Industrial Silencing Handbook

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INDUSTRIAL SILENCING HANDBOOK

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by
B.G. Golden
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March, 1982

ABSTRACT

This handbook is about noise—the noise produced by vents and blowdowns, engines, blowers, compressors and other sources common to the manufacturing, petroleum, gas, and chemical industries. Emphasis is placed on the source of noise, its cure and the standards by which it is judged.

Due to the general, often semi-technical, approach to many aspects of noise control presented in this handbook, many details and theoretical procedures have been omitted or condensed for simplicity. The procedures and guidelines presented have been prepared to aid in the solution of many of the most common industrial noise problems. Solutions to more complex problems may require more detail and study.

TABLE OF CONTENTS

1. INTRODUCTION TO NOISE 6
2. REVIEW OF THE FUNDAMENTALS 8
3. NOISE CRITERIA 24
   OSHA (Occupational Safety and Health Administration) 24
   ISO (International Organization for Standardization) .. 29
   NEMA (National Electrical Manufacturers Association) 30
4. TYPES OF NOISE CONTROL 32
   Silencers 32
   Acoustic Enclosures and Partial Barriers 36
   Acoustic Pipe Lagging 43
5. TYPICAL NOISE PROBLEMS AND RECOMMENDED SOLUTIONS 46
   Vents and Blowdowns to Atmosphere 48
   Engines 66
   Gas Turbines 80
   Rotary-Positive Blowers 86
   Centrifugal Compressors 90
   Reciprocating Air Compressors 92
   Vacuum Pumps 96
   Forced Draft Fans 99
   Pressure Regulators 104
   Gears, Motors and Drives — Mechanical Noise 108
6. CONCLUSION 109

APPENDIX I — BASIC FORMULAS 110
APPENDIX II — PIPING RESISTANCE 112
APPENDIX III — TYPICAL SILENCER FLOW AREA 115
APPENDIX IV — CONDENSED PROPERTIES—VARIOUS GASES 116
APPENDIX V — PROPERTIES OF DRY AIR 117
APPENDIX VI — PROPERTIES OF DRY SATURATED STEAM 119
APPENDIX VII — PROPERTIES OF SUPERHEATED STEAM 121
APPENDIX VIII — ACOUSTIC RESONANCE IN PIPING SYSTEMS 125
BIBLIOGRAPHY 126

Chapter 1
INTRODUCTION TO NOISE

Over the years, many neighborhoods that once were as quiet as the proverbial church mouse have fallen victim to both industrial and populous encroachment. As a result, numerous laws, regulations and ordinances have been enacted to control excessive noise. Worker compensation for job-related loss of hearing has increased tenfold since 1960, largely due to: (a) a growing worker awareness and concern about the physical effects of excessive noise, (b) the industrial trend to increased horsepower — with higher speeds, elevated pressures and temperatures — and increased noise and (c) the economy of layout and construction.

Worker exposure to excessive noise over an extended period may result in a permanent loss of hearing. Noise can also be hazardous as well as objectionable to nearby residents. With tighter controls now in effect, every effort should be made to eliminate or measurably reduce noise at its source in a logical and systematic manner. Developing and implementing a noise control program warrants the attention and involvement of all levels of engineering and operations management.

In the United States, industrial plants are legally responsible for worker safety, including control and abatement of excessive noise. One of the provisions of the Occupational Safety and Health Act (OSHA) of 1970 relates specifically to the occupational exposure of workers to noise. The maximum permissible noise level of 90 dBA (based on an eight-hour, time-weighted average) adopted by this Federal Safety Act is now the industry-wide standard used in most hearing conservation programs.

Under the Noise Control Act of 1972, the Environmental Protection Agency (EPA) was mandated by Congress to establish noise limits protective of public health and welfare with an adequate margin of safety. The EPA recommended a level of 75 dBA for eight-hour worker exposure and a level of 55 dBA outdoors. These recommendations are not expected to be adopted and enforced in the immediate future, as more study and analysis is needed.

In addition to federal regulatory efforts, states and cities have set maximum noise levels acceptable at plant property lines. These levels typically vary according to area zoning — heavy industrial, industrial, commercial, residential and hospital.

Foreign governments are dealing with their noise problems in a variety of ways:

- In Europe, as in the United States, controls are subject to considerable legal controversy and debate.
- The United Kingdom considers noise a nuisance, and worker recourse is largely through common law, not by governmental decree.
- French manufacturers are required to publish noise levels of their products for user review and evaluation.
- Sweden has a "foundation" for the protection of workers. Each employer is subject to a payroll tax for the improvement of the work environment.
- West Germany has the most stringent laws regulating noise. Control rests with state authorities.

In the United States, when workers are subjected to excessive noise, OSHA regulations require that administrative and/or engineering controls be applied. Where there is no practical engineering solution, administrative controls apply. Such controls may include reducing worker exposure time or requiring that workers in the area wear personal hearing protection — ear plugs or ear muffs. But whenever technically and economically feasible, engineering controls should be applied. Ideally, the results must: (a) reduce hearing risks to acceptable levels and (b) be cost effective. Amid the controversies surrounding this subject, there is one area of universal agreement — that effective noise control can result in improved worker communications, increased productivity, higher quality of work output and a quieter, safer environment as well.

Chapter 2
REVIEW OF THE FUNDAMENTALS

Sound is actually as old as creation itself, but the science of sound, acoustics, is closely associated with the growth of civilization. The basic theory of the origin, propagation and reception of sound was conceived at an early stage in the development of human thought. More than two thousand years ago, ancient Greek philosophers were convinced that sound originated in the motion of parts of the body. They believed that sound was transmitted through the air by some undefined bodily motions, and that this movement near the ear produced the sensation of hearing. Admittedly vague by present day standards, this early concept does not vary greatly from our modern theory of sound.

Formulation of the mathematics of sound is credited to Sir Isaac Newton who, in the early part of the 18th century, derived many of our laws of physics. But the 1877 publication of Lord Rayleigh's **The Theory of Sound** marked the beginning of sound technology as we know it today. Yet, there remain many unanswered questions in both the theory and application, making noise control today as much an art as it is an exact science.

A look at the fundamentals of noise must begin with a few basic definitions.

SOUND is the sensation of hearing.

NOISE is unwanted sound. Noise may be continuous, intermittent or erratic, depending upon the source. It annoys, distracts and generally upsets and interrupts the tranquility of an otherwise homogeneous and peaceful environment.

The **DECIBEL** (dB) is the unit of measurement used in noise control. It is a dimensionless unit used to express logarithmically the ratio of a sound level to a reference level. The reference level is usually 0.0002 microbar (μbar).

While the weakest sound an average person can detect is about 20 dB, the human ear can tolerate sound pressures up to 100,000 times higher! This is graphically shown in Figure 2-1.

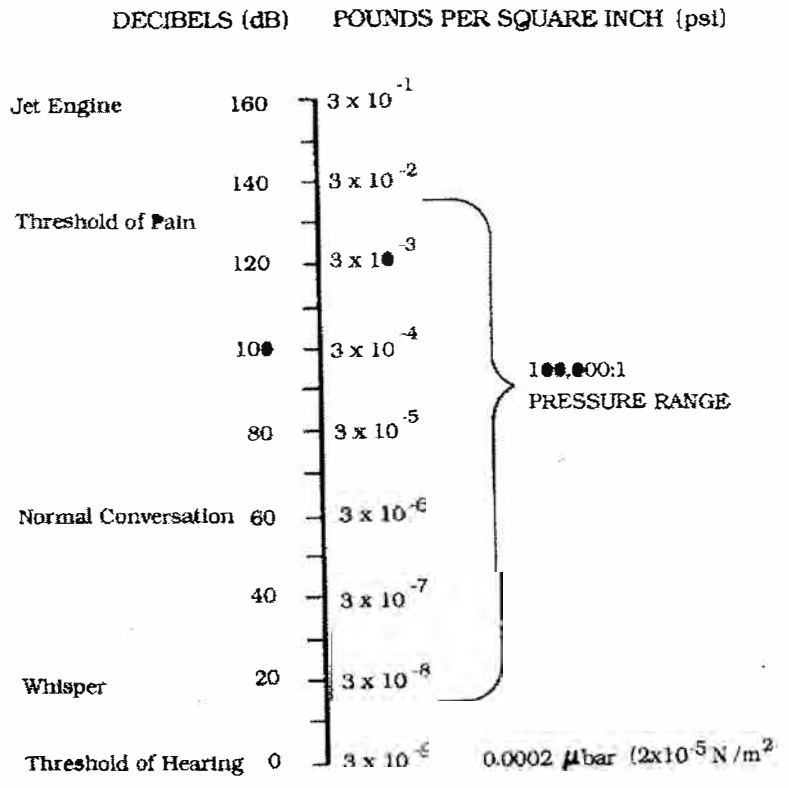


Figure 2-1. TYPICAL SOUND PRESSURE LEVELS

Audible frequencies are divided into **OCTAVE BANDS** for basic analysis. An octave band is a range of sound frequencies with an upper limit twice its lower limit (ratio of 2:1). The center (or geometric mid-frequency) is the square root of the product of the upper and lower cutoff frequencies of a pass band. The preferred octave bands using 1,000 Hz (cps) as a reference are shown in Figure 2-2.

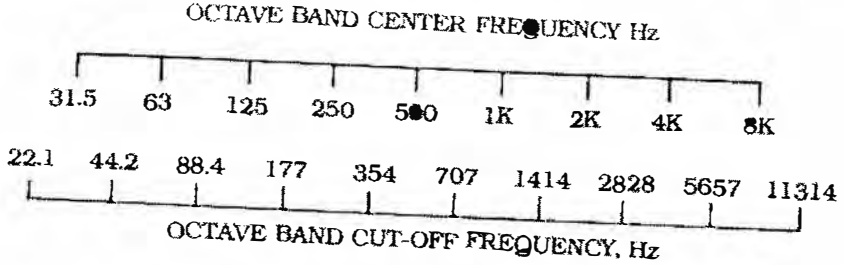


Figure 2-2. PREFERRED OCTAVE BANDS

Decibels are **COMBINED LOGARITHMICALLY** on an energy basis (not added directly as in: 95 dB + 95 dB ≠ 190 dB). Any two noise levels may be combined with an acceptable degree of accuracy, as shown in Figure 2-3.

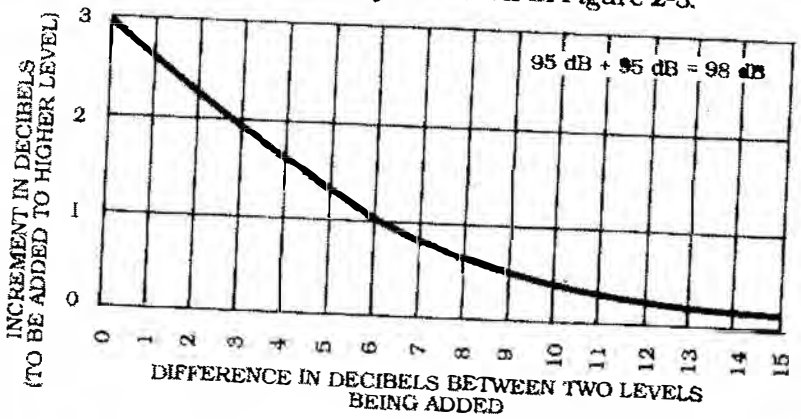


Figure 2-3. COMBINING NOISE LEVELS

Multiple noise levels may also be combined using the method shown in Figure 2-4 or, in the case of equal noise levels, by simply applying the correction shown in Table 2-1.

Theoretically, addition of equal sources in this manner must be assumed as a single point source. As long as the sources to be added are in close proximity, the results will be reasonably accurate and only somewhat conservative.

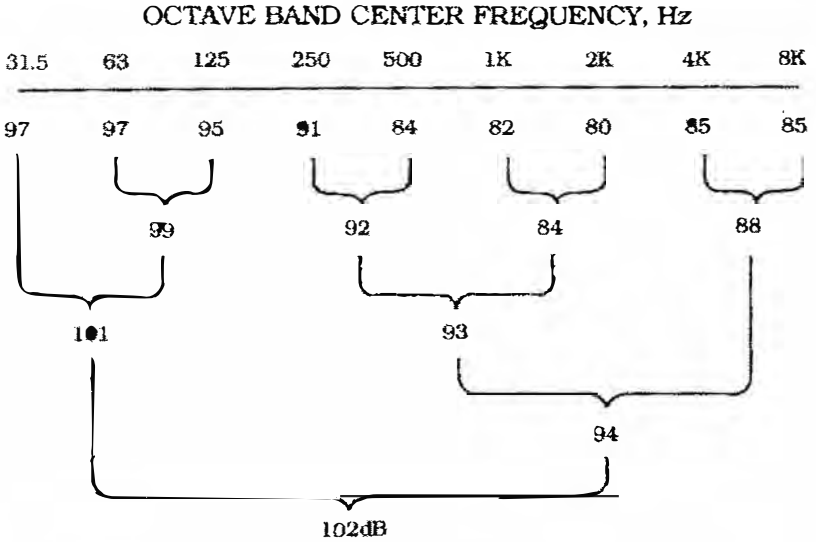


Figure 2-4. COMBINING OCTAVE BAND NOISE LEVELS

NUMBER OF EQUAL SOURCES	ADD
2	3
3	5
4	6
5	7
6-7	8
8	9
9-10	10
N	10 Log N. dB

Table 2-1. COMBINING MULTIPLE SOURCES OF EQUAL NOISE

The decibel represents a relative value or ratio, not an absolute value, and in noise control, the decibel level is stated as either sound pressure level (SPL or Lp) or sound power level (PWL or Lw).

The **SOUND PRESSURE LEVEL** (Lp) expressed in decibels is a ratio of a sound pressure to a reference pressure and is defined as:

$$Lp = 20 \text{ Log } \frac{P}{0.0002} \text{ dB, re } 0.0002 \ \mu \text{ bar (standard reference pressure)}$$

Noise (Lp) in a free field (based upon hemispherical propagation) is readily determined, where:

Lp (hemisphere) = Lp (sphere) + 3 dB ground reflection

$$Lp = Lw - 20 \text{ Log } r + 2.5 \text{ dB}$$

Where Lw = Sound Power Level (dB.)
r = distance from source (ft.)

Hemispherical propagation (divergence) of noise occurs where the source is near the ground or where the sound is directed horizontally and is a function of distance from the source. Note that in Table 2-2, when the distance from the source is either doubled or halved, the change in Lp is 6 dB.

Distance from source, ft.	Lw to Lp Conversion, dB
3	-7
5	-11
10	-17
20	-23
25	-25
50	-31
100	-37
200	-43
400	-49
8●●	-55
100●	-57

Table 2-2. HEMISPHERICAL PROPAGATION
(-20 Log r + 2.5 dB)

Lp, which is dependent upon distance and, to some extent, upon the acoustic environment is easily measured with a sound level meter. Lp values without reference to distance are essentially meaningless. To estimate the Lp at a distance greater than that at which it was measured:

$$Lp_{(\text{Desired Distance})} = Lp_{(\text{Distance})} - 20 \text{ Log } \frac{\text{Desired Distance}}{\text{Distance}}$$

Example: 120 dB measured at 3 ft. Find Lp at 100 ft.

$$Lp_{(100')} = 120 - 20 \text{ Log } (\frac{100}{3}) = 90 \text{ dB}$$

SOUND POWER LEVEL (Lw) is the measure of the total acoustic power radiated by a given source and is defined as:

$$Lw = 10 \text{ Log } \frac{W}{10^{-12}} \text{ dB, re } 10^{-12} \text{ watts}$$

The Lw is constant for a given source and is independent of the acoustic environment. It cannot be measured directly, but must be calculated. Lw can be described as roughly equal to the watt rating of a light bulb. Whereas Lp is comparable to the amount of light produced at a given distance from the bulb in any given environment. Just as more light is produced in a light colored room than in a dark colored room, Lp is higher in a reverberant (reflective) environment than in a room with highly sound-absorptive surfaces.

Most U.S. manufacturers publish or make available the sound power level (Lw) of their fans, turbines and other equipment. Where the actual unsilenced Lw or Lp is unavailable, one must rely on calculated or extrapolated data which are only an approximation.

When noise is emitted to the outdoors from the open end of an intake or exhaust stack (either vertical or horizontal), noise levels are higher directly in front of the stack than at the side. At positions directly in line with the stack opening (0°), noise levels are higher than if from a non-directional, unducted source free to propagate noise equally in all directions. The effect of **DIRECTIVITY** may be positive or negative, depending on the angle and the size of the stack. Average values are shown in Table 2-3.

Angle to Direction of Flow	Silencer Outlet Diameter, in.	Octave Band Center Frequency, Hz							
		63	125	250	500	1K	2K	4K	8K
0° (0°, 45°, 90°, 135°)	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 2-3. EFFECT OF DIRECTIVITY BASED ON ANGLE TO DIRECTION OF FLOW AND SIZE OF SILENCER OUTLET.

Ambient **BACKGROUND NOISE** is a complex phenomenon that fluctuates with time and proximity to roads, traffic, shielding from buildings and other factors. When vent blowdown and other noises are measured, care must be taken to record only the direct airborne noise from the source. Yet complete isolation of the background noise is not always possible. If the difference between the background and the source is less than 3 dB, the background is too high for an accurate measurement. If the difference is between 3 and 10 dB, a correction is necessary, as in Figure 2-5.

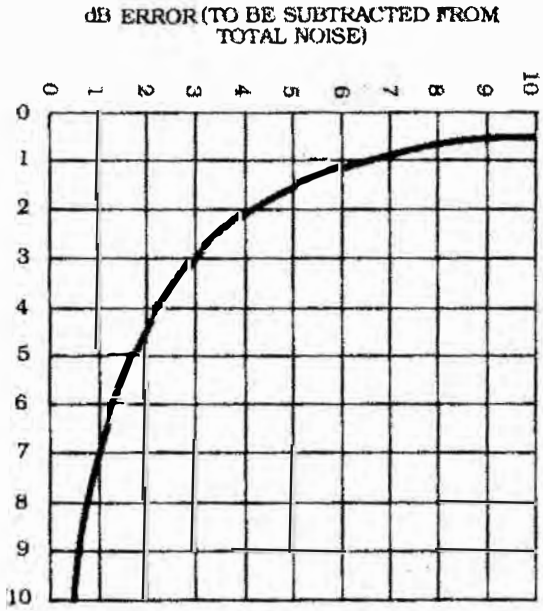


Figure 2-5. BACKGROUND CORRECTION.

In a **FREE FIELD**, without obstructions or reflections, the Lp is the same in every direction at equal distances from the source. This basic principle is not applicable, though, in near field conditions close to the source. Lp measurements should be taken at a distance of not less than the wave length of the lowest frequency of concern, or at a distance of at least twice the greatest dimension of the source.

Simply stated, noise diminishes in a free field — by 6 dB each time the distance from the source is doubled. But due to atmospheric conditions and natural or man-made obstacles, measured levels are generally lower than this amount. The additional reduction (in excess of the inverse square law) is referred to as **EXCESS AIR ATTENUATION** (molecular absorption) and is illustrated in Figure 2-6.

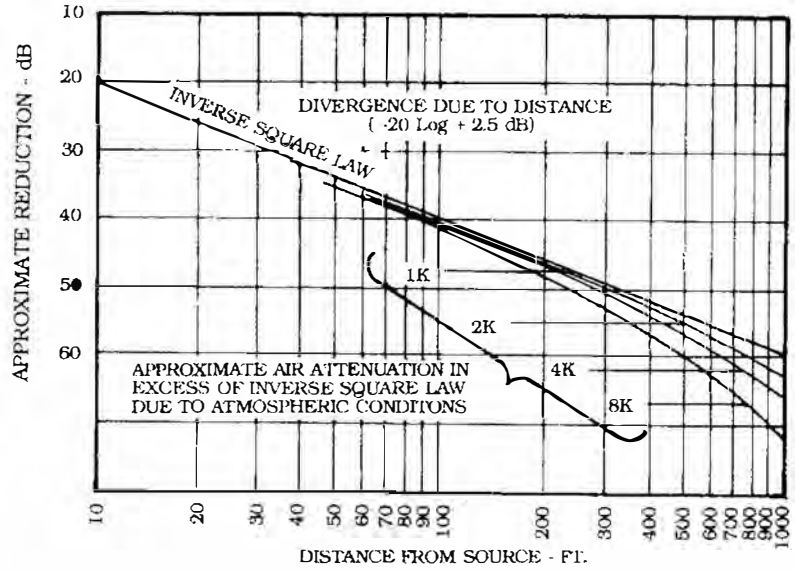


Figure 2-6. EXCESS AIR ATTENUATION

Sound absorption and sound transmission loss are the two basic performance characteristics of materials used in noise control. Manufacturers publish these values at selected frequency bands.

The **SOUND ABSORPTION** characteristics of a material is expressed in terms of absorption coefficients (α), the ratio of the energy absorbed by the surface of the material to the energy incident upon the surface. Materials which are highly absorptive have soft, porous surfaces and vary in both depth and density. The variance in absorption coefficients is shown in Table 2-4.

MATERIALS	Octave Band Center Frequency Hz					
	125	250	500	1k	2k	4k
Modular Acoustic Panel (4" thick)	0.74	1.05	1.14	1.11	1.01	1.04
Brick, Unglazed	0.03	0.03	0.03	0.04	0.05	0.07
Concrete	0.01	0.01	0.15	0.02	0.02	0.02
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
Glass (Ordinary Window)	0.35	0.25	0.18	0.12	0.07	0.04

Table 2-4. TYPICAL SOUND ABSORPTION COEFFICIENTS (α)

The **SOUND TRANSMISSION LOSS** (TL) in dB, through pipe walls or through a structure (panel or partition), is the difference between the incident sound power, Lw (1) and the transmitted sound power, Lw (2):

$$TL = Lw(1) - Lw(2)$$

TL is a function of mass, stiffness and damping, and is frequency dependent. Typically, it increases 6 dB per octave and 6 dB for each doubling of the mass. The approximate TL of various materials is shown in Table 2-5.

MATERIALS	OCTAVE BAND CENTER FREQUENCY - Hz							
	63	125	250	500	1K	2K	4K	8K
Modular Acoustic Panel (4" thk)	—	23	28	37	45	49	54	51
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 ⅛"	5	11	17	23	25	26	27	28
Glass ¼"	11	17	23	25	26	27	28	30
Sheet Aluminum 1/16"	1	7	13	19	23	25	26	27
Sheet Aluminum ⅛"	7	13	19	23	25	26	27	28
Sheet Aluminum ¼"	13	19	23	25	26	27	28	32
Sheet Steel 1/16"	9	15	21	27	33	38	39	39
Sheet Steel ⅛"	15	21	27	33	38	39	39	37
Sheet Steel ¼"	21	27	33	38	39	39	37	40
Pipe Walls ⅛"	6	11	16	22	27	32	37	42
Pipe Walls ¼"	11	16	22	27	32	37	42	47
Pipe Walls ⅜"	14	19	25	30	35	40	45	50
Pipe Walls ½"	16	22	27	32	37	42	47	52

Table 2-5. APPROXIMATE SOUND TRANSMISSION LOSS (TL) - dB

The human ear responds to amplitude (loudness) and frequency (pitch), and is not equally sensitive to all frequencies. It is most sensitive to frequencies between 500 and 3,000 Hz (cps), and less sensitive at extremely high and low frequencies. At 3,000 Hz, 90 dB sounds much louder than it does at 500 Hz.

An individual subjective reaction to noise is not always predictable. Typically, doubling of the **LOUDNESS** requires a 10 dB increase in sound pressure. A 3 dB increase is barely noticeable. Unfortunately, sound pressure levels (Lp) are not normally used to judge loudness, but various other methods for determining apparent loudness have been developed. The most common being (a) dBA and (b) sones. Both of these are loudness ratings. Other units of loudness are phons, and perceived noise level (PNdB).

SONES are calculated by adjusting each octave band level according to the ear's sensitivity in that band, and then adding the effect of all of the bands. Loudness comparisons with other noises are made linearly — a noise twice as loud as another must have a sone rating twice that of the other noise. For comparisons of sones and dBA levels (defined in Chapter 3), see Figure 2-7.

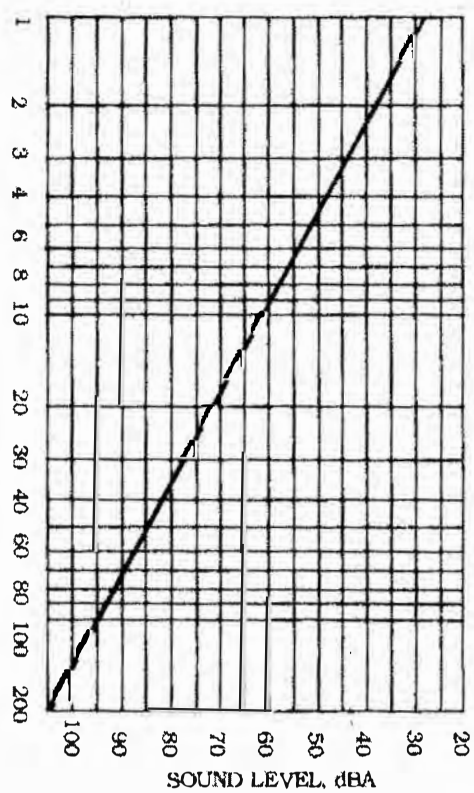


Figure 2-7. SONES - dBA CORRELATION.

Unlike outdoor free field conditions, sound within a room or building is reflected by the walls, floors and ceiling. Close to the source the **REVERBERANT EFFECT** is minimal since the direct sound is predominant. But away from the source, the reflected or reverberant sound may exceed the direct sound. The Lp within the reverberant region depends upon the acoustic Lw being radiated, the location of the source, the size and type construction of the room:

$$Lp = Lw + 10\ Log\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) + 10.5\ dB$$

Q = Directivity Factor
= 2 Small Free Standing Source in center of room
= 4 Small Source near wall
= 8 Small Source in corner
Large Source Q may vary with direction

r = Distance from Source (ft.)

R = Room Constant = $\frac{\alpha S}{1 - \alpha}$

S = Total surface area of room (sq. ft.)

α = Average sound absorption coefficient of the surfaces of the room at a given frequency.

Graphic solution of this formula is shown in Figure 2-8.

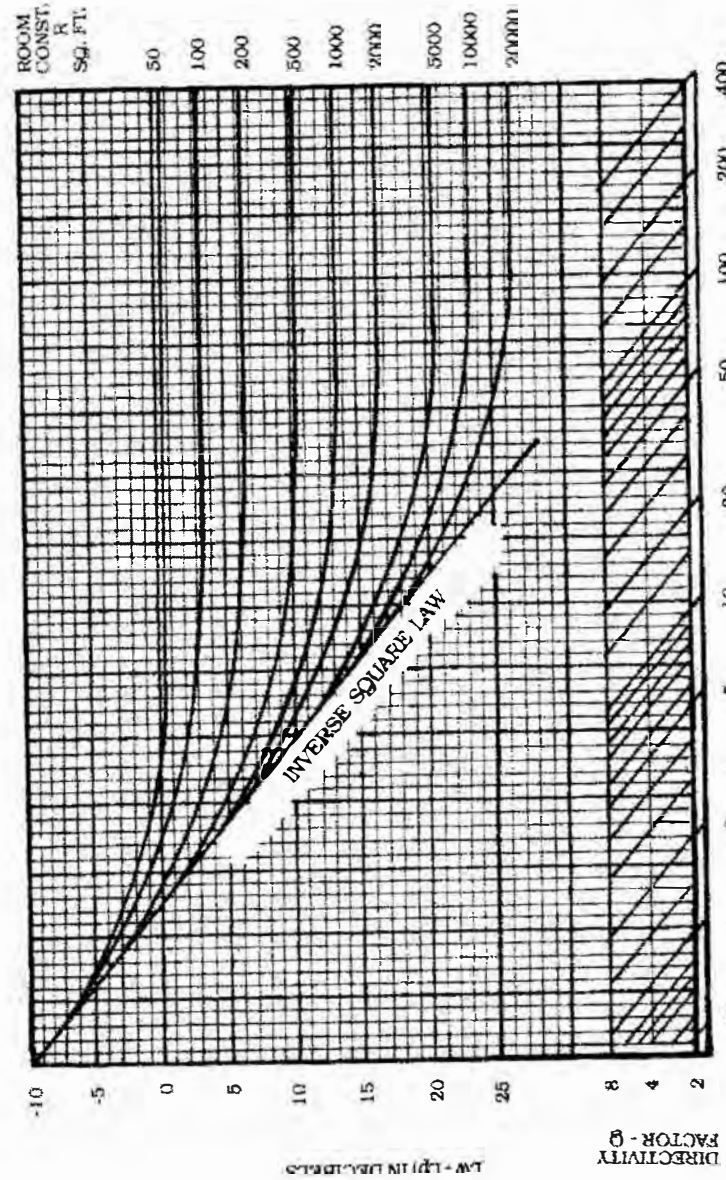


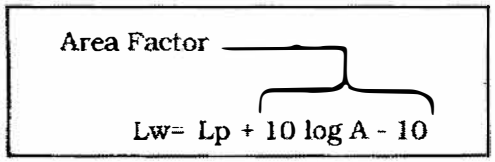
Figure 2-8. SOUND PRESSURE LEVEL (Lp) RELATIVE TO SOUND POWER LEVEL (Lw). DIFFERENT VALUES OF ROOM CONSTANT (R) AND DIRECTIVITY FACTOR (Q).

To reduce the reflected noise in a reverberant room or area, absorptive panels or materials may be added to the walls and ceiling. While absorption reduces the noise at a distance, it provides little or no reduction near the source. The noise reduction which is achieved by the addition of absorption may be approximated:

$$\text{Lp Reduction} = 10 \text{ Log } \frac{\text{S1 (after)}}{\text{S2 (before)}}$$

In the formula, S (Sabins) is simply the product of the surface area (square feet) of the reverberant space and its absorption coefficient ($\sim$). If the absorption in a room is doubled, the Lp can be expected to decrease by 3 dB. Addition of 4 times as much absorptive material will provide a 6 dB reduction. The practical limit is 6 to 8 dB.

The sound power level (Lw) that passes through an opening into or out of a room is approximately:



Where Lp is the sound pressure level in the room at or near the opening, and A is the cross-sectional area (square feet) of the opening. The propagation of high-frequency noise is particularly noticeable through small openings such as cutouts for pipes and cables. Large openings for ventilation must be fitted with silencers (sound traps).

Chapter 3
NOISE CRITERIA

In noise control, the term "criteria" is usually defined as the maximum allowable sound pressure level (Lp) at a **specific distance** from the source. Criteria may be expressed as a single overall value (all frequencies combined) or in individual octave bands.

Today, among the various noise criteria in existance, the most common are the dBA criteria established by OSHA, the ISO criteria developed by the International Organization for Standardization and the NEMA Standard (National Electrical Manufacturers Association).

OSHA (Occupational Safety and Health Act)

The standard sound level meter has three alternate weighting networks, designated as A, B and C, which selectively discriminate against low and high frequencies as shown in Figure 3-1. A sound level meter may also have a flat (no-weighting) response network.

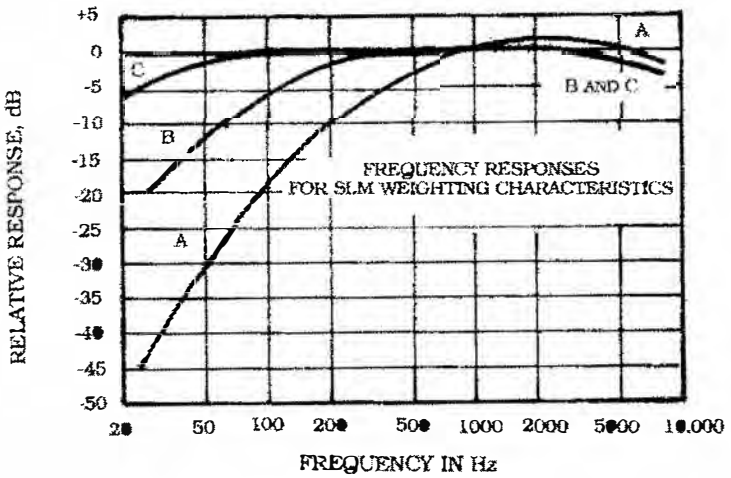


Figure 3-1. FREQUENCY RESPONSE CHARACTERISTICS FOR SOUND LEVEL METERS (ANSI-S1.4-1971)

The C scale measures the overall sound pressure level (essentially a flat response with all frequencies combined) and such readings may be referred to as dBC.

A-scale readings (dBA) are the most heavily weighted. The A scale approximates the ear's response to noise (500 to 6,000 Hz), and is now commonly used as an Industry and Government Standard in the evaluation of sound. Typical dBA levels are shown in Table 3-1.

dBA	Source	Perception/Hearing
140	Jet Engine at 25 ft.	Unbearable
130	High Pressure Safety Vent at 25 ft.	Threshold of pain
120	Large FD Fan Plenum Area	Uncomfortably loud
110	8000 hp Engine Exhaust at 25 ft.	
100	Compressor Building	Very loud
90	Boiler Room	
80	Pneumatic Drill	Loud
70	Commercial Area	
60	Normal Conversation	
50	Average Home	Comfortable
40	Nightime Residential Area	
30	Broadcast Studio	
20	Whisper	Barely Audible
10		
0		Threshold of hearing

Table 3-1. TYPICAL A-WEIGHTED SOUND LEVELS

The maximum occupational noise exposure levels set by OSHA under the Federal Safety Act are shown in Table 3-2. The "slow response" notation is helpful in obtaining an average for noise levels that may fluctuate by more than 4 dB.

Duration per Day (Hours)	Sound Level dBA (Slow Response)
8	90
6	92
4	95
3	97
2	100
1½	102
1	105
½	110
¼ or less	115

Table 3-2. PERMISSIBLE NOISE EXPOSURES (OSHA)

Note that OSHA permits a 5-dB increase in permissible levels for a reduction of 2:1 in exposure time (often referred to as the 5-dB exchange rate).

When daily exposure is composed of two or more periods of noise at different levels, the combined effect should be considered, rather than the individual effect of each. Mixed exposure must be considered if the sum of the following fractions exceeds unity:

$$C_1/T_1 + C2/T2 \ldots Cn/Tn$$

Cn indicates total time of exposure at a specified noise level. Tn indicates total time of exposure permitted at that level.

EXAMPLE: If a worker is exposed to a 90-dBA level for 5 hours and a 95-dBA level for 3 hours, the effect of the combined exposures is:

$$\frac{C1}{T1} + \frac{C2}{T2} = \frac{5}{8} + \frac{3}{4} = 1.375 > 1.0$$

Therefore, this exposure exceeds the allowable limits, and corrective noise abatement measures are required.

Although the basis of the noise exposure regulations is the A-weighted sound level (dBA), it's often desirable to convert dBA units into octave-band dBC equivalents for analysis purposes. An example is shown in Table 3-3.

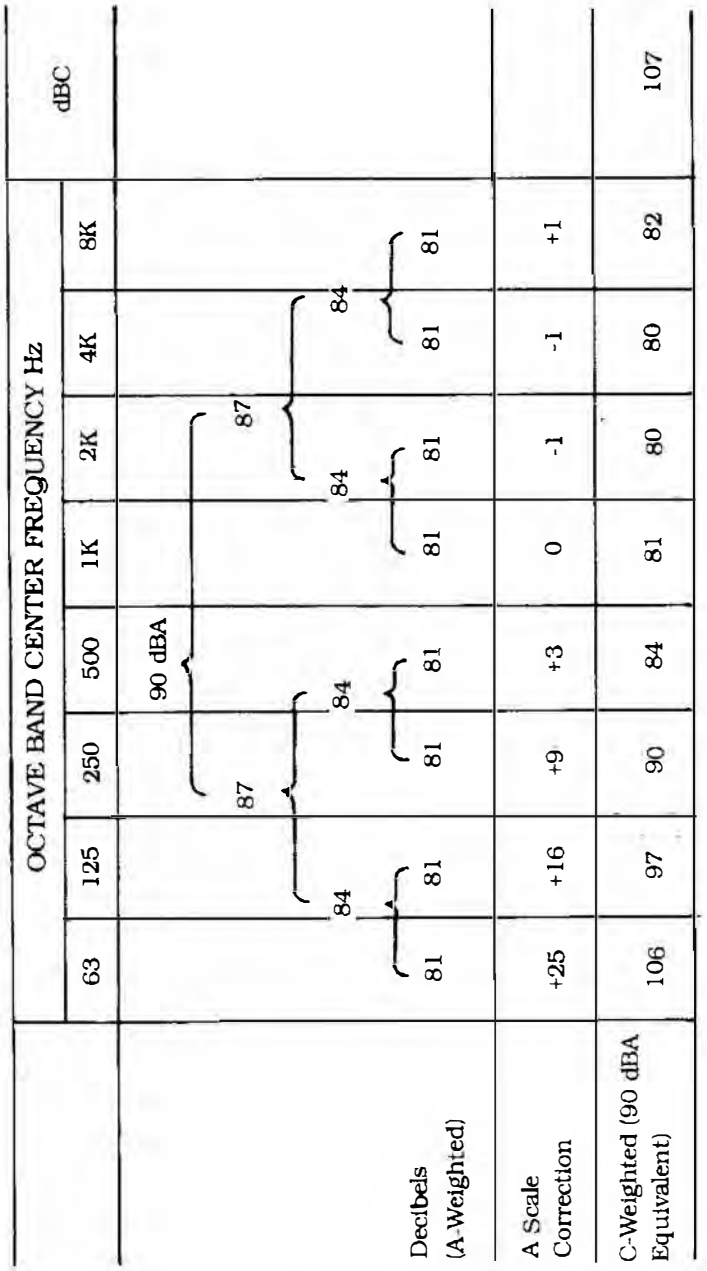


Table 3-3. TYPICAL 90 dBA vs C-WEIGHTED 90 dBA EQUIVALENT

Exposure to impulsive or impact noise should not exceed the 140-dB peak sound pressure level. This is the upper limit to which a person should be exposed, regardless of how brief the exposure might be.

Precautions must be taken when using dBA levels. Any number of octave-band combinations will produce the same overall dBA level. (Figure 3-2.), so, it's important to remember that sound should be combined and evaluated on an octave-band basis, and not merely on the basis of overall levels.

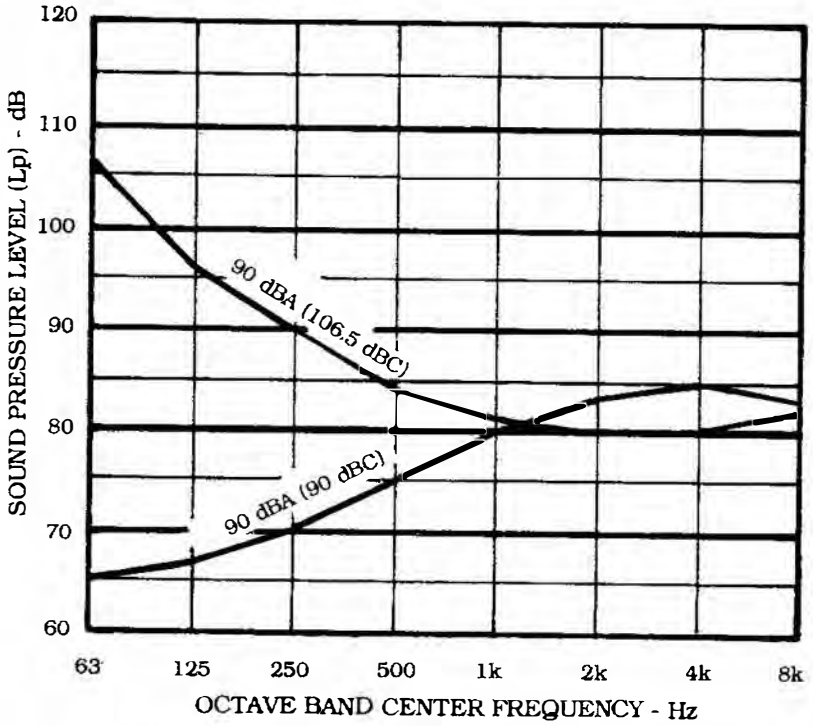


Figure 3-2. COMPARATIVE dBA LEVELS VS. OCTAVE-BAND LEVELS

ISO (International Organization for Standardization)

NR	OCTAVE-BAND CENTER FREQUENCY - Hz									dBA Equivalent
	31.5	63	125	250	500	1K	2K	4K	8K	
20	69	51.3	39.4	30.6	24.3	20	16.8	14.4	12.6	30
25	72.4	55.2	43.7	35.2	29.2	25	21.9	19.5	17.7	35
30	75.8	59.2	48.1	39.9	34.0	30	26.9	24.7	22.9	39
35	79.2	63.1	52.4	44.5	38.9	35	32.0	29.8	28.0	44
40	82.6	67.1	56.8	49.2	43.8	40	37.1	34.9	33.2	48
45	86	71	61.1	53.6	48.6	45	42.2	40	38.3	53
50	89.4	75	65.5	58.5	53.5	50	47.2	45.2	43.5	58
55	92.9	78.9	69.8	63.1	58.4	55	52.3	50.3	48.6	62
60	96.3	82.9	74.2	67.8	63.2	60	57.4	55.4	53.8	67
65	99.7	86.8	78.5	72.4	68.1	65	62.5	60.5	58.9	72
70	103.1	90.8	82.9	77.1	73	70	67.5	65.7	64.1	77
75	106.5	94.7	87.2	81.7	77.9	75	72.6	70.8	69.2	82
80	109.9	98.7	91.6	86.4	82.7	80	77.7	75.9	74.4	87
85	113.3	102.6	95.9	91	87.6	85	82.8	81	79.5	91
90	116.7	106.6	100.3	95.7	92.5	90	87.8	86.2	84.7	96
95	120.1	110.5	104.6	100.3	97.3	95	92.9	91.3	89.8	101
100	123.5	114.5	109	105	102.2	100	98	96.4	95	106
105	126.9	118.4	113.3	109.6	107.1	105	103.1	101.5	100.1	111
110	130.3	122.4	117.7	114.3	111.9	110	108.1	106.7	105.3	116
115	133.7	126.3	122	118.9	116.8	115	113.2	111.8	110.4	121
120	127.1	130.3	126.4	123.6	121.7	120	118.3	116.9	115.6	126
125	140.5	134.2	130.7	128.2	126.6	125	123.4	122	120.7	131

Table 3-4. ISO NOISE RATING (NR) CURVES

NEMA (National Electrical Manufacturers Association)

NEMA - Criteria (See Code Key Below)	OCTAVE BAND CENTER FREQUENCY - Hz								~dBA Equivalent
	63	125	250	500	1K	2K	4K	8K	
	SOUND PRESSURE LEVELS, Lp								
A	58	44	37	32	28	25	22	20	37
B	63	50	43	37	34	31	28	25	42
C	67	56	48	43	39	36	33	31	47
D	71	60	53	47	44	41	38	35	52
E	75	65	58	53	49	46	43	41	57
F	79	70	62	57	54	51	48	45	62
G	83	74	68	63	59	55	52	50	67
H	87	79	72	67	64	61	57	55	72

Urban—Residential

Peaking—daytime only	F
Peaking—nighttime only	D
Continuous—daytime only	E
Continuous—day and night	C

Urban—Nearby Industry

Peaking—daytime only	G
Peaking—nighttime only	E
Continuous—daytime only	F
Continuous—day and night	D

Area of Heavy Industry

Peaking—daytime only	H
Peaking—nighttime only	F
Continuous—daytime only	G
Continuous—day and night	E

Suburban—Residential

Peaking—daytime only	E
Peaking—nighttime only	C
Continuous—daytime only	D
Continuous—day and night	B

Very Quiet Suburban or Rural Residential

Peaking—daytime only	D
Peaking—nighttime only	B
Continuous—daytime only	C
Continuous—day and night	A

Table 3-5. NEMA CRITERIA

Chapter 4
TYPES OF NOISE CONTROL

Ideally, the optimum approach to noise control is the prevention (or reduction) of noise at its source before it becomes a problem. However, equipment re-design and/or operational changes necessary to accomplish this are seldom possible.

Effective noise control is normally achieved by means of: (a) isolation, (b) dissipation or (c) a combination of the two. This involves the use and application of sound absorption materials, acoustic shields and barriers (fixed and movable), acoustic enclosures and/or silencers.

Silencers

Typically, silencers are divided into three distinct categories: (a) the reactive "reflective" type, (b) the dissipative "absorptive" type and (c) a combination of these two basic types. (See Figure 4-1). Helmholtz resonators, side branch resonators and other silencers which are normally designed for a very narrow frequency range are not included in this discussion.

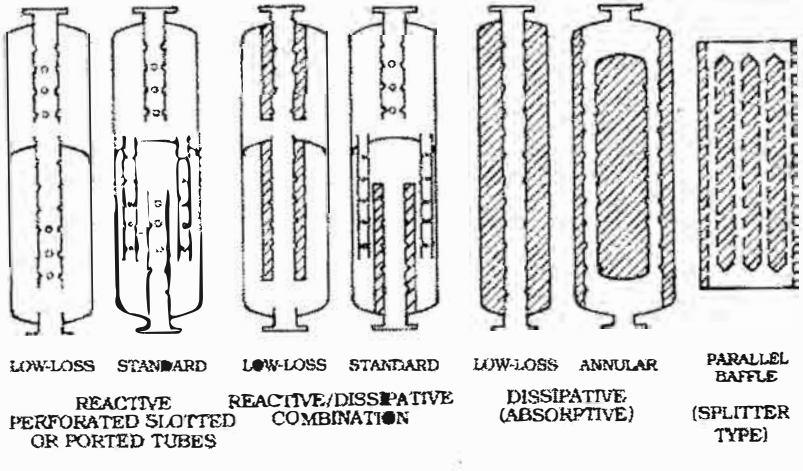


Figure 4-1. BASIC TYPES OF COMMERCIALLY AVAILABLE SILENCERS

The **reactive-type silencer** is generally restricted to relatively low-frequency applications, such as the intake (suction) and exhaust (discharge) of engines, blowers and compressors. It is largely dependent upon an area discontinuity (change) to reflect sound energy back to the source, and upon the dissipative effect of perforated, ported or slotted tubes for effective broad-band, low-frequency performance. Multi-chamber reactive silencers are available in: (a) straight-through tube arrangements for low-loss, pressure-drop applications and (b) in labyrinth-like, volume-tube configurations where pressure drop is not critical. Acoustic performance is a function of silencer diameter, overall volume and internal design. Absorptive material is not used.

A reactive-type silencer should be located as near the noise source as possible to avoid pipe resonances that may coincide with the fundamental frequency produced by the noise source. In cases where a close location is impossible or impractical, slight changes in pipe lengths may help to reduce pipe resonance. Even though critical pipe lengths can be calculated, there is no assurance that some degree of resonance will not occur.

The **dissipative-type silencer** is essentially a high-frequency, low-pressure drop attenuator. It depends on sound absorbing material to dissipate the sound energy and is usually applied on the intake (suction) and exhaust (discharge) of centrifugal compressors, FD fans, gas turbines, steam or process vents and similar equipment. Dissipative silencers are usually straight runs of acoustically lined piping or parallel baffles. Performance depends upon the internal design and the type of absorptive material used. The open flow area ranges from 25 to 75 percent, depending upon the required attenuation and allowable pressure drop.

The acoustic fill should be mineral wool, polyester, fiberglass or another durable, inert, vermin-proof, moisture-resistant material. For intake applications, the density of the acoustic fill is typically 2½ to 4 pounds per cubic foot packed under 10 percent compression to prevent voids. In exhaust service, the density is increased to a minimum of 6 pounds per cubic foot and, depending upon the temperature and unit velocity, the fill is normally protected with one or two wraps of fiberglass cloth or with one wrap of cloth and a stainless steel mesh screen, plus a perforated face sheet.

For service applications to 150°F, moisture and dust-proofing of the acoustical fill can be added by placing a sheet of polyethlene or mylar between the perforated face sheet and the acoustic fill. This reduces the acoustical performance, particularly in the upper frequency bands.

Generally, 3- to 4-inch-thick baffles are most effective in the higher frequencies, and thicker baffles are more effective in the lower frequencies. The silencer DIL is reduced with increased velocities. Conversely, there is an increase in silencer DIL when the noise and the flow are moving in the opposite directions — which is the case in all intake systems.

The **combination reactive/dissipative silencer** is functionally a reactive silencer with sound absorptive material to provide added high-frequency noise reduction. The perforated, slotted or ported tube applied to the reactive silencer acts as a dissipative element, reducing or eliminating troublesome pass-bands inherent to the basic reactive design. The performance of an effective reactive silencer is a result of both the dissipation and reflection of noise energy.

A selection of silencer type is based on both frequency content of the noise and the specific operation requirements.

Silencer performance or **DYNAMIC INSERTION LOSS** (DIL) is a function of the input intensity (amplitude) and frequency of the noise, silencer type and size, flow velocity, type of gas, temperature and surrounding environment. The greater the noise input, the greater the silencer DIL. Silencer performance is often categorized as being: (a) residential, (b) commercial or (c) industrial grade. (See Figure 4-2) However, generalized performance curves without regard to input amplitude, frequency and velocity are at best an approximation and should be avoided.

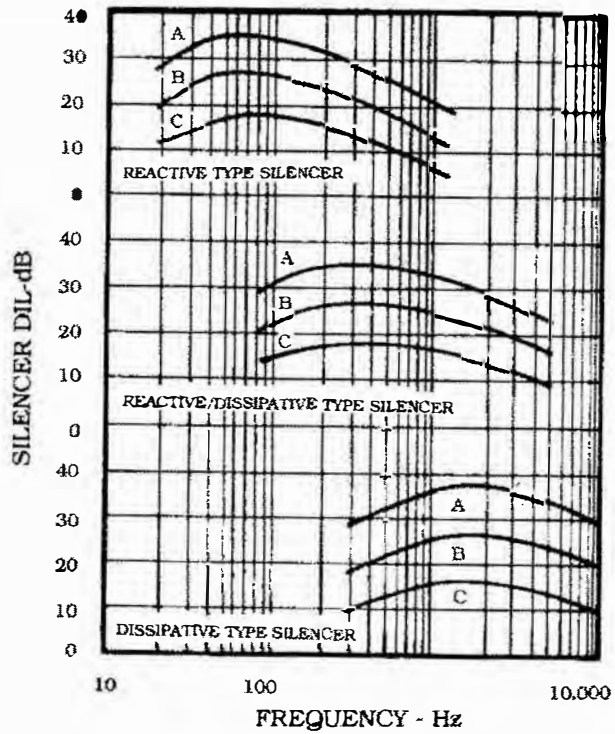


Fig. 4-2. GENERALIZED SILENCER PERFORMANCE CURVES
(A) Residential (B) Commercial (C) Industrial

DIL is the numerical difference in dB between two sound pressure levels measured at the same location, **under controlled flow conditions,** before and after installation of a silencer. Silencer insertion loss based on static conditions (no flow) is not a reliable measure of performance. Dynamic insertion loss is often relatively easy to measure, but it's difficult to calculate or predict theoretically.

Regardless of application and service, a silencer is rated not only on its effectiveness, but on its durability and life expectancy. No matter how fine a unit's performance, a structural failure can mean costly repairs or even complete replacement.

ATTENUATION is the difference in dB between two points in and along the path of sound propagation. In atmospheric systems, attenuation is the difference in sound levels, measured before and after the installation of a silencer. Attenuation, noise reduction (NR) and insertion loss are often used interchangeably in describing silencer performance.

Acoustic Enclosures and Partial Barriers

Acoustic enclosures of varying types and arrangements with and without ventilation are one of the most effective means for: (a) containment of excessive noise and (b) isolation of workers from the noise.

The most common acoustical enclosure is a modular arrangement of acoustical panels and connecting components. Commercial panels are available in four basic designs. (Figure 4-3).

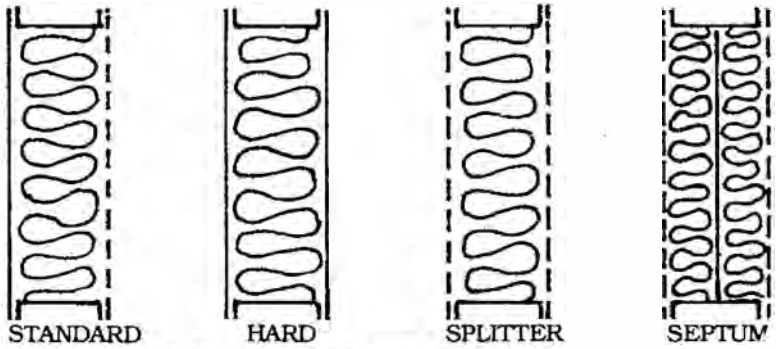


Figure 4-3. BASIC ACOUSTIC PANELS

The **standard panel** is the most common, providing both "semi-anechoic" absorption and dampening of high-frequency noise. The face sheet is perforated. Multiple layers of high-density sound absorption material may be applied for increased low-frequency noise reduction.

The **hard panel** is solid on both sides. Its use is limited but is very effective where noise isolation (or containment) is required.

The **splitter panel** is perforated on both sides and is used in many industrial applications such as an interior wall or partition.

The **septum panel** is similar in design to the splitter panel but with a solid divider in the center for added transmission loss (TL).

Flexible and functional, the panel concept has numerous applications in areas requiring acoustical treatment, such as:

a. Personnel enclosures
b. Equipment enclosures
c. Walls and partitions (fixed and movable)
d. Louvers and splitter-type silencers

To be acoustically effective, a panel must provide a high degree of both sound absorption and noise transmission loss. Many commercial modular panels are tested and rated by independent laboratories as shown in the Table 4-1 example.

	OCTAVE BAND CENTER FREQUENCY - Hz						
	125	250	500	1K	2K	4K	8K
Absorption Coefficient, α	0.74	1.05	1.14	1.11	1.01	1.04	1.00
Transmission Loss, dB	23	28	37	45	49	54	51

Table 4-1 TYPICAL PANEL PERFORMANCE
(Industrial Applications)

The design of connectors, windows and doors should be compatible with panel design. Ideally, panel connectors are effectively designed to prevent noise leaks between panels and to provide additional structural stability to the completed structure.

In contrast to enclosing noise-producing equipment, panels are also used to create quiet rooms — shelters to isolate workers from excessive noise. A partial enclosure, or even a simple noise barrier providing a reduction of 8 to 12 dBA may be all that's needed to meet a 90-dBA criteria.

Where additional noise containment is necessary, total enclosures can provide reductions of 25 to 35 dBA or more, depending on the amplitude and frequency of the noise source. Provision for ventilation and other airflow requirements, special panel cutouts and accessibility for inspection and maintenance may be required. In hazardous areas, two access doors with special panic-type latches may be required to meet OSHA safety requirements.

Standard modular panel construction for industrial applications varies with the intended service, but typically consists of:

a. 16 to 18 Ga. solid back and internal framing
b. 20 to 22 Ga. perforated face sheet
c. 2½ to 6 pounds per cubic foot density acoustic fill

The acoustic fill for most industrial applications is either fiberglass or mineral wool, which are inert, moisture-resistant, non-combustible materials. The fill is packed under not less than 5 percent compression to prevent voids.

Most commercial acoustic panels are unpainted galvanized sheet steel, zinc-coated primed steel or painted carbon steel. Manufacturers should be contacted for specific details of construction and for application assistance. (For various panel and enclosure configurations, see Figure 4-4.)

The fire resistance combustion rating of the acoustic fill for general industrial service should not be less than:

a. Flame spread - 15
b. Fuel contribution - 15
c. Smoke developed - 0

Most commercially developed acoustic panels (4 inches thick, 4 pounds per cubic foot density, with a 16 Ga. solid back and 22 Ga. perforated face sheet) have a Heat Transfer Factor of 0.07 BTU per hour per square foot per degree Fahrenheit temperature difference of standard air.

Total enclosures housing heat-producing equipment usually require ventilation, and silencers for both intake and exhaust. Depending upon the amount of heat to be removed, the number of air changes per minute ranges from 1 to 10. Ventilation may be forced or natural.

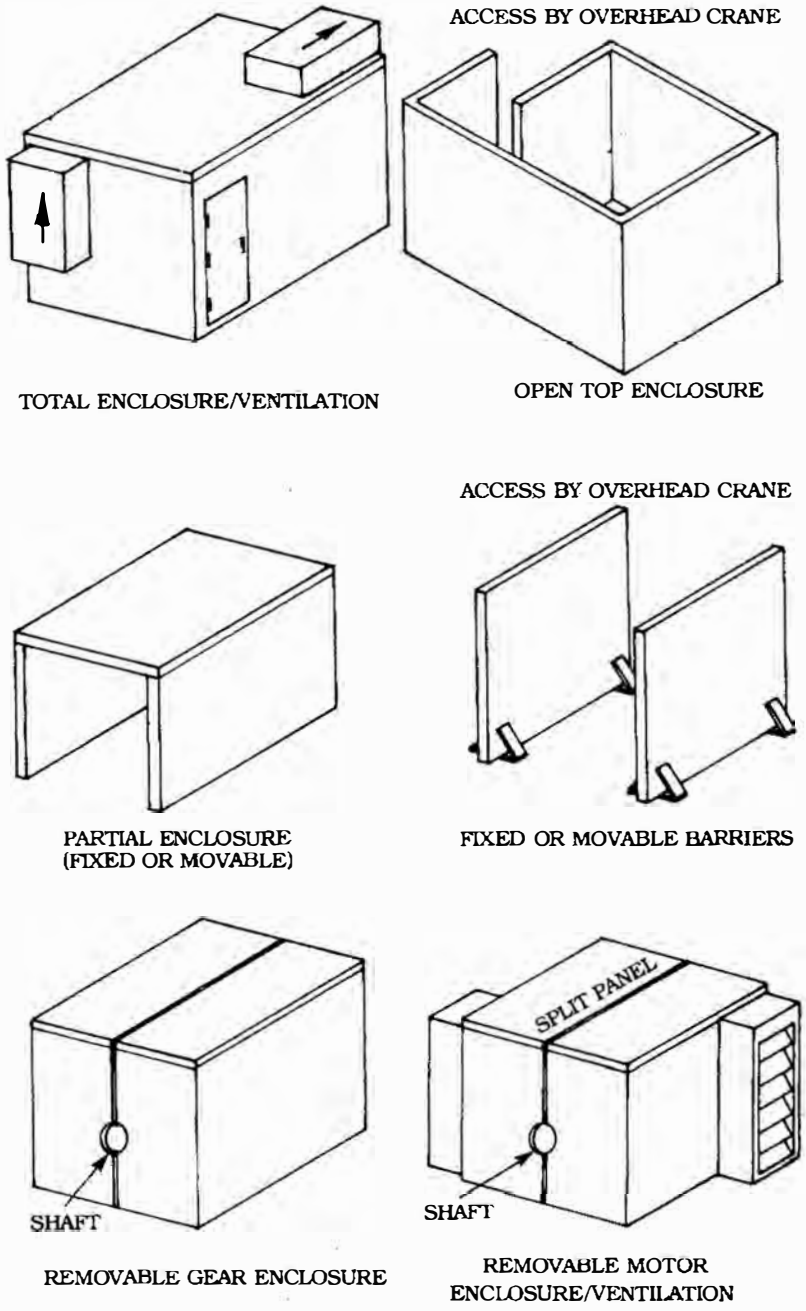


Figure 4-4. PANEL AND ENCLOSURE CONFIGURATIONS

When large-volume ventilation is required, forced ventilation is usually provided by fans installed on the exhaust side of the system. However, in hazardous gas service, the fans should be explosion-proof with non-sparking blades, and be installed on the intake in order to purge the enclosure.

Axial or centrifugal fans are generally used for moving large volumes of air against static pressures up to 2 inches H_2O. Higher-SP applications usually require vane-axial fans.

For an efficiently-designed forced ventilation system, the airflow requirements may be calculated as follows:

$$Q_V = \frac{F \times H}{\Delta T} + Q_E$$

Q_V = Required CFM

F = A factor involving air density and specific heat of air. This factor varies with altitude and air intake temperature (Table 4-2).

H = Total amount of heat (BTU/min) being dissipated into the enclosure. (1 hp ≃ 42.42 BTU/min.)

ΔT = Allowable Temperature rise within the enclosure - °F.

Q_E = Equipment intake CFM...engines, blowers, compressors, fans, etc. (where applicable).

Altitude (Ft)	AMBIENT TEMPERATURE - °F						
	0	20	40	60	80	100	120
0	48.3	50.4	52.6	54.7	56.8	58.9	61.0
1000	50.1	52.3	54.5	56.7	58.8	61.0	63.2
2000	52.0	54.3	56.5	58.8	61.0	63.3	65.6
3000	54.0	56.3	58.6	61.0	63.3	65.7	68.0
4000	56.0	58.4	60.8	63.3	65.7	68.2	70.6
5000	58.1	60.6	63.2	65.7	68.2	70.8	73.3

Table 4-2 VALUES OF FACTOR F $\left(\frac{\text{Cu. ft./lb.}}{0.24}\right)$

Natural (gravity) ventilation depends on wind velocity and/or temperature differential to create natural convection and move the air. The required free-throat area in natural ventilation systems may be approximated as:

$$A = \frac{Q}{26.4\,V + 9.4\,\sqrt{(h\,\Delta T)}}$$

A = Inlet throat free area - sq. ft. (assume inlet and outlet areas to be equal).
Q = CFM of Air to be removed.
V = Average wind velocity - MPH.
h = Vertical Height - Ft. (Distance between inlet and outlet).
ΔT = Temperature Rise - °F (Temperature difference between the inside air and the ambient outside air).

Acoustic Pipe Lagging

Acoustic lagging is the treatment of piping and equipment to prevent the radiation of noise in the surrounding area. Its effectiveness is measured in terms of transmission loss (TL) in decibels.

When pipe lagging is required, the piping should be acoustically insulated with a covering consisting of:

a. Fiberglass - 2½ to 6 pounds per cubic foot density
b. Mineral wool - 4 to 8 pounds per cubic foot density
c. Or other approved equal (foam-glass is not recommended as it tends to pulverize under loading and vibration)

When piping temperatures exceed those recommended for the above materials, calcium silicate insulation is usually added as a thermal barrier before the acoustic covering is applied. Cold piping may be provided with a base layer of closed cell rubber sheeting in lieu of the acoustic covering.

The acoustic covering generally has an external jacket of: (a) 26 to 28 Ga. galvanized steel, (b) 16 to 20 Ga. aluminum (preferred in low-temperature applications) or (c) a lead/aluminum laminate (26 Ga. aluminum bonded to 0.015 lead). Environments detrimental to aluminum may require stainless steel or coated aluminum.

When applying the acoustic covering, all circumferential joints of the covering should be staggered and sealed with an adhesive. Longitudinal seams and adjoining sections are to be firmly butted together and sealed. All gaps and voids are to be packed with loose insulation. Field cut the acoustic covering to snugly fit around irregular shapes, elbows, flanges and valves. The acoustic covering is then externally jacketed. Overlap the seams by about 2 inches and apply stainless bands on 9-10 inch centers (use of screws and rivets alone is not recommended).

The type, method and thickness of recommended acoustic treatment varies with the noise reduction requirements, piping temperature and other operating conditions as indicated in table 4-3. Where noise leaks exist at supports and other points in the system being treated, the most reduction that can typically be achieved is about 15 dBA.

Reduction Required (dBA)	Frequency Spectrum	Minimum Recommended Treatment*
10-15	High Frequency	2 in. thick, 4-6 lbs./cu. ft. acoustic covering.
10-15	Medium to Low Frequency	4 in. thick, 6 lbs./cu. ft. acoustic covering.
15-18	Medium to High Frequency	Two 2 in. layers of 6 lbs./cu. ft. single foil-faced absorptive material with facings inside.
-20	All Frequencies	Same as preceding except add 0.5 lb./ sq. ft. lead or lead vinyl (Septum) sheet between layers.
-25	All Frequencies	Same as preceding except add an additional 0.5 to 1.0 lb./sq. ft. sheet under outer jacket

* All absorptive materials should be covered with 26-28 gauge galvanized steel or 16-20 gauge aluminum (or equivalent) external jacket.

Table 4-3. TREATMENT vs. AVERAGE NOISE REDUCTION

Chapter 5
TYPICAL NOISE PROBLEMS AND RECOMMENDED SOLUTIONS

The sources of plant noise are as widely varied as the plant processes and operating facilities themselves, and no attempt is made here to list these sources by priority. Rather, the more common sources are individually discussed. Often the general solution to one problem can be applicable to another, yet separate noise sources in the same area may require entirely different solutions.

Where air, gas or steam flow is involved, as in turbines, engines, compressors, blowers, vents, and blowdowns, the most common noise control method is the installation of silencers. The proper selection and sizing of a silencer is of utmost importance to insure that pressure drop, acoustical performance (DIL), and other specific design criteria are met. Other noise controls are isolation and containment provided by acoustical enclosures, pipe lagging and similar methods.

There are many misconceptions about easy solutions to noise problems today. It's tempting to search for a simple modification or operational change as an after-the-fact solution. But rarely is such an approach cost effective.

One of the common myths in noise control is that all commercial silencers produce a costly pressure drop. Not so. Under optimum conditions, a silencer that is sized, designed and installed properly will create little or no measurable losses, and may even improve the efficiency of an engine or reciprocating compressor. Where pressure drop is critical, an acceptable limit can be designed into the silencer.

Another incorrect, but popular, belief is that acoustical treatment (lagging) of piping is a universal, inexpensive and convenient way to effectively reduce high-frequency pipe noise. Noise leaks and flanking paths, the complications of mechanical supports that must penetrate the lagging and connect to the structure, plus accessibility, maintenance and upkeep, can make a lagging system too costly to be practical. Also, field lagging rarely measures up to theoretical calculations.

In evaluating and selecting a silencing system, a manager too often relies upon the silencer specifications and warranty to insure a quiet operation. A common approach is to stipulate the maximum allowable noise levels and little more, leaving actual compliance up to the contractor or supplier. Of course, limits must be set, but performance specifications alone give little or no assurance that those levels will be met. Performance testing to verify the silencer guarantee is often complicated (or impossible) to conduct, without considerable expense and extensive evaluation.

As the first step in eliminating uncertainties, a manager should evaluate the various noise sources, individually and collectively, before issuing specifications. Unsilenced levels (either Lw or Lp) for all equipment should be obtained from the manufacturer and included in the specifications whenever possible. The silencer manufacturer then can and should guarantee the performance of a silencer, with all exceptions and limitations clarified. With that information, the manager may correctly analyze the overall system which may involve several sources of noise.

It is extremely important to evaluate the total system. For example, while a silencer might theoretically reduce the exhaust noise of an engine to 90 dBA at 10 feet, without effective silencing or isolation of the engine intake, casing and other sources of noise, the measured noise level in the area could actually be considerably higher.

Before the correct type and size of silencer can be determined for a given application, the required silencer DIL (attenuation) must be established. This is essentially the difference between the unsilenced Lp and the system criteria.

Determination of the unsilenced noise level (Lw or Lp), i.e., blowdown, engine exhaust, etc., is often extremely complex and is not attempted in this handbook. Actually, it is the responsibility of equipment manufacturers to establish and provide the unsilenced noise levels of their products, and many have been doing this for years. By the same token, it is the responsibility of the silencer manufacturer to provide the silencer DIL, and other pertinent information relating to overall performance of a particular silencer for a given application. In most of the applications that follow, the actual unsilenced levels used in the examples have been measured, calculated or provided by the manufacturer.

Vents and Blowdowns to Atmosphere

Vent and blowdown noise is a function of upstream pressure and temperature, type of gas being vented, the valve size and type, plus the effect of downstream piping.

Vent noise is relatively broad band, and of predictable high intensity (amplitude) in the immediate vicinity of the vent. Many theoretical aspects of vent and blowdown silencing are highly complex. Where usable theory has not yet been perfected or is not readily understood, the experience factor remains as important as scientific principles.

Silencer type and size is based upon:

- Service (continuous or intermittent)
- Amplitude (Lw or Lp) of the unsilenced noise
- Peak frequency (fp) of the unsilenced noise
- Specified noise criteria (required silencer DIL)
- Type of gas being vented
- Upstream pressure and temperature
- Flow rate and allowable pressure drop
- Cost considerations

A primary factor in vent silencing is to determine whether a noise source is continuous or intermittent. Continuous service is as implied — continuous venting under near-stable flow conditions. Intermittent noise is evaluated in terms of: (a) amplitude and frequency content, (b) duration of blowdown and (c) frequency of occurrence.

The formulas for calculating the noise produced by vents and blowdowns to atmosphere are, for the most part, empirical rather than purely theoretical. No single accepted method has yet been derived which includes all of the parameters that must be considered. So, actual measured Lp values, rather than calculated or extrapolated levels, should be obtained and used for analysis purposes whenever possible.

The peak frequency and spectrum shape of the unsilenced noise determines the silencer type. Low-pressure, large-valve vents to atmosphere generate low frequency noise. High-pressure, small-valve, high-velocity vents to atmosphere characteristically generate high frequency noise. No two vents are exactly the same. Piping downstream of the valve affects the spectrum shape. Larger piping causes a shift toward the lower frequencies as shown in Figures 5-1 and 5-2.

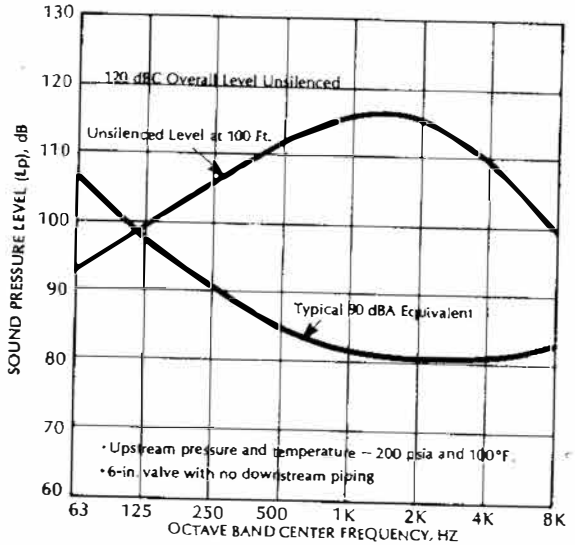


Figure 5-1. TYPICAL AIR VENT TO ATMOSPHERE

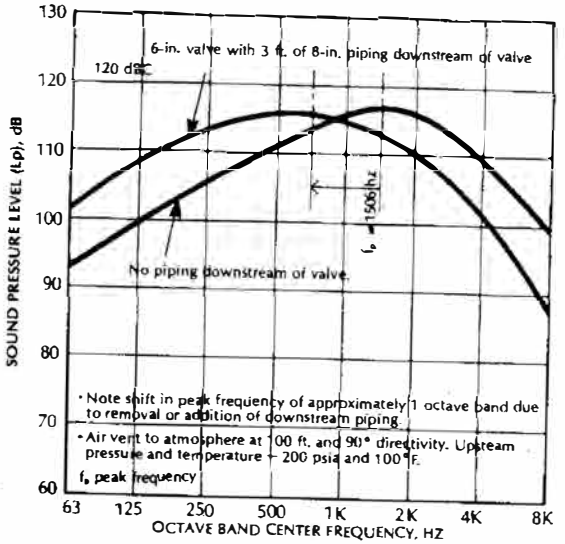


Figure 5-2. EFFECT OF EXIT PIPING DOWNSTREAM OF VALVE

Both the unsilenced noise (Lw) and the peak frequency (fp) have been calculated and tabulated (Table 5-1), covering a significant range of air-vent operating conditions. For gases other than air, the correction charts in Figures 5-3 and 5-4 may serve as a guide. Actual measured levels should be used when available.

Table 5-1. CALCULATED VENT POWER LEVELS (Lw) AND PEAK FREQUENCIES (fp) AIR VENTS TO ATMOSPHERE

Upstream Press. psia	Upstream Temp. °F	Valve Size (in.) Dia. — Overall vent Lw, dB re 10^{-12} watts, are shown in top of each square; vent peak frequency, fp Hz, are shown in bottom of each square.												
		1½	2	3	4	6	8	10	12	14	16	18	20	24
25	0	111 2796	113 2097	117 1398	119 1048	123 699	125 524	127 419	129 349	130 300	131 262	132 233	133 210	135 175
	100	114 3085	117 2314	120 1542	123 1157	126 771	129 578	131 463	132 386	133 331	135 289	136 257	137 231	138 193
	500	123 4039	126 3029	129 2019	132 1515	136 1010	138 757	140 606	142 505	143 433	144 379	145 337	146 303	148 252
	750	127 4534	130 3401	134 2267	136 1700	140 1134	142 850	144 680	146 567	147 486	148 425	149 378	150 340	152 283
50	0	134 4083	137 3062	140 2042	143 1531	146 1021	149 766	151 612	152 510	154 437	155 383	156 340	157 306	158 255
	100	138 4505	140 3379	144 2253	146 1689	150 1126	152 845	154 676	156 563	157 483	158 422	159 375	160 338	162 282
	500	145 5899	147 4424	151 2949	153 2212	157 1475	159 1106	161 885	163 737	164 632	165 553	166 492	167 442	169 369
	750	146 6622	149 4967	152 3311	155 2483	158 1656	161 1242	163 993	164 828	165 710	167 621	168 552	169 497	170 414
100	0	140 4888	143 3666	146 2444	149 1833	152 1222	155 917	157 733	158 611	160 524	161 458	162 407	163 367	164 306
100	100	143 5394	146 4045	149 2697	152 2023	155 1348	158 1011	160 809	161 674	163 578	164 506	165 449	166 405	167 337
	500	149 7062	152 5296	155 3531	158 2648	161 1765	164 1324	166 1059	167 883	169 757	170 662	171 588	172 530	173 441
	750	150 7928	153 5946	156 3964	159 2973	162 1982	165 1487	167 1189	168 991	170 849	171 743	172 661	173 595	174 496
200	0	142 5459	145 4095	148 2730	151 2047	154 1365	157 1024	159 814	160 682	161 585	163 512	164 455	165 409	
	100	145 6024	148 4518	151 3012	154 2259	157 1506	160 1129	162 904	163 753	165 645	166 565	167 502	168 452	
	500	152 7887	154 5915	158 3943	160 2958	164 1972	166 1479	168 1183	170 986	171 845	172 739	173 657	174 592	
	750	153 8855	156 6641	159 4427	162 3320	165 2214	168 1660	170 1328	171 1107	173 949	174 830	175 738	176 664	
300	0	143 5723	146 4292	149 2862	152 2146	155 1431	158 1073	160 858	161 715	163 613	164 537	165 477		
	100	147 6315	149 4736	153 3157	155 2368	159 1579	161 1184	163 947	165 789	166 677	167 592	168 526		
	500	154 8268	156 6201	160 4134	162 3100	166 2067	168 1550	170 1240	172 1033	173 886	174 775	175 689		
	750	155 9282	158 6962	161 4641	164 3481	167 2321	170 1740	172 1392	173 1160	175 995	176 870	177 774		

Upstream Press. psia	Upstream Temp. °F	Valve Size (in.) Dia.												
		1½	2	3	4	6	8	10	12	14	16	18	20	24
400	0	144 5886	147 4415	150 2943	153 2207	156 1472	159 1104	161 883	162 736	164 631	165 552			
	100	148 6495	150 4871	154 3247	156 2435	160 1624	162 1218	164 974	166 812	167 696	168 609			
	500	155 8503	157 6377	161 4252	163 3189	167 2126	169 1594	171 1275	173 1063	174 911	175 797			
	750	157 9547	159 7160	163 4773	165 3580	169 2387	171 1790	173 1432	175 1193	176 1023	178 895			
500	0	145 6001	148 4501	151 3000	154 2250	157 1500	160 1125	162 900	163 750	165 643				
	100	149 6621	151 4966	155 3311	157 2483	161 1655	163 1241	165 993	167 828	168 709				
	500	156 8669	159 6502	162 4335	165 3251	168 2167	171 1625	172 1300	174 1084	175 929				
	750	158 9733	161 7300	164 4866	167 3650	170 2433	173 1825	175 1460	176 1217	178 1043				
600	0	146 6088	149 4566	152 3044	155 2283	158 1522	161 1141	163 913	164 761					

600	100	149 6717	152 5038	155 3359	158 2519	161 1679	164 1259	166 1008	167 840					
	500	157 8795	159 6596	163 4397	165 3298	169 2199	171 1649	173 1319	175 1099					
	750	159 9874	162 7405	165 4937	168 3703	171 2468	174 1851	176 1481	177 1234					
800	0	147 6214	150 4661	153 3107	156 2330	159 1554	162 1165	164 932						
	100	151 6856	153 5142	157 3428	159 2571	163 1714	165 1286	167 1028						
	500	158 8977	161 6733	164 4489	167 3366	170 2244	173 1683	175 1347						
	750	161 10078	163 7559	167 5039	169 3779	173 2520	175 1890	177 1512						
1000	0	148 6303	151 4728	154 3152	157 2364	160 1576	163 1182							
	100	152 6955	154 5216	158 3477	160 2608	164 1739	166 1304							
	500	159 9106	162 6830	165 4553	168 3415	171 2277	174 1707							
	750	162 10223	164 7668	168 5112	170 3834	174 2556	176 1917							

Overall vent Lw, dB re 10^{-12} watts, are shown in top of each square; vent peak frequency, fp, Hz, are shown in bottom of each square.

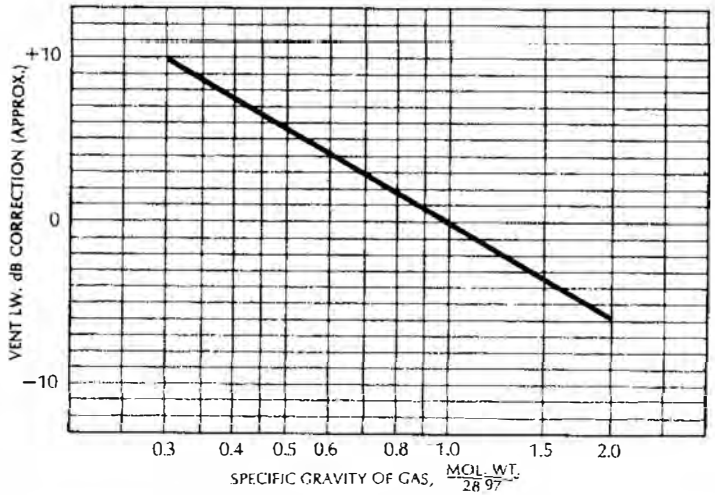


Figure 5-3. VENT POWER (Lw) CORRECTION

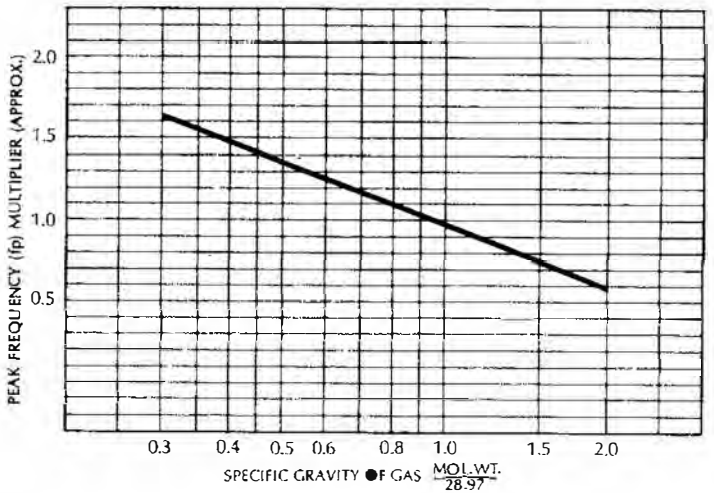


Figure 5-4. PEAK FREQUENCY (fp) MULTIPLIER

The Lw correction shown for gases other than air is based on the maximum flow that a valve will pass — at or slightly in excess of the critical pressure drop across the valve. The flow rate (ACFM) based on a 0.60 specific gravity gas through a given valve at a fixed pressure and temperature will be higher than that for air. For steam, the increase in noise is 3 to 4 dB. By throttling the steam valve until the two flows (ACFM) are equal, the steam vent noise will be reduced by an amount proportional to the reduction of the effective open port area of the valve.

Silencer exit velocities are based on the allowable Δ P and the silencing criteria. Typical operating velocities are:

a. Continuous vents -10,000-15,000 fpm
b. Intermittent blowdowns - 15,000-20,000 fpm

The acoustic limit is about 25,000 fpm.

To determine silencer and system Δ P based on actual operating conditions, use formulas in the APPENDIX.

Octave-band levels are determined by the peak frequency (fp), size and configuration of exit piping. For simplification of analysis, octave band corrections covering most applications have been generalized and grouped into two categories:

a. Downstream piping same size as valve, or no downstream piping (Table 5-2)
b. Downstream piping larger than valve (Table 5-3)

The individual octave band correction is subtracted from the overall (Lw) level.

Table 5-2. SPECTRUM CORRECTION
Valve with Same-Size or No Downstream Piping

Peak Frequence, (fp) Hz	Octave Band Center Frequency, Hz							
	63	125	250	500	1K	2K	4K	8K
187.5	−9	−5	−4	−8	−14	−20	−26	−32
219	−11	−5	−4	−7	−13	−19	−25	−31
250	−12	−6	−4	−6	−12	−18	−24	−30
312.5	−14	−8	−4	−5	−10	−16	−22	−28
375	−15	−9	−5	−4	−8	−14	−20	−26
437.5	−17	−11	−5	−4	−7	−13	−19	−25
500	−18	−12	−6	−4	−6	−12	−18	−24
625	−20	−14	−8	−4	−5	−10	−16	−22
750	−21	−15	−9	−5	−4	−8	−14	−20
875	−23	−17	−11	−5	−4	−7	−13	−19
1000	−24	−18	−12	−6	−4	−6	−12	−18
1250	−26	−20	−14	−8	−4	−5	−10	−16
1500	−27	−21	−15	−9	−5	−4	−8	−14
1750	−29	−23	−17	−11	−5	−4	−7	−13
2000	−30	−24	−18	−12	−6	−4	−6	−12
2500	−32	−26	−20	−14	−8	−4	−5	−10
3000	−33	−27	−21	−15	−9	−5	−4	−8
3500	−35	−29	−23	−17	−11	−5	−4	−7
4000	−36	−30	−24	−18	−12	−6	−4	−6
5000	−38	−32	−26	−20	−14	−8	−4	−5
6000	−39	−33	−27	−21	−15	−9	−5	−4
7000	−41	−35	−29	−23	−17	−11	−5	−4
8000	−42	−36	−30	−24	−18	−12	−6	−4

Table 5-3. SPECTRUM CORRECTION
Valve with Larger Downstream Piping

Peak Frequence, (fp) Hz	Octave Band Center Frequency, Hz							
	63	125	250	500	1K	2K	4K	8K
187.5	−5	−5	−8	−14	−23	−34	−45	−58
219	−5	−5	−7	−12	−21	−31	−42	−55
250	−6	−5	−6	−11	−19	−29	−40	−53
312.5	−7	−5	−5	−9	−16	−26	−36	−48
375	−7	−5	−5	−8	−14	−23	−34	−45
437.5	−8	−5	−5	−7	−12	−21	−31	−42
500	−9	−6	−5	−6	−11	−19	−29	−40
625	−11	−7	−5	−5	−9	−16	−26	−36
750	−12	−7	−5	−5	−8	−14	−24	−34
875	−14	−8	−5	−5	−7	−12	−21	−31
1000	−15	−9	−6	−5	−6	−11	−19	−29
1250	−17	−11	−7	−5	−5	−8	−14	−23
1500	−19	−12	−7	−5	−5	−8	−14	−23
1750	−21	−14	−8	−5	−5	−7	−12	−21
2000	−22	−15	−9	−6	−5	−6	−11	−19
2500	−25	−17	−11	−7	−5	−5	−9	−16
3000	−27	−19	−12	−7	−5	−5	−8	−14
3500	−29	−21	−14	−8	−5	−5	−7	−12
4000	−30	−22	−15	−9	−6	−5	−6	−11
5000	−33	−25	−17	−11	−7	−5	−5	−9
6000	−36	−27	−19	−12	−7	−5	−5	−8
7000	−38	−29	−21	−14	−8	−5	−5	−7
8000	−39	−30	−22	−15	−9	−6	−5	−6

The flow rate of vents and blowdowns to atmosphere (during critical flow pressure conditions) may be estimated using:

1. Air or Gas

$$SCFM\ (14.7\ psia\ and\ 60°F) = 17.8\ AP\sqrt{\frac{520}{(460 + °F)\ SG}}$$

A = Orifice Area Sq. In.
P = Initial Pressure, psia
SG = Specific Gravity of Gas

2. Steam
   a. Dry and Saturated

$$W\ (lbs/hr) = 51.43\ A\ P$$

   b. Wet Steam

$$W\ (lbs/hr) = 51.43\ A\ P\ Fm$$

$$Fm = \frac{1}{1\text{-}0.012m} \qquad m = \%\ moisture$$

   c. Superheated Steam

$$W\ (lbs/hr) = 51.43\ A\ P\ F\ s$$

$$Fs = \frac{1}{1 + 0.00065\ Ts} \qquad Ts = °F\ Superheat$$

In critical flow pressure applications, pressure drop across the valve in excess of the critical ΔP ($P_1 - P_2$) does not increase the flow but does increase the noise level. So, where possible, the ΔP across the valve should not exceed the critical ΔP. For steam, the critical pressure is 58 percent of the system pressure. For air, it is 53 percent of the system pressure. The ΔP across the silencer should not exceed either the critical flow pressure downstream of the valve, or the design pressure of the inlet diffuser.

The diffuser absorbs the impact of the vent pressure at the inlet of the silencer, distributes the flow preventing channeling and provides a shift or conversion of the low frequencies to the more easily attenuated high-frequency bands.

Selection of a suitable steam/air vent silencer for hogging jets or steam ejectors is based on the combined flow of the steam and the air. First, determine the quantity of steam in CFM, based on the temperature of the mixture. Second, determine the airflow, also in CFM, at the temperature of the mixture. The total CFM is the sum of the airflow and the steam flow.

The temperature of the mixture entering the silencer is a function of both the steam and air temperatures and quantities. This entrance temperature, if not given, must be determined from a heat balance based on the flow rates in pounds and the temperatures of both the steam and air. The steam discharge pressure is assumed to be atmospheric and the temperature before mixing, but **after expansion** is determined from the steam tables found in the APPENDIX.

Temperature of the **steam/air mixture**, if not stated, may be calculated:

$$Tm = \frac{2\ WsTs + Wa\ Ta}{Wa + 2\ Ws}$$

Where Tm = Temperature of mixture (°F)
Ts = Temperature of Steam after Expansion (°F)
Wa = Weight of Air (lbs/hr)
Ta = Temperature of entering air (°F)
Ws = Weight of Steam (lbs/hr)

Most steam/air ejectors discharging to atmosphere are designed for operation on steam of 40 psig and above. The higher the steam pressure and the larger the ejector, the noisier the discharge to atmosphere.

Steam/air ejector silencers are typically the reactive type with velocities ranging from 4,000 to 8,000 fpm, and allowable pressure drop of 5 to 10 inches H_2O, including exit loss to atmosphere. The silencer size should in no case be smaller than the ejector outlet for maximum ejector efficiency.

Silencer pressure drop, for steam/air mixtures, may be calculated by first determining the relative density of the mixture:

$$\Delta P = C \times \left(\frac{V}{4005}\right)^2 \times RD = "H_2O$$

Where C = Silencer coefficient including exit loss (1.0) to atmosphere - SEE APPENDIX

V = Silencer velocity (fpm)

$$RD = \text{Relative Density} = \frac{0.222\ (Wa + Ws)}{\text{Total CFM}}$$

An ejector is essentially a pump which moves a fluid from a low pressure to a higher pressure using the energy of a motivating fluid at a still higher pressure. A typical application is the removal of air from a condenser by maintaining a vacuum on a dryer.

The approximate BLOWDOWN TIME required to vent a pressurized system to atmosphere (where critical flow conditions exist across the valve) may be determined:

$$t\ (sec) \simeq \frac{5.5\ V}{A}\left(\ell n\ \frac{Pa}{Patm}\right)\sqrt{\frac{SG}{ZTa}}$$

V = Pressurized volume (cu. ft.)

A = Effective Area (Valve throat area x Flow Coefficient), Sq. In.

SG = Specific Gravity of Gas

Z = Compressibility Factor (at initial pressure)

Ta = Initial Temperature, °Abs. (460° + °F)

Pa = Initial Line Pressure, psia

Patm = Atmospheric Pressure, psia

ℓn = Log base e

Use flow coefficient of 0.85 unless the actual coefficient of the valve is known.

Caution is the key word to remember in silencing vents and blowdowns from high pressures. Silencer design, especially in the inlet section and across the diffuser, must be sufficiently strong to withstand the forces exerted during each blowdown cycle, plus a nominal safety factor.

Typical vent and blowdown applications:

- Safety valves
- Station and pipeline blowdowns
- Steam ejectors
- Switch valves (reversing valves)
- Autoclaves
- Hogging jets
- Snort valves
- Process control valves
- Compressor and blower bypass valves
- Boiler start-up purge systems

The type of vent or blowdown silencer is determined by the application, required performance and intended service. The size is generally based on design and velocity considerations.

Most vents produce predictably high-intensity noise which must be effectively silenced — not only to protect workers from possible hearing loss, but to foster improved communications, increased productivity and a quieter neighborhood environment overall.

EXAMPLE 1

TYPE: Air Vent to Atmosphere SERVICE: Continuous
OPERATING CONDITIONS: Flow - 30,000 ACFM at 14.7 pasia and 500°F
Upstream pressure and temperature: 100 psig and 500°F
Valve size: 4"
Pipe size: 8" Sch-- 40

SILENCER CRITERIA: Max. allowable Lp, -90 dBA @ 100 Ft.

SILENCER RECOMMENDATION: Burgess LCV-18/8 inlet

$$\text{Velocity: } \frac{30{,}000}{1.715} = 17{,}493 \text{ fpm}$$

$$\Delta P = 0.663 \times 10 \times \left(\frac{17{,}493}{4005}\right)^2 \times \frac{28.9}{960} = 3.81 \text{ psi}$$

ACOUSTIC ANALYSIS

LINE	DESCRIPTION	REFERENCE	OCTAVE BAND CENTER FREQUENCY (Hz) 63	125	250	500	1000	2000	4000	8000
	Over-all vent Lw 158 dB re 10^{-12}w	Table 5-1	158 + 0 = 158 dB							
2	Vent peak frequency 2648 Hz	Table 5-1								
3	Spectrum Correction	Table 5-3	-25	-17	-11	-7	-5	-5	-9	-16
4	Calc. octave band vent Lw	Line 1 combine w/line 3	133	141	147	151	153	153	149	142
5	Divergence to 100 Ft.	Table 2-2	-37	-37	-37	-37	-37	-37	-37	-37
6	Vertical Directivity 18 Ø at 135°	Table 2-3					-1	-3	-7	-13
7	Excess Air Attentuation	Fig. 2-6								-1
8	Calc. vent Lp * @ 100 Ft.	Line 4 combine w/lines 5.6 & 7	96	104	110	114	115	113	105	91
9	Silencer DIL, dB	Silencer Manufacturer	-16	-22	-25	-29	-34	-35	-33	-25
10	Silencer Lp @ 100 Ft.	Line 8 combine w/ line 9	80	82	85	85	81	78	72	66
11	"A" scale correction	Standard sound level meter	-25	-16	-9	-3	0	+1	+1	-1
12	Silenced 86 dBA	Line 10 combine w/line 11 Fig 2-3	55	66	76	82	81	79	73	65

*Enter actual measured values when available.

EXAMPLE 2

TYPE: Steam safety blowdown SERVICE: Intermittent (2 hours per day)
OPERATING CONDITIONS: 50,000 Lbs/ Hr, super-heated steam
Upstream pressure and temperature 300 psia and 600°F
Valve size: 3"
Pipe size: 6"

SILENCER CRITERIA: Max. allowable Lp, 100 dBA @ 50 Ft.

From steam tables, enthalpy (h) @ 300 psia and 600 °F = 1314.7 BTU/Lb.
At atmospheric conditions, the specific volume (v) of the steam, based on the enthalpy which remains constant, is 41.1 cu. ft./Lb. The temperature is reduced to approx. 558°F.

$\frac{50{,}000 \times 41.1}{60}$ 34,250 ACFM @ 14.7 psia and ~558°F

SILENCER RECOMMENDATION: Burgess ACV-18/6 inlet

$$\text{Velocity: } \frac{34{,}250}{1.715} = 19\,970 \text{ fpm}$$

$$\Delta P = 0.663 \times 11.5 \times \left(\frac{19\,970}{4\,005}\right)^2 \times \frac{18}{1018} = 3.35 \text{ psi}$$

ACOUSTIC ANALYSIS

LINE	DESCRIPTION	REFERENCE	OCTAVE BAND CENTER FREQUENCY (Hz) 63	125	250	500	1000	2000	4000	8000
1	Over-all vent Lw 164 dB re 10^{-12}W	Table 5-1 Fig. 5-3	← 160 + 4 = 164 dB →							
2	Vent peak frequency 5638 Hz	Table 5-1 Fig. 5-4	4337 x 1.30 = 5638 Hz							
3	Spectrum Correction	Table 5-3	-35	-26	-18	-12	-7	-5	-5	-8
4	Calc. octave band vent Lw *	Line 1 combine w/line 3	129	138	146	152	157	159	159	156
5	Divergence to 50 ft.	Table 2-2	-31	-31	-31	-31	-31	-31	-31	-31
6	Vertical Directivity 18"ϕ at 135°	Table 2-3					-1	-3	-7	-13
7	Calc. vent Lp @ 50 ft *	Line 4 combine w/lines 5 & 6	98	107	115	121	125	125	121	112
8	Silencer DIL, dB	Silencer Manufacturer	-18	-22	-26	-29	-33	-37	-37	-31
9	Silencer Lp @ 50 Ft.	Line 7 combine w/line 8	80	85	89	92	92	88	84	81
10	"A" scale correction	Standard sound lever meter	-25	-16	-9	-3	0	+1	+1	-1
11	Silenced 96 dBA	Line 9 combine w/line 10; Fig. 2-3	55	69	80	89	92	89	85	80

* Enter actual or measured values when available.

Engines

All internal combustion engines produce noise — some more or less than others — and 2- and 4-cycle engines of equivalent hp and rpm produce roughly the same overall noise levels. Yet the frequency distribution and amplitude of the noise may vary significantly as shown in Figure 5-5.

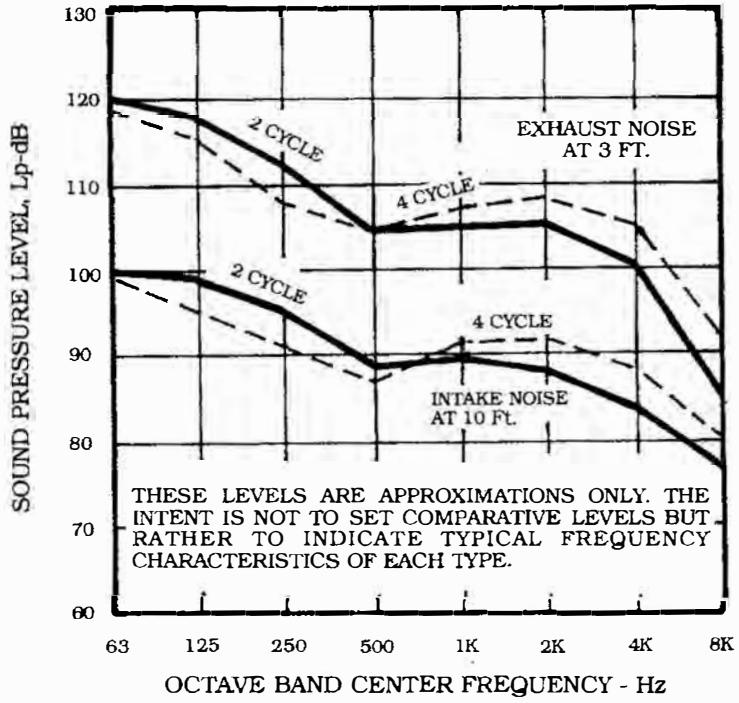


Figure 5-5. TYPICAL UNSILENCED NOISE LEVELS - FREQUENCY AND AMPLITUDE (2,000-hp Turbocharged Engine)

Power produced by an internal combustion engine is the result of combustion produced in the cylinders. The resultant noise is a function of horsepower, speed (rpm), scavenging means and type of ignition. As depicted in Figure 5-6, engine noise increases primarily with hp. The overall increase is about 3 dB each time the engine hp is doubled.

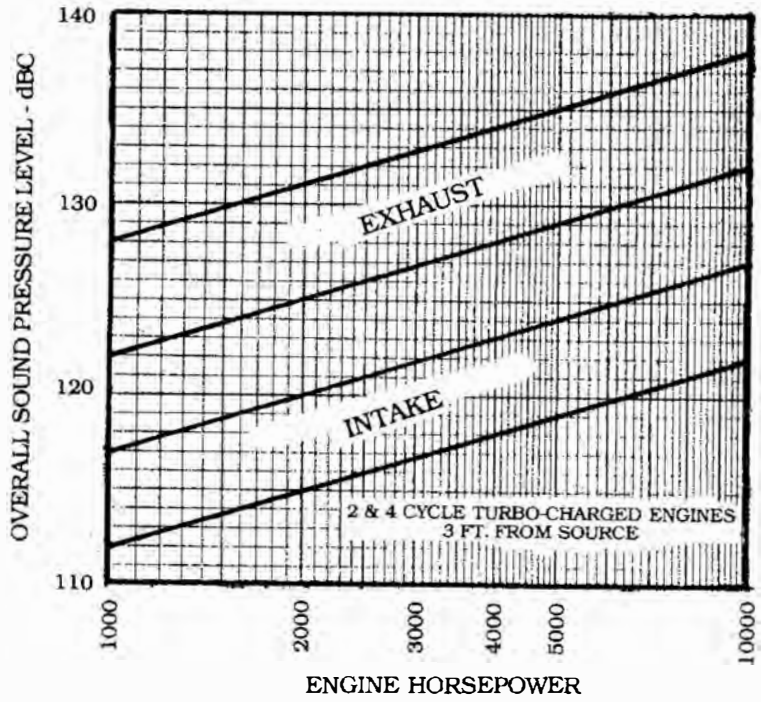


Figure 5-6. TYPICAL UNSILENCED NOISE LEVELS - INCREASING HORSEPOWER

The predominant sources that make up engine noise are:

a. Exhaust
b. Air intake
c. Cooling fan
d. Mechanical (casing)

Exhaust noise includes the various noise sources of the exhaust system. Besides the noise of the exhaust gas at the exit of the silencer or tailpipe to atmosphere, radiated noise from the silencer shell and piping must also be considered.

Air intake noise includes all noise sources of that system — the air filter, ducting and piping, as well as the air inlet itself.

Though seldom predominent, cooling fan noise may be a factor in critical applications. Mechanical radiated noise is produced by sources associated with mechanical components of the engine. Casing or mechanical noise is a result of acoustical radiation or vibration of the engine block, cover and accessories.

When predominant frequencies produced by an engine coincide with the natural frequencies of the exhaust system, there is usually a marked increase in noise and vibration. In acoustics, this is known as resonance. Theoretically, it is possible to avoid critical lengths of piping by "tuning" the exhaust system. However, actual results are often disappointing, for the optimum effects of tuning are usually marginal. The adverse effects of resonant pipe lengths can best be avoided by installing the silencer as close to the engine as possible, since short, unrestricted piping is less likely to respond acoustically. There are a few instances, however, where scavenging of an engine (certain 1- or 2-cylinder, low-rpm engines) will be markedly improved by tuning the piping. Specific pipe lengths, where applicable, must be determined by actual tests and should be provided by the engine manufacturer.

The fundamental frequency at which intake and exhaust piping may be excited depends upon the angular velocity of the engine crankshaft, the number of cylinders and their crank angle spacing, and whether the engine is two or four cycle. When the crank angle interval is not uniform, more than one fundamental frequency may develop. The valve action, non-sinusoidal motion of the engine pistons, non-linearity of the exhaust gas, curvature of the compression and expansion chambers — all act to distort the pressure waves resulting from the combustion within the cylinders and from the reciprocating action of the pistons. Other factors, such as scavenging and turbocharging, contribute to the behavior pattern of the fundamental and the development of the frequency spectrum.

The air intake system filters the air and provides the conduit (piping) to the engine. Nearly every engine is provided with an intake filter; the type and efficiency depend upon engine size, hp and operating environment.

Intake noise is largely dependent upon the scavenging means and is usually 8 to 10 dB less than the exhaust levels.

Engine intake silencers are grouped into three general types:

1. Multi-chamber (reactive-type) silencer
   a. 4-cycle, naturally-aspirated
   b. 2-cycle, positive-scavenged
2. Absorption (dissipative-type) silencer
   a. 2- and 4-cycle, centrifugal blower-scavenged
   b. 2- and 4-cycle, exhaust-turbocharged
3. Combination multi-chamber/absorption silencer
   a. 2-cycle, exhaust turbocharged
   b. 2-cycle, gear-driven, centrifugal blower-scavenged

The intake silencer size for 4-cycle, naturally-aspirated and 2-cycle, pump-scavenged engines is determined by both the intake slug volume and velocity. Silencer sizing for most other engines is based on velocity and allowable pressure drop (slug volume is not considered).

Intake slug volume = piston displacement of one cylinder x volumetric efficiency of the engine:

$$SV_i = \frac{\pi B^2}{4} \times S \times \frac{VE\ (\%)}{100} = \text{cu. in.}$$

Where B = cylinder bore, in.
S = cylinder stroke, in.

The intake slug volume sets the silencer size by volume (the minimum intake slug volume to silencer volume ratio is typically about 10:1). The second consideration is that of airflow (velocity and allowable pressure drop). Actual intake airflow (ACFM) = displacement CFM of the engine x volumetric efficiency.

4-cycle engines:

$$ACFM = \frac{\pi B^2 \times S \times N \times rpm \times VE\ (\%)}{4 \times 1728 \times 2 \times 100}$$

2-cycle engines:

$$ACFM = \frac{\pi B^2 \times S \times N \times rpm \times VE\,(\%)}{4 \times 1728 \times 100}$$

Where N = Number of Cylinders

The average intake silencer velocity, based on engine type and number of cylinders, is shown in Table 5-4.

Engine Type	Intake Velocity (fpm)
4-cycle,4 or more cylinders	4000 to 6000
2-cycle, 2 or more cylinders	4000 to 6000
4-cycle, 2 and 3 cylinders	2000 to 3000
2-cycle, 1 cylinder	2000 to 3000
4-cycle, 1 cylinder	1000 to 1500

Table 5-4 AVERAGE INTAKE SILENCER VELOCITIES

Engine exhaust is the primary source of engine noise, generally peaking at multiples of the fundamental engine firing frequency of the engine. Turbocharged engines develop a high-frequency component at one or more discreet frequency components. Even so, they do not normally produce as much noise as non-turbocharged engines. The difference in overall sound levels is about 6 dB for equivalent hp engines. The exhaust-driven turbocharger removes some of the sound energy from the gas, resulting in less noise. Yet noise from a turbocharged engine may be more objectionable due to high frequencies produced by the turbocharger, even though the overall noise amplitude is less.

Engine exhaust silencing best suited for most engines, and most commonly used in industry, falls into three categories:

1. Standard multi-chamber (reactive-type) silencer
   a. 4-cycle, naturally-aspirated
   b. 4-cycle, turbocharged where ΔP permits
   c. 2-cycle, positive blower-scavenged
2. Low-loss, straight-through, multi-chamber (reactive) silencer
   a. 2- and 4-cycle, exhaust-turbocharged
3. Combination multi-chamber/dissipative silencer
   a. 2- and 4-cycle, exhaust-turbocharged for critical high-performance applications

Exhaust flow and allowable pressure drop generally determine the size of the exhaust silencer, but again slug volume may be a consideration:

$$SVe = SV_1 \times \frac{460 + °F}{530} = \text{cu. in.}$$

The exhaust silencer size by volume is based on an exhaust slug volume-to-silencer ratio of about 10:1. The second consideration is that of exhaust flow (ACFM) and allowable pressure drop:

$$\text{Exhaust ACFM} = \text{Intake ACFM} \times \left(\frac{460 + °F}{530}\right)$$

So exhaust silencer size is determined first by slug volume and second, by velocity. The larger of the two sizes is recommended. The average exhaust silencer velocity based on engine type and rpm is shown in Table 5-5.

For 2- and 4-cycle turbocharged engines, exhaust silencer sizing is dictated by the engine exhaust connection size, exhaust gas flow and pressure drop. Slug volume is not a consideration.

Engine Type	Engine RPM	Exhaust Velocity (fpm)
2 and 4 Cycle Exhaust		
Low Speed	<350	4000 to 7000
Medium Speed	350 to 1200	6000 to 9000
High Speed	>1200	8000 to 10,000
Turbocharged	all speeds	5000 to 10,000

Table 5-5. AVERAGE EXHAUST SILENCER VELOCITIES

For silencer and system ΔP calculations based on actual operation conditions, use formulas in the APPENDIX.

Effective silencer DIL is limited only by the self-noise characteristics of a particular silencer design. Flow-generated self noise increases with velocity and silencer size as shown in Figures 5-7 and 5-8 and Table 5-6. There is a significant reduction in self-noise at elevated temperatures.

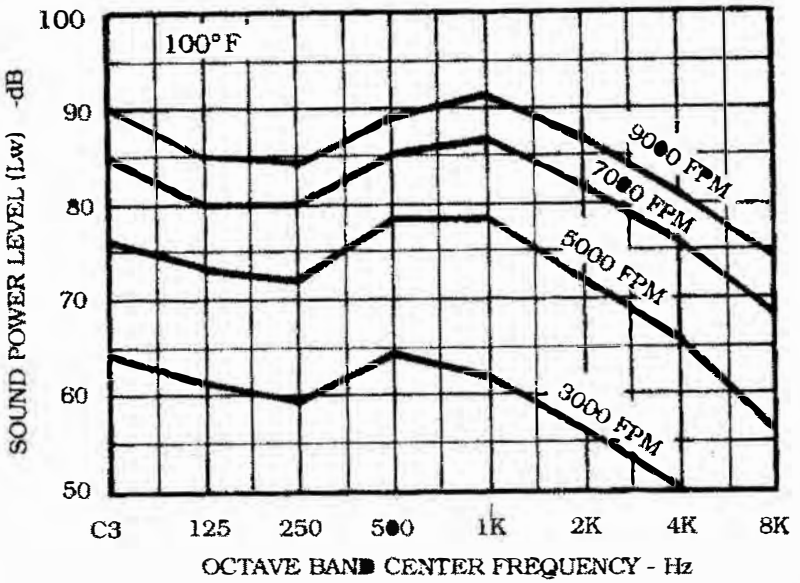


Figure 5-7. TYPICAL REACTIVE SILENCER SELF NOISE (6" SIZE MULTI-CHAMBER/STANDARD OFFSET TUBES)

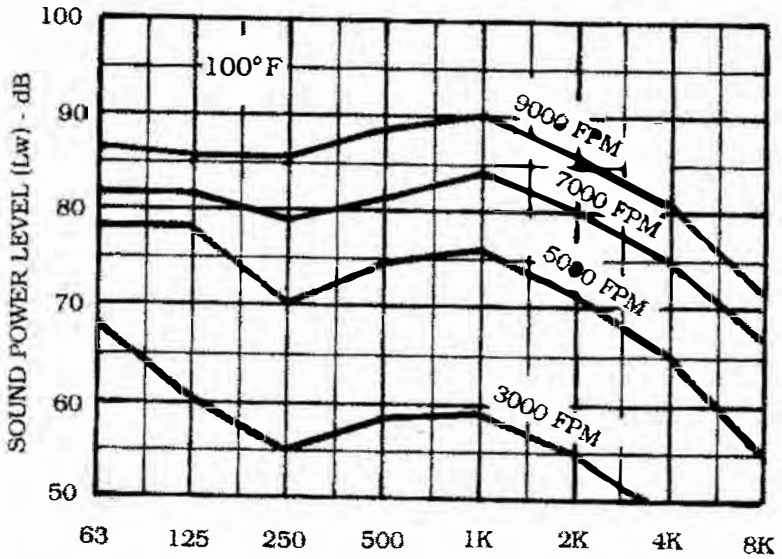


Figure 5-8. TYPICAL REACTIVE SILENCER SELF NOISE (6" SIZE MULTI-CHAMBER, LOW-LOSS, STRAIGHT-THROUGH DESIGN)

SILENCER SIZE (NOM)	3	6	12	18	24	36	48
Lw ADJUSTMENT, ~dB	-6	0	+6	+9	+12	+15	+18

Table 5-6. APPROXIMATE SELF NOISE CORRECTION BY SILENCER SIZE

A tailpipe is recommended for added silencer DIL in the first band and possibly the second. Optimum length is based on a ¼ wave length, which theoretically cannot support a standing wave. A pipe open at both ends will resonate at ½ wave lengths and so, should be avoided. Due to variations in engine rpm and temperature, specific tuning of the tailpipe should not be attempted.

Tailpipes of 2 to 10 diameters in length are recommended, but where possible, the length should not exceed:

$$\text{Max. Length (ft.)} = \frac{C}{4\,(f_1)} - \frac{D}{2}$$

(D/2: End Correction)

C = Speed of sound (fps) = $49.09\,(460 + {}^\circ F)^{1/2}$

f_1 = Fundamental Firing Frequency (Hz) f_1 (2 cycle) = $\frac{\text{No. of Cyl's x rpm}}{60}$

D = Tailpipe Dia. (ft.)

f_1 (4 cycle) = $\frac{\text{No. of Cyl's x rpm}}{2 \times 60}$

Contrary to popular belief, there is little to be gained by angle-cutting tailpipes or vent stacks. While a bias (bevel) cut provides no additional reduction of noise, it is more expensive than a square-cut end and the pipe is more vulnerable to vibration and stress.

Stationary engines are usually installed in a building or enclosure with intake (for larger engines) and exhaust (for all engines) ducted to the outside. At higher operating speeds, the engine and turbocharger casing may produce excessive noise within the building. Acoustic treatment (lagging) may be required for portions of the intake and exhaust piping and the turbocharger housing to prevent radiation of high-frequency noise. The interior walls of the building may also require some form of treatment. But rather than resorting to extensive wall and ceiling treatment to reduce the noise in multi-engine installations, such as compressor stations, a system of movable acoustical walls is often used to isolate and protect workmen during extended periods of maintenance and overhaul of individual units.

Normally about one-third of the fuel energy produced by internal combustion engines and gas turbines is released as exhaust heat to the atmosphere. In many cases, it's not only possible, but economically desirable, to recover a high percentage of this heat, converting the otherwise-wasted energy to useful work. Recovered heat can be used to heat buildings, heat water for general usage, produce low-pressure steam and many other energy-saving applications. Combination heat-recovery silencers will provide:

a. Increased silencing due to the reduction of the gas volume due to cooling
b. Heat recovery up to 1,500 BTU/BHP/HR
c. Lower initial cost (as compared to individual units)

The heat-recovery silencer may operate wet or dry. The heating elements are typically rolled pipe coils or multiple runs of straight piping. Extended surface fintubing is the most efficient. The elements are generally removable for inspection and maintenance and should be drained when operating dry.

Although the silencer does not require code construction or certification, heat recovery elements, connecting headers and piping are typically of code construction and are hydrostatically-tested to 50 psig or more, depending on actual operating conditions.

When a heat-recovery silencer is used for low-pressure steam, a steam chest with pumps, controls, gauges, valves and piping is required as shown in the diagram of Figure 5-9.

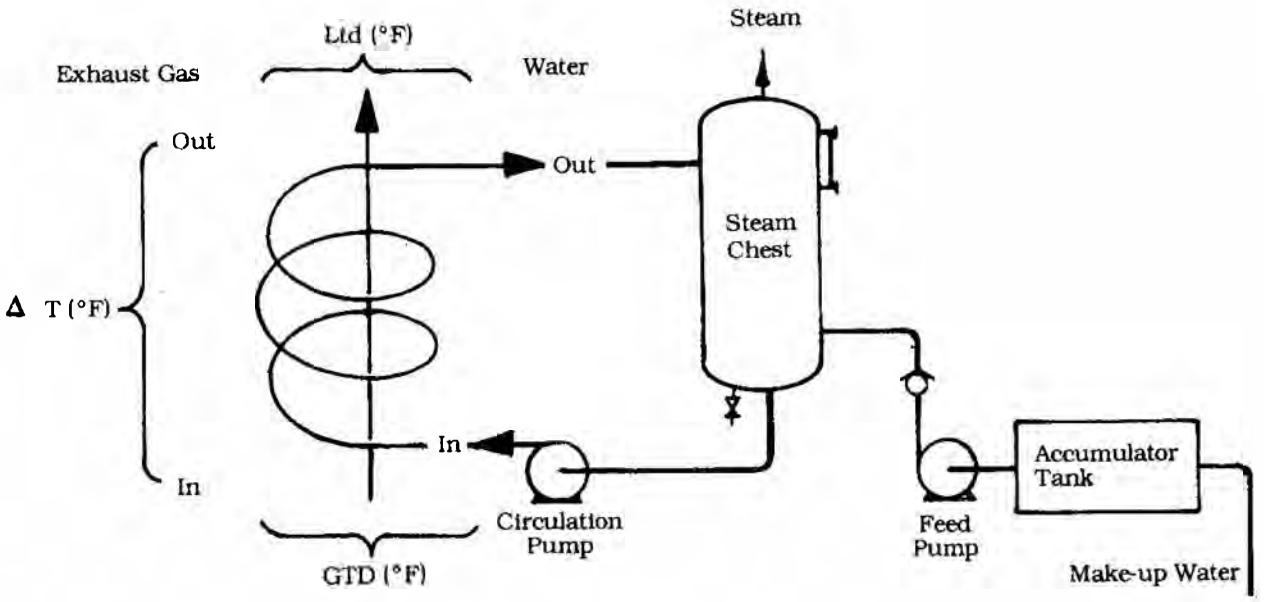


Figure 5-9. TYPICAL SCHEMATIC OF HEAT RECOVERY SILENCER SYSTEM

Max. Heat Available = Exhaust Gas (#/Hr.) x 0.24 x ΔT = BTU/hr.

Heat Recovered = LMTD x TSA x U = BTU/hr.

$$LMTD = \frac{GTD - LTD}{\ell n \frac{GTD}{LTD}}$$

TSA = Total Outside surface area (sq. ft.) of recovery coil or tubes

U = Overall heat transfer coefficient

ℓn = Log Base e

Most heat-recovery silencer systems are custom-designed to meet specific customer requirements.

EXAMPLE 3

TYPE: Engine intake and exhaust SERVICE: Continuous (day and night)
AREA CLASSIFICATION: Urban/nearby industry

SILENCER CRITERIA: Max. allowable Lp, NEMA D at 300 ft.

ENGINE DATA: 2,500 hp, 2 cycle, turbo-charged, 6 cylinder, 300 rpm

INTAKE / 20" ϕ CONNECTION

9,500 cfm at 14.7 psia and 40°F
ΔP = 1.0" H_2O max. (silencer only)

EXHAUST / 24" ϕ CONNECTION

23,000 cfm at 14.7 psia and 750°F
ΔP = 2.0" H_2O max. (silencer only)

SILENCER RECOMMENDATION

Burgess ACB-20 (dissipative type)
Velocity: $\frac{9,500}{2.123}$ = 4,475 fpm
Silencer $\Delta P = 0.75 \left(\frac{4,475}{4,005}\right)^2 \left(\frac{530}{500}\right) = 1.0"\ H_2O$

Burgess BMA-T-24 (reactive, straight-thru low ΔP design)
Velocity: $\frac{23,000}{3.071}$ = 7,489 fpm
Silencer $\Delta P = 1.25 \left(\frac{7,489}{4,005}\right)^2 \left(\frac{530}{1210}\right) = 1.9"\ H_2O$

ACOUSTIC ANALYSIS

			OCTAVE BAND CENTER FREQUENCY (Hz)							
LINE	DESCRIPTION	REFERENCE	63	125	250	500	1000	2000	4000	8000
1	Unsilenced intake Lp at 300 ft.	Engine Manufacturer	72	68	60	53	56	55	52	45
2	Filter DIL~dB	Filter Manufacturer		-1	-3	-4	-5	-5	-5	-5
3	Silencer DIL~dB	Silencer Manufacturer	-5	-10	-17	-23	-26	-25	-20	-13
4	Silenced intake Lp	Line 1 combined w/line 2 & 3	67	57	40	26	25	25	27	27
5	Unsilenced exhaust Lp at 300 ft.	Engine Manufacturer	82	80	72	63	63	60	55	50
6	Silencer DIL~dB	Silencer Manufacturer	-20	-30	-28	-21	-19	-19	-19	-20
7	Silenced exhaust Lp	Line 5 combined w/line 6	62	50	44	42	44	41	36	30
8	Total combined silenced L_p at 300 ft.	Line 4 combined w/line 7 (Fig. 2-3)	68	58	45	42	44	41	37	32
9	Allowable L_p max.	NEMA D (Table 3-5)	72	60	53	47	44	41	38	35

Gas Turbines

Most gas turbines are noisy and require noise control. The extent of the silencing depends upon the turbine size (hp), location, duty cycle and the silencing criteria. Both worker exposure and neighborhood noise levels must be considered, with worker hearing protection, communication and safety the major concerns.

Gas turbine systems are usually supplied by the turbine, generator or compressor manufacturer as a package complete with silencing and other related equipment.

The major sources of gas turbine noise are:

a. Intake
b. Exhaust
c. Casing (radiated)
d. Structural resonances

See Figure 5-10 for a comparative example of intake, exhaust and casing noise.

Intake noise is created by the interaction of the gas compressor blades and the stator, and is a function of pressure increase, number of blades and blade-tip speed. The overall level of the intake noise is somewhat less than that produced by the exhaust, although its high-frequency content sounds much louder than the higher-amplitude, lower-frequency exhaust noise.

Exhaust noise from the power turbine is the predominant source. Its increase in amplitude and shift to the lower frequencies is due to the combustion process. Casing noise is normally about 10 dB below the combined intake and exhaust levels.

The generator driven by the power turbine in electric power generation is not a major noise source. Nor is the gas compressor used in pipeline systems. However, when a gas generator and power turbine are enclosed, extension of the housing to include the electric generator or gas compressor should be considered.

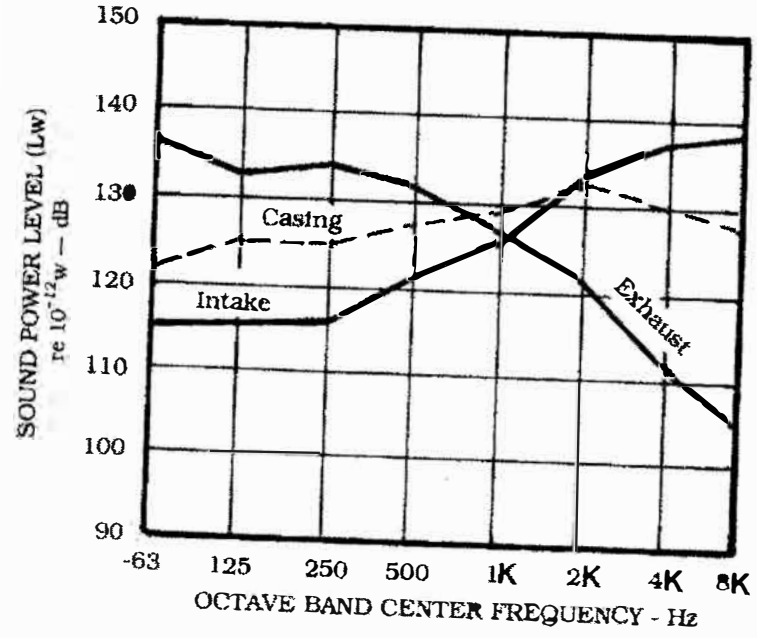


Figure 5-10. TYPICAL UNSILENCED NOISE LEVELS (16,000 Bhp GAS TURBINE PIPELINE COMPRESSOR UNIT)

The intake airflow to the gas turbine is usually stated in pounds per second. Conversion to ACFM is a function of ambient pressure and temperature:

Intake Flow = Lbs./Sec. x Specific Volume x 60 =ACFM

$$\text{Specific Volume} = \frac{10.72 \times Ta}{28.9 \times Pa} = \text{cu. ft./lb.}$$

The exhaust ACFM may be calculated by correcting for the elevated exhaust temperature:

$$\text{Exhaust Flow} = \text{Intake ACFM} \times \frac{460 + {}^{\circ}F}{530} \text{ ACFM}$$

Air filter and inlet plenums provide various amounts of noise reduction, ranging from 6 to 10 dB depending upon the frequency of the noise and the type of filter used. Heat-recovery systems will also provide a reduction, but generally of a lesser amount. Further, any analysis should take into account the effect of the multiple sources.

The major considerations in the application of noise control for gas turbines are:

a. Aerodynamic requirements (airflow distribution and pressure drop)
b. Acoustic requirements (total system)
c. Structural requirements (including corrosion, thermal expansion, seismic and windload design, etc.)

The intake system must be properly designed and installed so it does not restrict airflow at the turbine inlet. The same applies to the exhaust system. Typical pressure drop limits are shown in Figure 5-11.

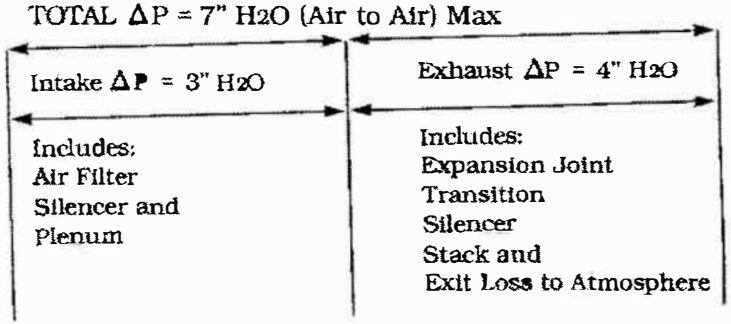


Figure 5-11. TYPICAL PRESSURE DROP COMPONENT LIMITS

For silencer and system ΔP calculations based on actual operating conditions, use formulas in the APPENDIX.

Gas turbines normally require high-performance, low-loss panel-type dissipative silencers on both the intake and exhaust systems. Sizing is based upon allowable pressure drop, silencing criteria and required connection size. Depending upon the internal design of the exhaust silencer, its velocity limit is about 15,000 fpm. Actual panel thickness and spacing between panels is determined by the service and frequency content of the noise.

A gas turbine silencing system, when provided as a complete package, often includes (but is not limited to) the following:

1. Air inlet system
   a. Filters (multi-stage) — inertial and media type
   b. Filter house/access ladder and platform
   c. Silencer and plenum assembly/blow-in door and pressure drop limit switch
   d. Structural support
2. Gas generator and power turbine enclosure/access doors and provision for turbine removal
3. Exhaust system
   a. High temperature expansion joint
   b. Transition assembly
   c. Silencer assembly
   d. Stack extension
   e. Structural support
4. Secondary air (ventilation) fire dampers
5. Optional automatic fire protection (Halon or CO_2) system
6. Optional enclosure for electric generator or centrifugal compressor

EXAMPLE 4

TYPE: Gas turbine intake and exhaust silencers
AREA CLASSIFICATION: Heavy industry

SERVICE: Continuous (daytime only)

SILENCER CRITERIA: NEMA G at 400 ft.

TURBINE DATA: 4,000 hp gas turbine/compressor

INTAKE / CONNECTION 32" ϕ

27,500 cfm at 14.7 psia and 80°F
Allowable ΔP: 1.0" H_2O max. total

SILENCER RECOMMENDATION

Burgess M-12-36 x 48 x 48/outlet transition
(dissipative splitter type silencer: 50% open)

Velocity: $\frac{27,500}{6}$ = 4,583 fpm (throat)

Silencer ΔP = 0.5" H20 (silencer only)

EXHAUST / CONNECTION 24" x 72"

69,250 cfm at 14.7 psia and 900°F
Allowable ΔP: 2.0" H_2O max. total

Burgess H-12-48 x 72 x 84/inlet transition
(dissipative splitter type silencer: 33--1/3% open)

Velocity: $\frac{69,250}{8.00}$ = 8,650 fpm (throat)

Silencer ΔP = 1.0" H20 (silencer only)

ACOUSTIC ANALYSIS

-20 log 400 + 2.5 = -50 dB

LINE	DESCRIPTION	REFERENCE	OCTAVE BAND CENTER FREQUENCY (Hz) 63	125	250	500	1000	2000	4000	8000
1	Intake Lw	Turbine manufacturer	109	112	115	118	120	123	123	121
2	Filter	Filter manufacturer		-1	-2	-3	-4	-5	-5	-5
3	Divergence to 400 ft.	Table 2-2	-50	-50	-50	-50	-50	-50	-50	-50
4	Excess air attenuation	Fig. 2-6							-1	-2
5	Silencer DIL, dB	Silencer manufacturer	-6	-8	-12	-20	-27	-27	-22	-17
6	Silenced intake, Lp	Combine lines 1 thru 5	53	53	51	45	39	41	45	47
7	Exhaust Lw	Turbine manufacturer	142	145	145	143	140	140	136	130
8	Vertical directivity	Table 2-3				-1	-3	-5	-9	-14
9	Divergence to 400 ft.	Table 2-2	-50	-50	-50	-50	-50	-50	-50	-50
10	Excess air attenuation	Fig. 2-6							-1	-2
11	Silencer DIL, dB	Silencer manufacturer	-15	-20	-30	-45	-45	-45	-30	-20
12	Silenced exhaust, Lp	Combine lines 7 thru 11	77	75	65	47	42	40	46	44
13	Total combined Lp at 400 ft.	Combine lines 6 & 12; Fig. 2-3	77	75	65	49	44	43	48	49
14	Allowable Lp max.	NEMA G; Table 3-5	83	74	68	63	59	55	52	50

Rotary-Positive Blowers

The most common types of rotary-positive blowers are:

a. Lobe
b. Vane
c. Helical and Axial

Each type normally requires both intake and discharge silencers. The intake noise is not as high as that produced by the discharge but, when open to atmosphere, the intake is the major noise source, since much of the discharge noise is contained by the piping system. In the immediate area of the air intake, unsilenced noise levels of 110 to 115 dBA are not uncommon.

Capacities for **lobe-type blowers** are normally expressed in actual CFM at inlet pressure and temperature conditions. Maximum discharge pressures, varying with the blower size and application, are typically less than 15 psig.

Blower sizes are described in inches, e.g., 12 x 25. The first number represents the timing gear diameter, and the second, the length of the rotor. Both the noise and pulsation produced are functions of the peripheral velocity of the timing gear, which is numerically equal to the product of the gear circumference (feet) and the blower rpm. Noise and pulsation produced by rotary positive blowers inherently reach a critical level at gear pitch-line velocities of about:

a. Intake - 3,300 fpm
b. Discharge - 2,700 fpm

In silencing, reference to these critical pitch-line velocities is noted as being "above" or "below" the blower transition speed (rpm). Selection of silencer type is based on blower transition speed, which is a function of gear diameter and blower rpm as shown in Table 5-7.

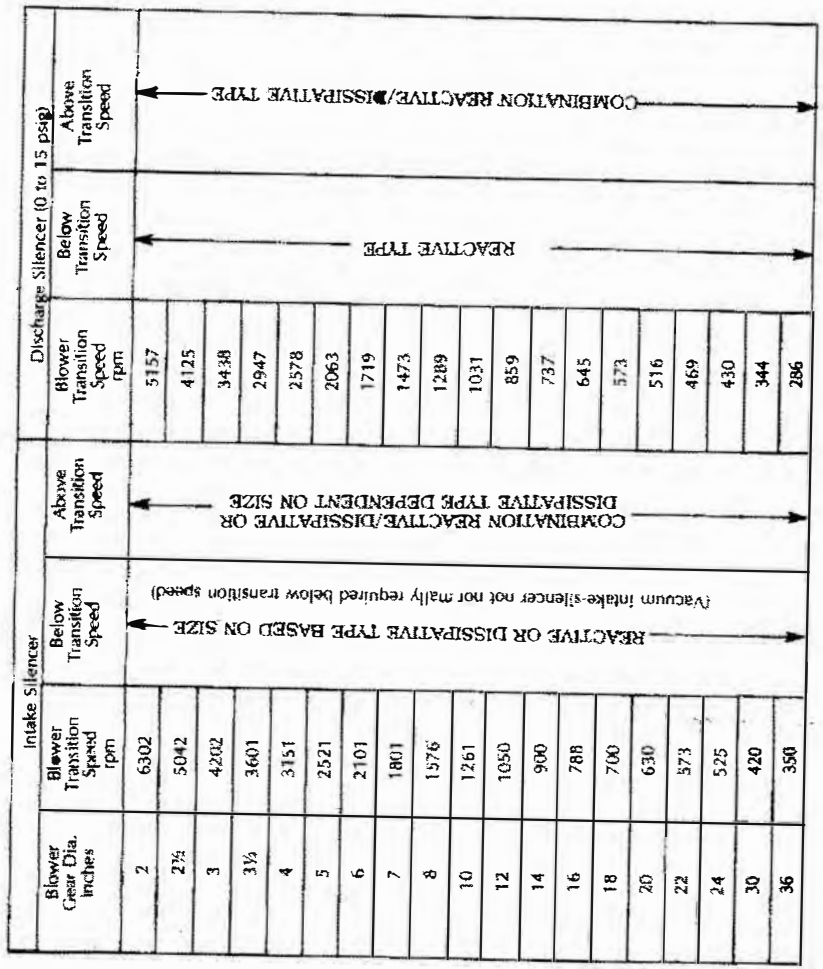


Table 5-7. SELECTION OF SILENCER TYPE BASED ON BLOWER TRANSITION SPEED

$$\text{Blower Transition Speed (rpm)} = \frac{\text{critical PLV}}{0.262 \times \text{gear diameter}}$$

Silencer sizing is generally based on the blower connection or pipe size with velocities to 5500-6000 fpm, where silencer type and pressure drop permit. Acceptable silencer pressure drops range between:

a. Intake - 8 to 10 inches H_2O (atmospheric)
b. Discharge - 1 to 2 percent of the absolute operating pressure

Silencers for small blowers, which normally do not produce high-amplitude, low-frequency noise, are sized on the basis of flow (velocity) and allowable pressure drop. Silencer volume is not a consideration.

Silencers for large blowers that inherently produce high-amplitude, low-frequency noise and pulsation are sized on the basis of: (a) blower displacement, (b) flow and (c) pressure drop. The blower displacement is relatively small compared to the mass flow rate of the blower, yet it can (and often does) affect silencer performance. Where blower displacement is a consideration, the silencer volume-to-blower displacement ratio needed to provide acceptable silencing ranges between 15 and 20:1, depending on the service and performance criteria to be met.

Large, low-speed blowers require reactive-type silencers. High-speed blowers operating "above" transition speed require combination reactive/dissipative type silencers to reduce both the high-amplitude, broad-band noise and low-frequency pulsation:

a. Intake (2 chambers)
b. Discharge (2 or 3 chambers)

Dissipative-type silencers are successfully used on the intake of small blowers. In addition to the silencer, the high-speed blower may require acoustical lagging or effective isolation of the blower casing.

Large, high-speed blowers may require partial or total isolation of both the casing and driver to prevent the radiation of excessive high-frequency noise. Isolation can be achieved with a partial or total housing around the blower and silencers. When totally enclosed, the blower often requires forced ventilation. In addition, acoustical lagging of the piping may be required to meet OSHA levels at 3 feet.

Silencer requirements for **vane-type blowers** are the same as those for the lobe type except at "above transition" speeds where there may be considerable carry-over of lubricating oil in the discharge piping. Oil carry-over often prevents effective use of a dissipative-type silencer. A reactive-type silencer with external acoustic lagging is normally required.

Silencer requirements for the **helical** and **axial-type compressors** are again essentially the same as that for the lobe type. "Above transition" speed requires a combination reactive/dissipative silencer for added reduction of high frequencies to prevent shell and pipe ring.

High-speed, axial-type compressors normally produce both high-amplitude, high-frequency noise and pulsation. So silencers of special design and construction are required. A dissipative or reactive-type silencer may be used on the intake. However, the dissipative-type is the more effective. If the specification precludes use of internal "pack" material in contact with the gas stream, the application requires a reactive-type silencer with external acoustic lagging. Also, the severity of the noise and pulsation produced by the discharge may prohibit the use of a dissipative-type silencer. In such cases, a reactive-type silencer of extra heavy internal construction (with external acoustical lagging) to ASME code requirements may be the solution. Typical operating pressures range from 25 to 300 psig. Silencer velocities may run as high as 6,000 fpm, depending on the service, silencer type and allowable pressure drop.

For silencer and system ΔP calculations based on actual operating conditions, use formulas in the APPENDIX.

Silencer manufacturers should be contacted for specific silencer recommendations for all high-speed, axial-flow compressors, as well as other applications and services where operating pressure exceeds 15 psig.

Centrifugal Compressors

Centrifugal and axial-flow compressors may be single- or multiple-stage, covering a wide range of capacities — from 200 to 150,000 CFM — and pressures to over 5,000 psig. The centrifugal compressor in the process industry is typically driven by an electric motor or steam turbine, while, in gas transmission, the centrifugal is powered by a gas turbine or engine.

The noise produced by centrifugal compressors may exceed 120 dBA, depending on size, rpm and service, and consists primarily of discrete tones interdispersed over a broad-band frequency spectrum of lesser intensity. The maximum amplitude occurs at the blade-passing frequency and its second harmonic:

$$\text{Blade Passing Frequency} = \frac{N \times \text{rpm}}{60} = \text{Hz}$$

Where N = Number of rotating blades.

Large high-speed compressors may require partial or total isolation of the compressor casing, gear unit and/or driver to prevent radiation of high-frequency noise. Isolation can be in the form of acoustical lagging, a partial housing or total housing around the combined unit.

The importance of proper sizing of intake and discharge piping for centrifugal and axial-flow compressors cannot be over-emphasized. Large inlet piping does not necessarily produce less noise. (Actually it may increase the noise at the intake.) But larger discharge piping (reduced velocities) is an effective way to reduce piping noise.

Both intake and discharge silencing is normally required. However, in many atmospheric intake services, only intake silencers are needed. The most effective silencer is the low pressure-drop, dissipative type. All silencers should be close-coupled to the compressor when possible to prevent excessive pipe-radiated noise.

Silencer velocities normally range from 3,000 to 6,000 fpm. Maximum velocity acoustically should not exceed 7,500 fpm. Actual silencer sizing depends on the allowable pressure drop and size of piping. Maximum drop is generally 2 to 4 inches H_2O depending on the size of the compressor and the operating conditions.

The predominant compressor noise source is the atmospheric intake. The compressor itself is normally of heavy cast material which minimizes the radiation of casing noise. The other major noise source is the discharge, which may radiate through the piping, headers and the entire discharge system to create a serious and demanding noise problem in adjoining areas.

Centrifugal compressor noise is produced primarily by blade-tip turbulence which is a function of horsepower, speed (rpm), mass flow, discharge pressure, number of stages of compression, type of gas, piping velocities and inline restrictions — elbows, reducers and the like.

High-speed compressors are inherently noisier than low-speed units. And too, doubling of the horsepower typically results in a noise increase of 4 to 5 dB. The compressor manufacturer can provide the unsilenced Lw or Lp needed for acoustic analysis and silencer selection.

Silencer and piping guidelines for large high-speed blowers and compressors should include, but not necessarily be limited to, the following considerations:

1. Straight runs of piping should be used where possible, avoiding excessive use of elbows, reducers and other restrictive components.

2. For best overall performance, the silencer should be bolted directly to the compressor. Depending on size and service, an expansion joint and possibly an elbow at the compressor connection, may be necessary. (Even this limited amount of piping may require external acoustical lagging to prevent radiated noise.) Additional lengths of piping between the compressor and the silencer are not recommended.

3. The silencer may be a dissipative-type or reactive/dissipative combination, depending on blower or compressor type and service.

4. Where internal acoustic fill is not permitted, the silencer may be reactive (multi-chamber) type with external acoustic lagging. The entire length of the silencer shell and the silencer head on the compressor side should be acoustically lagged and jacketed by the silencer manufacturer.

5. Typically the lagging is 2- to 4-inch-thick fiberglass (3 to 6 pounds per cubic foot in density) or mineral wool (4 to 8 pounds density) with aluminum or steel jacket (overlapped and banded tightly).

6. External acoustic lagging of the silencer may be required to meet the 90-dBA OSHA criteria at 3 feet, regardless of whether the silencer is close-coupled or installed some distance from the compressor. In addition, effective acoustic isolation of the compressor housing and drive motor may be required.

7. Pipe sizing, silencer type and arrangement and method of support are normally reviewed and approved by the blower/compressor manufacturer.

Reciprocating Air Compressors

The noise produced by a reciprocating air compressor is a function of horsepower — the number of cylinders, bore and stroke (whether single-acting or double-acting), speed (rpm), type of gas and compression ratio. The compressor size (hp) and/or compression ratio are the major factors, and noise is generated during both the intake and discharge cycle of operation.

Reciprocating compressors are widely used in plant and process air service, i.e., single-stage, single-cylinder (DA), at 600 to 900 rpm. The predominant component noise sources are:

a. Air intake (atmospheric)
b. Discharge piping
c. Casing and driver (engine, motor, etc.)

The air intake system is typically provided with a filter and silencer or a filter/silencer combination. The filter may be either a media impingement or inertial type as recommended by the compressor manufacturer. The silencer must be a multi-chamber reactive type.

Reciprocating compressors produce both noise and pulsation. Unsilenced, large-bore, low-speed air compressors generate an audible intake noise that is almost always a problem. In addition, it can produce low-frequency noise known as "airborn pulsation," causing walls, windows and doors to vibrate, even at considerable distances from the installation.

When the intake is not provided with the proper type and size silencer, the adverse effects of pulsation may starve the cylinder, reduce the volumetric efficiency of the compressor, reduce the efficiency of the filter and increase both the intake and radiated piping noise. Comparative pulsation levels are shown in Figure 5-12.

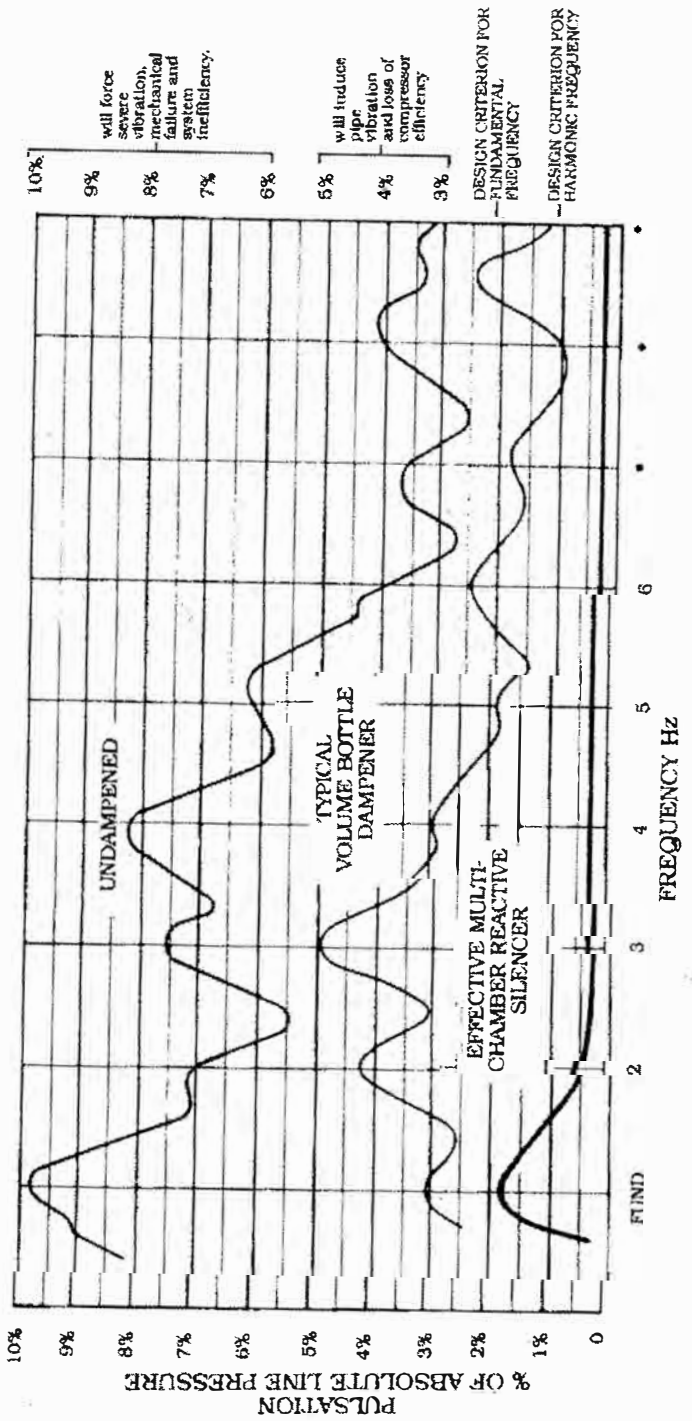


Figure 5-12. TYPICAL RECIPROCATING COMRPESSOR PULSATION LEVELS

Pulsation is as evident in the intake piping of a reciprocating compressor as it is in the discharge. Consider: (a) the intake side of a compressor with the piston receding from the discharge, preparatory to calling for more air, and (b) the reflected pulse in the pipe moving away from the intake valve of the compressor, pulling a pressure void at the cylinder. When these two conditions are simultaneous, the line pressure at the intake valve is below normal for the duration of the intake cycle. The reduced pressure condition (when the air surges into the compressor intake as the valve opens) decreases volumetric efficiency since the cylinder is charged with gas having lower-than-desired density.

When the predominant frequencies produced by a reciprocating compressor coincide with the natural frequencies of the intake system, an adverse noise and pulsation condition can develop. To avoid this:

a. The length of intake piping should be held to a minimum - not to exceed 2/3 of the ¼ wave (λ) length of the fundamental compressor frequency f1. Avoid the ¼ λ and integers thereof.

b. The silencer should be "close-coupled" to the cylinder where possible.

Close-coupling generally implies not more than 3 pipe diameters or a maximum of 18 inches of connecting pipe between the compressor connection and the silencer. This ratio of pipe length to diameter generally keeps the compressor valve-to-silencer distance within ten diameters, resulting in reduced noise levels with no measurable net pumping losses across the system.

In instances where a compressor has more than one first stage cylinder, the cylinder crank angles must be taken into account when sizing a special manifold-type silencer for multi-cylinder applications.

The intake slug volume (SVI) = displacement of the cylinder x volumetric efficiency of the compressor:

$$SV_i = \frac{\pi B^2}{4} \times S \times \frac{VE(\%)}{100} = \text{cu. in.}$$

Where B = Cylinder Bore, in.
S = Cylinder Stroke, in.

Intake slug volume sets the silencer size by volume for single-cylinder compressors. The intake slug volume-to-silencer volume ratio is typically 10 to 13:1:

	Normal (Based on 3.5:1 ℓ/d ratio)	**Minimum** (Based on 5.0:1 ℓ/d ratio)
Typical Silencer I.D. =	$\left(\frac{V_s}{2.618}\right)^{1/3}$	$\left(\frac{V_s}{3.796}\right)^{1/3}$

Where the V_s is the required silencer volume, cu. in.
ℓ = silencer length
d = silencer diameter

The second consideration is that of airflow (velocity and allowable pressure drop). Air is drawn into a single-acting cylinder once every revolution. A double-acting cylinder takes in two charges per revolution.

Actual intake airflow (ACFM) = displacement CFM of the compressor x volumetric efficiency:

$$ACFM = \frac{\pi B^2 \times S \times rpm \times N \times A \times VE(\%)}{4 \times 1728 \times 100}$$

Where N = Number of cylinders
A = Cylinder Action Single-action = 1
Double-action = 2

Average intake silencer velocities are:

a. Single-acting cylinder — 2,000-3,000 fpm

b. Double-acting cylinder — 4,000-5,000 fpm

The silencer sizing, as determined by volume and airflow is then compared. The larger of the two sizes is recommended. The selection of silencer type and design is based on the required noise reduction (DIL).

The total intake system pressure drop — including the filter, silencer and piping — is generally limited to 6 to 10 inches H_2O. In order to meet the air supply requirements of a compressor (100-hp and above), the pressure drop across the silencer should not exceed 3 to 6 inches H_2O.

Most process gas applications, including air discharge silencers (dampeners), require special design and code considerations. For these services, contact the silencer or compressor manufacturer.

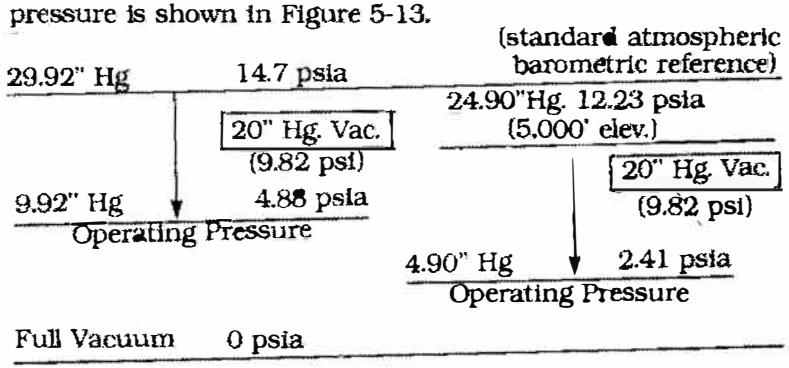
Vacuum Pumps

The most common of the industrial vacuum pumps requiring noise control are:

a. Water-sealed and dry-type rotary
b. Reciprocating piston
c. Steam jet ejector

Most vacuum pumps pull a vacuum on the closed intake system and discharge directly to atmosphere. Capacities are expressed in terms of intake conditions, and should be stated for both atmospheric start-up and normal operating vacuum conditions. Fluid flow is given in gpm.

Vacuum measurements alone are not sufficient to establish the system operating pressure. Simultaneous barometric reference is required. The effect of barometric pressure is shown in Figure 5-13.



14.7 psia = 29.92" Hg. = 406.90" H_2O
1 psi = 2.04" Hg. = 27.68" H_2O
1" Hg. = 0.49 psi = 13.60" H_2O

Figure 5-13 EFFECT OF BAROMETRIC PRESSURE ON 20" Hg VACUUM

Water-sealed rotary pumps generally do not require intake silencing, but most require separators for liquid removal prior to entry into the pump. A separator, while not a silencer, will provide up to 10 dBA reduction, depending upon operating conditions. Both the separator and intake silencer/separator combination (when required) should be designed and constructed for full vacuum (29.92 inches Hg). The vacuum separator provides corrosion protection for the vacuum pump.

The intake (vacuum) separator is typically a high-efficiency, low-pressure drop design/centrifugal type with a low-side inlet, top or high-side outlet and bottom drain. The size of the **intake separator** depends upon the airflow and the quantity of liquid to be removed:

a. Maximum velocity for optimum liquid removal is approximately 6,000 fpm, based on actual CFM at intake (vacuum) condition.
b. Water flow rate (gpm) seldom exceeds 1 gallon per 15 cubic feet of air. Higher liquid rates require a larger separator.

The water outlet (drain) must be sized to handle the required flow and to match the water removal pump. A barometric leg may be used as a liquid seal, and the water removed by gravity if sufficient vertical height is available.

Water-sealed pumps use water as a seal in vacuum service, discharging the liquid, along with the airflow, to atmosphere. A discharge separator/silencer is required. In installations without an intake separator, where both seal-water and inlet water are discharged, the separator/silencer may have to be over-sized. If the over-capacity condition is caused by excessive intake water, an intake separator is needed as excess water through the pump increases the power cost.

The size of the **discharge separator/silencer** is also based on the airflow and the quantity of liquid:

a. Maximum velocity for optimum liquid removal is approximately 6,000 fpm at maximum flow which occurs at start-up.
b. If complete separation is not required and the amount of water to be removed is relatively low, a silencer with an over-sized drain may be sufficient. Maximum velocity is approximately 7,000 fpm at start-up conditions and 5,500 fpm at operating conditions. At the velocities indicated, pressure drop is normally not a consideration.

The drain must be sized to accommodate the large quantity of seal water being discharged with the air and piped to a sump. A water seal in the sump is recommended. Height of the water seal "barometric leg" is determined by the pressure measured in the bottom chamber of the separator/silencer.

The selection of the separator/silencer type depends upon the percent of free liquid that is to be removed and the required silencer DIL.

Centrifugal-type discharge separators/silencers are required for the removal of large amounts of seal water. Efficiency requirements range from 90 to 99 percent of the entrained liquid, depending on the service and application. High-amplitude, broad-band noise from the vacuum pump and discharge to atmosphere requires either a reactive or reactive/dissipative upper chamber, based on the rpm of the vacuum pump and/or the silencing criteria.

Dry-vacuum rotary pumps may require both intake (vacuum) and discharge silencers. The sizing parameters and procedures covered under rotary-positive blowers apply:

a. Determine the silencer type based on the blower intake and discharge transition speed.
b. Silencer size is based on flow (ACFM) and blower displacement.

Reciprocating, piston-type vacuum pumps ordinarily do not require intake silencers except in circumstances where the line piping might starve the pump.

Discharge silencers are sized and selected on the basis of both slug volume and flow, similar to that indicated for the 2-cycle engine. The discharge noise is typically low-frequency, requiring a 2- or 3-chamber reactive silencer, depending upon the silencing criteria.

The displacement CFM and pump volumetric efficiency should be obtained from the pump manufacturer or may be calculated:

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

Discharge flow to atmosphere, under operating conditions, is a function of the operating vacuum and may be expressed:

$$\text{Discharge CFM} = \frac{29.92 - \text{"Hg Vac}}{29.92} \times \text{Displacement cfm} \times \frac{VE}{100}$$

The maximum start-up velocity, based on discharge CFM for atmospheric inlet conditions, should not exceed 7,000 fpm. Maximum velocity for operating conditions, based on discharge CFM, should not exceed 3,000 fpm. At the maximum velocities indicated, silencer pressure drop is not normally a consideration.

The largest of the sizes determined by slug volume, start-up and operating conditions is recommended.

Forced Draft Fans

The intake of large, centrifugal, forced draft (FD) fans is a top priority noise source in many power plants, often creating a worker exposure problem as well as a neighborhood annoyance.

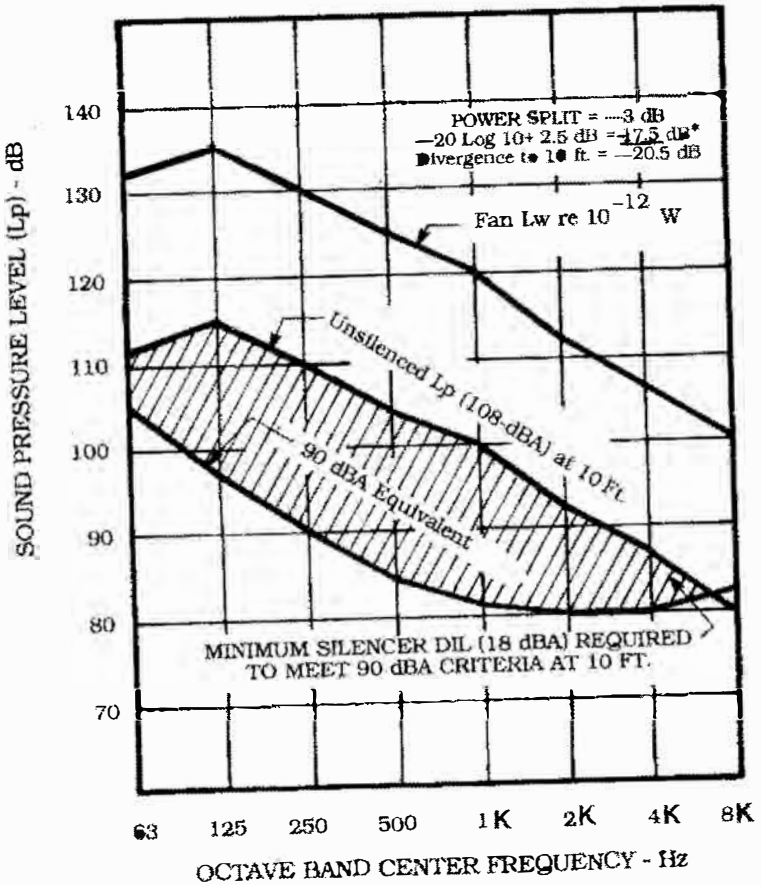
FD fan noise radiates equally into the intake and discharge system. The intake is usually open to atmosphere and is, by far, the most predominant noise source since the discharge noise is largely contained in 3/16-inch or heavier reinforced ducting. Discharge silencing is usually not required.

All FD fans are noisy. The larger double-inlet FD fan often produces noise in excess of 120 dBA. While the smaller single-inlet fan is less noisy, all FD fans require effective silencing of the intake noise. Unsilenced inlet noise from the single-inlet fan shown in Figure 5-14 is 108 dBA. As indicated in the example, broad-band silencing is needed to reach 90 dBA at 10 feet.

Air turbulence and broad-band noise are generated by the rotating fan blades. Discrete tones are caused by the interaction of the blades and the housing. The maximum amplitude occurs at the blade-pass frequency and its second harmonic:

$$\text{Blade pass frequency} = \frac{N \times \text{rpm}}{60} = \text{Hz}$$

N - Number of rotating blades



*Based on free field divergence

Figure 5-14. SINGLE-INLET FD FAN INTAKE NOISE ANALYSIS

Fan noise increases with:

a. Static pressure
b. Reduced fan efficiency
c. Fan size (CFM)
d. Higher tip speed

Fan noise may be effectively reduced by providing:

a. Intake and discharge (where applicable) silencers
b. Fan vibration isolators
c. Duct liners
d. Acoustic lagging or absorbent coatings

Where possible, a fan should be located on a roof or other remote spot, away from work areas. Use of high velocity systems should be avoided to prevent velocity-induced noise and vibration.

Generally, FD fan intake silencers are the dissipative type, whereas induced draft (ID) fan discharge silencers are a special design with the acoustical fill set back out of the gas flow to prevent solids in the gas stream from clogging the perforated facing.

Silencers are normally selected on the basis of maximum allowable pressure drop at rated flow, and the silencing criteria. Maximum silencer velocity acoustically is about 7,500 fpm.

A silencer may be either cylindrical or rectangular. The latter is often more economical in the larger sizes. An outlet plenum is generally needed (depending on the silencer design) so as not to reduce fan capacity. Properly-designed silencers may be installed directly at the fan intake and discharge, provided that the airflow is evenly distributed and that the allowable pressure drop is not exceeded.

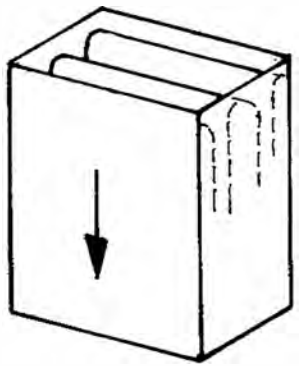
Silencers for large volume fans which operate at relatively low static pressures are sized for pressure drops of 0.20 to 0.50 inch H_2O Drops of up to 3 inches H_2O may be acceptable for smaller fans. Rainhoods for vertical downflow silencers add ~0.25 inch H_2O to the system pressure drop. For silencers and system ΔP calculations based on actual operating conditions use formula in the APPENDIX.

The silencers must be of all-welded construction and adequately stiffened for rugged, heavy-duty fan service.

In addition to the silencers, a large fan located in a critical area may require an acoustic enclosure around the fan motor, fan and the ducting in the immediate area of the fan. The enclosure should consist of absorbent panels to prevent reverberent noise build-up, and the arrangement and dimensions should be such that the normal airflow to the fan is not restricted. The TL of the enclosure (plenum) must be compatible with the silencer DIL. Figure 5-15 shows the most common silencer and plenum arrangements for both single- and double-inlet FD fans.

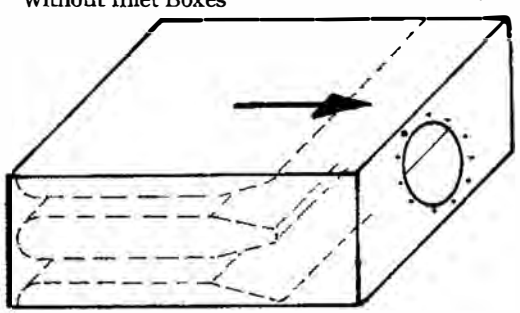
1. Fans/Inlet Boxes

Rectangular Silencer (Vertical Downflow or horizontal)



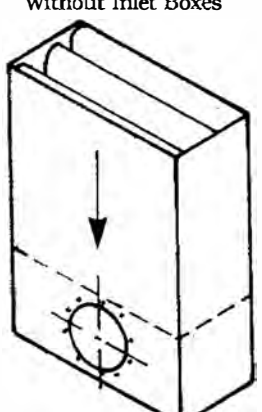
2. Single Inlet Fans, Without Inlet Boxes

Cylindrical or Rectangular Silencer/Outlet Plenum to match fan connection. (Horizontal)



3. Single Inlet Fans Without Inlet Boxes

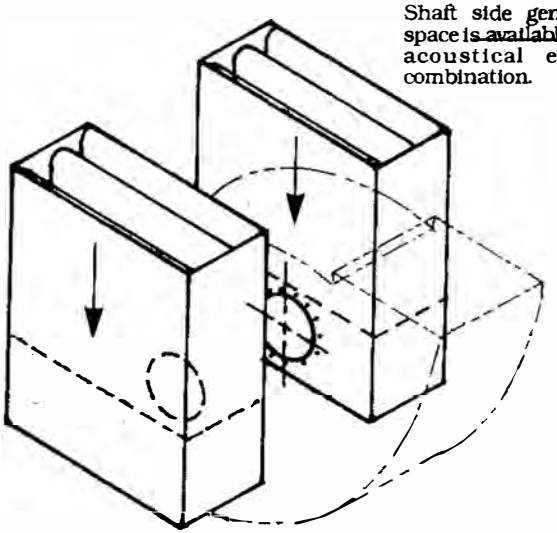
Rectangular Silencer/Outlet Plenum to match fan connection (vertical downflow or horizontal)



4. Double Inlet Fans Without Inlet Boxes

Non-shaft side, typically same as 2 and 3.

Shaft side generally requires (where space is available) a special silencer and acoustical enclosure (plenum) combination.



5. Large multiple double-inlet fans in Series

Common inlet plenum/ parallel baffle type inlet silencer

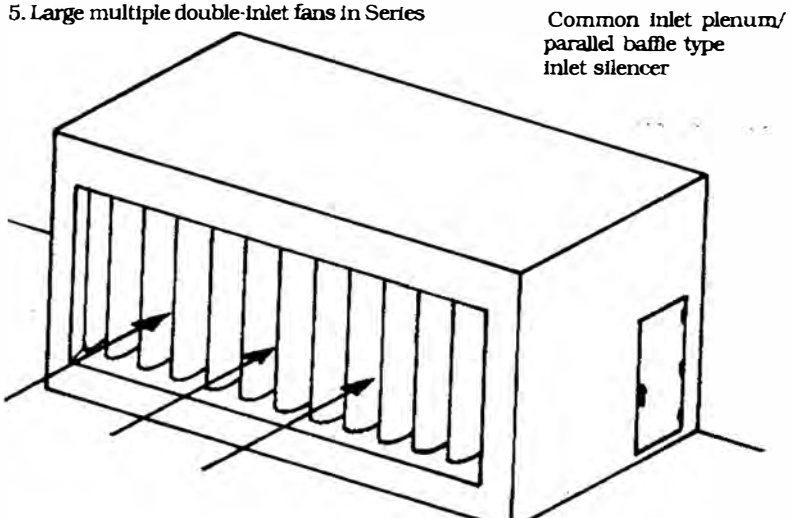


Figure 5-15. TYPICAL FD FAN SILENCER ARRANGEMENTS

Pressure Regulators

The major source of regulator noise is flow turbulence, created by rapid expansion of gas as in high-pressure vent and blowdown service. The extent of this aerodynamic noise and turbulence is primarily a function of the pressure expansion across the regulator. In critical flow applications, both the gas flow and resultant noise increases until the critical pressure ratio is reached. Beyond this ratio there is no further increase in flow, yet the noise level continues to increase with increased ratios. A series of pressure reductions, each below the critical ratio, will produce less noise. However, the cost would probably be prohibitive due to the increased complexity of such a system:

a. At critical "sonic" flows, the noise is most predominant in the downstream piping.
b. At sub-critical flows which produce less noise, the noise is propagated about equally upstream and downstream.

Effective control of valve and regulator noise may involve one or both of the following treatments:

a. Source Treatment — The reduction of noise at its source may be in the form of multiple orifices incorporated into the regulator design or placed near the regulator outlet.
b. Path Treatment — (See Figure 5-16.) Reduction of noise between the source and receiver may consist of: (1) in-line silencers, (2) heavy-weight piping and/or (3) acoustic pipe lagging.

This discussion is limited to the path treatment only, leaving source treatment to the manufacturer.

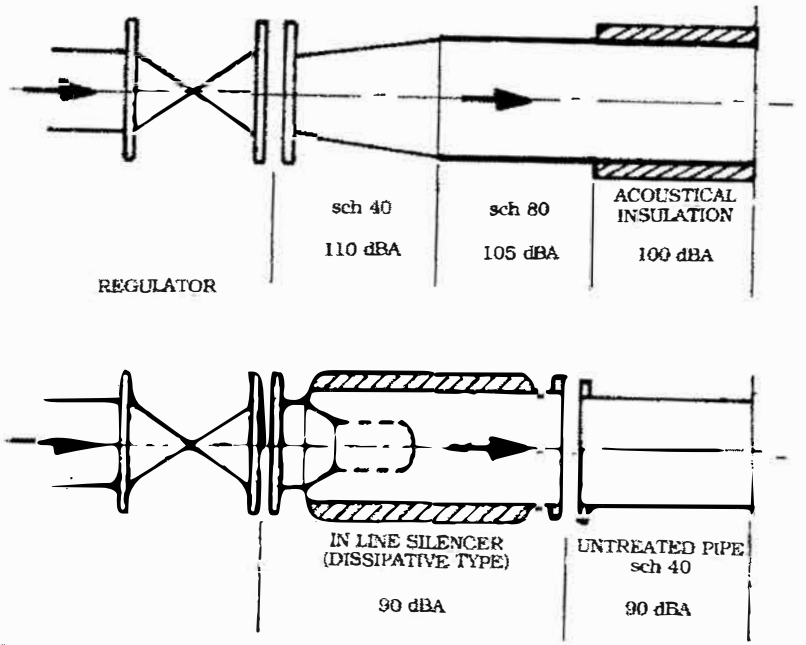


Figure 5-16. VARIOUS PATH TREATMENTS FOR REGULATOR NOISE

In critical flow applications, the unsilenced noise in the discharge piping may exceed 120 dBA. A level of 110 dBA is not uncommon, as indicated in Figure 5.17.

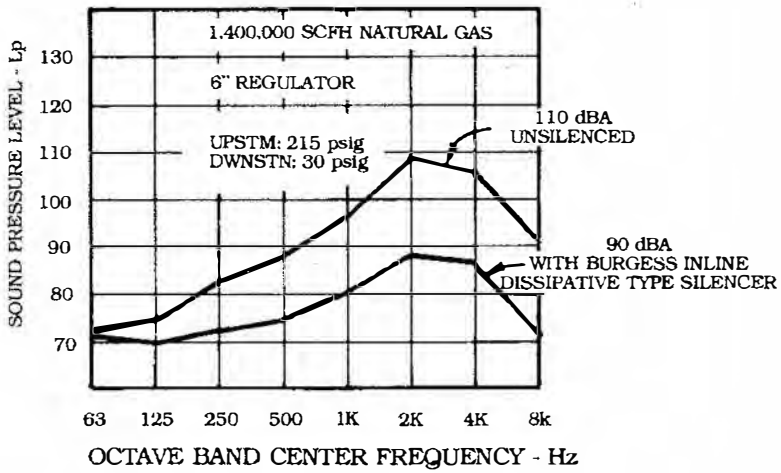


Figure 5-17. TYPICAL HIGH-FREQUENCY REGULATOR NOISE

When the regulator is located inside a room or building, with workers in the immediate vicinity, downstream and possibly upstream silencers are normally required. The regulator itself may require acoustical lagging or total isolation in the form of a removable enclosure.

For a regulator installed outdoors, a downstream silencer plus acoustic lagging of the upstream piping is normally adequate. In addition, in critical locations, the regulator area may require partial isolation in the form of an open-top sound barrier.

Acoustic lagging of the piping, in lieu of installing silencers, is seldom totally effective. However, in short runs of above-ground piping where the radiating surface area is limited, or where the radiated noise level is borderline, acoustic lagging alone is sometimes acceptable.

A dissipative-type silencer is normally required. In critical flow applications, where increased silencer pressure drop is not a limiting factor, a multi-port, multi-stage inlet diffuser is recommended for improved silencer performance.

Silencer and piping velocities normally range from 3,000 to 6,000 fpm. The upper limit acoustically is 9,000 to 10,000 fpm to prevent regeneration of noise in the piping, and to ensure against loss of internal pack material into the piping system.

Generally, the gas flows are stated at standard conditions (SCFD at 14.7 psia and 70°F), in which case, the actual flow and velocities must be calculated:

$$\text{Actual flow} = \frac{\text{SCFD} \times \text{Ta} \times Z}{51{,}918 \times \text{Pa}} = \text{ACFM}$$

WHERE
SCFD = Std. cu. ft./Day
ACFM = Actual cu. ft./Min.
Ta = (460 + °F)
Pa = PSIA
Z = Gas Compressibility Factor

Gears, Motors and Drives — Mechanical Noise

Gear noise is generally related to gear design, manufacturing tolerances and operation. Accuracy of tooth design and shaft alignment must be maintained to prevent metal-to-metal impact noise. However, the major source of noise is not gear inaccuracy, but friction which is a function of the gear speed, specific tooth loading and tooth-passing frequency:

$$\text{Tooth-passing Frequency} = \frac{N \times \text{rpm}}{60} = \text{Hz}$$

N = Number of Gear Teeth

Gear noise is a composite of the tooth-passing frequency and a number of the higher harmonics, metal-to-metal impact and structural resonances of the system.

A simple acoustic enclosure — one that is easily and quickly removed for inspection and maintenance of the gear unit — may be required to reduce noise to acceptable levels.

Couplings can generate both noise and vibration due to speed and misalignment. A removable shroud (sleeve) will usually eliminate the noise problem, and provide worker protection from the rotating shaft. In hazardous areas, the acoustic shroud should be of non-metallic, non-sparking material.

Electric **motors** may also be noisy and require silencing. The cooling fan noise is typically the predominant source. In a large 3,600-rpm motor, the fan can produce noise in the frequency range approximating that of a siren. Other sources of motor noise are the interaction of the rotor and stator, dynamic unbalance and housing resonances.

Dissipative silencing may be built into the motor housing by the manufacturer, or silencers may be supplied separately and ducted to intake and discharge openings in the housing. A removable ⊓-shaped acoustical enclosure completely housing the motor is sometimes the more practical solution. Acoustic louvers and/or dissipative silencers should be provided to insure adequate air for cooling.

Chapter 6
CONCLUSION

This handbook has discussed only a few of the major sources of industrial noise. A multiplicity of other noise sources require similar controls, but plants differ so widely in complexity and operation that each one must be considered individually for the particular noise sources present.

Noise control should be an important consideration in the planning and layout of any new facility. Retrofit cost alone (not including downtime or production loss) can be many times more costly than the initial installation of a noise control system.

In summary, effective noise control (as presently mandated by federal, state and local agencies) creates numerous benefits. It improves worker communications, increases production, improves quality of work output, reduces the chance of accidents and injury, reduces the potential for occupational hearing loss among workers and, in general, improves the plant relationship with its neighbors in the community.

APPENDIX I
BASIC FORMULAS

1. BASIC FLOW FORMULAS FOR AIR AND GASES —

$$Vs = \frac{Va\ Pa\ Ts}{Ps\ Ta\ Z} \qquad Va = \frac{Vs\ Ps\ Ta\ Z}{Pa\ Ts}$$

$$SCFM = 36.054\ ACFM\ \frac{Pa}{Ta\ Z} \qquad ACFM = \frac{SCFM\ Ta\ Z}{36.054\ Pa}$$

$$SCFH = 2163\ \ ACFM\ \frac{Pa}{Ta\ Z} \qquad ACFM = \frac{SCFH\ Ta\ Z}{2.163\ Pa}$$

$$SCFD = 51.918\ ACFM\ \frac{Pa}{Ta\ Z} \qquad ACFM = \frac{SCFD\ Ta\ Z}{51.918\ Pa}$$

Where Pa = Operating Pressure, psia
Ps = Standard Pressure, 14.7 psia
Ta = Operating Temperature, °Absolute (460 + °F)
Ts = Standard Temperature, °Absolute (460 + 70)
Z = Gas Compressibility Factor (if not known use 1.
Vs = SCFM at 14.7 psia and 70°F.
Va = ACFM

2. BASIC SILENCER FLOW FORMULA —

$$ACFM = 3590 \times A \sqrt{\frac{\Delta P \times Ta \times Z}{C \times Pa \times MWa}}$$

Where A = Silencer open area (Sq. ft.)
$\Delta P = H_2O$

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

3. BASIC SILENCER ΔP FORMULA —

$$\text{Silencer } \Delta P = C \times \left(\frac{V}{4005}\right)^2 \times \frac{MWa}{MWs} \times \frac{Pa}{Ps} \times \frac{Ts}{Ta} \times \frac{1}{Z}$$

Where ΔP = Pressure Drop, "H_2O
C = Silencer Coefficient
V = Exit Velocity, fpm
MWa = Molecular weight of the gas
MWs = Molecular weight of standard air = 28.9

$$\Delta P = \frac{"H_2O}{27.68} = psi$$

In most atmospheric air applications the formula may be reduced to -

$$\Delta P = C \times \left(\frac{V}{4005}\right)^2 \times \left(\frac{530}{460 + °F}\right) = "H_2O$$

ΔP intake system = Δ P Filter (or entrance loss) + Δ P silencer + ΔP piping

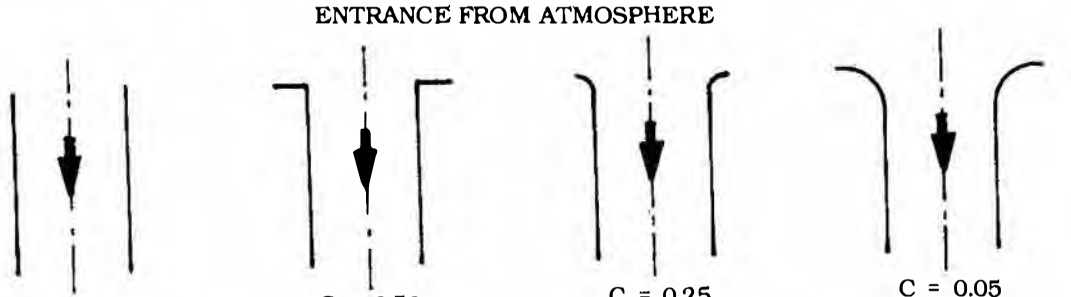
ΔP exhaust system = Δ P Silencer + Δ P piping + exit loss

Silencer Coefficient C varies with Silencer type and design -

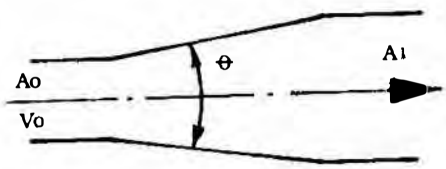
	C Values (Range)
Dissipative (inline)	0.25 - 0.75
Reactive type Low Loss	0.25 - 1.50
Standard	3.5 - 6.0

Contact the Silencer manufacturer for actual Silencer C values.

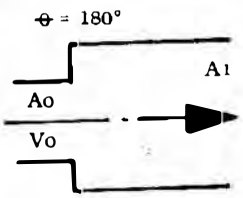
ENTRANCE FROM ATMOSPHERE



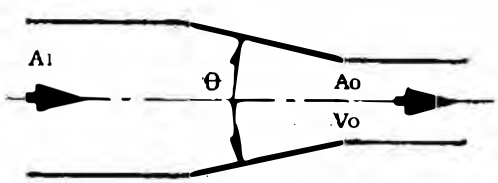
TRANSITION
(DIVERGENT FLOW)



Ao/A1	Co θ DEGREES						
	10	20	30	40	60	90	180
0	.15	.36	.65	.92	1.2	1.1	1.0
0.10	.12	.29	.52	.75	.93	.89	.83
0.20	.10	.23	.41	.59	.74	.70	.65
0.30	.07	.18	.31	.40	.57	.54	.50
0.40	.06	.13	.23	.33	.41	.39	.37
0.50	.04	.06	.10	.15	.18	.17	.16
0.60	.03	.06	.10	.15	.18	.17	.16



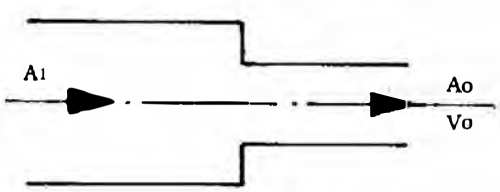
TRANSITION
(CONVERGENT FLOW)



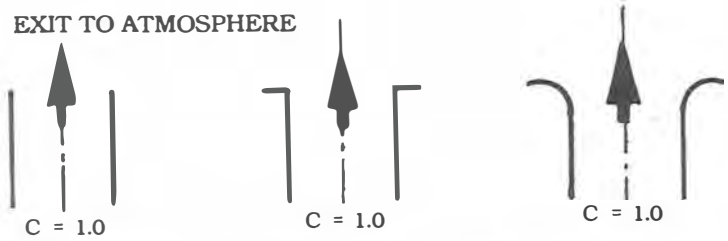
Ao/A1	Co θ DEGREES					
	0	12	30	40	50	60
0.1	0	.01	.02	.03	.05	.07
0.2	0	.01	.02	.03	.05	.07
0.3	0	.01	.02	.03	.05	.06
0.4	0	.01	.02	.03	.04	.06
0.5	0	.01	.01	.02	.04	.05
0.6	0	.01	.01	.02	.03	.04
0.7	0	0	.01	.02	.03	.04
0.8	0	0	.01	.01	.02	.03
0.9	0	0	0	.01	.01	.01

ABRUPT CONTRACTION

Ao/A1	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
Co	.45	.40	.35	.30	.25	.20	.15	.10	.05

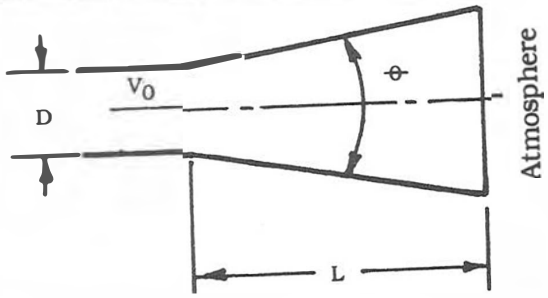


EXIT TO ATMOSPHERE



DIVERGENT CONE (TO ATMOSPHERE)

L/D	Co θ-DEGREES					
	6	8	10	15	20	30
1.0	1.0	.90	.80	.61	.55	.58
1.5	.86	.73	.64	.50	.47	.60
2.0	.73	.60	.52	.41	.42	.62
3.0	.54	.42	.36	.31	.36	.65
4.0	.42	.34	.29	.26	.33	.69
6.0	.25	.22	.20	.25	.33	.67



SMOOTH PIPING

$$C_O = 0.0148 \left(\frac{L}{D}\right)$$

ELBOWS - C_0

	LONG RADIUS	SHORT RADIUS
SMOOTH	0.15	0.22
3 piece	0.34	0.42
5 piece	0.24	0.33

PIPING ELBOWS AND PIPE LENGTHS SHOULD BE MINIMIZED TO PREVENT EXCESSIVE PRESSURE DROP.

APPENDIX III
TYPICAL SILENCER FLOW AREA

Silencer Nominal Size (inches)	Area (Sq. Ft.)
1	0.006
1½	0.014
2	0.023
2½	0.033
3	0.051
3½	0.069
4	0.103
5	0.156
6	0.225
8	0.381
10	0.599
12	0.850
14	1.028
16	1.350
18	1.715
20	2.123
22	2.575
24	3.071
26	3.581
28	4.162
30	4.787
32	5.455
36	6.922
42	9.393
48	12.306

APPENDIX IV
CONDENSED PROPERTIES - VARIOUS GASES

GAS	SYMBOL	MOLECULAR WEIGHT	RATIO OF SPECIFIC HEATS K = Cp/Cv
Air		28.97	1.40
Water	H_2O	18.02	1.33
Ammonia	NH_3	17.03	1.31
Argon	A	39.94	1.67
Carbon Monoxide	CO	28.01	1.40
Carbon Dioxide	CO_2	44.01	1.30
Chlorine	Cl_2	70.91	1.37
Helium	H_e	4.00	1.66
Hydrogen	H_2	2.02	1.41
Nitrogen	N_2	28.02	1.40
Oxygen	O_2	32.00	1.40
Methane	CH_4	16.04	1.31
Ethane	C_2H_6	30.07	1.19
Propane	C_3H_8	44.09	1.14
Ethylene	C_2H_4	28.05	1.25
Hydrogen Sulfide	H_2S	34.08	1.33
Sulfur Dioxide	SO_2	64.06	1.24

APPENDIX V
PROPERTIES OF DRY AIR

(BAROMETER PRESSURE 29.92 "H_g)

Temp. °F.	Specific Volume cu. ft. per lb.	Temp. Factor Ratio to Volume at 70° F.	Relative Density (std. air equals unity)	Temp. °F.	Specific Volume cu. ft. per lb.	Temp. Factor Ratio to Volume at 70° F.	Relative Density (std. air equals unity)
0	11.58	.868	1.1522	400	21.86	1.623	.6163
10	11.84	.887	1.1277	410	21.91	1.642	.6092
20	12.09	.906	1.1042	420	22.15	1.660	.6023
30	12.34	.925	1.0816	430	22.40	1.679	.5965
40	12.59	.943	1.0600	440	22.66	1.698	.5889
50	12.84	.962	1.0392	450	22.91	1.717	.5824
60	13.09	.981	1.0192	460	23.17	1.736	.5761
70	13.34	1.000	1.0000	470	23.42	1.755	.5699
80	13.60	1.019	.9815	480	23.67	1.774	.5638
90	13.85	1.038	.9636	490	23.93	1.793	.5579
100	14.10	1.057	.9464	500	24.17	1.811	.5521
110	14.34	1.075	.9296	510	24.42	1.830	.5464
120	14.60	1.094	.9136	520	24.67	1.849	.5408
130	14.85	1.113	.8983	530	24.93	1.868	.5354
140	15.11	1.132	.8833	540	25.18	1.887	.5300
150	15.36	1.151	.8689	550	25.43	1.906	.5248
160	15.61	1.170	.8548	560	25.69	1.925	.5196
170	15.87	1.189	.8413	570	25.93	1.943	.5145
180	16.10	1.207	.8281	580	26.18	1.962	.5096
190	16.36	1.226	.8154	590	26.42	1.980	.5048
200	16.61	1.245	.8030	600	26.69	2.000	.5000
210	16.87	1.264	.7910	610	26.94	2.019	.4953
220	17.12	1.283	.7794	620	27.20	2.038	.4907
230	17.37	1.302	.7681	630	27.45	2.057	.4862
240	17.63	1.321	.7571	640	27.69	2.075	.4818
250	17.88	1.340	.7465	650	27.94	2.094	.4775
260	18.12	1.358	.7361	660	28.20	2.113	.4732
270	18.37	1.377	.7260	670	28.45	2.132	.4690
280	18.63	1.396	.7162	680	28.70	2.151	.4649
290	18.88	1.415	.7067	690	28.96	2.170	.4609
300	19.14	1.434	.6974	700	29.21	2.189	.4569
310	19.39	1.453	.6883	710	29.46	2.208	.4530
320	19.64	1.472	.6795	720	29.70	2.226	.4492
330	19.90	1.491	.6709	730	29.96	2.245	.4454
340	20.14	1.509	.6625	740	30.21	2.264	.4417
350	20.39	1.528	.6543	750	30.46	2.283	.4380
360	20.64	1.547	.6463	760	30.72	2.302	.4344
370	20.90	1.566	.6386	770	30.97	2.321	.4309
380	21.15	1.585	.6306	780	31.22	2.340	.4274
390	21.40	1.604	.6235	790	31.47	2.358	.4240

APPENDIX V (Cont'd)
PROPERTIES OF DRY AIR

Temp. °F	Specific Volume cu.ft.	Temp. Factor Ratio to per lb. °F	Relative Density (std. air Volume unity)
800	31.72	2.377	.4206
810	31.97	2.396	.4173
820	32.23	2.415	.4141
830	32.48	2.434	.4109
840	32.73	2.453	.4077
850	32.98	2.472	.4046
860	33.24	2.491	.4015
870	33.48	2.509	.3985
880	33.73	2.528	.3955
890	33.99	2.547	.3926
900	34.24	2.566	.3897
910	34.49	2.585	.3869
920	34.75	2.604	.3841
930	35.00	2.623	.3813
940	35.26	2.642	.3786
950	35.50	2.660	.3759
960	35.75	2.679	.3732
970	36.00	2.698	.3706
980	36.26	2.717	.3681
990	36.51	2.736	.3655
1000	36.76	2.755	.3630
1010	37.02	2.774	.3605
1020	37.25	2.792	.3581
1030	37.51	2.811	.3557
1040	37.76	2.830	.3533
1050	38.02	2.849	.3510
1060	38.27	2.868	.3487
1070	38.54	2.888	.3464
1080	38.78	2.906	.3442
1090	39.03	2.925	.3419
1100	39.27	2.943	.3397
1110	39.53	2.962	.3376
1120	39.78	2.981	.3354
1130	40.03	3.000	.3333
1140	40.28	3.019	.3313
1150	40.54	3.038	.3292
1160	40.79	3.057	.3272
1170	41.03	3.075	.3252
1180	41.29	3.094	.3232
1190	41.54	3.113	.3212
1200	41.79	3.132	.3193
1210	42.05	3.151	.3174
1220	42.30	3.170	.3155
1230	42.55	3.189	.3136
1240	42.81	3.208	.3118
1250	43.05	3.226	.3099
1260	43.30	3.245	.3081
1270	43.56	3.264	.3064
1280	43.81	3.283	.3045
1290	44.06	3.302	.3029
1300	44.32	3.321	.3011
1310	44.57	3.340	.2994
1320	44.81	3.358	.2978
1330	45.06	3.377	.2961
1340	45.32	3.396	.2944
1350	45.57	3.415	.2938
1360	45.82	3.434	.2912
1370	46.06	3.453	.2896
1380	46.33	3.472	.2880
1390	46.58	3.491	.2865
1400	46.82	3.509	.2849
1410	47.08	3.528	.2834
1420	47.33	3.547	.2819
1430	47.59	3.566	.2804
1440	47.84	3.585	.2789
1450	48.09	3.604	.2775
1460	48.35	3.623	.2760
1470	48.60	3.642	.2746
1480	48.84	3.660	.2732
1490	49.09	3.679	.2718
1500	49.35	3.698	.2704
1510	49.60	3.717	.2690
1520	49.85	3.736	.2677
1530	50.11	3.755	.2663
1540	50.36	3.774	.2650
1550	50.60	3.792	.2637
1560	50.85	3.811	.2624
1570	51.11	3.830	.2611
1580	51.36	3.849	.2598
1590	51.62	3.868	.2585

Altitude in Feet Above Sea Level	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000	15000	20000
Barometric Pressure in Inches of Mercury	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.38	20.58	16.89	13.75

APPENDIX VI
PROPERTIES OF DRY SATURATED STEAM

TEMP. F t	ABS. PRESS. psia P	SPECIFIC VOLUME SAT. LIQUID V_f	SPECIFIC VOLUME EVAP. V_{fg}	SPECIFIC VOLUME SAT. VAPOR V_g	ENTHALPY SAT. LIQUID h_f	ENTHALPY EVAP. h_{fg}	ENTHALPY SAT. VAPOR h_g	ENTROPY SAT. LIQUID s_f	ENTROPY EVAP. s_{fg}	ENTROPY SAT. VAPOR s_g	TEMP. F t
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	110
120	1.6924	0.01620	203.27	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	120
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	130
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	140
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	150
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	160
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	170
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109	180
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	190
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	200
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	210
212	14.696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	212
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440	220
230	20.780	0.01684	19.365	19.382	198.23	958.6	1157.0	0.3387	1.3901	1.7288	230
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	240
250	29.825	0.01700	13.804	13.821	216.48	945.5	1164.0	0.3675	1.3323	1.6998	250
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	260
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	270
280	49.203	0.01726	8.626	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	280
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	290

APPENDIX VI (Cont'd.)
PROPERTIES OF DRY SATURATED STEAM

TEMP. F t	ABS. PRESS. psia P	SPECIFIC VOLUME			ENTHALPY			ENTROPY			TEMP. F t
		SAT. LIQUID Vf	EVAP. Vfg	SAT. VAPOR Vg	SAT. LIQUID hf	EVAP. hfg	SAT. VAPOR hg	SAT. LIQUID Sf	EVAP. Sfg	SAT. VAPOR Sg	
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350	300
310	77.68	0.01755	5.609	5.626	279.92	902.6	1182.5	0.4504	1.1727	1.6231	310
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	320
330	103.06	0.01776	4.289	4.307	300.68	887.0	1187.7	0.4769	1.1233	1.6002	330
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	340
350	134.63	0.01799	3.324	3.342	321.63	870.7	1192.3	0.5029	1.0754	1.5783	350
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	360
370	173.37	0.01823	2.606	2.625	342.79	853.5	1196.3	0.5286	1.0287	1.5573	370
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	380
390	220.37	0.01850	2.0651	2.0836	364.17	835.4	1199.6	0.5539	0.9832	1.5371	390
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	400
410	276.75	0.01878	1.6512	1.6700	385.83	816.3	1202.1	0.5788	0.9386	1.5174	410
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	420
430	343.72	0.01910	1.3308	1.3499	407.79	796.0	1203.8	0.6035	0.8947	1.4982	430
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	440
450	422.6	0.0194	1.0799	1.0993	430.1	774.5	1204.6	0.6280	0.8513	1.4793	450
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	460
470	514.7	0.0198	0.8811	0.9009	452.8	751.5	1204.3	0.6523	0.8083	1.4606	470
480		0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	480
490	621.4	0.0202	0.7221	0.7423	476.0	726.8	1202.8	0.6766	0.7653	1.4419	490
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	500
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	520
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1196.2	0.7374	0.6568	1.3942	540
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	560
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	580
600	1542.9	0.0236	0.2432	0.2666	617.0	548.5	1165.3	0.8131	0.5176	1.3307	600
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	620
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	640
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	660
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	680
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389	700
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580	705.4

APPENDIX VII
PROPERTIES OF SUPERHEATED STEAM

ABS. PRESS. psia (Sat. Temp.)		TEMPERATURE — DEGREES FAHRENHEIT												
		200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400
1 (101.74)	v	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	929.1	988.7	1107.8	1227.0
	h	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1585.2	1637.7	1745.7	1857.5
	s	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4625	2.4952	2.5566	2.6137
5 (162.24)	v	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	185.79	197.71	221.6	245.4
	h	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1585.1	1637.7	1745.7	1857.4
	s	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.2851	2.3178	2.3792	2.4363
10 (193.21)	v	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	92.88	98.84	110.77	122.69
	h	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2	1585.0	1637.6	1745.6	1857.3
	s	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2086	2.2413	2.3028	2.3598
14.969 (212.00)	v	.....	30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	63.19	67.25	75.37	83.48
	h	.....	1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1584.8	1637.5	1745.5	1857.3
	s	.....	1.8180	1.8743	1.9261	1.9734	2.0170	2.0958	2.1319	2.1662	2.1989	2.2603	2.3174	
20 (227.96)	v	.....	22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	46.42	49.41	55.37	61.34
	h	.....	1191.6	1239.2	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1584.7	1637.4	1745.4	1857.2
	s	.....	1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1321	2.1648	2.2263	2.2834
40 (267.25)	v	.....	11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	23.20	24.69	27.68	30.66
	h	.....	1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1584.3	1637.0	1745.1	1857.0
	s	.....	1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0555	2.0883	2.1498	2.2069
60 (292.71)	v	.....	7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	15.453	16.451	18.446	20.44
	h	.....	1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1583.8	1636.6	1744.8	1856.7
	s	.....	1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0106	2.0434	2.1049	2.1621
80 (312.03)	v	.....	.....	6.220	7.020	7.797	8.562	9.322	10.077	10.830	11.582	12.332	13.830	15.325
	h	.....	.....	1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1583.4	1636.2	1744.5	1856.5
	s	.....	.....	1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	1.9787	2.0115	2.0731	2.1303
100 (327.81)	v	.....	.....	4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.259	9.860	11.060	12.258
	h	.....	.....	1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1582.9	1635.7	1744.2	1856.2
	s	.....	.....	1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9538	1.9867	2.0484	2.1056
120 (341.25)	v	.....	.....	4.081	4.636	5.165	5.683	6.195	6.702	7.207	7.710	8.212	9.214	10.213
	h	.....	.....	1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1582.4	1635.3	1743.9	1856.0
	s	.....	.....	1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9335	1.9664	2.0281	2.0854
140 (353.02)	v	.....	.....	3.468	3.954	4.413	4.861	5.301	5.738	6.172	6.604	7.035	7.895	8.752
	h	.....	.....	1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1581.9	1634.9	1743.5	1855.7
	s	.....	.....	1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9163	1.9493	2.0110	2.0683
160 (363.53)	v	.....	.....	3.008	3.443	3.849	4.244	4.631	5.015	5.396	5.775	6.152	6.906	7.656
	h	.....	.....	1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1581.4	1634.5	1743.2	1855.5
	s	.....	.....	1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9014	1.9344	1.9962	2.0535

v — Specific Volume, cu. ft. per lb. h — Enthalpy, BTU per lb. s — Entropy, BTU per ° F per lb.

APPENDIX VII (Cont'd.)
PROPERTIES OF SUPERHEATED STEAM

ABS. PRESS. psia (Sat. Temp.)		TEMPERATURE — DEGREES FAHRENHEIT												
		200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400
180 (373.06)	v			2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.129	5.466	6.136	6.804
	h			1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1581.0	1634.1	1742.9	1855.2
	s			1.5745	1.6373	1.6894	1.7356	1.7776	1.8167	1.8534	1.8882	1.9212	1.9831	2.0404
200 (381.79)	v			2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.613	4.917	5.521	6.123
	h			1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1580.5	1633.7	1742.6	1855.0
	s			1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.8763	1.9094	1.9713	2.0287
220 (389.86)	v			2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.191	4.467	5.017	5.565
	h			1206.5	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	1580.0	1633.3	1742.3	1854.7
	s			1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8656	1.8987	1.9607	2.0181
240 (397.37)	v			1.9276	2.247	2.533	2.804	3.068	3.327	3.584	3.839	4.093	4.597	5.100
	h			1202.5	1264.5	1318.2	1371.5	1423.2	1474.8	1526.9	1579.6	1632.9	1742.0	1854.5
	s			1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8558	1.8888	1.9510	2.0084
260 (404.42)	v				2.063	2.330	2.582	2.827	3.067	3.305	3.541	3.776	4.242	4.707
	h				1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1579.1	1632.5	1741.7	1854.2
	s				1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8467	1.8799	1.9420	1.9995
280 (411.05)	v				1.9047	2.156	2.392	2.621	2.845	3.066	3.286	3.504	3.938	4.370
	h				1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1578.6	1632.1	1741.4	1854.0
	s				1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8383	1.8716	1.9337	1.9912
300 (417.33)	v				1.7675	2.005	2.227	2.442	2.652	2.859	3.065	3.269	3.674	4.078
	h				1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1578.1	1631.7	1741.0	1853.7
	s				1.5701	1.6288	1.6751	1.7184	1.7582	1.7954	1.8305	1.8638	1.9260	1.9835
350 (431.72)	v				1.4923	1.7036	1.8960	2.084	2.266	2.445	2.622	2.798	3.147	3.493
	h				1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	1577.0	1630.7	1740.3	1853.1
	s				1.5481	1.6070	1.6563	1.7009	1.7408	1.7777	1.8130	1.8463	1.9086	1.9663
400 (444.59)	v				1.2851	1.4779	1.6508	1.8161	1.9767	2.134	2.290	2.445	2.751	3.055
	h				1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1575.8	1629.6	1739.5	1852.5
	s				1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.7977	1.8311	1.8936	1.9513

APPENDIX VII (Cont'd.)
PROPERTIES OF SUPERHEATED STEAM

ABS. PRESS. psia (Sat. Temp.)		TEMPERATURE — DEGREES FAHRENHEIT												
		500	550	600	620	640	660	680	700	800	900	1000	1200	1400	1600
450 (456.28)	v	1.1231	1.2155	1.3005	1.3332	1.3652	1.3967	1.4278	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
	h	1238.4	1272.0	1302.8	1314.6	1326.2	1337.5	1348.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
	s	1.5095	1.5437	1.5735	1.5845	1.5951	1.6054	1.6153	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
500 (467.01)	v	0.9927	1.0800	1.1591	1.1893	1.2188	1.2478	1.2763	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
	h	1231.3	1266.8	1298.6	1310.7	1322.6	1334.2	1345.7	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
	s	1.4919	1.5280	1.5588	1.5701	1.5810	1.5915	1.6016	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
550 (478.94)	v	0.8852	0.9686	1.0431	1.0714	1.0989	1.1259	1.1523	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
	h	1223.7	1261.2	1294.3	1306.8	1318.9	1330.8	1342.5	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
	s	1.4751	1.5131	1.5451	1.5568	1.5680	1.5787	1.5890	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
600 (486.21)	v	0.7847	0.8753	0.9463	0.9729	0.9988	1.0241	1.0489	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
	h	1215.7	1255.5	1289.9	1302.7	1315.2	1327.4	1339.3	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
	s	1.4586	1.4990	1.5323	1.5443	1.5558	1.5667	1.5773	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
700 (503.10)	v		0.7277	0.7934	0.8177	0.8411	0.8639	0.8860	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
	h		1243.2	1280.6	1294.3	1307.5	1320.3	1332.8	1345.0	1403.2	1459.0	1513.9	1623.5	1734.8	1848.5
	s		1.4722	1.5084	1.5212	1.5333	1.5449	1.5559	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
800 (518.23)	v		0.6154	0.6779	0.7006	0.7223	0.7433	0.7635	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
	h		1229.8	1270.7	1285.4	1299.4	1312.9	1325.9	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.6
	s		1.4467	1.4863	1.5000	1.5129	1.5250	1.5366	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
900 (531.98)	v		0.5264	0.5873	0.6089	0.6294	0.6491	0.6680	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
	h		1215.0	1260.1	1275.9	1290.9	1305.1	1318.8	1332.1	1393.9	1451.8	1508.1	1619.3	1731.6	1846.3
	s		1.4216	1.4653	1.4800	1.4938	1.5066	1.5187	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
1000 (544.61)	v		0.4535	0.5140	0.5350	0.5546	0.5733	0.5912	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
	h		1198.3	1248.8	1265.9	1281.9	1297.0	1311.4	1325.3	1385.2	1443.2	1505.1	1617.3	1730.0	1845.0
	s		1.3961	1.4450	1.4610	1.4757	1.4893	1.5021	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
1100 (556.31)	v		0.4532	0.4738	0.4929	0.5110	0.5281	0.5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031	
	h		1236.7	1255.3	1272.4	1288.5	1303.7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8	
	s		1.4251	1.4425	1.4583	1.4728	1.4862	1.4989	1.5535	1.5996	1.6405	1.7130	1.7775	1.8363	
1200 (567.22)	v		0.4016	0.4222	0.4410	0.4586	0.4752	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101	
	h		1223.5	1243.9	1262.4	1279.6	1295.7	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5	
	s		1.4052	1.4243	1.4413	1.4568	1.4710	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263	
1400 (587.10)	v		0.3174	0.3390	0.3580	0.3753	0.3912	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8648	
	h		1193.0	1218.4	1240.4	1260.3	1278.5	1295.5	1369.1	1433.1	1493.2	1608.9	1723.7	1840.0	
	s		1.3639	1.3877	1.4079	1.4258	1.4419	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083	

v — Specific Volume, cu. ft. per lb. h — Enthalpy, BTU per lb. s — Entropy, BTU per °F per lb.

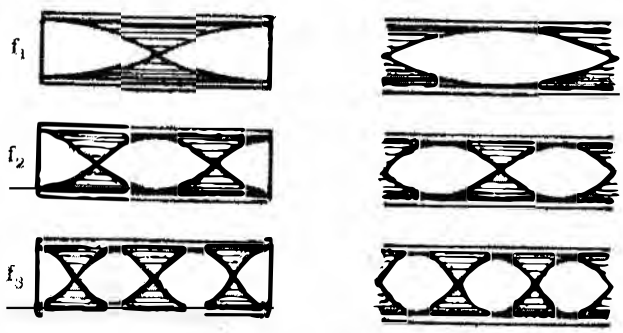
APPENDIX VII (Cont'd.)
PROPERTIES OF SUPERHEATED STEAM

ABS. PRESS. psia (Sat. Temp.)		TEMPERATURE — DEGREES FAHRENHEIT													
		500	550	600	620	640	660	680	700	800	900	1000	1200	1400	1600
1600 (604.90)	v				0.2733	0.2936	0.3112	0.3271	0.3417	0.4034	0.4553	0.5027	0.5906	0.6738	0.7545
	h				1187.8	1215.2	1236.7	1259.6	1278.7	1358.4	1425.3	1487.0	1604.6	1720.5	1837.5
	s				1.3489	1.3741	1.3952	1.4137	1.4303	1.4964	1.5476	1.5914	1.6669	1.7328	1.7926
1800 (621.03)	v					0.2407	0.2597	0.2760	0.2907	0.3502	0.3986	0.4421	0.5218	0.5968	0.6693
	h					1185.1	1214.0	1238.5	1260.3	1347.2	1417.4	1480.8	1600.4	1717.3	1835.0
	s					1.3377	1.3638	1.3855	1.4044	1.4765	1.5301	1.5752	1.6520	1.7185	1.7786
2000 (635.82)	v					0.1936	0.2161	0.2337	0.2488	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
	h					1145.8	1184.9	1214.8	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
	s					1.2945	1.3300	1.3564	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
2500 (668.13)	v							0.1484	0.1686	0.2294	0.2710	0.3081	0.3678	0.4244	0.4784
	h							1132.3	1176.8	1303.6	1387.8	1458.4	1585.3	1706.1	1826.2
	s							1.2687	1.3073	1.4127	1.4772	1.5273	1.6086	1.6775	1.7389
3000 (695.36)	h								1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
	s								1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
3296.2 (705.40)	v									0.1583	0.1981	0.2288	0.2806	0.3267	0.3703
	h									1250.5	1355.2	1434.7	1589.8	1694.6	1817.2
	s									1.3568	1.4309	1.4874	1.5742	1.6452	1.7060
3500	v								0.0306	0.1364	0.1762	0.2058	0.2546	0.2977	0.3381
	h								758.5	1224.9	1340.7	1424.5	1583.3	1689.8	1813.6
	s								0.9515	1.3241	1.4127	1.4723	1.5615	1.6336	1.6968
4000	v								0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
	h								753.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s								0.9347	1.2757	1.3827	1.4482	1.5417	1.6184	1.6795
4500	v								0.0276	0.0798	0.1226	0.1500	0.1917	0.2273	0.2602
	h								753.5	1113.9	1286.5	1388.4	1540.8	1673.5	1800.9
	s								0.9235	1.2204	1.3529	1.4253	1.5235	1.5990	1.6640
5000	v								0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
	h								748.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s								0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499
5500	v								0.0262	0.0465	0.0880	0.1143	0.1516	0.1825	0.2106
	h								741.3	985.6	1224.1	1349.3	1518.2	1657.0	1788.1
	s								0.9090	1.1093	1.2930	1.3821	1.4908	1.5699	1.6369

APPENDIX VIII
ACOUSTIC RESONANCE IN PIPING SYSTEMS

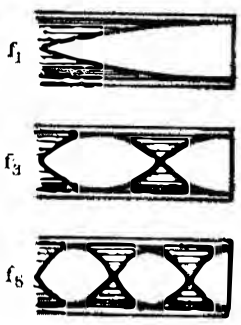
½λ RESONANCE

¼λ RESONANCE



PIPE CLOSED AT BOTH ENDS

PIPE OPEN AT BOTH ENDS



PIPE CLOSED AT ONE END AND OPEN AT OTHER END

$$f = \frac{nC}{2L}$$

f = RESONANT FREQUENCY, hz.
n = 1,2,3 (FUNDAMENTAL, 2nd and 3rd HARMONICS)
$C \approx 223\left(\frac{kT}{MW}\right)^{1/2}$ = SPEED OF SOUND, fps
L = EFFECTIVE LENGTH, FT.

$$f = \frac{(2n-1)C}{4L}$$

n = 1,3,5 (FUNDAMENTAL, 3rd and 5th HARMONICS)

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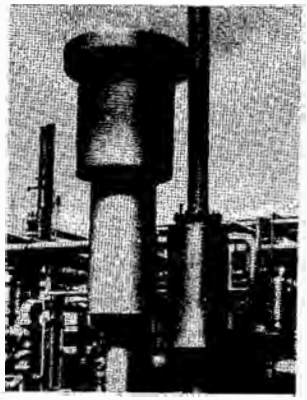
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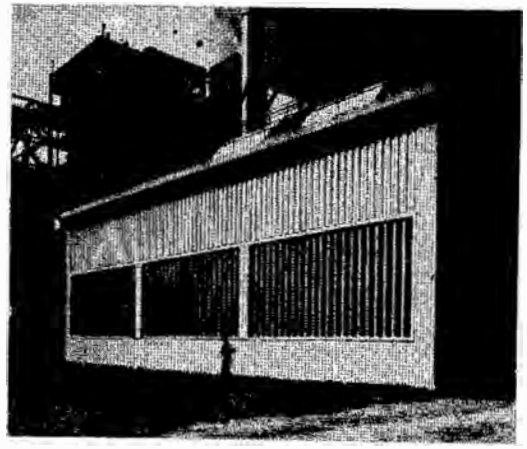
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BURGESS-MANNING INDUSTRIAL SILENCERS







Burgess-Manning has supplied a wide variety of silencing products for industrial applications since the early 1920's. Today, with this broad experience along with continued research and product development efforts, we offer you one of the broadest lines of standard silencers to meet almost all industrial silencing requirements. Our experienced engineering staff is continuing to expand and improve our products and is available to work with you on standard or custom designed equipment to meet your specific requirements. Standard or custom silencers outlined here can be manufactured to AWS or ASME code requirements in carbon or stainless steel and aluminum in our own facilities to strict quality standards.

PRODUCT APPLICATION GUIDE
Air filters, filter/silencers, silencers

PRODUCTS

Atmospheric Service	Air Filters			Air Filter-Silencers					Silencers									
	BFV	BD	BPF	SDF	BF	BFVS	BDS	BPFS	BMI	BMAI	BMC	BMB	BMA	BEO	CA	LCA	ACB	ACA
Engines																		
Intake																		
Air Filter	X	X	X															
Filter-silencer				X	X	X	X	X										
Silencer									X	X						X	X	X
Exhaust																		
Silencer											X	X	X	X				
Spark Arrestor Silencer																		
Gas Turbines																		
Intake																		
Air Filter	X	X	X															
Filter-silencer				X	X	X	X	X										
Silencer																X	X	X
Exhaust																		
Silencer																		
Rotary Positive Blowers																		
Intake																		
Air Filter	X	X	X															
Filter-silencer				X	X	X	X	X										
Silencer									X	X					X			
Discharge Silencer											X	X	X	X				
Rotary Vacuum Blowers																		
Intake Separator-Silencer																		
Discharge Separator-Silencer																		
Vents																		
Silencer													X	X	X	X	X	X
Compressors																		
Centrifugal																		
Atmospheric Intake																		
Air Filter	X	X	X															
Filter-silencer				X	X	X	X	X										
Silencer																	X	X
Discharge Silencer																	X	X
Reciprocating																		
Atmospheric Intake																		
Air Filter	X	X	X															
Filter-silencer				X	X	X	X	X										
Silencer									X	X								

PRODUCTS

Atmospheric Service	Silencers						Vent Silencers					Water Separators & Separator Silencers					Spark Arrestor Snubbers	
	BMSI	ADSC	BMSS	BEDA	BDM + BMB	BDM + BMA	LCV	ACV	DA	BMSV	Stack Inserts	WVK	WSV	WSS	WSDA	BMBS	DPSC	DPSS
Engines																		
Intake																		
Air Filter																		
Filter-silencer																		
Silencer		X																
Exhaust																		
Silencer				X														
Spark Arrestor Silencer																	X	X
Gas Turbines																		
Intake																		
Air Filter																		
Filter-silencer																		
Silencer																		
Exhaust																		
Silencer																		
Rotary Positive Blowers																		
Intake																		
Air Filter																		
Filter-silencer																		
Silencer	X																	
Discharge Silencer			X	X	X	X												
Rotary Vacuum Blowers																		
Intake Separator-Silencer												X	X					
Discharge Separator-Silencer														X	X	X		
Vents																		
Silencer			X	X			X	X	X	X	X							
Compressors																		
Centrifugal																		
Atmospheric Intake																		
Air Filter																		
Filter-silencer																		
Silencer																		
Discharge Silencer																		
Reciprocating																		
Atmospheric Intake																		
Air Filter																		
Filter-silencer																		
Silencer																		

VENT AND BLOWDOWN SILENCERS

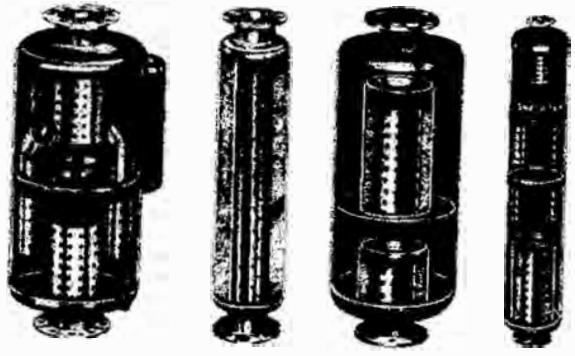
Burgess-Manning vent and blowdown silencers have been developed through years of extensive laboratory and field testing to attenuate the high noise levels associated with high pressure gas and steam vents and blowdowns to atmosphere. The noise levels generated are a function of the pressure, valve and pipe size, temperature and type of gas being discharged. Heavy duty construction features have been designed into these silencers for extended life in the severe conditions existing on high pressure vent applications. Inlet diameters range from 2 to 60 inches. For application, sizing, and dimensional information on standard silencer models ask for Bulletin 10-1-2.





INTERNAL COMBUSTION ENGINE SILENCERS

Burgess-Manning engine silencers are designed for use on engine intake and exhaust systems for 2/4-cycle naturally aspirated, turbocharged reciprocating engines. Standard silencers are available to meet hospital, residential, commercial, industrial, or your specific acoustic requirements. Spark arrester silencers are available for marine or hazardous applications. Silencer inlet diameters range from 1" to 66"; sizes 12" and below are stocked in many models. For application, sizing and dimensional information on standard engine silencer models, including two spark-arrester silencer models, ask for Bulletin 10-2-2.





RECIPROCATING COMPRESSOR SILENCERS

Burgess-Manning air intake silencers and pulsation dampener are designed to reduce the air intake noise and pressure pulses on reciprocating compressors to acceptable levels. Typically requiring reactive-type silencing for the low frequencies involved, the Burgess-Manning multi-chamber silencers provide the required broad band attenuation needed to meet the specified levels. For pressure discharge applications, silencers/pulsation dampeners are available to meet your application requirements. Standard silencer inlet diameters range from 1" to 48"; sizes 12" and below are stocked. For application, sizing, and dimensional information, ask for Bulletin 10-3-2.





CENTRIFUGAL COMPRESSOR SILENCERS

Burgess-Manning silencers are designed for atmospheric intake and discharge and pressure discharge applications. Being characteristically high-frequency sound, Burgess-Manning centrifugal compressor intake and discharge silencers are designed to attenuate this noise to meet your particular requirements. Typically absorptive designed silencers are utilized since they are highly effective in the higher frequency bands. Designed with a minimum of airflow restriction, the pressure drop across the silencers is minimized. Standard silencer inlet diameters range from 1" to 66"; for application, sizing, and dimensional information, ask for Bulletin 10-4-2.





ROTARY POSITIVE BLOWER SILENCERS

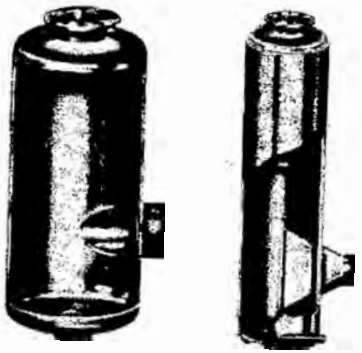
Burgess-Manning silencers are designed for atmospheric air intake and pressure discharge applications. Developed to reduce the pulsations and noise associated with rotary lobe, screw and helical screw type blowers. Through many years of field and lab experience Burgess-Manning has developed reactive and reactive-absorptive silencers specifically designed to reduce the shell radiated noise and the inlet and discharge pulse generated noise levels associated with blower systems. Special externally acoustically lagged units are available to meet the most stringent requirements. Standard silencer inlet diameters range from 1" to 48"; sizes 12" and below are stocked. For application, sizing and dimensional information, ask for Bulletin 10-5-2.





ROTARY VACUUM PUMP SEPARATOR-SILENCERS

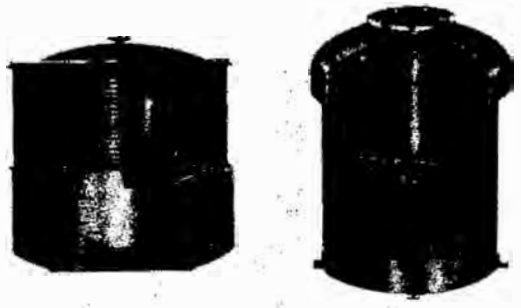
Rotary vacuum pump separators and separator silencers are designed for full vacuum service on liquid seal vacuum pump applications. Burgess-Manning separators are designed for use on the inlet side where typically silencing is not required, and separator silencers are available for the discharge application. Standard separators are available to remove up to 99% of free liquid. Standard silencer/separator inlet diameters range from 1" to 30"; sizes 10" and below are stocked. For application, sizing and dimensional information, ask for Bulletin 10-6-2.





AIR INTAKE FILTER/FILTER SILENCERS

Burgess-Manning air intake filters and filter/silencers are designed for use on internal combustion engines, rotary positive blowers, reciprocating and centrifugal compressors. Filters and filter/silencers are available in both single-stage or dual-stage series. Filter media can be disposable, non-abrasive, non-corrosive porushair or reuseable metalik steel-mesh. Second stage filters use felt media. Standard filter and filter/silencer inlet diameters range from 4" to 42"; all replacement elements maintained in stock. For application, sizing, and dimensional information, ask for Bulletin 40-1-2.





FAN SILENCERS

Burgess-Manning fan silencers are designed to reduce the intake and discharge noise caused by axial and centrifugal mechanical draft fans. The standard and custom designed silencers are selected or designed to reduce noise to sound levels specified by the customer. Typical applications include forced and induced draft fans for power plants and steel mills. For additional information, contact Burgess-Manning.



GAS TURBINE SILENCER SYSTEMS

Burgess-Manning designs and manufactures intake and exhaust silencers, intake filters, and evaporative cooling systems for use on gas turbines. Since each turbine installation is typically unique, Burgess-Manning custom-designs systems and sub-systems to meet specific project and/or environmental requirements. Major components include single or multi-stage filters; demisters and evaporative coolers, intake air silencers, secondary air intake silencers, acoustical enclosures and modular exhaust silencers. Customer applications have ranged from skid-mounted 2,000 HP turbine drivers to 85,000 kw generator systems. For additional information, contact Burgess-Manning.



Notes

Notes



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