop for Inlet and Discharge of Low Pressure Centrifugal Air Compressors (Based on Velocity of 6,000 fpm)

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SECTION F

RECIPROCATING AIR COMPRESSORS

Reciprocating compressors, when drawing air from the atmosphere, require reactive (chamber-type) inlet silencers or filter silencers of adequate volume to reduce both noise and pulsation.

The application of discharge silencers, surge bottles or pulsation dampeners for closed pressure systems and silencer sizing for multiple-cylinder air compressors is a rather complex process. In most instances the application engineer will want to get assistance from the silencer manufacturer.

The UNIVERSAL SILENCER multi-chamber reactive-type inlet silencer (UCI/UCD, URB/URD series) is an effective broadband silencer which functions over a wide range of compressor operation and frequencies. It has all of the performance characteristics of the classic low-pass filter, while the dissipative effect of its perforated tubes adds mid-range frequency attenuation.

Reciprocating compressors used in plant and air service range in size from 50 to over 3,000 hp. Higher hp usually run at reduced rpm's (1,200 to 600 rpm and less). The major sources of reciprocating compressor noise are

- Air inlet (from atmosphere),
- Discharge piping,
- Compressor cylinders, and
- Housing and drive unit.

Reciprocating compressors produce both low-frequency noise and pulsation which often results in pulsation-induced vibration. Unsilenced, large-bore, low-speed air compressors produce high-amplitude, low-frequency noise often referred to as "air-borne" pulsation which may cause walls, windows and doors to vibrate, even at considerable distances from the source.

The adverse effects of pulsation may starve the cylinders, resulting in reduced volumetric efficiency and power loss. Accordingly, the inlet silencer should be installed at the compressor (direct coupled, where possible). Except for very small compressors, when the inlet piping is open to atmosphere, it should not be terminated inside the room or enclosure housing the compressor. This, when coupled with the reverberant effect of the housing, will invariably create a noise problem.

The air filter may be either an oil bath, media impingement or inertial type. When a UNIVERSAL SILENCER filter/silencer combination such as the RF series (Bulletin 241) is used, it most likely will be installed outdoors, requiring an extended length of inlet piping between the silencer and the compressor. In order to prevent a resonant piping condition, the piping system must avoid the troublesome pipe lengths discussed in Section 6C. Whenever the pulsation frequency produced by the compressor coincides with the natural frequency of the inlet piping, adverse noise and vibration can develop.

The selection and sizing of the inlet silencer for reciprocating air compressors is generally based on a combination of operational installation costs and environmental considerations, including

- Air flow requirement (velocity and pressure drop and inlet slug-volume (SV)),
- Size, weight and arrangement limitations,
- Acoustic and economic requirements.

The inlet SV is equal to the displacement of the cylinder times the volumetric efficiency of the compressor and represents the short-term air flow demand of the compressor, as shown in the following equation.

$$\text{Inlet } SV = \frac{\pi B^2 \times S \times VE}{4 \times 100}$$

Where

SV = *slug-volume in cubic inches,*
B = *cylinder bore in inches,*
S = *cylinder stoke in inches, and*
VE = *volumetric efficiency (average = 85%).*

The ACFM is equal to the displacement CFM of the compressor times the volumetric efficiency.

$$\text{Inlet } ACFM = \frac{\pi B^2 \times S \times N \times A \times RPM \times VE}{4 \times 1{,}728 \times 100}$$

Where

RPM = *compressor speed in revolutions per minute,*
N = *number of cylinders, and*
A = *action,*
single action = 1,
double action = 2.

Using another equation

$$\text{Inlet } ACFM = SCFM \left(\frac{T_a}{530}\right) = \frac{Q\,(V_g)}{60}$$

Where

$SCFM$ = *standard **CFM** at 14.7 psia and 70°F,*
T_a = *operating temperature, °R,*
Q = *air flow by weight in pounds per hour, and*
V_g = *specific volume of air at operating temperature in cubic feet per pound.*

Silencers are sized by first calculating the SV requirement and then determining the velocity and pressure drop. In most instances, the larger of the two is the recommended size.

The SV to silencer volume ratio is typically 10:1 (Table 6-16). The more critical applications may require added silencer volume for increased silencer insertion loss and pulsation control.

Under atmospheric inlet conditions

$$Velocity(V) = \frac{ACFM}{A} = 174\sqrt{\left(\frac{\Delta P\, T_a}{C}\right)}$$

and

$$Silencer\ \Delta P = C\left(\frac{V}{4005}\right)^2\left(\frac{530}{T_a}\right)$$

Where

C = *silencer coefficient,*
V = *silencer velocity in feet per minute,*
A = *silencer flow area in square feet,*
ΔP = *pressure drop in "H_2O, and*
T_a = *inlet temperature, °R (460 + °F).*

Recommended silencer velocities for different types of compressors are:

- Single acting compressor 2,000-3,000 fpm
- Double acting compressor 4,000-6,000 fpm (rated at 5,500 fpm)

For silencer capacity (ACFM vs. pressure drop) see Table 6-17.

Silencer Size	Maximum SV (cu in)*		
	UCI/UCD	URB/URD	RF
1	—	25.0	—
1 1/2	—	60.5	—
2	—	120	—
2 1/2	—	180	—
3	—	280	—
3 1/2	—	470	—
4	—	640	—
5	—	1,080	—
6	—	1,500	—
8	1,900	3,100	3,250
10	3,300	5,700	6,700
12	5,300	8,400	10,500
14	6,200	14,600	11,600
16	10,300	22,400	18,000
18	15,500	30,400	21,500
20	16,600	32,500	30,000
22	23,800	41,200	33,500
24	32,900	49,400	46,500
26	25,000	66,000	49,000
28	47,500	86,000	65,500
30	61,500	112,000	85,500

**Calculated Silencer volume ÷ 10*

Table 6-16. Silencer Sizing Based on Slug-Volume (SV)

	Silencer Velocity								
	2,000	**2,500**	**3,000**	**3,500**	**4,000**	**4,500**	**5,000**	**5,500**	**6,000**
Temperature (°F)	**Pressure Drop, H_2O (Entrance Loss from Atmosphere not Included)**								
0	1.20	1.89	2.72	3.69	4.84	6.10	7.55	9.12	10.86
70	1.05	1.64	2.36	3.21	4.20	5.30	6.55	7.92	9.43
100	0.99	1.55	2.23	3.04	3.97	5.01	6.20	7.50	8.92
Silencer Size	**Flow Rate (ACFM)**								
1	11	14	17	19	22	25	27	30	33
1½	24	30	36	42	48	54	60	66	72
2	44	55	66	77	88	99	110	121	132
2½	68	85	102	119	136	153	170	187	204
3	98	122	147	172	196	220	245	270	294
3½	134	168	201	235	268	302	335	369	402
4	174	218	261	305	348	392	435	479	522
5	272	340	408	476	544	612	680	748	816
6	392	490	588	686	784	882	980	1,078	1,176
8	698	873	1,047	1,222	1,396	1,571	1,745	1,920	2,094
10	1,100	1,375	1,650	1,925	2,200	2,475	2,750	3,025	3,300
12	1,580	1,975	2,370	2,765	3,160	3,555	3,950	4,345	4,740
14	2,140	2,675	3,210	3,745	4,280	4,815	5,350	5,885	6,420
16	2,800	3,500	4,200	4,900	5,600	6,300	7,000	7,700	8,400
18	3,600	4,500	5,400	6,300	7,200	8,100	9,000	9,900	10,800
20	4,400	5,500	6,600	7,700	8,800	9,900	11,000	12,100	13,200
22	5,200	6,500	7,800	9,100	10,400	11,700	13,000	14,300	15,600
24	6,200	7,750	9,300	10,850	12,400	13,950	15,500	17,050	18,600
26	7,400	9,250	11,100	12,950	14,800	16,650	18,500	20,350	22,200
28	8,600	10,750	12,900	15,050	17,200	19,350	21,500	23,650	25,800
30	9,800	12,250	14,700	17,150	19,600	22,050	24,500	26,950	29,400

Table 6-17. Inlet Silencer Capacity and Pressure Drop (ΔP) (UCI, URB/URD and RF Series)

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SECTION G
VENTS AND BLOWDOWNS TO ATMOSPHERE

Experience has shown that effective reduction of valve and regulator noise is generally most complex and is often difficult to apply. This is particularly true when quieting large, high-pressure vents and blowdowns to atmosphere. However, an abbreviated and simplified vent and blowdown noise analysis procedure is followed in this section which may be used as a basic guide in silencer application with emphasis on

- Vent silencer analysis,
- Silencer analysis and performance, and
- Special applications.

Typical vent and blowdown silencer applications include:

- Safety valves
- Station and pipeline blowdowns
- Steam ejectors
- Switch valves (reversing valves)
- Autoclaves
- Process control valves
- Compressor and blower by-pass valves
- Boiler start-up and purge

Vent and blowdown silencer applications are seldom a simple catalog selection. Usually, the purchaser provides detailed operational and performance requirements up front. The following information is needed for system analysis, silencer selection, sizing and pricing:

- Type of gas
- Molecular weight or specific gravity of gas
- Mass flow (lbs/hr, ACFM or SCFM)
- Upstream pressure and temperature
- Valve type and size
- Unsilenced noise levels (when available)
- Allowable pressure drop
- Noise criteria (octave band preferred)
- Installation (vertical or horizontal)
- Piping size and arrangement
- Required options

Silencer selection and sizing is generally based on velocity, required acoustical performance and pressure drop. Typical silencer velocities are

- Continuous — 10,000 to 15,000 fpm, and
- Intermittent — Up to 18,000 fpm.

The acoustical and structural limit is about 25,000 fpm. The velocity is determined from the following equation.

$$\text{Silencer Velocity} = \frac{ACFM}{A}$$

Where

$ACFM$ = *actual cubic feet per minute (usually at 14.7 psia), and*
A = *silencer open flow area in square feet.*

Normally the valve manufacturer or the design engineer specifies the flow rate. If this is not given and/or a verification is desired, then the **maximum flow rate** during the critical flow pressure cycle of vents and blowdowns to atmosphere may be approximated using Figure 6-16.

Air and gas blowdown or vent

$$SCFM = 17.8\,AP\sqrt{\frac{520}{T\,SG}}$$

Where

$SCFM$ = *standard cubic feet per minute* (14.7 *psia and* 70°F),
A = *value open area in square inches* x *flow coefficient*,
P = *upstream pressure*, *psia*,
T = *upstream temperature*, °R(460 + °F),
SG = *specific gravity of gas* = $\frac{MW}{28.97}$,
and if flow coefficient is not known, use 0.85.

Dry and saturated steam blowdown or vent

$$W\,(lbs/hr) = 51.43\,AP$$

Wet steam blowdown or vent

$$W\,(lbs/hr) = 51.43\,AP\,F_m$$

Where

W = *Mass flow of steam in pounds per hour*,

$$F_m = \frac{1}{1\text{-}0.012\,m}$$

m = % *of moisture*.

Superheated steam blowdown or vent

$$W\,(lbs/hr) = 51.43\,APF_s$$

$$F_s = \frac{1}{1 + 0.00065\,T_s}$$

$$T_s = {}^{\circ}F \text{ Superheat}$$

Figure 6-16. Maximum Flow Rates

BASIC UNSILENCED POWER LEVELS

In critical flow applications, both the gas flow and the resultant noise level increase until the critical pressure ratio is reached. Beyond this ratio, there is no further increase in flow velocity, yet the noise level continues to increase with increased pressure ratios.

The pressure drop of the vent silencer is rarely important, but when added to the atmospheric pressure, should not exceed the critical flow pressure downstream of the valve. Otherwise, the flow rate may be adversely effected or the silencer may be subjected to higher pressures than appropriate.

Non-critical, low-pressure vent and blowdown applications usually do not require or permit using an inlet diffuser due to pressure drop limitations. When a diffuser cannot be used, either a chamber or absorptive-type silencer may be used. Silencer sizing then is based on the ΔP allowable and/or silencer self-generated noise.

The overall sound power level (L_W) under critical vent flow conditions may be approximated by the equations that follow.

For air and gases (except steam), use

$$L_w = 10 \log (PA) + 20 \log \frac{T}{SG} + 85 \text{ dB}$$

For steam (both saturated and superheated), use

$$L_w = 17 \log W + 50 \log T - 85 \text{ dB}$$

Where

L_W = *overall vent sound power level, dB (re 10^{-12} watt),*
P = *upstream pressure, psia,*
T = *upstream temperature, °R(460 + °F),*
A = *effective value open area in square feet,*
SG = *specific gravity of gas (molecular weight / 28.97), and*
W = *steam mass flow in pounds per hour.*

Always use measured L_W or L_P values when available.

Vent peak frequency (air, gas and steam) is used to establish the spectral shape and the peak frequency. The following equations can be used to calculate the frequency and other parameters.

$$f_m = \frac{0.4(a_j)\,M_j}{d_m}$$

Where

f_m = *peak frequency, in Hertz,*
d_m = *valve throat diameter in feet, and*
a_j = *speed of sound at valve exit.*

$$a_j = 223\sqrt{\frac{kT_j}{MW}}$$

T_j = *Temperature at jet*

$$T_j = \frac{T}{1 + \frac{k-1}{2}(M_j)^2}$$

M_j = *Maximum Mach number of jet*

$$M_j = \sqrt{\frac{2}{k-1}\left[\left(\frac{P}{P_a}\right)^{\frac{k-1}{k}} - 1\right]}$$

MW = *molecular weight of gas,*
P = *upstream pressure, psia,*
P_a = *ambient pressure, psia,*
T = *upstream temperature, °R, and*
k = *ratio of specific heats = 1.4 for air.*

$$k = \left(\frac{C_p}{C_v}\right) \text{ of gas}$$

The constant of 0.4 is based on actual test data (venting air from 100 psig to atmosphere).

Once the overall vent sound power (L_W) and the vent peak frequency (f_m) have been determined, the next step in the analysis cycle is to apply the spectrum corrections as provided in either Table 6-18 or 6-19, since most vent exits are direct to atmosphere with no piping downstream of the valve or larger downstream piping.

Peak Frequency (f_m), Hz	Octave Band Center Frequency (Hz)							
	63	125	250	500	1k	2k	4k	8k
250	-13	-6	-4	-6	-12	-18	-24	-30
500	-18	-12	-6	-4	-6	-12	-18	-24
1,000	-24	-18	-12	-6	-4	-6	-12	-18
2,000	-30	-24	-18	-12	-6	-4	-6	-12
4,000	-36	-30	-24	-18	-12	-6	-4	-6
8,000	-40	-36	-30	-24	-18	-12	-6	-4

Table 6-18. Spectrum Correction (dB) — Valve with Same Size or No Downstream Piping

Peak Frequency (f_m), Hz	Octave Band Center Frequency (Hz)							
	63	125	250	500	1k	2k	4k	8k
250	-6	-5	-7	-12	-19	-29	-40	-50
500	-9	-6	-5	-6	-11	-19	-29	-40
1,000	-15	-9	-6	-5	-6	-11	-19	-29
2,000	-22	-15	-9	-6	-5	-6	-11	-19
4,000	-30	-22	-15	-9	-6	-5	-6	-11
8,000	-40	-30	-22	-15	-9	-6	-5	-6

Table 6-19. Spectrum Correction (dB) — Valve with Larger Downstream Piping

The peak frequency (f_m) correction in octave bands is subtracted from the overall sound power level (L_W) to obtain octave band levels for analysis.

Silencer self noise (SN) is the noise generated by the gas flow through the exit portion of the silencer and is the lowest noise level that can be achieved at a given flow velocity through the silencer. The silencer SN should not exceed the stated criteria. SN can be reduced by reducing the velocity, which for a given flow, requires a larger silencer.

Silencer SN may be approximated, using the following empirical procedure.

$$\text{Silencer SN } (L_w) = 10 \log\left[(A)\left(\frac{MW}{T}\right)(V)^{6.2}\right] - 38 \text{ dB}$$

Where

SN = *silencer self noise in decibels,*
A = *silencer flow area in square feet,*
MW = *molecular weight of gas,*
T = *absolute temperature of gas, °R, and*
V = *velocity of gas in feet per second.*

Silencer SN and the unsilenced vent sound power are combined on a power basis. The octave band SN spectrum is relatively flat over the entire frequency spectrum region of practical interest beginning with the 125 Hz band which is approximately 3 dB less than the overall level as shown in Table 6-20.

Hz	125	250	500	1k	2k	4k	8k
dB	-3	-10	-13	-12	-11	-10	-13

Table 6-20. Adjustment to Obtain Self Noise Spectrum

This spectrum is generally applicable to silencers of 25-50% open area with 3-6" silencer tubes or gap widths.

SPECIAL CONSIDERATIONS

The blowdown time required for blowdown of a pipeline section or pressurized reservoir can be estimated using the following equation.

$$t = \frac{5.5\,V}{A} \sqrt{\left(\frac{SG}{T_o}\right)} \log_e\left(\frac{P_o}{P_a}\right)$$

Where

t = *blowdown time in seconds,*
V = *pressured volume in cubic feet,*
A = *valve open area in square feet* x *flow coefficient,*
SG = *special gravity of gas* (MW/28.97),
T_o = *initial temperature,* °R,
P_o = *initial line pressure, psia, and*
P_a = *atmospheric pressure, psia.*

Use 0.85 if valve flow coefficient is not known.

An optional multi-port orifice may be placed in the inlet nozzle of the silencer at the factory to provide a staged reduction in pressure for both noise and flow control. The increase in pressure drop across the silencer in this manner usually does not exceed 100 psi.

Secondary restrictive diffusers are used either to reduce the flow rate while meeting a required blowdown time requirement or to maintain a desired back pressure while passing a specified flow. Always submit any vent silencer application requiring restrictive diffusers to the factory for review. It is recommended that all piping leading up to the silencer be rated for the full pressure upstream of the control valve. The customer should be asked to confirm that the control valve will not malfunction from excessive back pressure caused by installing a silencer with a restrictive diffuser.

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SECTION H
PRESSURE REGULATORS

Most pressure regulator (PR) applications are for regulation of either natural gas at city gates, steam used for heating or other processes.

Regulator noise is caused by the rapid expansion of gas or steam, as in high pressure vents and blowdowns to atmosphere. In critical flow applications, both the gas flow and the noise increase until the critical pressure ratio is reached. The critical pressure ratio occurs when the flow through the orifice is at sonic velocity. Once the critical pressure is reached, the noise increases only with an increase in flow.

The inlet piping is usually the same size as the regulator inlet. The regulator inlet velocity is generally limited to a maximum of approximately 10,000 fpm. The enlarged downstream piping is again limited to about the same maximum velocity. When the pressure is reduced, the gas or steam volume increases. Accordingly, the downstream piping is enlarged to reduce both velocity and flow noise.

The unsilenced noise level (L_W) of a pressure regulator is usually provided by the regulator manufacturer for a given application. However, when available, actual measured unsilenced noise levels should be obtained and used to determine the silencer IL needed to meet the silencing criteria.

PR noise is inherently high-frequency and most often requires a low ΔP dissipative-type silencer, such as the SU5 series.

In critical "sonic" flow applications, radiated piping noise usually predominates. Non-critical, subsonic flow conditions produce less noise, which is propagated about equally between the upstream and downstream piping. Left unsilenced, the room (or area) noise may become unbearable.

All PR silencers are custom designed for pressure service and usually follow ASME pressure vessel code design and construction. It is common practice to base the design pressure of the downstream silencer on the higher upstream pressure. ***Any time the design pressure is based on the lower pressure, it is mandatory that a fail-safe pressure control system be provided.***

If the PR noise is **unacceptable** in the immediate vicinity of the regulator valve, but **acceptable** downstream, consider acoustical treatment of the regulator itself, or partial or total enclosure of the regulator, and optional installation of a low ΔP flow orifice at the entrance of the enlarged downstream piping.

In addition, a portion of both the upstream and downstream piping may require acoustic lagging.

If the PR noise is **acceptable** in the immediate vicinity of the regulator valve, but is **excessive** downstream, consider installation of a low ΔP silencer at the regulator outlet.

For maximum reduction of all sources of PR noise in critical applications, consider acoustical treatment of the regulator itself, or total enclosure of the regulator, and installation of both inlet and outlet low ΔP silencers.

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SECTION I

CENTRIFUGAL FANS (INDUSTRIAL-TYPE)

Large centrifugal fans produce high levels of sound and usually require both inlet and outlet silencers or at least, some form of acoustical treatment. Applications include:

- Primary air fans, when installed in an indoor work area, may require both inlet and outlet silencers (dissipative-type).
- Forced draft (FD) fans generally require only inlet silencers or an inlet silencer/plenum combination (dissipative-type).
- Induced draft (ID) fans usually require only an outlet silencer (dissipative-type). However, when the air flow contains fly ash or other combustion contaminants, a self-cleaning or non-clogging resonator/dissipative-type silencer is required.

Fan noise varies with

1. Static pressure,
2. Fan efficiency,
3. Air flow, and
4. Fan speed (rpm), including the blade tone component.

The major sources of centrifugal fan noise are

1. Air inlet and outlet of primary air fans,
2. The inlet of FD fans and the outlet of ID fans,
3. Fan casing (and ducting), and
4. Drive motor.

BASIC UNSILENCED POWER LEVEL

The blade tone component is the predominant source of centrifugal fan noise. Secondly, a random, lower intensity, broadband noise spectrum is produced as a result of flow turbulence within the fan itself and from the attached piping.

As noted, both inlet and outlet silencers may be required, depending upon the size of the fan, service and application. Larger fans may also require a transition and/or plenum for proper distribution of the air flow. Also, some fans (including the drive motor) located in critical areas may have to be totally enclosed and have provisions for air flow, silencing and accessibility.

Fan vibration may be transmitted to the building structure, but vibration isolators usually solve this problem.

The fan manufacturer normally provides the unsilenced sound power levels (L_W) of its fans in octave bands. However, when the unsilenced sound levels are not available, the sound level (L_W) of the fan may be estimated, using the following equation.

$$L_w = L_w(B) + 10 \log Q + 20 \log P + B_t$$

or

$$L_w = L_w(B) + L_w(C)$$

Where

L_w = *total sound power level (dB, re 10^{-12}watt),*
$L_w(B)$ = *basic sound power level of fans from Table 6-21,*
Q = *flow rate (ACFM),*
P = *static pressure ("H_2O),*
$L_w(C)$ = $10 \log Q + 20 \log P + B_t$
B_t = *blade tone component (dB, Table 6-21) is added* ***only*** *to the octave band containing the blade frequency,*
f_b = *blade frequency* = $\frac{RPM \times N}{60}$,
RPM = *fan speed, and*
N = *number of blades.*

When using this procedure for estimating the sound power of a fan, it is assumed that the fan is well-designed, properly maintained and is operating at or near its rated efficiency.

SPECIAL CONSIDERATIONS

When possible, fans should be located away from the work area. Increased fan speeds and high system velocities should be avoided to prevent vibration and radiated noise.

When the silencer is installed at the inlet or outlet of the fan, it must not overly obstruct the flow or apply excessive pressure drop. Selection criteria for silencers are given in Figure 6-17.

Cylindrical dissipative-type silencers with a concentric plug or rectangular dissipative-type silencers with parallel baffles are the most common types of fan silencers. Both types may require an evase exit arrangement for pressure regain and to insure uniform flow.

All primary air and FD fan applications requiring inlet and outlet silencers other than UNIVERSAL SILENCER Models U2, SU3, SU4 and SU5 are custom designed. This also applies to transitions, inlet boxes and plenum assemblies, which are special designs.

ID fan applications normally require a resonator/dissipative combination or non-pack resonator-type silencer. The UNIVERSAL SILENCER ET series may be offered for the outlet of the smaller ID fans when non-pack is specified.

Fan Description	Blade-Tone Component (B_t)	Octave Band Center Frequency (Hz)							
Centrifugal		63	125	250	500	1k	2k	4k	8k
Air Foil Blade	3 dB	35	35	34	32	31	26	18	10
Backward Curved Blade	3 dB	35	35	34	32	31	26	18	10
Radial Blade	5 – 8 dB	48	45	45	43	38	33	30	29
Forward Curved Blade	2 dB	40	38	38	34	28	24	21	15
Tubular	4 – 6 dB	46	43	43	38	37	32	28	25
Vane-Axial	6 – 8 dB	42	39	41	42	40	37	35	25
Tube-Axial	6 – 8 dB	44	42	46	44	42	40	37	30
Propeller	5 – 7 dB	51	48	49	47	45	45	43	31

Table 6-21. Basic Sound Power Levels of Fans (dB)

Selection of silencer type is based on —

1. Type of fan
2. Service and application
3. Silencing criteria

Silencer sizing is based on —

1. Flow rate (Table 6-22)
2. Allowable pressure drop ("H_2O)
3. Silencing criteria
4. Fan connection or duct size (inches)
5. Cost

Silencer velocities —

1. Dissipative-type (6,000 fpm (rated))
2. Resonator-type (4,000-5,000 fpm)

Silencer pressure drop —

1. Small fans (1-1/2" to 3 "H_2O)
2. Large fans (0.15" to 0.5 "H_2O)

Lower silencer and system velocities will provide increased silencer insertion loss and reduced system noise.

Figure 6-17. Silencer Selection Criteria

Silencer Velocity (fpm)	3,000	3,500	4,000	4,500	5,000	5,500	6,000	6,500
Silencer Model	**Pressure Drop ("H_2O at 70°F)**							
U2	0.14	0.19	0.25	0.31	0.39	0.47	0.56	0.66
SU3, SU4	0.31	0.42	0.55	0.69	0.86	1.04	1.23	1.45
SU5	0.42	0.57	0.75	0.95	1.17	1.41	1.68	1.97
Silencer Size	**Flow Rate (ACFM)**							
8	1,047	1,222	1,386	1,570	1,745	1,920	2,094	2,268
10	1,650	1,925	2,200	2,475	2,750	3,025	3,300	3,575
12	2,370	2,765	3,160	3,555	3,950	4,345	4,740	5,135
14	3,210	3,745	4,280	4,815	5,350	5,885	6,420	6,955
16	4,200	4,900	5,600	6,300	7,000	7,000	8,400	9,100
18	5,400	6,300	7,200	8,100	9,000	9,900	10,800	11,700
20	6,600	7,700	8,800	9,900	11,000	12,100	13,200	14,300
22	7,800	9,100	10,400	11,700	13,000	14,300	15,600	16,900
24	9,300	10,850	12,400	13,950	15,500	17,050	18,600	20,150
26	11,100	12,950	14,800	16,650	18,500	20,350	22,200	24,050
28	12,900	15,050	17,200	19,350	21,500	23,650	25,800	27,950
30	14,700	17,150	19,600	22,050	24,500	26,950	29,400	31,850
32	16,800	19,600	22,400	25,200	28,000	30,800	33,600	36,400
26	21,300	24,850	28,400	31,950	35,500	39,050	42,600	46,150
42	28,800	33,600	38,400	43,200	48,000	52,800	57,600	62,400
48	37,800	44,100	50,400	56,700	63,000	69,300	75,600	81,900
54	47,700	55,650	63,600	71,550	79,500	87,450	95,400	103,350
60	58,800	68,600	78,400	88,200	98,000	107,800	117,600	127,400

Table 6-22. Silencer Capacity vs. Pressure Drop

N O T E S

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Circumferences & Areas of Circles

Dia. (in)	Circ. (in)	Area in²	Area ft²	1/ft²	Dia. (in)	Circ. (in)	Area in²	Area ft²	1/ft²
¼	.785	.049	.0003	2933	28	87.965	615.75	4.276	.2339
½	1.571	.196	.0014	733.4	29	91.106	660.52	4.587	.2180
¾	2.356	.442	.0031	325.9	30	94.248	706.86	4.909	.2037
1	3.142	.785	.0055	183.3	31	97.389	754.77	5.241	.1908
1 ¼	3.927	1.227	.0085	117.3	32	100.531	804.25	5.585	.1790
1 ½	4.712	1.767	.0123	81.489	33	103.673	855.30	5.940	.1684
2	6.283	3.142	.0218	45.837	34	106.814	907.92	6.305	.1586
2 ½	7.854	4.909	.0341	29.336	35	109.956	962.11	6.681	.1497
3	9.425	7.069	.0491	20.372	36	113.097	1,017.90	7.069	.1415
3 ½	10.996	9.621	.0668	14.967	37	116.239	1,075.20	7.467	.1339
4	12.566	12.566	.0873	11.459	38	119.381	1,134.10	7.876	.1270
4 ½	14.137	15.904	.1104	9.054	40	125.664	1,256.60	8.726	.1146
5	15.708	19.635	.1364	7.334	42	131.947	1,385.40	9.621	.1039
5 ½	17.279	23.759	.1650	6.061	44	138.230	1,520.50	10.559	.0947
6	18.850	28.274	.1963	5.093	46	144.513	1,661.90	11.541	.0866
6 ½	20.420	33.183	.2304	4.340	48	150.796	1,809.60	12.567	.0796
7	21.991	38.485	.2673	3.742	50	157.080	1,963.50	13.635	.0733
7 ½	23.562	44.179	.3068	3.259	52	163.363	2,123.70	14.748	.0678
8	25.133	50.215	.3491	2.865	54	169.646	2,290.20	15.904	.0629
8 ½	26.704	56.745	.3941	2.538	56	175.929	2,463.00	17.104	.0585
9	28.274	63.617	.4418	2.264	58	182.212	2,642.10	18.348	.0545
9 ½	29.845	70.882	.4922	2.032	60	188.496	2,827.40	19.635	.0509
10	31.416	78.540	.5454	1.833	62	194.779	3,019.10	20.966	.0477
11	34.558	95.033	.6600	1.515	64	201.062	3,217.00	22.340	.0448
12	37.699	113.100	.7854	1.273	66	207.345	3,421.20	23.758	.0421
13	40.841	132.730	.9217	1.085	68	213.628	3,631.70	25.220	.0397
14	43.982	153.940	1.0690	.9354	70	219.911	3,848.50	26.726	.0374
15	47.124	176.710	1.2270	.8149	72	226.195	4,071.50	28.274	.0354
16	50.265	201.060	1.3960	.7162	74	232.478	4,300.80	29.867	.0335
17	53.407	226.980	1.5760	.6344	76	238.761	4,536.50	31.503	.0317
18	56.549	254.470	1.7670	.5659	78	245.044	4,778.40	33.183	.0301
19	59.690	283.530	1.9690	.5079	80	251.327	5,026.50	34.906	.0286
20	62.832	314.160	2.1820	.4584	82	257.611	5,281.00	36.674	.0273
21	65.973	346.360	2.4050	.4158	84	263.894	5,541.80	38.485	.0260
22	69.115	380.130	2.6400	.3788	86	270.177	5,808.80	40.339	.0248
23	72.257	415.480	2.8850	.3466	88	276.460	6,082.10	42.237	.0237
24	75.398	452.390	3.1420	.3183	90	282.743	6,361.70	44.178	.0226
25	78.540	490.870	3.4090	.2934	92	289.027	6,647.60	46.164	.0217
26	81.681	530.930	3.6870	.2712	94	295.310	6,939.80	48.193	.0207
27	84.823	572.560	3.9760	.2515	96	301.593	7,238.20	50.265	.0199

Properties of Various Gases (at 60° F, 14.7 psia)

Gas	Symbol	C_p/C_v	Sp. gr. air = 1.00	Mol. wt.	Lb per cu ft	Cu ft per lb	Boiling point at atmos. press., °F	Crit. temp., °F	Crit. press. psia
Acetylene	C_2H_2	1.30	0.91	26.02	0.07	14.53	–118	96	910
Air	–	1.41	1.00	28.98	0.08	13.06	–317	–221	546
Ammonia	NH_3	1.32	0.60	17.03	0.05	22.18	–28	270	1,638
Argon	A	1.67	1.38	39.94	0.11	9.46	–302	–187	705
Benzene	C_6H_6	1.08	2.69	78.05	0.21	4.85	176	551	700
Butane	C_4H_{10}	1.11	2.07	58.08	0.15	6.51	31	307	528
Butylene	C_4H_8	1.11	1.94	56.06	0.15	6.75	21	291	621
Carbon dioxide	CO_2	1.30	1.53	44.00	0.12	8.59	–109	88	1,072
Carbon disulfide	CS_2	1.20	2.63	76.12	0.20	4.97	115	523	1,116
Carbon monoxide	CO	1.40	0.97	28.00	0.07	13.50	–313	–218	514
Carbon tetrachloride	CCl_{14}	1.18	5.33	153.83	0.41	2.46	170	541	661
Carbureted water gas	–	1.35	0.41	–	–	–	–	–	–
Chlorine	Cl_2	1.33	2.49	70.91	0.19	5.33	–30	291	1,118
Dichloromethane	CH_2Cl_2	1.18	3.01	84.93	0.22	4.46	105	421	1,490
Ethane	C_2H_6	1.22	1.05	30.05	0.08	12.59	–127	90	717
Ethyl Chloride	C_2H_5Cl	1.13	2.37	64.50	0.17	5.87	54	370	764
Ethylene	C_2H_4	1.22	0.97	28.03	0.07	13.50	–155	50	747
Flue gas	–	1.40	–	–	–	–	–	–	–
Freon (F-12)	CCl_2F_2	1.13	4.52	120.91	0.32	3.13	–21	233	580
Helium	He	1.66	0.14	4.00	0.01	94.51	–452	–450	33
Hexane	C_6H_{14}	1.08	2.74	86.11	0.23	4.39	156	454	433
Hexylene	C_6H_{14}	–	2.92	84.09	0.22	4.50	–	–	–
Hydrogen	H_2	1.41	0.07	2.02	0.01	188.62	–423	–400	188
Hydrogen chloride	HCl	1.41	1.27	36.46	0.10	10.37	–121	124	1,198
Hydrogen sulfide	H_2S	1.30	1.19	34.08	0.09	11.10	–75	212	1,306
Isobutane	C_4H_{10}	1.11	2.02	58.08	0.15	6.51	14	273	543
Isopentane	C_5H_{12}	1.07	2.50	72.09	0.19	5.25	82	370	483
Methane	CH_4	1.32	0.55	16.03	0.04	23.63	–258	–116	672
Methyl Chloride	CH_3Cl	1.20	1.79	50.48	0.13	7.49	–11	289	966
Naphtalene	$C_{10}H_8$	–	4.42	128.06	0.34	2.95	–	–	–
Natural gas (1) (app. avg.)	–	1.27	0.67	19.46	0.05	19.45	–	–80	670
Neon	Ne	1.64	0.70	20.18	0.05	18.78	–410	–380	389
Nitric oxide	NO	1.40	1.04	30.01	0.08	12.61	–240	–137	954
Nitrogen	N_2	1.41	0.97	28.02	0.07	13.46	–320	–232	492
Nitrous oxide	N_2O	1.31	1.53	44.02	0.12	8.60	–129	98	1,053
Oxygen	O_2	1.40	1.11	32.00	0.08	11.82	–297	–182	730
Pentane	C_5H_{12}	1.06	2.47	72.09	0.19	5.25	97	387	485
Phenol	C_6H_5OH	–	3.27	94.05	0.25	4.02	360	786	889
Propane	C_3H_8	1.15	1.56	44.06	0.12	8.59	–48	204	632
Propylene	C_3H_6	1.15	1.45	42.05	0.11	9.00	–52	198	661
Refinery gas (1) (app. avg.)	–	1.20	–	–	–	–	–	–	–
Sulfur dioxide	SO_2	1.26	2.26	64.06	0.17	5.90	14	315	1,141
Water vapor (steam)	H_2O	1.33(2)	0.62	18.02	0.05	21.00	212	706	3,206

(1) To obtain exact characteristics of natural gas and refinery gas, the exact constituents must be known.
(2) This value is given at 212°F. All others are at 60°F. Authorities differ slightly; hence all data are average results.

Temperature Conversion Table

Note: The temperature to be converted is entered in column X.
To obtain a reading in °C use the left column;
for conversion to °F use the right column.

°F = 9/5*(°C+ 40)- 40 °R = °F + 459.7

°C = 5/9*(°F+ 40)- 40 °K = °C + 273.2

°C	X	°F	°C	X	°F
-273	-459.4		48.9	120	248
-268	-450		54.4	130	266
-240	-400		60.0	140	284
-212	-350		65.6	150	302
-184	300		71.1	160	320
-157	-250	-418	76.7	170	338
-129	-200	-328	82.2	180	356
-101	-150	-238	87.8	190	374
-73.3	-100	-148	93.3	200	392
-45.6	-50	-58	98.9	210	410
-42.8	-45	-49	100	212	414
-40.0	-40	-40	104	220	428
-37.2	-35	-31	110	230	446
-34.4	-30	-22	116	240	464
-31.7	-25	-13	121	250	482
-28.9	-20	-4	149	300	572
-26.1	-15	5	177	350	662
-23.2	-10	14	204	400	752
-20.6	-5	23	232	450	842
-17.8	0	32	260	500	932
-15.0	5	41	288	550	1,022
-12.2	10	50	316	600	1,112
-9.4	15	59	343	650	1,202
-6.7	20	68	371	700	1,292
-3.9	25	77	399	750	1,382
-1.1	30	86	427	800	1,472
0.0	32	89	454	850	1,562
1.7	35	95	482	900	1,652
4.4	40	104	510	950	1,742
7.2	45	113	538	1,000	1,832
10.0	50	122	566	1,050	1,922
12.8	55	131	593	1,100	2,012
15.6	60	140	621	1,150	2,102
18.3	65	149	649	1,200	2,192
21.1	70	158	677	1,250	2,282
23.9	75	167	704	1,300	2,372
26.7	80	176	732	1,350	2,462
29.4	85	185	762	1,400	2,552
32.2	90	194	788	1,450	2,642
35.0	95	203	816	1,500	2,732
37.8	100	212	871	1,600	2,912
40.6	105	221	927	1,700	3,092
43.3	110	230	982	1,800	3,272

Conversion Factors

Length Units

U.S. System				Metric System		
mil	in	ft	yd	µm	mm	m
1	.001	.00008333	.00002778	25.4	.0254	.0000254
1,000	1	.083333	.027778	25,400	25.4	.0254
12,000	12	1	.3333	304,800	304.8	.3048
36,000	36	3	1	914,400	914.4	.9144
.03937	.00003937	.000003281	1,760	1	.001	.000001
39.37	.03937	0.003281	.000001094	1,000	1	.001
39,370	39.37	3.2808	.001094	1,000,000	1,000	1

1 mile	=	5,280 ft	=	1.609.3 m
1 kilometer	=	1,000 m	=	0,62137 mi
1 angstrom	=	.0000000001 m	=	0.000000003937 in
1 hand	=	4 in	=	101.6 mm
1 fathom	=	6 ft	=	1.8288 m
1 rod	=	5.5 yd	=	5.0292 m
1 furlong	=	220 yd	=	201.17 m
1 naut mi	=	1.1508 mi	=	1.8520 km
1 league	=	3 mi	=	4.828 km

Area Units

U.S. System				Metric System		
in^2	ft^2	yd^2	acres	mm^2	m^2	km^2
1	0.007	.0007716	.0000001594	645.16	.0006452	6.452×10^{-10}
144	1	.11111	.00002296	92,903	.092903	.0000000929
1,296	9	1	.0002066	836,127	.83613	.0000008361
6,272,640	43,560	4,840	1	4,046,856,422	4,047	.004047
.001550	0.00001	.000001196	2.471×10^{-10}	1	.000001	10^{-12}
1,550	11	1.196	.02471	1,000,000	1	.000001
1,550,003,100	10,764,000	1,196,000	247.1	10^{-12}	1,000,000	1

Conversion Factors

Volume Units

U.S. System				Metric System		
gal	bbl	in^3	ft^3	cm^3 or mL	L	m^3
1	.03274	231	.13368	3,785.4	3.7854	.003785
30.55	1	7,056	4.0833	115,630	115.63	.11563
.004329	.0001417	1	.0005787	16.387	.01639	.00001639
7.4805	.2449	1,728	1	28,317	28.317	.02832
.0002642	.000008648	.06102	.00003531	1	.001	.000001
.26417	.008648	61.024	.03531	1,000	1	.001
264.17	8.6484	61024	35.315	1,000,000	1,000	1

1 gallon (Imp.)	=	1.20095 gal (US)	=	4.5461 L
1 barrel (US)	=	31.5 gal (US)	=	119.24 L
1 hogshead	=	63 gal (US)	=	238.48 L
1 cu ft	=	volume of 62.427 lb of water at 4°C		

Velocity Units

U.S. System					Metric System		
in/sec	fps	fpm	mph	knots	cm/s	kph	m/s
1	0.08333	5	.05682	.04937	2.54	.09144	.0254
12.00	1	60	.68182	.59247	30.48	1.0973	.3048
0.20	.01667	1	.01136	.009875	.5080	.01829	.00508
17.60	1.4667	88	1	.86896	44.704	1.6093	.44704
20.25	1.6878	101.27	1.1508	1	51.444	1.852	.51444
0.39	.03281	1.9685	.02237	.01944	1	.036	.01
10.94	.91134	54.681	.62137	.53996	27.7778	1	.27778
39.37	3.2808	196.85	2.2369	1.9438	100	3.6	1

Conversion Factors

Pressure or Stress Units

U.S. System					Metric System			
psf	in H_2O	in Hg	psi	atm	dy/cm^2	Pa	mm H_2O	mm Hg
1	.19255	.01414	.006944	.0004725	478.8	47.88	4.8908	.35913
5.1934	1	.07343	.03607	.002454	2486.6	248.66	25.4	1.8651
70.726	13.619	1	.49115	.03342	33864	3386.4	345.91	25.4
144	27.728	2.036	1	.06805	68948	6894.8	704.28	51.715
2116.2	407.48	29.921	14.696	1	1013000	101300	10350	760
.002089	.0004022	.00002953	.0000145	.0000009869	1	.1	.76131	.000750
.02089	.004022	.0002953	.000145	.000009869	10	1	7.6131	.007501
.20446	.03937	.002891	.00142	.00009662	1.3135	.13135	1	.07343
2.7845	.53616	.03937	.01934	.001316	1333.2	133.32	13.619	1

Note: H_2O @ 68°F or 20°C and 62.3205 lb/ft^3 Hg @ 32°F or 0°C and 848.714 lb/ft^3

1 bar = .9869 atm = 10,000 Pa

1 ksi = 1,000 psi = 6.8947 MPa

1 Pa = 1N/m^2

1 torr = 1mm Hg

Work, Energy and Heat Units

U.S. System			Metric System		
ft - lb	Btu	hp - hr	J	cal	kW - hr
1	.001285	.000000505	1.3558	.32383	.0000003766
778.17	1	.000393	1055.1	252.0	.0002931
1,980,000	2,544.4	1	2,685,000	641,200	.7457
.73756	.0009478	.0000003725	1	.23885	.0000002778
3.088	.003968	.00000156	4.1868	1	.000001163
2,655,000	3,412.1	1.341	3,600,000	859,800	1

1 J = 1 W - s = 1 N - m

1 erg = 1 dyne - cm = 10^{-7} J

1 therm = 10,000 Btu

1 kcal = 1,000 cal

Conversion Factors

Volume Flow Rate Units

U.S. System		Metric System	
gpm	cfm	m^3/hr	m^3/s
1	0.13368	0.22712	0.00006309
7.4805	1	1.6990	0.0004719
4.4029	0.58858	1	0.0002778
15,850	2118.9	3,600	1

Density Units

U.S. System		Metric System		
lb/ft^3	lb/in^3	g/m^3	g/cm^3	kg/m^3
1	0.0005787	16,018	0.016018	16.018
1,728	1	27,680,000	27.680	27,680
0.00006243	0.00000003613	1	0.000001	0.001
62.428	0.03613	1,000,000	1	1,000
0.062428	0.00003613	1,000	.001	1

1 gr	=	64.799 mg		
1 dr	=	27.344 gr	=	1.7719 g
1 oz	=	16 dr	=	28.350 g
1 lb	=	16 oz	=	453.60 g
1 lb	=	7,000 gr		
1 gr/ft^3	=	2,2884 g/m^3		

Air Density Ratios — Corrections for Pressure Drop

					Altitude in Feet Above Sea Level													
					0	500	1000	1500	2000	2500	3000	4000	5000	6000	7000	8000	9000	10000
	Pressure in psia																	
Air Temp.	30	25	20	15	14.696	14.43	14.18	14.16	13.66	13.42	13.17	12.69	12.23	11.78	11.34	10.91	10.51	10.11
	Pressure in Inches of Mercury																	
	61.08	50.90	40.72	30.54	29.92	29.38	28.86	28.83	27.82	27.32	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
°F	Air density ratios — Specific Gravity of Standard Air at 70°F = 1.000 = 0.075 lb/ft^3																	
-40	2.577	2.147	1.718	1.288	1.262	1.239	1.217	1.216	1.174	1.152	1.131	1.090	1.050	1.012	0.974	0.937	0.902	0.868
-30	2.517	2.097	1.678	1.258	1.233	1.211	1.189	1.188	1.146	1.126	1.105	1.065	1.026	0.988	0.951	0.916	0.881	0.848
-20	2.459	2.049	1.640	1.230	1.205	1.183	1.162	1.161	1.120	1.100	1.080	1.040	1.003	0.966	0.930	0.895	0.861	0.829
-10	2.405	2.004	1.603	1.202	1.178	1.157	1.136	1.135	1.095	1.076	1.056	1.017	0.980	0.944	0.909	0.875	0.842	0.810
0	2.352	1.960	1.568	1.176	1.152	1.132	1.111	1.110	1.071	1.052	1.033	0.995	0.959	0.924	0.889	0.856	0.824	0.793
10	2.302	1.918	1.535	1.151	1.128	1.107	1.088	1.087	1.049	1.030	1.011	0.974	0.939	0.904	0.870	0.838	0.806	0.776
20	2.254	1.878	1.503	1.127	1.104	1.084	1.065	1.064	1.027	1.008	0.990	0.954	0.919	0.885	0.852	0.820	0.789	0.760
30	2.208	1.840	1.472	1.104	1.082	1.062	1.043	1.042	1.006	0.988	0.970	0.934	0.900	0.867	0.835	0.803	0.773	0.744
40	2.164	1.803	1.443	1.082	1.060	1.041	1.022	1.021	0.986	0.968	0.950	0.915	0.882	0.850	0.818	0.787	0.758	0.729
50	2.121	1.768	1.414	1.061	1.039	1.020	1.002	1.001	0.966	0.949	0.932	0.898	0.865	0.833	0.802	0.772	0.743	0.715
60	2.081	1.734	1.387	1.040	1.019	1.001	0.983	0.982	0.948	0.931	0.914	0.880	0.848	0.817	0.787	0.757	0.729	0.701
70	2.041	1.701	1.361	1.021	1.000	0.982	0.965	0.964	0.930	0.913	0.896	0.864	0.832	0.801	0.772	0.743	0.715	0.688
80	2.004	1.670	1.336	1.002	0.981	0.964	0.947	0.946	0.913	0.896	0.880	0.848	0.817	0.787	0.757	0.729	0.702	0.675
90	1.967	1.639	1.311	0.984	0.964	0.946	0.929	0.929	0.896	0.880	0.864	0.832	0.802	0.772	0.744	0.716	0.689	0.663
100	1.932	1.610	1.288	0.966	0.946	0.929	0.913	0.912	0.880	0.864	0.848	0.817	0.788	0.759	0.730	0.703	0.677	0.651
120	1.865	1.554	1.244	0.933	0.914	0.897	0.881	0.880	0.850	0.834	0.819	0.789	0.760	0.732	0.705	0.679	0.653	0.628
140	1.803	1.503	1.202	0.902	0.883	0.867	0.852	0.851	0.821	0.807	0.792	0.763	0.735	0.708	0.682	0.656	0.631	0.608
160	1.745	1.454	1.163	0.872	0.855	0.839	0.824	0.824	0.795	0.780	0.766	0.738	0.711	0.685	0.660	0.635	0.611	0.588
180	1.690	1.409	1.127	0.845	0.828	0.813	0.799	0.798	0.770	0.756	0.742	0.715	0.689	0.664	0.639	0.615	0.592	0.570
200	1.639	1.366	1.093	0.820	0.803	0.788	0.774	0.774	0.747	0.733	0.720	0.693	0.668	0.644	0.620	0.596	0.574	0.552
225	1.579	1.316	1.053	0.790	0.774	0.760	0.746	0.745	0.719	0.706	0.693	0.668	0.644	0.620	0.597	0.575	0.553	0.532
250	1.524	1.270	1.016	0.762	0.746	0.733	0.720	0.719	0.694	0.681	0.669	0.645	0.621	0.598	0.576	0.554	0.534	0.513
275	1.472	1.226	0.981	0.736	0.721	0.708	0.695	0.695	0.670	0.658	0.646	0.623	0.600	0.578	0.556	0.535	0.515	0.496
300	1.423	1.186	0.949	0.712	0.697	0.685	0.673	0.672	0.648	0.637	0.625	0.602	0.580	0.559	0.538	0.518	0.498	0.480

Properties of Saturated Steam with Temperature

Temp.	Pressure	Density	Specific Volume			Enthalpy		
			Liquid	Evap.	Vapor	Liquid	Evap.	Vapor
°F	psia	lb/ft³	ft³/lb	ft³/lb	ft³/lb	Btu/lb	Btu/lb	Btu/lb
32	0.0885	0.0003	0.01602	3306	3306	0.00	1075.8	1075.8
35	0.1000	0.0003	0.01602	2947	2947	3.02	1074.1	1077.1
40	0.1217	0.0004	0.01602	2444	2444	8.05	1071.3	1079.3
45	0.1475	0.0005	0.01602	2036	2036	13.06	1068.4	1081.5
50	0.1781	0.0006	0.01603	1703	1703	18.07	1065.6	1083.7
55	0.2141	0.0007	0.01603	1431	1431	23.07	1062.7	1085.8
60	0.2563	0.0008	0.01604	1207	1207	28.06	1059.9	1088.0
65	0.3056	0.0010	0.01605	1021	1021	33.05	1057.1	1090.2
70	0.3631	0.0012	0.01606	867.8	867.9	38.04	1054.3	1092.3
75	0.4298	0.0014	0.01607	740.0	740.0	43.03	1051.5	1094.5
80	0.5069	0.0016	0.01608	633.1	633.1	48.01	1048.6	1096.6
90	0.6982	0.0021	0.01610	468.0	468.0	57.99	1042.9	1100.9
100	0.9492	0.0029	0.01613	350.3	350.4	67.97	1037.2	1105.2
110	1.275	0.0038	0.01617	265.3	265.4	77.94	1031.6	1109.5
120	1.692	0.0049	0.01620	203.25	203.27	87.92	1025.8	1113.7
130	2.222	0.0064	0.01625	157.32	157.34	97.90	1020.0	1117.9
140	2.889	0.0081	0.01629	122.99	123.01	107.89	1014.1	1122.0
150	3.718	0.0103	0.01634	97.06	97.07	117.89	1008.2	1126.1
160	4.741	0.0129	0.01639	77.27	77.29	127.89	1002.3	1130.2
170	5.992	0.0161	0.01645	62.04	62.06	137.90	996.3	1134.2
180	7.510	0.0199	0.01651	50.21	50.23	147.92	990.2	1138.1
190	9.339	0.0244	0.01657	40.94	40.96	157.95	984.1	1142.0
200	11.526	0.0297	0.01663	33.62	33.64	167.99	977.9	1145.9
210	14.123	0.0359	0.01670	27.8	27.82	178.05	971.6	1149.7
212	14.696	0.0373	0.01672	26.78	26.80	180.07	970.3	1150.4
220	17.186	0.0432	0.1677	23.13	23.15	188.13	965.2	1153.4
230	20.780	0.0516	0.01684	19.36	19.382	198.23	958.8	1157.0
240	24.969	0.0613	0.01692	16.31	16.323	208.34	952.2	1160.5
250	29.825	0.0724	0.01700	13.80	13.821	218.48	945.5	1164.0
260	35.429	0.0850	0.01709	11.75	11.763	228.64	938.7	1167.3
270	41.858	0.0994	0.01717	10.04	10.061	238.84	931.8	1170.6
280	49.203	0.1157	0.01726	8.628	8.645	249.06	924.7	1173.8
290	57.556	0.1340	0.01735	7.444	7.461	259.31	917.5	1176.8
300	67.013	0.1547	0.01745	6.449	6.466	269.59	910.1	1179.7
320	89.660	0.2035	0.01765	4.896	4.914	290.28	894.9	1185.2
340	118.01	0.2640	0.01787	3.770	3.788	311.13	879.0	1190.1
360	153.04	0.3382	0.01811	2.939	2.957	332.18	862.2	1194.4
380	195.77	0.4283	0.01836	2.317	2.335	353.45	844.6	1198.1
400	247.31	0.5368	0.01864	1.845	1.863	374.97	826.0	1201.0
420	308.83	0.6667	0.01894	1.481	1.500	396.77	806.3	1203.1
440	381.59	0.8217	0.01926	1.198	1.217	418.90	785.4	1204.3
460	466.9	1.0056	0.0196	0.9748	0.9944	441.4	763.2	1204.6
480	566.1	1.2237	0.0200	0.7972	0.8172	464.4	739.4	1203.7
500	680.8	1.4817	0.0204	0.6545	0.6749	487.8	713.9	1201.7
520	812.4	1.7876	0.0209	0.5385	0.5594	511.9	686.4	1198.2
540	962.5	2.1510	0.0215	0.4434	0.4649	536.6	656.6	1193.2
560	1133	2.5853	0.0221	0.3647	0.3868	562.2	624.2	1186.4
580	1326	3.1085	0.0228	0.2989	0.3217	588.9	588.4	1177.3
600	1543	3.7481	0.0236	0.2432	0.2668	617.0	548.5	1165.5
620	1787	4.5434	0.0247	0.1995	0.2201	646.7	503.6	1150.3
640	2060	5.5617	0.0260	0.1538	0.1798	678.6	452.0	1130.5
660	2365	6.9348	0.0278	0.1165	0.1442	714.2	390.2	1104.4
680	2708	8.9686	0.0305	0.0810	0.1115	757.3	309.9	1067.2
700	3094	13.1406	0.0369	0.0392	0.0761	823.3	172.1	995.4
705.4	3206.2	19.8807	0.0503	0	0.0503	902.7	0	902.7

Adapted from the data of J.H. Keenan and F.G. Keyes, *Thermodynamic Properties of Steam*, John Wiley & Sons, Inc., 1938.

Properties of Saturated Steam with Pressure

Pressure	Temp.	Density	Specific Volume		Enthalpy		
			Liquid	Vapor	Liquid	Evap.	Vapor
psia	°F	lb/ft³	ft³/lb	ft³/lb	Btu/lb	Btu/lb	Btu/lb
10	193.21	0.0260	0.01659	38.42	161.17	982.10	1,143.3
11	197.75	0.0285	0.01662	35.14	165.73	979.30	1,145.0
12	201.96	0.0309	0.01665	32.40	169.96	976.60	1,146.6
13	205.88	0.0333	0.01667	30.06	173.91	974.20	1,148.1
14	209.56	0.0357	0.01670	28.04	177.61	971.90	1,149.5
14.696	212.00	0.0373	0.01672	26.80	180.07	970.30	1,150.4
15	213.03	0.0380	0.01672	26.29	181.11	969.70	1,150.8
16	216.32	0.0404	0.01674	24.75	184.42	967.60	1,152.0
17	219.44	0.0428	0.01677	23.39	187.56	965.50	1,153.1
18	222.41	0.0451	0.01679	22.17	190.56	963.60	1,154.2
19	225.24	0.0474	0.01681	21.08	193.42	961.90	1,155.3
20	227.96	0.0498	0.01683	20.09	196.16	960.10	1,156.3
21	230.57	0.0521	0.01685	19.19	198.79	958.40	1,157.2
22	233.07	0.0544	0.01687	18.38	201.33	956.80	1,158.1
23	235.49	0.0567	0.01689	17.63	203.78	955.20	1,159.0
24	237.82	0.0590	0.01691	16.94	206.14	953.70	1,159.8
25	240.07	0.0613	0.01692	16.30	208.42	952.10	1,160.6
26	242.25	0.0636	0.01694	15.72	210.62	950.70	1,161.3
27	244.36	0.0659	0.01696	15.17	212.75	949.30	1,162.0
28	246.41	0.0682	0.01698	14.66	214.83	947.90	1,162.7
29	248.40	0.0705	0.01699	14.19	216.86	946.50	1,163.4
30	250.33	0.0727	0.01701	13.75	218.82	945.30	1,164.1
35	259.28	0.0834	0.01708	11.99	227.91	939.20	1,167.1
40	267.25	0.0953	0.01715	10.50	236.03	933.70	1,169.7
45	274.44	0.1064	0.01721	9.401	243.36	928.60	1,172.0
50	281.01	0.1174	0.01727	8.515	250.09	924.00	1,174.1
55	287.07	0.1284	0.01732	7.787	256.30	919.60	1,175.9
60	292.71	0.1394	0.01738	7.175	262.09	915.50	1,177.6
65	297.97	0.1503	0.01743	6.655	267.50	911.60	1,179.1
70	302.92	0.1611	0.01748	6.206	272.61	907.90	1,180.6
75	307.60	0.1719	0.01753	5.816	277.43	904.50	1,181.9
80	312.03	0.1827	0.01757	5.472	282.02	901.10	1,183.1
85	316.25	0.1935	0.01761	5.168	286.39	897.80	1,184.2
90	320.27	0.2042	0.01766	4.896	290.56	894.70	1,185.3
95	324.12	0.2150	0.01770	4.652	294.56	891.70	1,186.2
100	327.81	0.2256	0.01774	4.432	298.40	888.80	1,187.2
110	334.77	0.2470	0.01782	4.049	305.66	883.20	1,188.9
120	341.25	0.2682	0.01789	3.728	312.44	877.90	1,190.4
130	347.32	0.2894	0.01796	3.455	318.81	872.90	1,191.7
140	353.02	0.3106	0.01802	3.220	324.82	868.20	1,193.0
150	358.42	0.3317	0.01809	3.015	330.51	863.60	1,194.1
175	370.76	0.3843	0.01824	2.602	343.60	852.80	1,196.4
200	381.79	0.4371	0.01839	2.288	355.36	843.00	1,198.4
225	391.79	0.4897	0.01852	2.042	366.09	833.80	1,199.9
250	400.95	0.5423	0.01865	1.844	376.00	825.10	1,201.1
300	417.33	0.6481	0.01890	1.543	393.84	809.00	1,202.8
350	431.72	0.7541	0.01913	1.326	409.69	794.20	1,203.9
400	444.59	0.8613	0.01930	1.161	424.00	780.50	1,204.5
500	467.01	1.0778	0.01970	0.9278	449.40	755.00	1,204.4
600	486.21	1.2990	0.02010	0.7698	471.60	731.60	1,203.2
700	503.10	1.5258	0.02050	0.6554	491.50	709.70	1,201.2
800	518.23	1.7584	0.02090	0.5687	509.70	688.90	1,198.6
900	531.98	1.9976	0.02120	0.5006	526.60	668.80	1,195.4
1,000	544.61	2.2442	0.02160	0.4456	542.40	649.40	1,191.8
1,100	556.31	2.4994	0.02200	0.4001	557.40	630.40	1,187.8
1,200	567.22	2.7632	0.02230	0.3619	571.70	611.70	1,183.4

Adapted from the data of J.H. Keenan and F.G. Keyes, *Thermodynamic Properties of Steam*, John Wiley & Sons, Inc., 1938.

Sheet Metal and Wire Gauges and Weights

Gauge	Manfrs' Std. Gauge[1] Plate & Sheet			U.S. Std. Gauge[2] Plate & Sheet			U.S. Steel Wire Gauge[3] Wire		
	Thickness	Steel	Galvanized	Thickness	Steel	316SS	Diameter	Steel	Copper
Number	inch	lb/ft²	lb/ft²	inch	lb/ft²	lb/ft²	inch	lb/100 ft	lb/100 ft
0000000	½	20.4	–	½	20.00	20.66	0.4900	64.11	72.63
000000	15/32	19.12	–	15/32	18.75	19.37	0.4615	56.87	64.43
00000	7/16	17.85	–	7/16	17.50	18.08	0.4305	49.48	56.06
0000	13/32	16.57	–	13/32	16.25	16.79	0.3938	41.41	46.91
000	⅜	15.3	–	⅜	15.00	15.50	0.3625	35.09	39.75
00	11/32	14.02	–	11/32	13.75	14.21	0.3310	29.25	33.14
0	5/16	12.75	–	5/16	12.50	12.91	0.3065	25.08	28.42
1	9/32	11.47	–	9/32	11.25	11.62	0.2830	21.38	24.23
2	¼	10.2	–	0.2656	10.62	10.98	0.2625	18.40	20.84
3	0.2391	10.0	10.16	0.2500	10.00	10.33	0.2437	15.86	17.97
4	0.2242	9.375	9.531	0.2344	9.375	9.686	0.2253	13.55	15.35
5	0.2092	8.75	8.906	0.2188	8.750	9.040	0.2070	11.44	12.96
6	0.1943	8.125	8.281	0.2031	8.125	8.395	0.1920	9.840	11.15
7	0.1793	7.50	7.656	0.1875	7.500	7.749	0.1770	8.360	9.480
8	0.1644	6.875	7.031	0.1719	6.875	7.103	0.1620	7.010	7.940
9	0.1495	6.25	6.406	0.1562	6.250	6.457	0.1483	5.870	6.650
10	0.1345	5.625	5.781	0.1406	5.625	5.812	0.1350	4.870	5.510
11	0.1196	5.00	5.156	0.1250	5.000	5.166	0.1205	3.880	4.390
12	0.1046	4.375	4.531	0.1094	4.375	4.520	0.1055	2.970	3.370
13	0.0897	3.75	3.906	0.0938	3.750	3.874	0.0915	2.240	2.530
14	0.0747	3.125	3.281	0.0781	3.125	3.229	0.0800	1.710	1.940
15	0.0673	2.813	2.969	0.0703	2.812	2.906	0.0720	1.380	1.570
16	0.0598	2.50	2.656	0.0625	2.500	2.583	0.0625	1.040	1.180
17	0.0538	2.250	2.406	0.0562	2.250	2.325	0.0540	0.779	0.882
18	0.0478	2.00	2.156	0.0500	2.000	2.066	0.0475	0.602	0.683
19	0.0418	1.75	1.906	0.0438	1.750	1.808	0.0410	0.449	0.509
20	0.0359	1.50	1.656	0.0375	1.500	1.550	0.0348	0.323	0.366
21	0.0329	1.375	1.531	0.0344	1.375	1.421	0.0318	0.269	0.305
22	0.0299	1.25	1.406	0.0312	1.250	1.291	0.0286	0.218	0.247
23	0.0269	1.125	1.281	0.0281	1.125	1.162	0.0258	0.178	0.201
24	0.0239	1.00	1.156	0.0250	1.000	1.033	0.0230	0.141	0.160
25	0.0209	0.875	1.031	0.0219	0.875	0.904	0.0204	0.111	0.126
26	0.0179	0.75	0.906	0.0188	0.750	0.775	0.0181	0.087	0.099
27	0.0164	0.688	0.844	0.0172	0.688	0.710	0.0173	0.080	0.091
28	0.0149	0.625	0.781	0.0156	0.625	0.646	0.0162	0.070	0.079
30	0.0120	0.50	0.656	0.0125	0.500	0.517	0.0140	0.052	0.059

Notes:

1. Manufacturers' Standard Gauges for Steel Products are thickness gauges. Weights are based on steel of 0.2833 lb/cu in and galvanizing of 0.0174 oz/sq in. Galvanized sheets are 0.0037" thicker than uncoated sheets. All thicknesses of ¼" or more are plates and are unrelated to gauge numbers. Refer to manufacturers' data sheets for tolerances and other options.
2. United States Standard Gauges for Iron & Steel Products are weight gauges based on wrought iron with a density of 0.2778 lb/cu in. Weights for stainless steel are based on densities of 0.287 lb/cu in. All thicknesses of 3/16" or more are plates and thicknesses of 5/16" or more are unrelated to gauge numbers.
3. United States Steel Wire Gauges are identical American Steel & Wire Gauges, Roebling Wire Gauges, and Washburn & Moen Wire Gauges. Weights are based on steel with a density of 0.2833 lb/cu in.

Weights of Sheet and Plate Steel — Pounds/Foot Length

	Width of Carbon Steel							
Thickness	12	24	36	48	60	72	84	96
0.1875	7.65	15.30	22.95	30.60	38.25	45.90	53.55	61.20
0.2500	10.20	20.40	30.60	40.80	51.00	61.20	71.40	81.60
0.3125	12.75	25.50	38.25	51.00	63.75	76.50	89.25	102.0
0.3750	15.30	30.60	45.90	61.20	76.50	91.80	107.1	122.4
0.4375	17.85	35.70	53.55	71.40	89.25	107.1	125.0	142.8
0.5000	20.40	40.80	61.20	81.60	102.0	122.4	142.8	163.2
0.5625	22.95	45.90	68.85	91.80	114.8	137.7	160.7	183.6
0.6250	25.50	51.00	76.50	102.0	127.5	153.0	178.5	204.0
0.6875	28.05	56.10	84.15	112.2	140.3	168.3	196.4	224.4
0.7500	30.60	61.20	91.80	122.4	153.0	183.6	214.2	244.8
0.8125	33.15	66.30	99.45	132.6	165.8	198.9	232.1	265.2
0.8750	35.70	71.40	107.10	142.8	178.5	214.2	249.9	285.6
0.9375	38.25	76.50	114.8	153.0	191.3	229.5	267.8	306.0
1.0000	40.80	81.60	122.4	163.2	204.0	244.8	285.6	326.4
1.2500	51.00	102.0	153.0	204.0	255.0	306.0	357.0	408.0
1.5000	61.20	122.4	183.6	244.8	306.0	367.2	428.4	489.6
1.7500	71.40	142.8	214.2	285.6	357.0	428.4	499.8	571.2
2.0000	81.60	163.2	244.8	326.4	408.0	489.6	571.2	652.8
2.2500	91.80	183.6	275.4	367.2	459.0	550.8	642.6	734.4
2.5000	102.0	204.0	306.0	408.0	510.0	612.0	714.0	816.0
2.7500	112.2	224.4	336.6	448.8	561.0	673.2	785.4	897.6
3.0000	122.4	244.8	367.2	489.6	612.0	734.4	856.8	979.2

	Width of Stainless Steel							
Thickness	12	24	36	48	60	72	84	96
0.1875	7.75	15.50	23.25	31.00	38.75	46.49	54.24	61.99
0.2500	10.33	20.66	31.00	41.33	51.66	61.99	72.32	82.66
0.3125	12.92	25.83	38.75	51.66	64.58	77.49	90.41	103.3
0.3750	15.50	31.00	46.49	61.99	77.49	92.99	108.5	124.0
0.4375	18.08	36.16	54.24	72.32	90.41	108.5	126.6	144.6
0.5000	20.66	41.33	61.99	82.66	103.3	124.0	144.6	165.3
0.5625	23.25	46.49	69.74	92.99	116.2	139.5	162.7	186.0
0.6250	25.83	51.66	77.49	103.3	129.2	155.0	180.8	206.6
0.6875	28.41	56.83	85.24	113.7	142.1	170.5	198.9	227.3
0.7500	31.00	61.99	92.99	124.0	155.0	186.0	217.0	248.0
0.8125	33.58	67.16	100.7	134.3	167.9	201.5	235.1	268.6
0.8750	36.16	72.32	108.5	144.6	180.8	217.0	253.1	289.3
0.9375	38.75	77.49	116.2	155.0	193.7	232.5	271.2	310.0
1.0000	41.33	82.66	124.0	165.3	206.6	248.0	289.3	330.6
1.2500	51.66	103.3	155.0	206.6	258.3	310.0	361.6	413.3
1.5000	61.99	124.0	186.0	248.0	310.0	372.0	433.9	495.9
1.7500	72.32	144.6	217.0	289.3	361.6	433.9	506.3	578.6
2.0000	82.66	165.3	248.0	330.6	413.3	495.9	578.6	661.2
2.2500	92.99	186.0	279.0	372.0	464.9	557.9	650.9	743.9
2.5000	103.3	206.6	310.0	413.3	516.6	619.9	723.2	826.6
2.7500	113.7	227.3	341.0	454.6	568.3	681.9	795.6	909.2
3.0000	124.0	248.0	372.0	495.9	619.9	743.9	867.9	991.9

Standard Pipe Data

Nominal Size	Outside Diameter	Standard, Schd 40 or Schd 40S			Extra Strong, Schd 80 or Schd 80S		
		Wall	Inside Dia	Weight	Wall	Inside Dia	Weight
in	in	in	in	lb/ft	in	in	lb/ft
¼	0.540	0.088	0.364	0.4249	0.119	0.302	0.535
⅜	0.675	0.091	0.493	0.5677	0.126	0.423	0.739
½	0.840	0.109	0.622	0.8511	0.147	0.546	1.088
¾	1.050	0.113	0.824	1.1310	0.154	0.742	1.474
1	1.315	0.133	1.049	1.6790	0.179	0.957	2.172
1 ¼	1.660	0.140	1.380	2.2730	0.191	1.278	2.997
1 ½	1.900	0.145	1.610	2.7180	0.200	1.500	3.632
2	2.375	0.154	2.067	3.6530	0.218	1.939	5.023
2 ½	2.875	0.203	2.469	5.7940	0.276	2.323	7.662
3	3.500	0.216	3.068	7.5770	0.300	2.900	10.250
3 ½	4.000	0.226	3.548	9.1100	0.318	3.364	12.510
4	4.500	0.237	4.026	10.790	0.337	3.826	14.990
5	5.563	0.258	5.047	14.620	0.375	4.813	20.780
6	6.625	0.280	6.065	18.980	0.432	5.761	28.580
8	8.625	0.322	7.981	28.560	0.500	7.625	43.390
		Standard (Schd 40S*)			Extra Strong (Schd 80S*)		
10	10.75	0.365*	10.02	40.49	0.50*	9.75	54.74
12	12.75	0.375*	12.00	49.57	0.50*	11.75	65.42
14	14.00	0.375	13.25	54.58	0.50	13.00	72.10
16	16.00	0.375	15.25	62.59	0.50	15.00	82.78
18	18.00	0.375	17.25	70.60	0.50	17.00	93.46
20	20.00	0.375	19.25	78.61	0.50	19.00	104.10
22	22.00	0.375	21.25	86.62	0.50	21.00	114.80
24	24.00	0.375	23.25	94.63	0.50	23.00	125.50
26	26.00	0.500	25.00	136.20	0.75	24.50	202.30
28	28.00	0.500	27.00	146.90	0.75	26.50	218.30
30	30.00	0.500	29.00	157.60	0.75	28.50	234.30
32	32.00	0.500	31.00	168.20	0.75	30.50	250.30
34	34.00	0.500	33.00	178.90	0.75	32.50	266.40
36	36.00	0.500	35.00	189.60	0.75	34.50	282.40
42	42.00	0.500	41.00	221.60	0.75	40.50	330.50
48	48.00	0.500	47.00	253.70	0.75	46.50	378.50
		Schedule 40 ONLY			Schedule 80 (Schd 80S*)		
10	10.75	0.365	10.02	40.49	0.593	9.56	64.34
12	12.75	0.406	11.94	53.53	0.687	11.38	88.52
14	14.00	0.438	13.12	63.45	0.750	12.50	106.10
16	16.00	0.500	15.00	82.78	0.843	14.31	136.50
18	18.00	0.562	16.88	104.70	0.937*	16.13	170.80
20	20.00	0.593	18.81	122.90	1.031	17.94	208.90
22	22.00	0.653	20.69	148.90	1.150	19.70	256.10
24	24.00	0.687	22.63	171.10	1.218*	21.56	296.40
26	26.00	0.750	24.50	202.30	1.500	23.00	392.50
28	28.00	0.750	26.50	218.30	1.500	25.00	424.60
30	30.00	0.750	28.50	234.30	1.500	27.00	456.60
32	32.00	0.750	30.50	250.30	1.500	29.00	488.70
34	34.00	0.750	32.50	266.40	1.500	31.00	520.70
36	36.00	0.750	34.50	282.40	1.500	33.00	552.80
42	42.00	0.750	40.50	330.50	1.500	39.00	648.90
48	48.00	0.750	46.50	378.50	1.500	45.00	745.00

ANSI Standard Flanges and Plate Flanges Drilled to Class 125/150

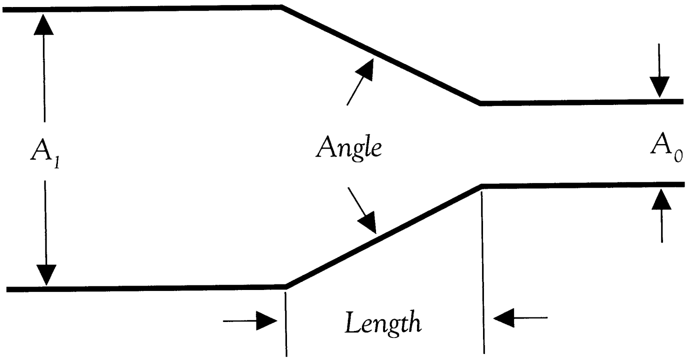
Nominal Size	Class 125 & Plate Flanges Drilled to ANSI Pattern						Class 250/300				
	Plate	Flange		Bolts and Bolt Circle			Flange		Bolts and Bolt Circle		
	Thickness	Diameter	Thickness	Diameter	Number	Size	Diameter	Thickness	Diameter	Number	Size
in	in	in	in	in		in	in	in	in		in
1	.375	4.25	.4375	3.12	4	.500	4.88	.6875	3.500	4	.625
1 ¼	.375	4.62	.5000	3.50	4	.500	5.25	.7500	3.875	4	.625
1 ½	.375	5.00	.5625	3.88	4	.500	6.12	.8125	4.500	4	.750
2	.375	6.00	.6250	4.75	4	.625	6.50	.8750	5.000	8	.625
2 ½	.375	7.00	.6875	5.50	4	.625	7.50	1.0000	5.875	8	.750
3	.375	7.50	.7500	6.00	4	.625	8.25	1.1250	6.625	8	.750
3 ½	.375	8.50	.8125	7.00	8	.625	9.00	1.8125	7.250	8	.750
4	.375	9.00	.9375	7.50	8	.625	10.00	1.2500	7.875	8	.750
5	.375	10.00	.9375	8.50	8	.750	11.00	1.3750	9.250	8	.750
6	.500	11.00	1.0000	9.50	8	.750	12.50	1.4375	10.625	12	.750
8	.500	13.50	1.1250	11.75	8	.750	15.00	1.6250	13.000	12	.875
10	.500	16.00	1.1875	14.25	12	.875	17.50	1.8750	15.250	16	1.000
12	.500	19.00	1.2500	17.00	12	.875	20.50	2.0000	17.750	16	1.125
14	.500	21.00	1.3750	18.75	12	1.000	23.00	2.1250	20.250	20	1.125
16	.500	23.50	1.4375	21.25	16	1.000	25.50	2.2500	22.500	20	1.250
18	.500	25.00	1.5625	22.75	16	1.125	28.00	2.3750	24.750	24	1.250
20	.500	27.50	1.6875	25.00	20	1.125	30.50	2.5000	27.000	24	1.250
22	.500	29.50	1.8125	27.25	20	1.250	33.00	2.6250	29.250	24	1.500
24	.500	32.00	1.8750	29.50	20	1.250	36.00	2.7500	32.000	24	1.500
26	.750	34.25	2.0000	31.75	24	1.250	38.25	3.1250	34.500	28	1.375
28	.750	36.50	2.0625	34.00	28	1.250	40.75	3.3750	37.000	28	1.375
30	.750	38.75	2.1250	36.00	28	1.500	43.00	3.6250	39.250	28	1.500
32	1.000	41.75	2.2500	38.50	28	1.500	45.50	3.8750	41.500	28	1.875
34	1.000	43.75	2.3125	40.50	32	1.500	47.50	4.0000	43.500	28	1.875
36	1.000	46.00	2.3750	42.75	32	1.500	50.00	4.1250	46.000	32	2.000
42	1.000	53.00	2.6250	49.50	36	1.500	50.75	4.6875	47.500	32	1.500
48	1.000	59.50	2.7500	56.00	44	1.500	57.75	5.2500	54.000	32	1.375
54	1.250	66.25	3.0000	62.75	44	1.750	65.25	6.0000	61.000	28	2.250
60	1.250	73.00	3.1250	69.25	52	1.750	71.25	6.4375	67.000	32	2.250

Bolt holes are ⅛" larger than the bolt. Sizes larger than 24 are not covered by ANSI B16.5 and are listed from various sources as industry standards. See the manufacturer for additional information.

Pressure Loss Coefficients

Conical or Plane Diffuser in Line, Includes Sudden Expansion

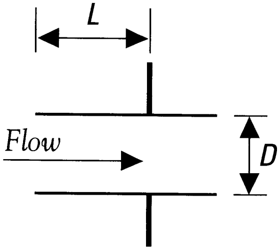
A_0/A_1	Angle in degrees													
	3	6	8	10	12	14	16	20	24	30	40	60	90	180
0.000	0.03	0.08	0.12	0.15	0.19	0.23	0.28	0.37	0.46	0.62	0.90	1.15	1.10	1.02
0.050	0.03	0.07	0.10	0.14	0.17	0.21	0.25	0.33	0.42	0.56	0.82	1.04	0.99	0.92
0.075	0.03	0.07	0.10	0.13	0.16	0.20	0.24	0.31	0.40	0.53	0.77	0.99	0.95	0.88
0.100	0.03	0.06	0.09	0.12	0.16	0.19	0.22	0.30	0.37	0.50	0.73	0.93	0.89	0.83
0.150	0.02	0.06	0.08	0.11	0.14	0.17	0.20	0.26	0.33	0.45	0.65	0.84	0.79	0.74
0.200	0.02	0.05	0.07	0.10	0.12	0.15	0.18	0.23	0.30	0.39	0.58	0.74	0.70	0.65
0.250	0.02	0.05	0.06	0.09	0.11	0.13	0.15	0.21	0.26	0.35	0.51	0.65	0.62	0.58
0.300	0.02	0.04	0.06	0.07	0.09	0.11	0.13	0.18	0.23	0.30	0.44	0.57	0.54	0.50
0.400	0.01	0.03	0.04	0.05	0.07	0.08	0.10	0.13	0.17	0.22	0.33	0.41	0.39	0.37
0.500	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.09	0.12	0.15	0.23	0.29	0.28	0.25
0.600	0.01	0.01	0.02	0.02	0.03	0.04	0.04	0.06	0.07	0.10	0.14	0.18	0.17	0.16
0.800	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.04	0.06	0.08	0.06



Conical or Plane Contraction in Line, Includes Sudden Contraction

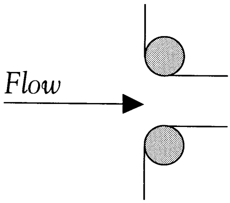
A_0/A_1	Angle in degrees								
	0	10	20	30	40	60	100	140	180
0.000	0.50	0.47	0.45	0.43	0.41	0.40	0.42	0.45	0.50
0.050	0.48	0.45	0.43	0.41	0.39	0.38	0.40	0.43	0.48
0.075	0.46	0.43	0.42	0.40	0.38	0.37	0.39	0.42	0.46
0.100	0.45	0.42	0.41	0.39	0.37	0.36	0.38	0.41	0.45
0.150	0.43	0.40	0.38	0.37	0.35	0.34	0.36	0.38	0.43
0.200	0.40	0.38	0.36	0.34	0.33	0.32	0.34	0.36	0.40
0.250	0.38	0.35	0.34	0.32	0.31	0.30	0.32	0.34	0.38
0.300	0.35	0.33	0.32	0.30	0.29	0.28	0.29	0.32	0.35
0.400	0.30	0.28	0.27	0.26	0.25	0.24	0.25	0.27	0.30
0.500	0.25	0.24	0.23	0.22	0.21	0.20	0.21	0.23	0.25
0.600	0.20	0.19	0.18	0.17	0.16	0.16	0.17	0.18	0.20
0.800	0.10	0.09	0.09	0.09	0.08	0.08	0.08	0.09	0.10

Intakes from Atmosphere



Projecting or Flush Pipes

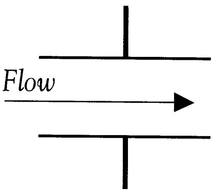
$L/D > 0.5$	$C = 1.0$
$L/D = 0.5$	$C = 0.75$
$L/D < 0.5$	$C = 0.5$



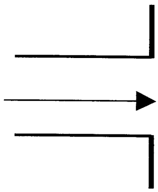
Rounded Entrances

slightly rounded	$C = 0.23$
well rounded	$C = 0.05$

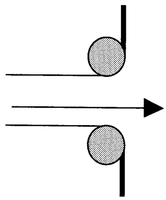
Exits and Exhausts to Atmosphere



projecting pipe



sharp edge (flush)



rounded

$C = 1.0$

Silencer Model	C	Silencer Model	C
EN1, 2, 3, 4	4.00	URD	4.20
EN5 < 12	4.00	SD	4.20
EN5 > 10	5.30	SURS	5.30
ET2	0.50	RD	4.20
ET4, 5	1.00	—	—
ES2, 3, 4	4.20	GTE3	0.60
U2, 5	0.25	GTE4	0.75
SU3, 4	0.85	GTE6	0.90
SU5	0.75	HV5	11.25
URB	4.20	HV10	11.50
UCI	4.20	HV15	11.75
RIS	4.20	HV20	12.00
UCD	4.20	HV25	12.25

Perforated Grids – C Coefficients Based on Free Stream Velocity

Thickness/ Hole Size		Open Area of Screen or Perforated Metal																
t/d		0.02	0.04	0.06	0.08	0.10	0.15	0.20	0.23	0.25	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00
0.00	1.35	6900	1670	718	390	241	98.0	50.1	35.7	29.00	18.10	8.05	3.91	1.95	0.94	0.41	0.13	0.00
0.05	1.32	6830	1653	710	386	239	97.0	49.6	35.4	28.70	17.90	7.97	3.87	1.93	0.93	0.40	0.13	0.00
0.10	1.28	6735	1630	701	381	235	95.7	48.9	34.9	28.30	17.70	7.86	3.82	1.91	0.92	0.40	0.13	0.00
0.20	1.22	6595	1596	686	373	231	93.7	47.9	34.1	27.70	17.30	7.69	3.74	1.87	0.91	0.40	0.13	0.00
0.30	1.16	6454	1562	671	365	226	91.7	46.9	33.4	27.20	16.90	7.53	3.66	1.83	0.89	0.39	0.13	0.01
0.40	1.10	6314	1528	657	357	221	89.7	45.9	32.7	26.60	16.60	7.37	3.59	1.80	0.87	0.38	0.13	0.01
0.50	0.97	6004	1452	624	339	210	85.2	43.6	31.1	25.20	15.80	7.01	3.41	1.71	0.84	0.37	0.12	0.01
0.60	0.84	5693	1377	592	322	199	80.8	41.3	29.4	23.90	14.90	6.64	3.24	1.62	0.80	0.35	0.12	0.01
0.70	0.63	5189	1255	539	293	181	73.6	37.6	26.8	21.80	13.60	6.04	2.95	1.48	0.73	0.33	0.12	0.01
0.80	0.42	4685	1133	487	265	163	66.3	33.9	24.2	19.60	12.20	5.44	2.66	1.34	0.66	0.30	0.11	0.02
1.00	0.24	4258	1030	442	240	148	60.2	30.8	21.9	17.80	11.10	4.95	2.42	1.22	0.61	0.28	0.11	0.02
1.50	0.09	3919	948	407	221	137	55.5	28.4	20.2	16.40	10.30	4.57	2.25	1.15	0.58	0.28	0.11	0.03
2.00	0.02	3775	913	392	213	132	53.5	27.4	19.5	15.80	9.90	4.43	2.19	1.13	0.58	0.28	0.12	0.04
Circular Wire Screen		2402	577	247	133	82.1	33.1	17.0	12.1	9.90	6.28	2.97	1.60	0.92	0.54	0.30	0.13	0.00

Adapted from the data of I.E. Idelchik, *Handbook of Hydraulic Resistance*, Translated from Russian, U.S. Atomic Energy Commission and National Science Foundation, Washington, D.C. 1966.

APPENDIX - XIII

Average Sound Absorption Coefficient (α)						
Materials	Octave Band Center Frequency-Hz					
	125	250	500	1k	2k	4k
Fiberglass Pack	1.00	1.16	1.15	1.00	0.98	0.98
Thermafiber Pack	0.77	1.14	1.15	1.04	1.04	0.94
4" Thick Acoustic Panel	0.75	1.01	1.11	1.06	1.02	0.95
Brick (Unglazed)	0.03	0.03	0.03	0.04	0.05	0.07
Concrete, Smooth	0.01	0.01	0.15	0.02	0.02	0.02
Concrete, Coarse	0.36	0.44	0.31	0.29	0.39	0.25
Concrete, Painted	0.10	0.05	0.06	0.07	0.09	0.08
Spray-On Mineral Fiber, 1" Thick	0.16	0.45	0.70	0.90	0.90	0.85
Gypsum Board, Typical	0.29	0.10	0.05	0.04	0.07	0.09
Wood	0.15	0.11	0.10	0.07	0.06	0.07
Steel Panels	0.06	0.06	0.04	0.03	0.03	0.04
Carpet, Heavy on Concrete with Pad	0.20	0.06	0.14	0.37	0.60	0.65
Glass (Ordinary)	0.35	0.25	0.18	0.12	0.07	0.04

APPENDIX - XIV

Approximate Sound Transmission Loss (dB) of Various Materials								
Materials	Octave Band Center Frequency							
	63	125	250	500	1k	2k	4k	8k
4" Thick Acoustic Panel	18	22	29	40	50	55	57	58>
Concrete - 4"	32	34	35	37	42	49	55	60
Concrete - 8"	34	36	38	43	50	56	61	66
Wood 1"	11	16	18	19	20	26	32	37
Wood 2"	15	17	19	20	26	32	37	41
Glass 1/8"	5	11	17	23	25	26	27	28
Glass 1/4"	11	17	23	25	26	27	28	30
Sheet Aluminum 1/16"	1	7	13	19	23	25	26	27
Sheet Aluminum 1/8"	7	13	19	23	25	26	27	28
Sheet Aluminum 1/4"	13	19	23	25	26	27	28	32
Sheet Steel 1/16"	9	15	21	27	33	38	39	39
Sheet Steel 1/8"	15	21	27	33	38	39	39	37
Sheet Steel 1/4"	21	27	33	38	39	39	37	40
Sheet Lead 1/16"	13	19	25	31	37	43	49	53
Sheet Lead 1/8"	19	25	31	37	43	49	53	55
Sheet Lead 1/4"	25	31	37	43	49	53	55	55

GLOSSARY

(Condensed Listing)

A-weighted Sound Level is weighted sound pressure level obtained by the use of metering characteristics and the A-weighting specified in American National Standard Sound Levels Meters for Measurement of Noise and Other Sounds (ANSI S1.4-1983, 1985).

Absorption Coefficient (α) is the ratio of sound energy absorbed by a surface of a medium (or material) exposed to a sound field to the sound energy incident on the surface.

Absorptive (Dissipative)-Type Silencer utilizes acoustic fill to provide silencing in higher frequency applications.

Acoustic Fill is a medium (or materials) such as fiberglass and mineral-wool of various densities and depths that have relatively high sound absorption characteristics.

Ambient Noise is the all-encompassing noise that is present within a given environment.

Anechoic Room (Free Field Room) is a room whose boundaries effectively absorb the sound affording essentially free-field conditions.

Annoyance is an individual's subjective reaction to noise, usually a negative reaction.

ANSI — The American National Standards Institute is the coordinating body of standards and specifications in the United States.

Attenuation is the energy loss of a silencer or other device. Ten times the logarithm of the ratio of incident power to transmitted power.

Audible Frequency Range — For purposes of engineering evaluation, only those frequencies between about 30 and 10,000 Hz are considered to be audible.

Background Noise is the noise that is present within a given environment excluding noise from a specific source being evaluated or measured.

Broadband Noise is noise that is essentially flat over the entire frequency range. See narrow band noise.

Combination Reactive/Absorptive Silencer utilizes both volume or chambers and acoustical fill for broadband noise reduction.

Combining Decibels — Decibels are combined logarithmically on an energy basis (not added arithmetically).

Critical Speed is the speed of a rotation system that corresponds to the resonant or natural frequency of the system.

Day/Night Sound Level (L_{dn}) is the 24-hour, time averaged, A-weighted sound level obtained by adding 10 dBA to the sound levels from 10 p.m. to 7 a.m.

Decibel (dB) is the standard by which noise is measured and evaluated. The decibel is a dimensionless unit used to express logarithmically the ratio of one sound to another either in terms of sound power or sound pressure.

dBA — One of the three basic sound level meter weighting networks (A, B, and C scales) that provide different response characteristics to noise as a function of frequency. The A scale is the most heavily weighted against low frequencies and most nearly approximates the frequency response of the human ear. "A" scale decibel readings are often referred to as "dBA" levels.

Direct Sound Path is straight-line propagation of noise from the source to the listener in a free field and follows the inverse square law.

Directivity is the difference in sound measured under free field conditions with a uniform source and that measured at the same distance, but at varying angles from the actual source.

Divergence is the spreading (reduction) of sound as a function of the distance from the source.

Dynamic Insertion Loss (DIL) is the reduction of noise provided by a given silencer under actual operating conditions. The difference in sound level without and with the silencer.

Equivalent Diameter is the diameter of a pipe (or silencer inlet nozzle) whose area is equal to the flow area of the silencer itself.

Excess Air Attenuation (in dB) is the attenuation of sound due to atmospheric molecular absorption and is in excess of the inverse square law.

Far Field (Free Field) is the point in the sound path where the sound propagates equally in all directions in accordance with the inverse square law and sound pressure level decreases by 6 dB for each doubling of distance from the source. See Near Field.

Frequency is the number of cyclical variations per second (cps) or Hz.

Frequency Spectrum is a quantity of sound pressures (dB) expressed as a function of frequency.

Fundamental Frequency is the lowest component frequency of periodic quantity.

Hard Room is a room in which the surfaces have very low values of sound absorption and, therefore, are highly reflective providing little or no sound absorption. See Reverberant Sound.

Harmonic Frequency is the frequency of a component of a periodic quantity and is an integral multiple of the fundamental frequency.

Hemispherical Divergence of sound occurs when the source is relatively large and/or is near the ground under otherwise free field conditions (Hemispherical divergence = spherical divergence -3 dB).

Hertz (Hz) is the unit of frequency in cycles per second.

ISO — The International Standards Organization is the coordinating body of international standards and specifications.

Insertion Loss (IL) is the reduction of noise due to the insertion of a silencer or other device in the system. Under actual operating conditions, insertion loss is referred to as dynamic insertion loss (DIL).

Inverse Square Law — Under free field conditions, sound varies inversely with the square of the distance from the source. When doubling or halving of the distance from the source there is a change of 6 dB in the sound pressure level.

Loudness is not a physical measurement of noise, but rather is a psycho-acoustic response to noise. The unit of loudness is the sone.

Loudness Level of a sound is numerically equal to the sound pressure (dB), relative to 0.0002 microbar, of a simple tone of 1,000 Hz, which is judged by the listener to be equivalent in loudness.

Near Field is close to the source where only small changes in sound pressure occur with changes in position (distance) from the source. A region in which pseudo-sound, (i.e., non-propagating pressure waves) are found.

Noise, simply defined, is unwanted sound.

Noise Abatement is the same as noise reduction.

Noise Control is the prevention or lessening of the noise before it is generated.

Noise Criteria (NC) is the maximum noise levels that are allowed at a specific location or distance from a noise source in a given environment.

Noise Reduction (NR) is the reduction of noise after it is produced.

Octave Bands are frequency bands where the upper limit of each band is twice the lower limit. Octave bands are identified by their center frequency (geometric-mean-frequency).

One-Third Octave Bands are the same as octave bands except each octave is divided into one-third octaves.

Preferred Frequencies are those of either octave or one-third octave bands as defined in ANSI S1.6-1984.

Phon is a unit of loudness level (see loudness level).

Pure Tone is a sound emitted at a single frequency.

Reactive-Type Silencer is a chamber (volume)-type silencer or muffler (low-frequency attenuator) with no acoustic fill.

Receiver is a person (or persons) or equipment affected by noise.

Resonance of a system in forced oscillation exists when any change (however small in the frequency of excitation) causes a decrease in the response of the system.

Reverberant Sound is reflected sound from the ceilings, walls and other surfaces within a confined space. In a confined area, close to the source, direct sound is usually predominant, but at a distance reverberant sound may be predominant.

Self Noise (SN) is the noise generated by air or gas flow through a silencer with source noise excluded.

Soft Room is a room in which its surfaces have very high values of sound absorption.

Sone is a unit of loudness. By definition, a simple tone of 1,000 Hz, 40 dB above a listener's threshold, produces a loudness of 1 sone.

Sound is an oscillation in pressure in an elastic medium, which is capable of producing the sensation of hearing. Also, the sensation of hearing caused by a pressure oscillation.

Sound Absorption Coefficient (α) is the dimensionless ratio of sound energy absorbed by a given surface to that incident upon the surface.

Sound Power Level (L_W) in decibels is 10 times the logarithm to the base 10 of the ratio of a given power to a reference power. (The reference power is usually 10^{-12} watt.)

Sound Pressure Level (L_p) in decibels is 20 times the logarithm to the base 10 of the ratio of a sound pressure to a reference pressure. (The reference pressure is usually 0.0002 microbar or 20 micropascals.)

Sound Transmission Loss (TL) is a logarithmic ratio of the sound power on one side of a partition (or silencer shell) to the sound power radiated by the other side.

Speech Interference Level (SIL) is the arithmetic average of the sound pressure centered at 500, 1,000 and 2,000 Hz range.

Spherical Divergence of sound occurs in most applications where the source is relatively small and a free field exists and the area is free of obstructions and reflective surfaces (spherical divergence = hemispherical divergence + 3 dB).

Threshold of Hearing is the lowest continuous sound pressure that will create an auditory sensation for the average person (and is defined as the weakest sound pressure detectable by an average person which is about 0.0002 microbar at 1,000 Hz).

Wavelength (λ) is the length of one complete cycle of a sound wave.

$$\lambda = \left(\frac{\text{speed of sound}}{\text{frequency}}\right) = \frac{c}{f}$$

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