CHAPTER 37

CONTROL OF NOISE EXPOSURE

Vaughn H. Hill

DEFINITION OF PROBLEM

Measure Noise Level

From an engineering control standpoint, the first step in a hearing conservation program is to measure the noise levels in all working areas. Areas in which the noise level does not exceed 90 dBA* need not be considered further since noise reduction is not required.

Determine Exposure Time

In areas where the noise level does exceed 90 dBA, a study should be made to determine the actual worker exposure time. Then using this exposure time and the measured noise level, one can determine whether or not the government standards are exceeded. Details regarding the use of government criteria are given in Chapter 25. It may be desirable to use a dosimeter to determine the actual daily noise exposure for comparison with government criteria. Dosimeters are discussed in Chapter 25.

Evaluate Extent of Hazard

If the combination of noise level and exposure time indicate that government criteria are exceeded, an evaluation should be made to determine the most economical solution to the problem. Considerations for making such an evaluation are: (1) reduction of noise level, (2) reduction of exposure time, (3) segregation of worker from noise, (4) substitution of more quiet machine or process or (5) provision of worker with personal protection such as ear muffs or plugs. Under the Occupational Safety and Health Act this latter option is available only if others fail. Usually all of the following will be involved in making the best evaluation: (1) management, (2) medical, (3) personnel, (4) manufacturing, (5) engineering and (6) maintenance. This chapter will be limited to the problem of reducing noise in working environments.

CONSIDERATION OF ALL POSSIBLE MEANS OF NOISE CONTROL

In a subject as broad as industrial noise control, it is impractical to discuss all possible solutions to all problems. Therefore, typical problems of the type occurring most commonly in industry will be discussed with the hope that the reader will acquire an understanding of the principles of noise control that will guide him in solving a much wider variety of problems.

The mode of attacking a noise problem is somewhat analogous to that of controlling any environmental hazard. Appropriate control measures include such things as change in plant layout and design, substitution of less hazardous method, reduction of the hazard at its source, and reduction of the hazard after it has left its point of origin.

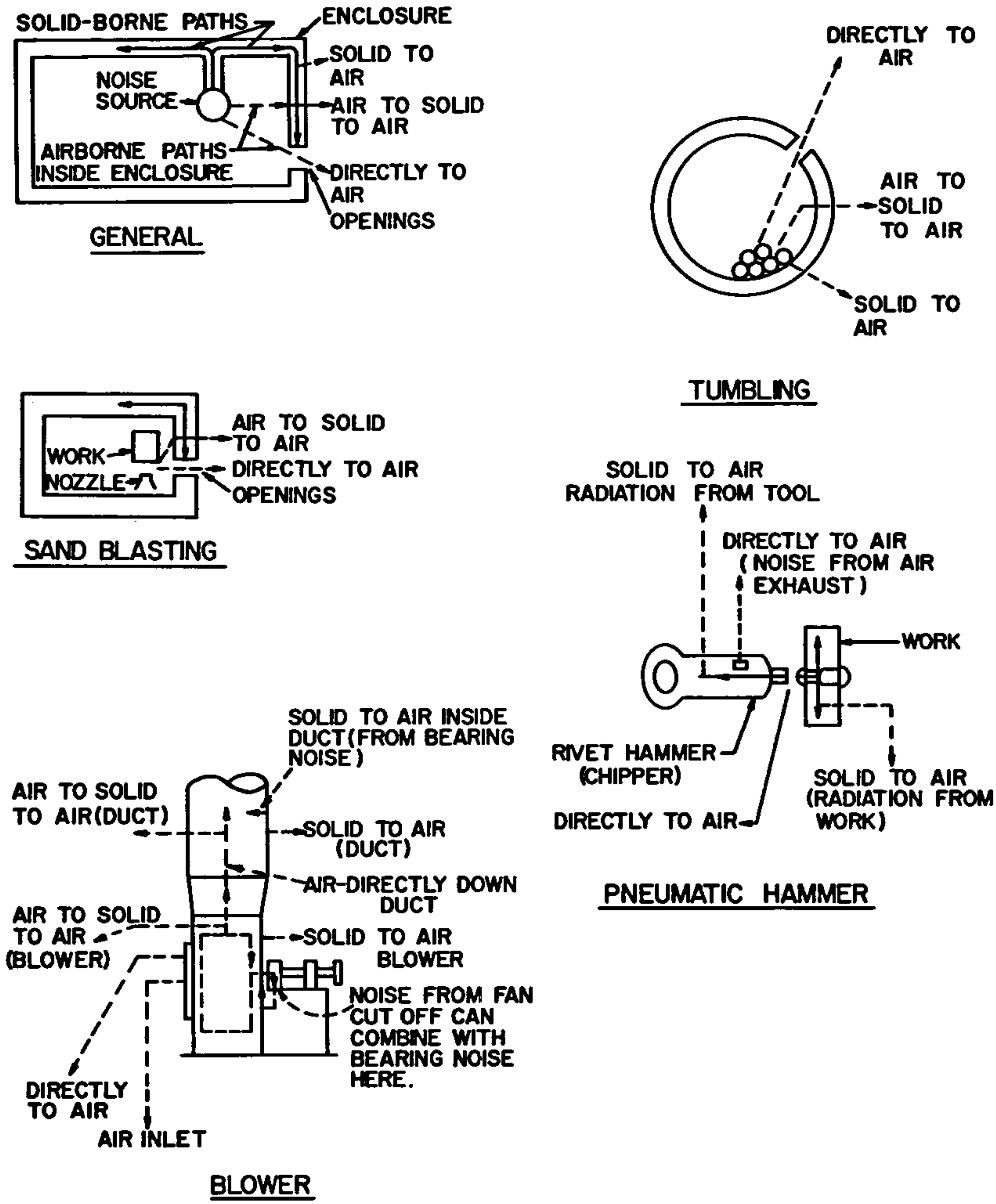
In analyzing a noise problem one must consider that sound from a source can travel by more than one path to the point at which it becomes objectionable. Therefore, noise flow diagrams such as shown by Figure 37-1 are a definite aid to accurate analysis of a given problem. For instance, this shows that sound sources inside enclosures can have (a) direct radiation of sound through openings in the enclosure, (b) sound radiation from the enclosure due to solid borne vibration from the source and (c) indirect radiation from the enclosure, that is, airborne from the source to the inside of the enclosure and subsequent reradiation from the outside of the enclosure. The problem is to determine which paths carry the most sound energy and then select appropriate methods of obtaining the desired reductions along those paths.

It has proved helpful to follow a planned method of analysis so that no possible control measure is overlooked. The following outline can be used in making such an analysis:

Noise Control Analysis Outline

- I. Plant Planning
- II. Substitution
  - A. Use quieter equipment
  - B. Use quieter process
  - C. Use quieter material
- III. Modification of the Noise Source
  - A. Reduce driving force on vibrating surface
    1. Maintain dynamic balance
    2. Minimize rotational speed
    3. Increase duration of work cycle
    4. Decouple the driving force
  - B. Reduce response of vibrating surface
    1. Add damping
    2. Improve bracing
    3. Increase stiffness
    4. Increase mass
    5. Shift resonant frequencies
  - C. Reduce area of vibrating surface
    1. Reduce overall dimensions
    2. Perforate surface
  - D. Use directionality of source
  - E. Reduce velocity of fluid flow
  - F. Reduce turbulence

*Note: This figure may be changed by regulation or law. Check for the current standard requirement.



Tyzzer, F. G.: Reducing industrial noise. Amer. Ind. Hyg. Assoc. J. 14:264-95, 1953.

Figure 37-1. Noise Flow Diagrams.

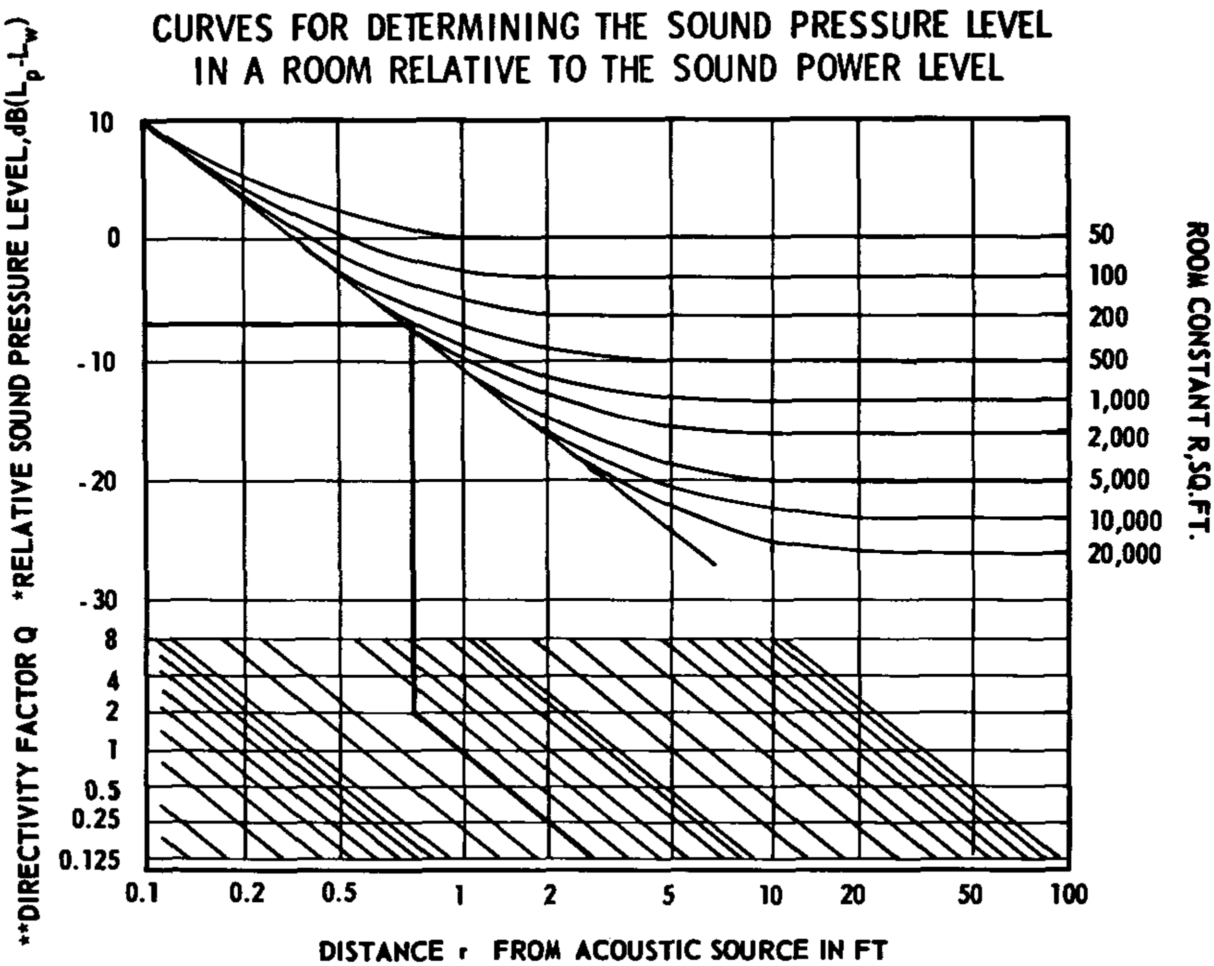


Figure 37-?. Curves for Determining the Sound Pressure Level in a Room Relative to the Sound Power Level.

IV. Modification of the Sound Wave
   A. Confine the sound wave
   B. Absorb the sound wave
      1. Absorb sound within the room
      2. Absorb sound along its transmission path.

Examples of many of these possible control measures will be illustrated in this chapter.

Plant Planning

Noise specifications. An immediate step essential to those concerned with noise control is to stop buying new noise problems. Minimum essential designs for processes and equipment must involve light, high speed machines, high pressures, high flow velocities, light building structures and minimum floor space. All of these can lead to noise problems if limits are not specified. Noise specifications are a *must* for new equipment. To properly use noise specifications one must understand Figure 37-2. This graph shows the relationship between sound power level (L_w) and sound pressure level (L_p) and their relationship to distance from the source (r), directivity factor (Q) and room constant (R).

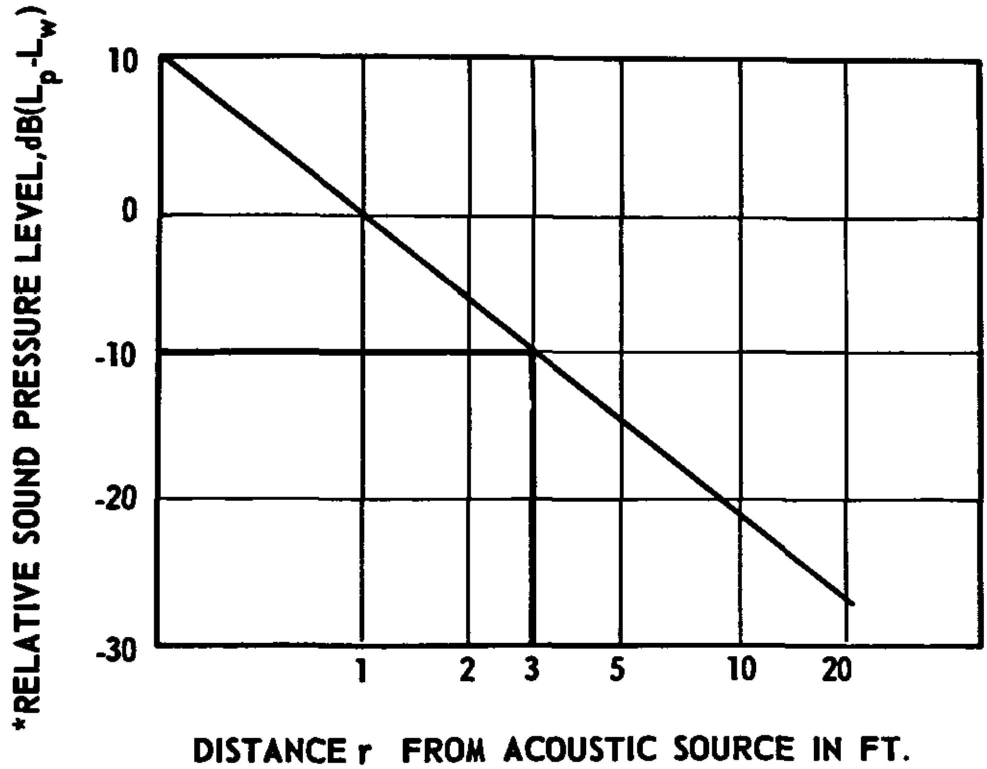
L_w is total energy of the sound source and is independent of the distance or environment. It is calculated, not measured. See Chapter 23 ("Physics of Sound") for further discussion.

L_p is sound energy flow per unit area at some distance (r) from the source. It varies with distance from the source, directivity and room constant. Therefore, these environmental conditions must be specified when expressing noise in terms of sound pressure level.

Free field radiation. When a sound source is in a free field (where there are no reflections) it will diminish with the square of the distance from the source. The relation between L_w and L_p in this case is shown by Figure 37-3. L_p can be measured with a sound level meter or analyzer.

Room constant (R). R is a measure of the ability of a room to absorb sound. It can be calculated as follows:

FREE FIELD NOISE RADIATION POINT SOURCE – NONDIRECTIONAL



$$L_w = L_p + 20 \, \mathrm{Log}_{10} r + 0.5 - T$$

Figure 37-3. Free Field Noise Radiation Point Source — Nondirectional.

$$R = \frac{\bar{\alpha}\, S_t}{1 - \bar{\alpha}} \text{sq. ft.}$$

where S_t = total area of room bounding surfaces in sq. ft.

$\bar{\alpha}$ = average sound absorption coefficient of room bounding surfaces

$$= \frac{S_1 \alpha 1 + S_2 \alpha 2 + \ldots . S_n \alpha n}{S_1 + S_2 = \ldots\ldots S_n}$$

where S_1, S_2, etc. = area of each absorbing surface in sq. ft.

α 2, α 2, etc. = corresponding coefficients of absorption

R can be estimated from Figure 37-4 as an alternative to the calculation.

Directivity factor (Q). Q is a measure of the degree to which sound is concentrated in a certain direction rather than radiated evenly in a full spherical pattern. Directivity factors for typical radiation patterns are shown in Figure 37-5. They are actually portions of spherical radiation patterns as related to the surface area of a sphere which is $4\pi r^2$.

For spherical free field radiation, that is, where there are no reflections, the radiation pattern is illustrated at the upper left of Figure 37-5 and Q=1.

For hemispherical radiation, such as shown at the upper right, or in areas where the sound source is near the center of the floor in a large room the same sound source would produce twice the concentration of sound energy at a point on the surface of the hemisphere as it would on the surface of a sphere of the same radius. The surface area of the hemisphere is $4\pi r^2/2$ and Q=2.

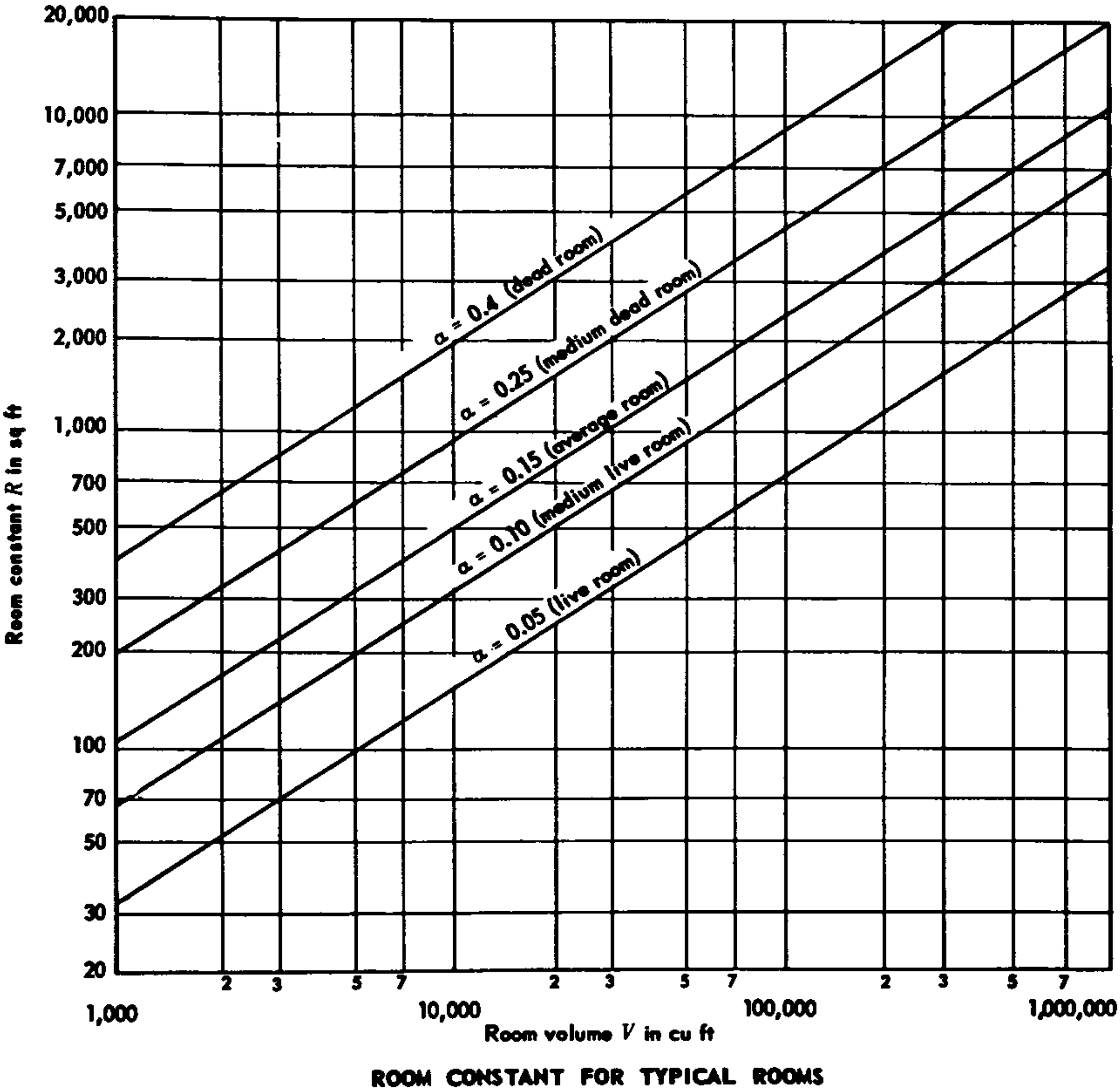
Similarly, if the sound source is near the intersection of the floor and a wall of a room such that 1/4 spherical radiation exists as shown at the lower left, the radiating area can be expressed by $4\pi r^2/4$ and Q=4.

Similarly, if the sound source is near the intersection of the floor and two walls of a room such that 1/8 spherical radiation exists as shown at the lower right, the radiating area can be expressed by $4\pi r^2/8$ and Q=8.

The source might also have a directional noise radiation pattern indicated by the vendor. If so, this would have to be taken into account in addition to the environmental radiation pattern discussed above.

This is a simplified presentation of directivity, but should be sufficient for most industrial situations.

Noise measurements made in the vendor's test laboratory can be modified to estimate the



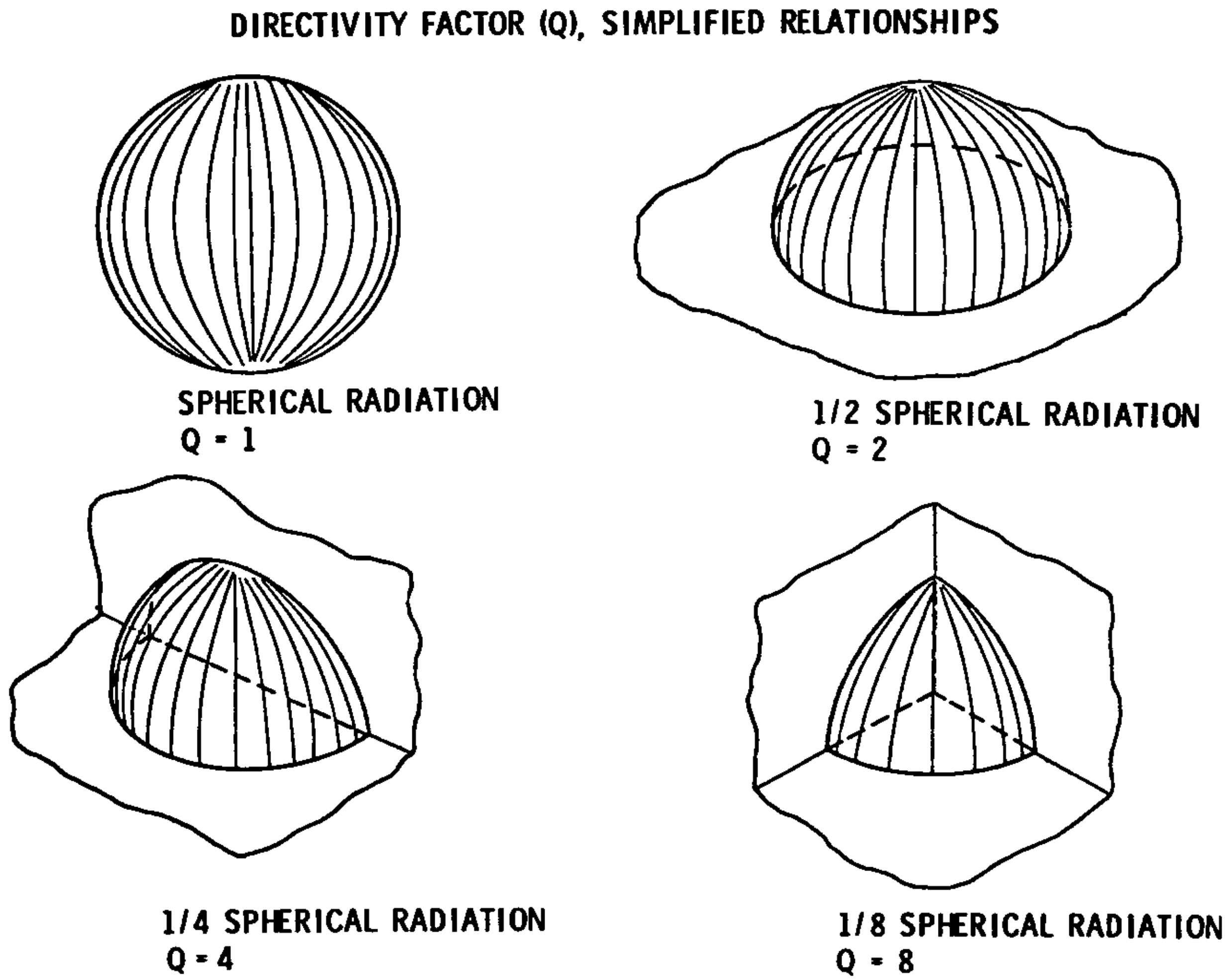
Beranek, L. L. (ed): Noise and Vibration Control. New York, McGraw-Hill, p. 277.

Figure 37-4. Room Constant for Typical Rooms.

levels in the purchaser's intended environment by the use of Figure 37-2. The power level of the noise source will be the same in both locations. The sound pressure level in the purchaser's environment will be the difference between L_p — L_w in the vendor's shop as compared to L_p — L_w in the purchaser's environment. L_w is determined from previous calculations by working through Figure 37-2 and the measured L_p. Then L_p for the purchaser's environment can be determined by working through Figure 37-2 for the new conditions.

Substitution. Sometimes it is possible to substitute a quieter machine, process or material. It is very likely that noise was not considered at the design stage of existing projects (plants). For new projects it will probably be less expensive to buy quiet equipment than noisy equipment that will require noise reduction treatment. Figures 37-6 through 10 illustrate some substitute possibilities.

Quieter materials. The materials used in the construction of buildings, machines, pipes, chutes and containers have a vital relation to noise control. Some materials and structures have high internal damping; others have little and ring when struck. These latter are the potential trouble makers and should be avoided where the possibility of vibrational excitation is involved. Ringing can be re-



duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-5. Directivity Factor (Q), Simplified Relationships.

duced by damping the material or reducing the exciting impact by means of resilient bumpers. Figures 37-11 and 12 illustrate the use of quieter materials. Damping will be discussed later in this chapter.

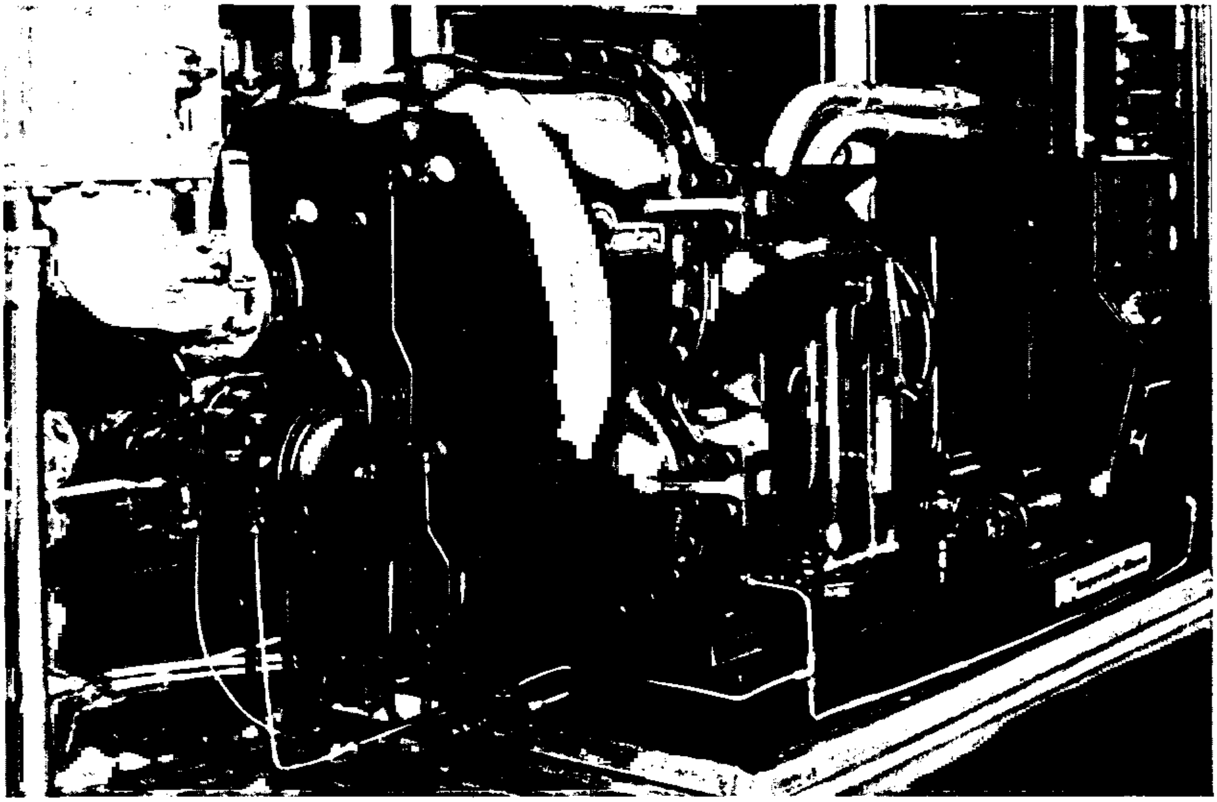
Modification of the Noise Source

There are two basic noise sources: (1) vibrating surfaces and (2) fluid flow. In either case, usually, the nearer the source one can affect treatment, the less expensive will be that treatment because it will be minimum in size.

Direct sound away from area of interest. Many industrial sound sources are directional, that is, they radiate more sound in one direction than in others. Common examples of directional sources are intake and exhaust (vent) openings, partially enclosed sources, and large sheet metal surfaces. It is sometimes possible to utilize directionality of the source to provide noise control in a particular region of the sound field. This type of control is achieved by directing the source so that a minimum in the sound field occurs at the point or area of interest. A typical example is a vertical vent stack that directs the sound above the populated area, or a vent stack cut off at an angle to direct the sound to one side. When the sound source is in a room it is not possible to achieve worthwhile noise reduction by source direction when the point of interest lies in the reverberant portion of the sound field. For enclosed areas containing little sound absorption the reverberant field may extend to within a few feet of the source, and direction of the source will have little effect on the sound levels throughout most of the area. Under these conditions there will be some advantage in directing the source to an area of highly absorbent material, for this effectively reduces the source strength as far as the remainder of the room is concerned. Figure 37-13 is an example where sound has been directed away from the point of interest.

Reduce vibrating surface. This type of noise source will consist of a driving force, coupled to a sound radiating surface. Control at the source may then consist of reduction of the driving force, reduction of the radiating surface response to the driving force or reduction of the radiating efficiency of the vibrating surface.

Reduce driving force. The driving (vibration ex-



Ingersoll Rand, Allentown, Pennsylvania.

Figure 37-6. QUIETER EQUIPMENT — CENTRIFUGAL COMPRESSOR: This high speed centrifugal multi stage compressor has a heavy cast case which encloses the impellers and interstage cooling system. It also encloses the noise so that operating areas around the compressor do not exceed 90 dbA provided motor and external piping noise is controlled. This compressor is a good example of a machine well designed for noise control.

citing) force is often the result of unbalance or eccentricity in a rotating piece of equipment. Such forces increase with increase in rotational speed, therefore the speed should be kept to a minimum. Figure 37-14 is an example of the effect of speed reduction. A large machine running at slower speed might be a better selection as far as noise is concerned. Certainly eccentricity and balance should be checked to be sure they are within normal tolerance. Good alignment, lubrication, and bearing maintenance are also important in minimizing noise. Figure 37-15 shows the noise reduction achieved by improving maintenance on a blower system. Driving forces can also be caused by reciprocating members such as pistons or rams.

Impact type driving forces are produced in

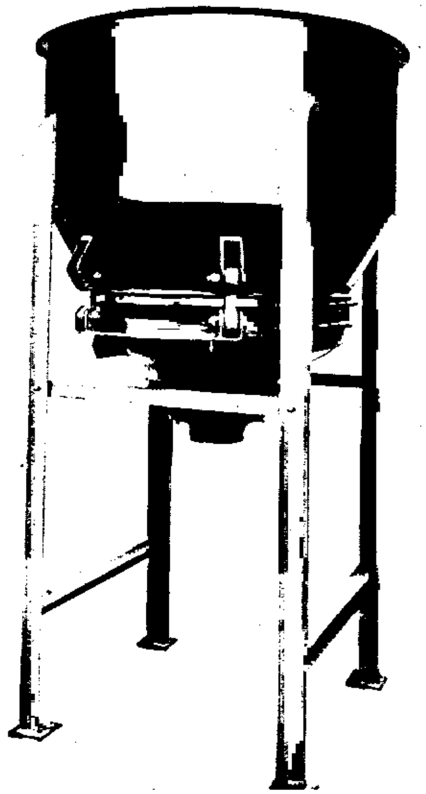
Figure 37-7. QUIETER EQUIPMENT — V-BELT DRIVE.

A rubber toothed type belt with flanged grooved pulleys was used to drive a pump. Noise levels were too high in the octave bands above 2400 Hz. The grooved pulleys and toothed belt were replaced with a V-belt drive. Reductions of more than 15 dB were obtained in the octave bands above 2400 Hz as shown below.

The frequency distribution of noise created by a toothed belt drive is dependent on the tooth passage rate — the higher the speed, the higher the frequency.

If a toothed type belt must be used, the noise could have been reduced by enclosing the belt and pulleys. The enclosure should be lined with sound absorbing materials which is effective for the frequency range of interest.

Frequency — Hz — octave band	20-75	75-150	150-300	300-600	600-1200	1200-2400	2400-4800	4800-9600	9600-19,200
Noise Reduction in dB	5	4	4	2	0	6	17	18	25



Vibra Screw Incorporated, Totowa, New Jersey.

Figure 37-8. QUIETER EQUIPMENT — VIBRATION ISOLATED HOPPER. An 8 ft. dia. hopper with electric solenoid type vibrator was creating excessive noise. A live bottom bin by Vibra Screw was installed as shown below and a noise reduction achieved as shown here. The noise reduction is due to the fact that the cone only is vibrated, there is much less vibratory power required and there is no metal to metal impacts.

Frequency—Hz—octave band—center frequency of band	63	125	250	500	1000	2000	4000	8000
Noise Reduction—dB	7	6	20	22	16	12	12	9

most metal or plastic fabricating operations such as punching, forging, riveting and shearing. Because of the short duration of most impact forces, considerable noise reduction can be achieved by modifying the system to provide a smaller force over a longer period of time. Figure 37-16 shows how this can be accomplished with a punch. Figure 37-17 illustrates this principle on a 48″ shear. Here the cutting blades are segmented and skewed to give a shear type cut.

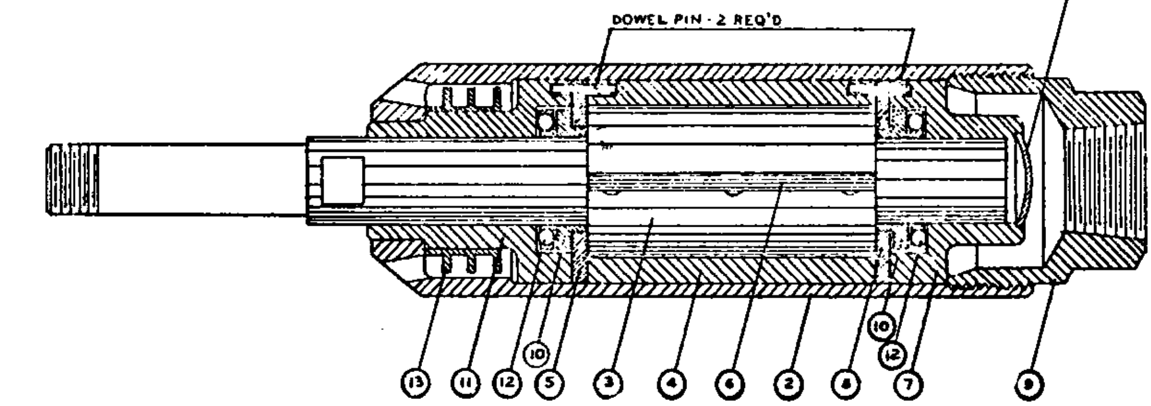
Impact type forces can also be reduced by providing resilient bumpers at the point of impact. Examples of this method include lining tumbling barrels, chutes, hoppers, stock guides, etc. Figures 37-11 and 12 illustrate this method of control.

It is a rare case where a machine causes sufficient vibration of a building to cause the building to radiate noise in excess of 90 dBA. However, a common pitfall about equipment mounting should be pointed out here. It is becoming common practice to mount machines and their drives on a common base of steel weldments. This is fine for alignment and shipment, but for vibration producing machines such as cutters, pulverizers, grinders, blowers, compressors, the steel base can become a serious noise radiator. This problem can easily be overcome by constructing the steel base so that after installation it can be filled with nonshrinking concrete or sand. If it is desired to vibration-isolate the equipment, the isolators should be between the concrete filled steel base and the building floor. The heavy base is also desirable in this case so that center of gravity of the equipment will be lower, that is, nearer the level of the vibration isolation mounts. For stability's sake the level of the isolators should be as close as practical to the vertical center of gravity of the vibrating machine. Since good vibration isolators are readily available and manufacturing instructions for selection and installation are adequate for most cases, further discussion will not be given here. Vibration isolation is usually not necessary except near offices or control rooms or where process equipment dictate the need. Vibration isolation of equipment can cause the more serious problem of pipe line failure at the flexible joints in the lines required by the increased vibration of the isolated equipment.

Reduce response of vibrating surface. The response of a vibrating member to a driving force can be reduced by damping the member, improving its support, increasing its stiffness or increasing its mass. When the frequency of the driving force is equal to the natural frequency of the member being vibrated, large surface displacements are usually developed. This condition is known as resonance. Most mechanical structures have a family or series of resonances which are rather widely spaced in the low frequency range but are more closely spaced at higher frequencies. Because of the large surface displacements developed at resonance, there is usually increased noise radiation. Resonant vibration may be limited effectively by damping, decoupling or detuning by shifting the natural frequency. Optimizing a damping treatment is usually a complicated procedure best left to the experts if the cost can be justified. For many industrial problems it is satisfactory to use a simple rule of thumb approach. For the rough treatment typical of industrial environments, constrained layer damping is usually preferred. This means covering the vibrating surface with a thin sheet of damping material plus an outer covering of sheet metal. The sandwich so formed is cemented (both surfaces) and bolted together on 6″ to 8″ centers. The rule of thumb is that for vibrating panels having a thickness of up to 16 gauge, use an outer steel plate (restraining plate) of the same gauge as the vibrating plate. For vibrating plates of 16 gauge to ⅛″ thick, use a restraining plate of 16 gauge steel. For vibrating plates of ⅛″ to ¼″ thick, use a ⅛″ thick re-

FIGURE 37-9 | 115000-B.2 7/16

An Elliott No. 1380 tube cleaner (2-7/8" dia) created excessive noise. A new design, Elliott No. 115,000 was substituted. The new design had a built in muffler for the air discharge of the turbine drive as shown by the drawing. Noise reduction achieved is shown in the table.



14	CA PLUG	P-8161 K	1				
13	MUFFLER UNIT	115088	1				
12	THRUST BEARING	112054	2				
11	FRONT BEARING ASSEM	112050	1				
10	THRUST WASHER	112037	2				
9	MACHINE COUPLING	112031	1				
8	REAR PLATE	112027	1				
7	REAR BEARING ASSEM.	112009	1				
6	PADDLE	112008	5				TC-
5	FRONT PLATE	112007	1				TC-
4	SHELL	112006	1				TC
3	ROTOR	115005 A	1				TC-
2	BODY	115002:2 7/16	1				TC
1	ASSEMBLY	115000-B:2 7/16		—			TC-
	PART • 4E	PART No	REQ	CHG OUT	L	M	ENGR CHANGE

BODY DIAM 2 7/16
SHAFT THRD 3/4",
HOSE CONN 3/4" N.P.T.

TYPE "DBT" -MUFFLED
AIR DRIVEN MOTOR
(EXTRA LONG ROTOR)

FOR

BY | CK | DATE 6 13-66

ELLIOTT COMPANY
SPRINGFIELD OHIO
NEWARK N J

SE-
11500C :2 7/16

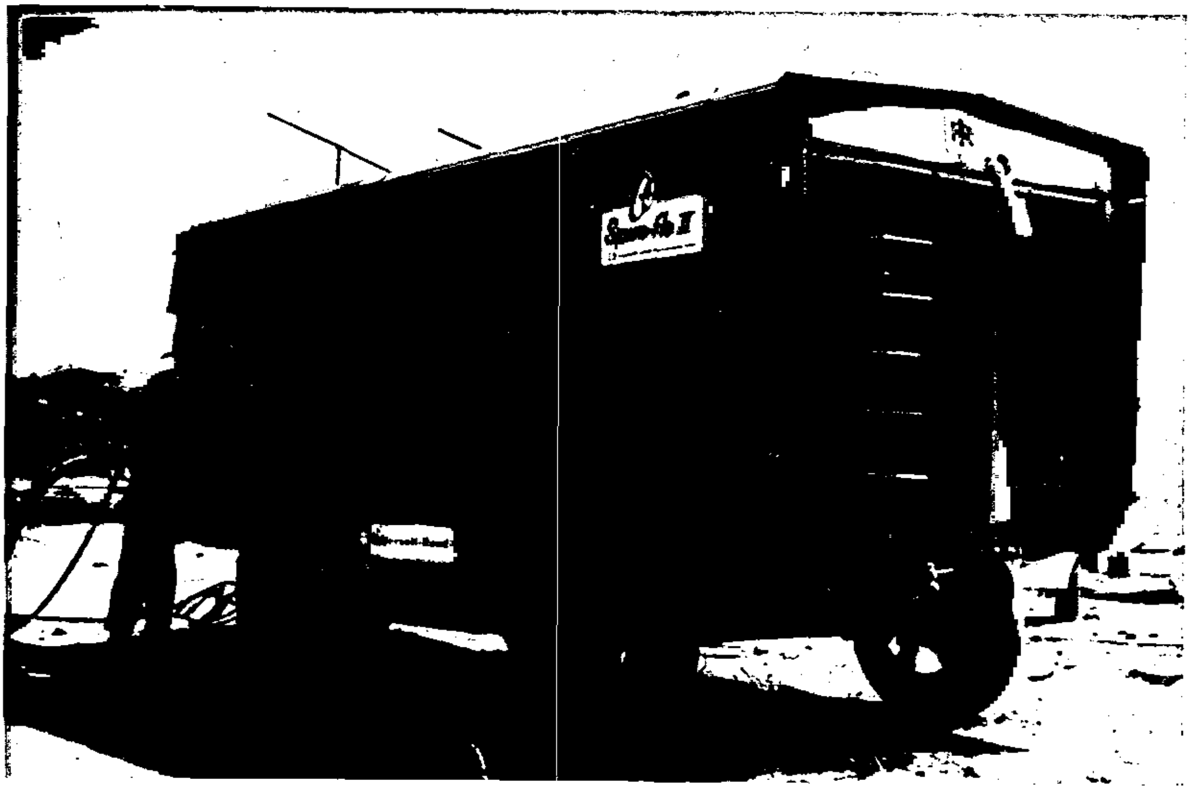
Elliott Company, Springfield, Ohio.

Figure 37-9. An Elliott No. 1380 tube cleaner (2-78" dia.) created excessive noise. A new design, Elliott No. 115,000 was substituted. The new design had a built-in muffler for the air discharge of the turbine drive as shown by the drawing. Noise reduction achieved is shown in the table.

straining plate. For vibrating plates of ¼" thick or heavier, use a ¼" thick restraining plate. The common damping materials are damping felt, elastomeric damping sheeting and sheet lead. All material selected should be compatible with the temperature and environmental conditions involved such as exposure to chemicals and oils. It is to be noted that the flat unsupported surfaces are the ones radiating the most noise. Corners of box-like structures, reinforcing bosses, etc., are so rigid they probably do not require damping. This simplifies the damping treatment because it eliminates many double curved surfaces which would be difficult to laminate. Manufacturers of damping materials can advise regarding the most effective use of their materials. Do not hesitate to use the heavy restraining plates suggested above. They make the panel significantly stiffer and reduce not only resonance but the driven response as well. The extra weight and stiffness might be the most important factor in reducing the noise. Figure 37-18 illustrates constrained layer damping and damping by means of filling structure with sand. Figure 37-19 illustrates noise control by increased mass and stiffness.

Reduce efficiency of noise radiating surface. The sound energy generated by vibrating surfaces depends not only upon the velocity of surface motion but also upon the area of the radiating surface. Because the displacement of most surfaces is limited by the constrained edges, the surface velocity will decrease with frequency, and the area must increase if constant sound output is to be maintained. Therefore, the effective radiation of low frequency sound is usually limited to large surfaces. Conversely, any surface of more than several square inches can effectively radiate sound at frequencies above 1000 Hz. In general, any regularly shaped area with one dimension greater than one-fourth wavelength can effectively radiate sound at the frequency corresponding to that wavelength in air.

Surfaces radiating low frequency sounds can sometimes be made less efficient radiators by di-



Ingersoll Rand, Allentown, Pennsylvania.

Figure 37-10. QUIETER EQUIPMENT — PORTABLE AIR COMPRESSOR: The Ingersoll Rand portable air compressor shown above was designed with noise control as a specification. All functional components of this diesel — engine powered machine including engine, compressor, mufflers, fuel tanks, receiver separator tank and frame are completely enclosed in an aluminum, glass fiber, sheet steel sandwich-panel material. Improved cooling by increased air flow through mufflers was required for this enclosed machine. The noise reduction achieved by this design is shown below.

Frequency — Hz — octave band center frequency of band	63	125	250	500	1000	2000	4000	8000
Noise reduction in dB	8	13	24	17	10	10	10	14

viding them into smaller segments or otherwise reducing the total area. The use of perforated or expanded metal can often result in less efficient sound radiation from sheet metal guards and cover pieces.

Turbulent fluid flow. A very common industrial noise source is high velocity fluid flow. The strange thing about it is that usually the velocity required by the industrial process is not high enough to create a serious noise problem. For example, where compressed air is used to clean or wipe a product, such as blowing water from a freshly extruded plastic, the noise source is the sonic velocity of the gas passing through the pressure reducing valve. The noise source is not the air velocity which wipes the water from the plastic. This problem (and many more similar ones) can be solved by placing a muffler just downstream from the

Figure 37-11. QUIETER EQUIPMENT — RESILIENT LINING FOR TUMBLING BARREL.

The tumbling of steel balls against the steel shell of a ball mill can produce excessive noise. By lining the steel with resilient material this noise will be reduced by a considerable amount. One such mill lined with rubber produced the noise reduction shown below.

Frequency — Hz — octave band — center frequency of band	63	125	250	500	1000	2000	4000	8000
Noise reduction — in dB	3	4	6	7	11	12	15	19

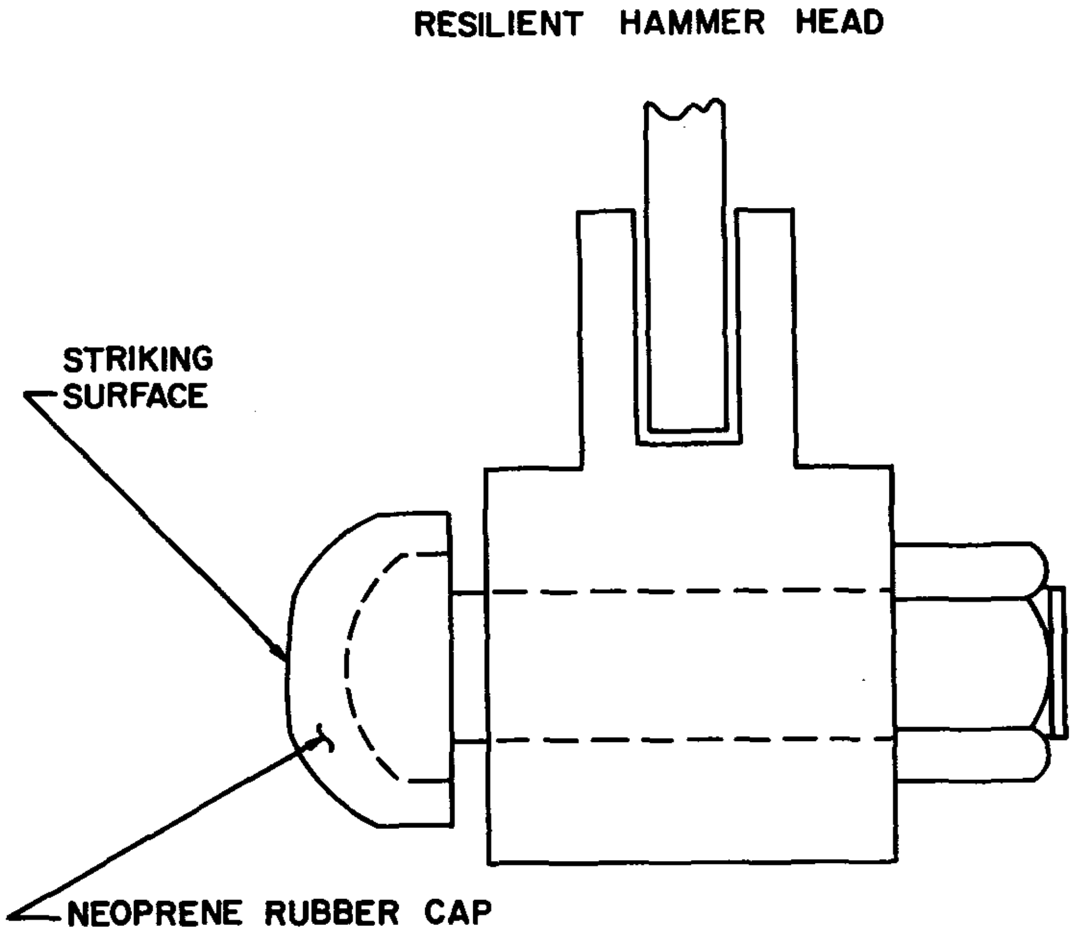


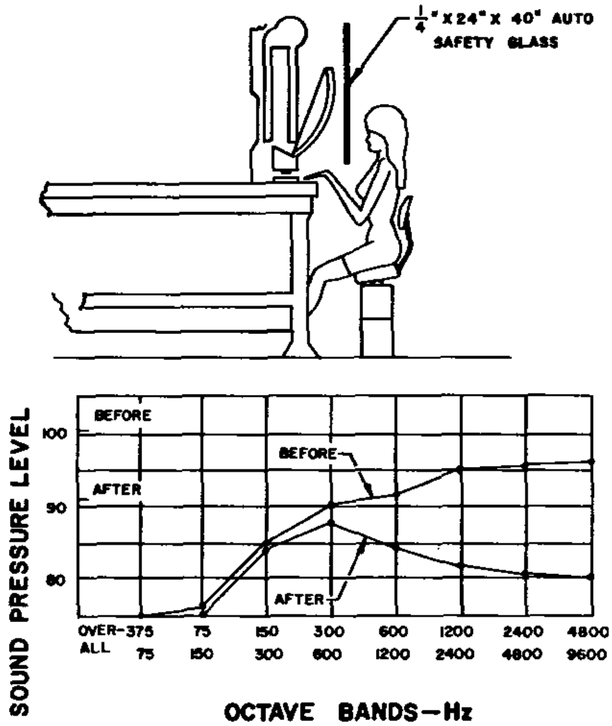
Figure 37-12. QUIETER MATERIAL — RESILIENT HAMMER HEADS: Rotary dryers commonly use hammers (Knockers) on the outside of the dryer shell to prevent product buildup on the inside. The metal to metal impact noise produced is usually objectionable. This noise can be reduced by providing resilient heads for the hammers. By providing sufficient striking area between hammer and shell, the resilient facing material can usually be made to transmit the desired vibration to the dryer shell without causing the objectionable metal to metal impact noise. In one case, the overall noise level was reduced 28 dB. Common materials used for the face of hammers are Adiprene®, neoprene, nylon, Fabreeka and rawhide.

NOTE: Lempco Automotive, Inc., supply nylon hammer heads and Garland Mfg. Co. supply rawhide hammer heads.

valve as shown by Figure 37-20. The muffler is to remove noise from the sonic velocity in the valve. Then with the nozzle downstream from the muffler designed for minimum velocity to do the job, there should be no noise problem. Velocities as high as 10,000 ft. per minute can be used without excessive noise, and even this velocity may not be needed for many processes. The rule is, for low noise don't use velocities higher than necessary. In particular, don't use sonic velocities. Beware of gas pressure reducing valves.

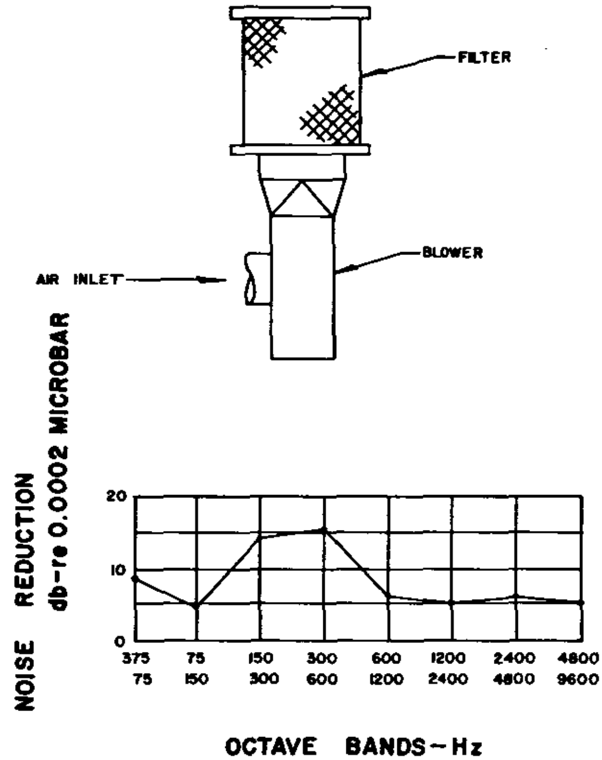
Where the ratio of upstream to downstream absolute pressures is 1.9 or greater, sonic velocity and excessive noise is produced — unless the reduction is controlled through the use of a special valve which avoids sonic velocity. A valve which accomplishes this achieves a gradual pressure drop and expanding volume such that sonic velocities are not reached. The valve consists of a stack of plates as shown by Figure 37-21 and each plate has small gas passages as shown by Figure 37-22. The high pressure gas enters from below as the stem rises. When the stem passes by the ID of a plate, the gas flows through the tortuous path to the plate O.D. where it is at the low pressure level. As the valve stem rises, more plates (and passages) are exposed and more volume of gas passes. This is called the "Drag Valve." It can be designed for any desired noise level and gas flow.

For existing pressure reducing valves or for applications where quiet valves cannot be used due to dirty gas or lack of economic justification,



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

Figure 37-13. NOISE SHIELD — DIRECT SOUND AWAY FROM POINT OF INTEREST. The use of shields between a noise source and an employee is usually quite effective when both the source and the employee are close to the shield and when the noise is predominantly high frequency. An example is shown on the punch press which uses compressed air jets to blow foreign particles from the die. The installation of ¼" thick safety glass shield gave the reduction shown in the graph.



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

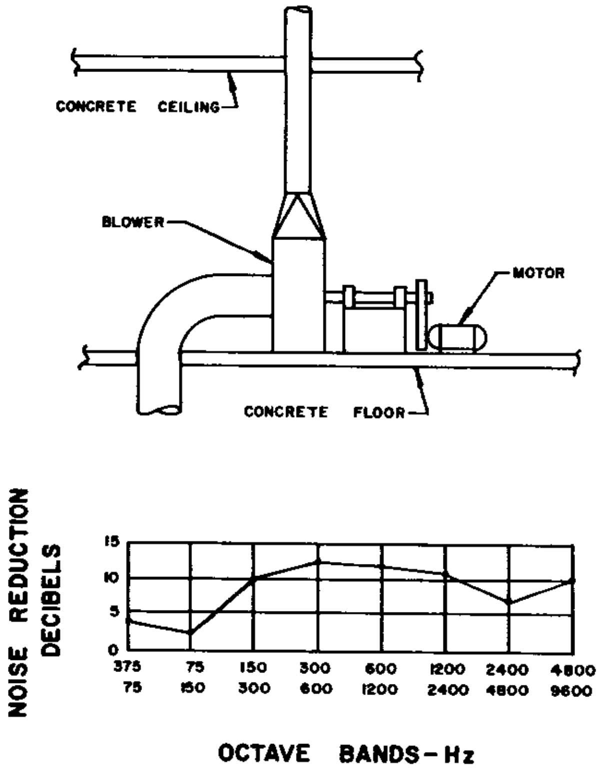
Figure 37-14. MINIMIZE ROTATIONAL SPEED. The blower for a vapor collection system produced excessive noise while moving 3600 cfm at 2.8" static pressure. It ran at 3450 rpm and had a 12.5" diameter material wheel. It discharged into a cylindrical filter consisting of 1.5" thick glass fiber compressed to 0.75". A quieter fan was selected and the noise reduction achieved is shown on the graph. The new blower ran at 900 rpm and had a 32.265" diameter air wheel. A material wheel was not required since only air and oil vapor were being handled.

other means of noise control are available. If commercial mufflers can be used, the noise can be controlled as shown by Figure 37-23. If mufflers cannot be used, external pipe covering can be applied as shown by Figure 37-24. This last treatment is not as economical as it might appear since sound can travel long distances down pipes with little attenuation and the pipe covering might be quite expensive. In addition, the sound might produce excessive vibration in downstream equipment and cause failure of such things as heat exchangers and packed columns or cause excessive noise radiation from suction bottles, etc.

To select mufflers for the usual type pressure reducing valves one must estimate the noise level just downstream from the valve. Unpublished work by K. U. Ingard provides a convenient method for doing this.

Figure 37-25 shows that by relating the absolute pressure drop ratio and the gas flow in lbs. per minute, the overall sound power can be determined. Then by means of Figure 37-26 the octave band power levels can be determined. In Figure 37-26 the zero line corresponds to the overall power level as determined from Figure 37-25. The frequency scale used in Figure 37-26 is normalized to the frequency $f_1 = 0.2$ c/d where c is the speed of sound in the gas and d the equivalent valve diameter. The frequency f represents the center frequency of the corresponding octave band. To illustrate the use of Figure 37-25 and 26, consider the following example:

Effective port area of valve	= 1 sq. in.
Mass flowrate	= 50 lb. per minute
Gas	= air at 190°F
Upstream pressure	= 100 psig
Downstream	= 48 psig



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

Figure 37-15. REDUCE DRIVING FORCE — IMPROVED MAINTENANCE (BLOWER). An exhauster running at 705 rpm, 6″ static pressure, and 13,800 cfm was badly out of balance and the bearings needed replacing. As a result the blower produced excessive noise. After balancing and installing new bearings the noise was reduced as shown by the graph.

Determine the octave band power levels generated at the valve discharge and the octave band sound pressure levels in the 3″ line just downstream from the valve.

From Figure 37-25 at 50 lb. per minute and a pressure ratio of 2.1 the overall power level (L_w) would be 127 dB. The equivalent valve port diameter for an effective area of 1 sq. in. is —

$$A = \pi \left[\frac{d}{2}\right]^2$$

$$\text{when } A = 1 \qquad d^2 = \frac{4}{\pi} \qquad d = \sqrt{\frac{4}{\pi}} = 1.13''$$

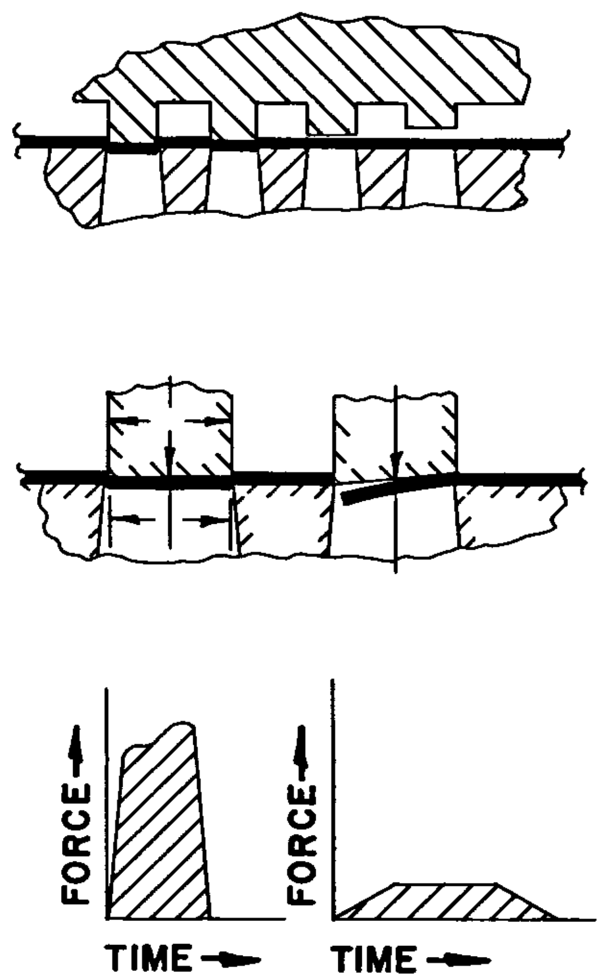
From Keenan & Kaye Gas Tables,[1] C = 1248 ft. per sec. for air at 190°F. Then:

$$f_1 = \frac{.2C}{d} = \frac{.2 \times 1248}{\frac{1.13}{12}} = \frac{.2 \times 1248 \times 12}{1.13} = 2650 \text{ Hz}$$

Checking the frequency ranges of the octave bands, we find that 2650 falls in the 6th octave band which has a frequency range of 1400 to 2800 Hz and a center frequency of 2000 Hz. Now referring to Figure 37-26 for $f/f_1 = 1$ and a pressure ratio of 2.1, the octave band L_w for the 6th octave band would be 127-7 or 120 dB. To obtain L_w for the other octave bands, consider them relabeled as follows:

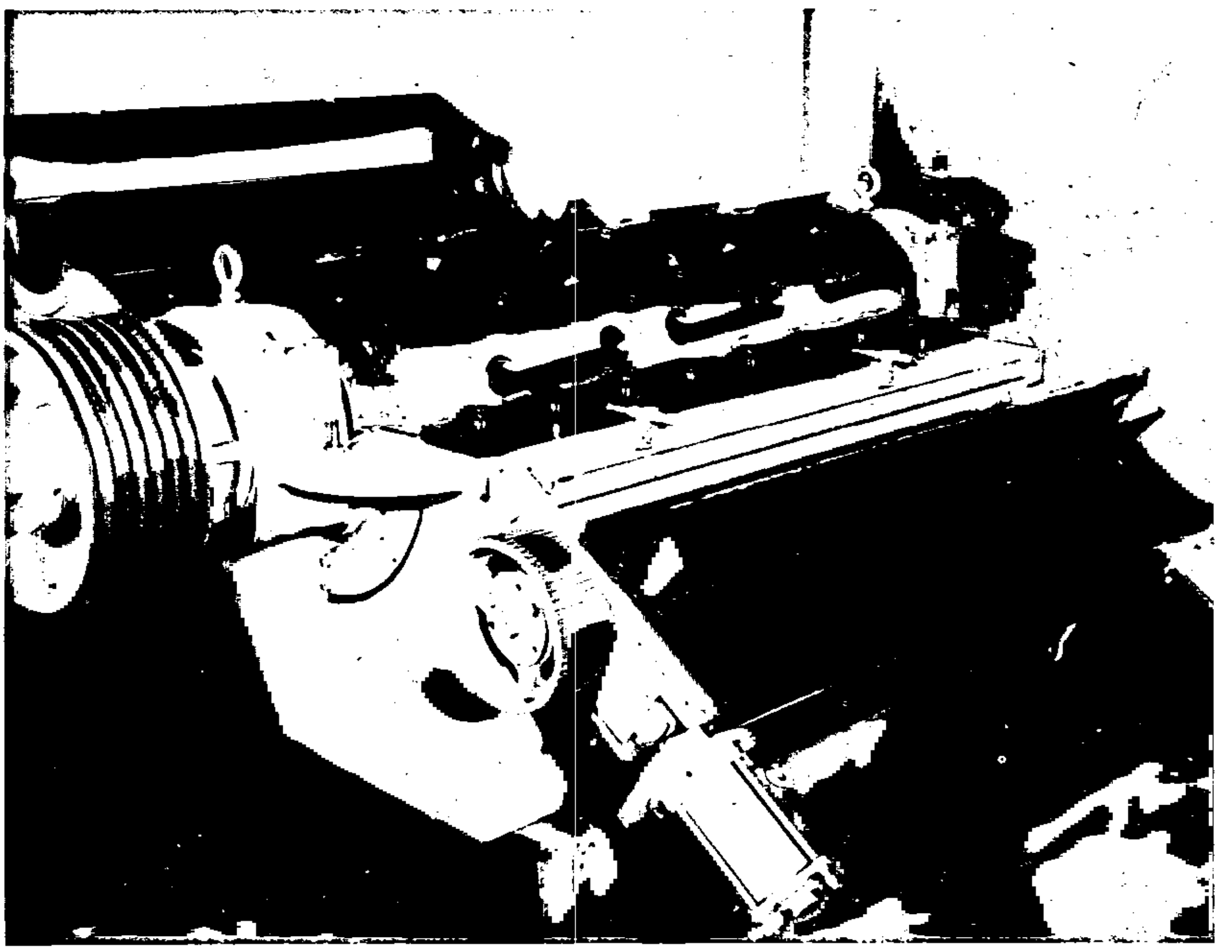
For the 7th octave band $f/f_1 = 2$
For the 8th octave band $f/f_1 = 4$
For the 5th octave band $f/f_1 = ½$
For the 4th octave band $f/f_1 = ¼$ etc.

The next step is to determine from Figure 37-26 the number of dB to be subtracted from the overall L_w to obtain the octave band power levels. This would result in the values shown in column 5 of Table 37-1. Subtracting column 5 from column 4 gives the octave band power levels shown



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

Figure 37-16. REDUCE DRIVING FORCE — SEGMENTED PUNCH AND SHEAR CUT. Illustration of Stepped Punches for Punching Several Holes at One Stroke of the Press. Schematic illustration of blanking operation, showing the effect of shear angle on the punch. The force-time diagram for each condition is shown.



duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-17. Reduce Driving Force — 48″ Film Cutter with Skewed, Segmented Blades.

Figure 37-18. REDUCE RESPONSE OF VIBRATING SURFACE BY DAMPING (EXTRUDER GEAR CASE).

duPont de Nemours & Co., Wilmington, Delaware.

The casing of a 2000 HP extruder gear was radiating excessive noise. The gear cover was ⅜″ steel. The base was 1″ steel with 1″ thick 9″ deep ribs. Measurements with an accelerometer showed that the ⅜″ steel and the 1″ steel were vibrating at approximately the same intensity. This indicated that all surfaces should be damped.

The ⅜″ steel was damped with ¼″ damping felt No. 11 N (by Anchor Packing Co.) plus an outer covering of ¼″ steel. The sandwich (⅜″ steel + ¼″ felt + ¼″ steel) was bolted together on 8″ centers.

The irregularity of the 1″ steel of the base made constrained layer damping (as used on the cover) impractical. Instead, ¼″ steel plate was welded to the 9″ deep ribs and the voids filled with sand. The photographs below show the gear before and after treatment. The table below shows the noise reduction achieved after treatment of only one of three units in the room.

Frequency — Hz — octave band — center frequency of band	63	125	250	500	1000	2000	4000	8000
Noise reduction — in dB	X	X	X	4	17	26	24	18

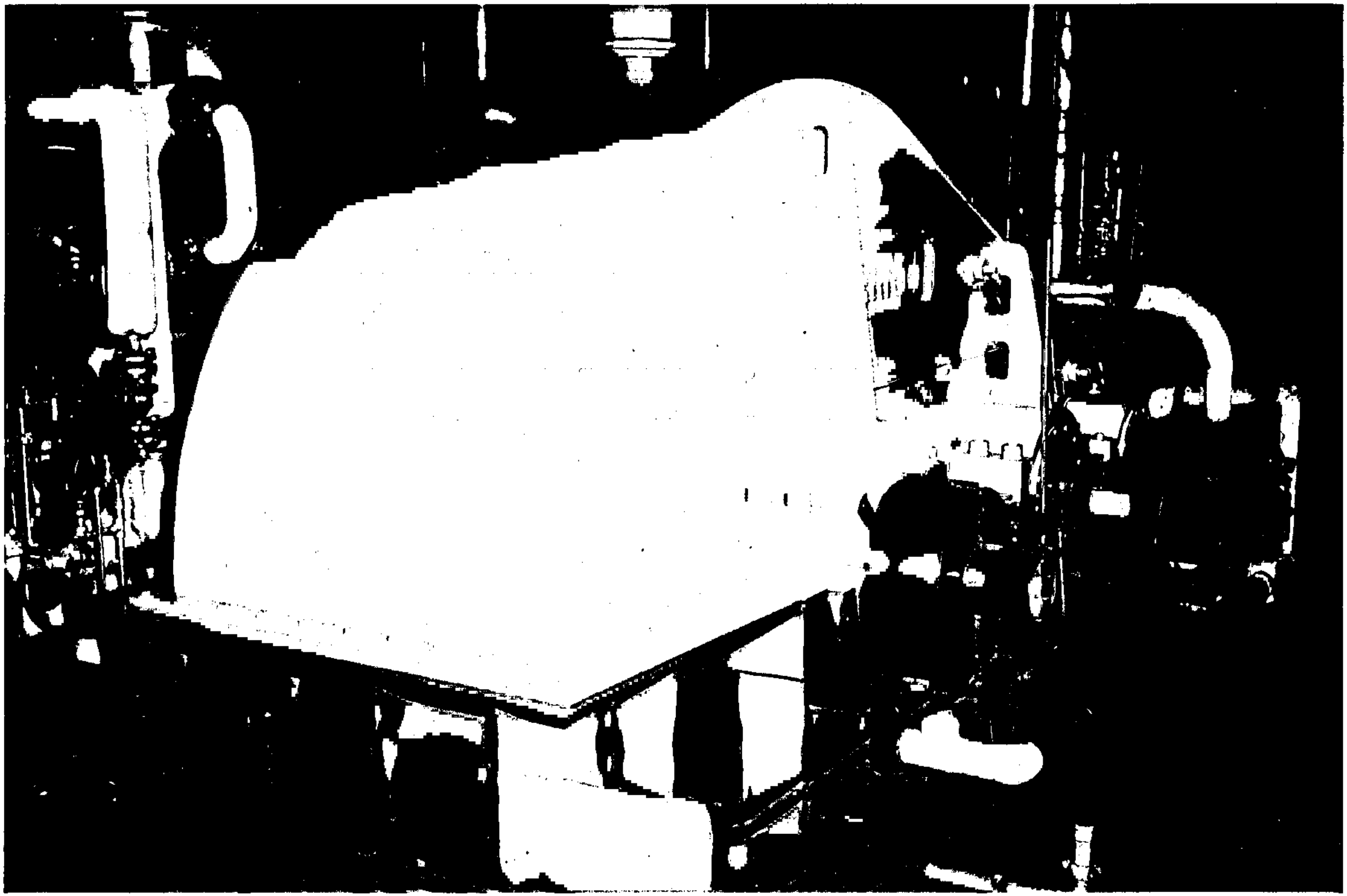


Figure 37-18. (BEFORE)

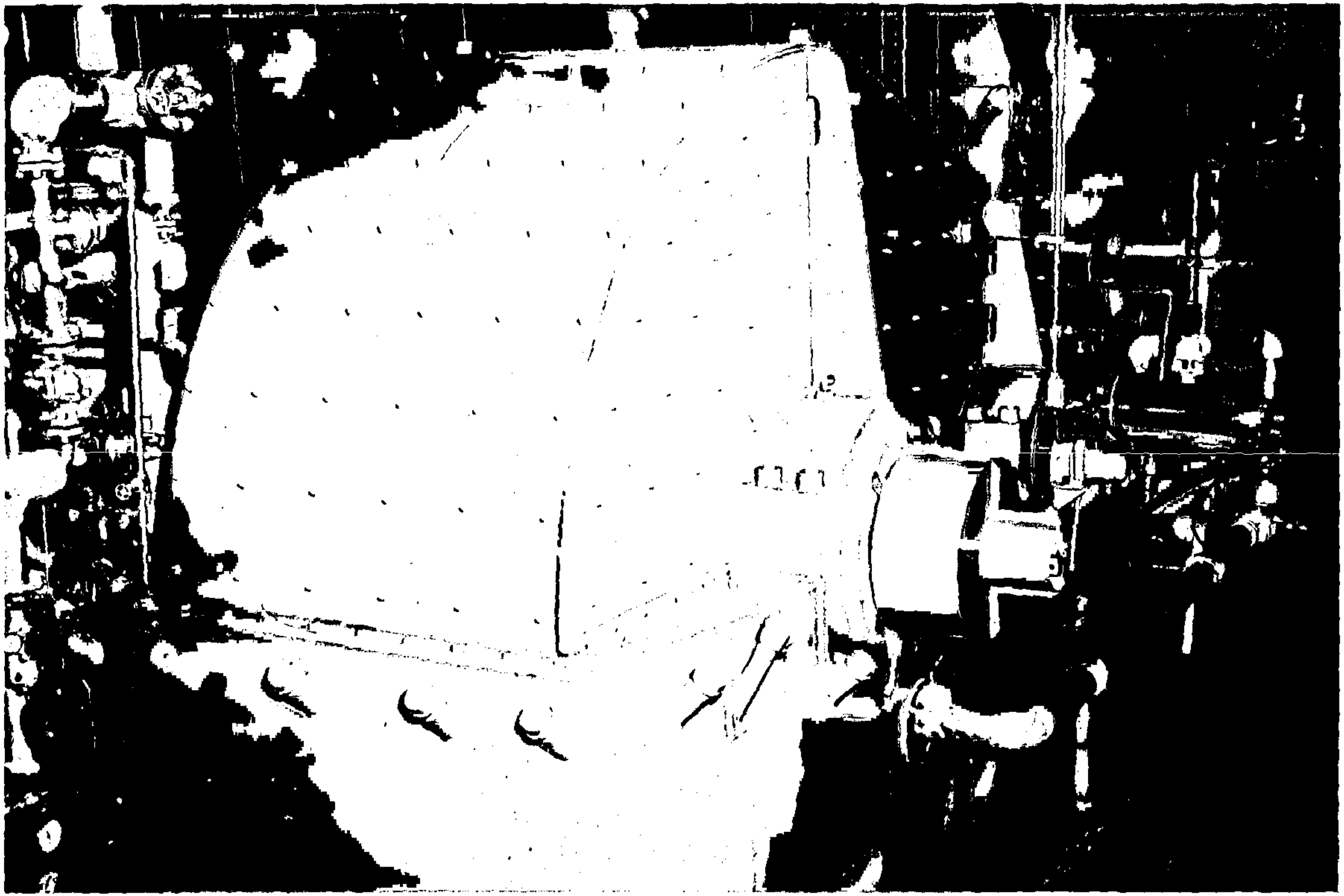
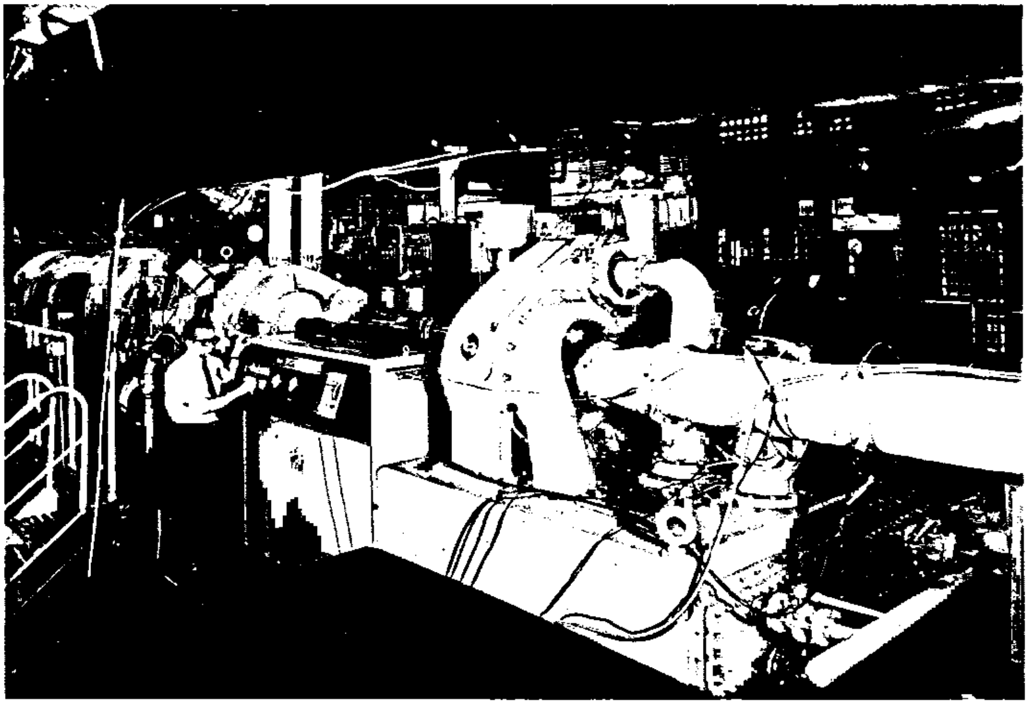
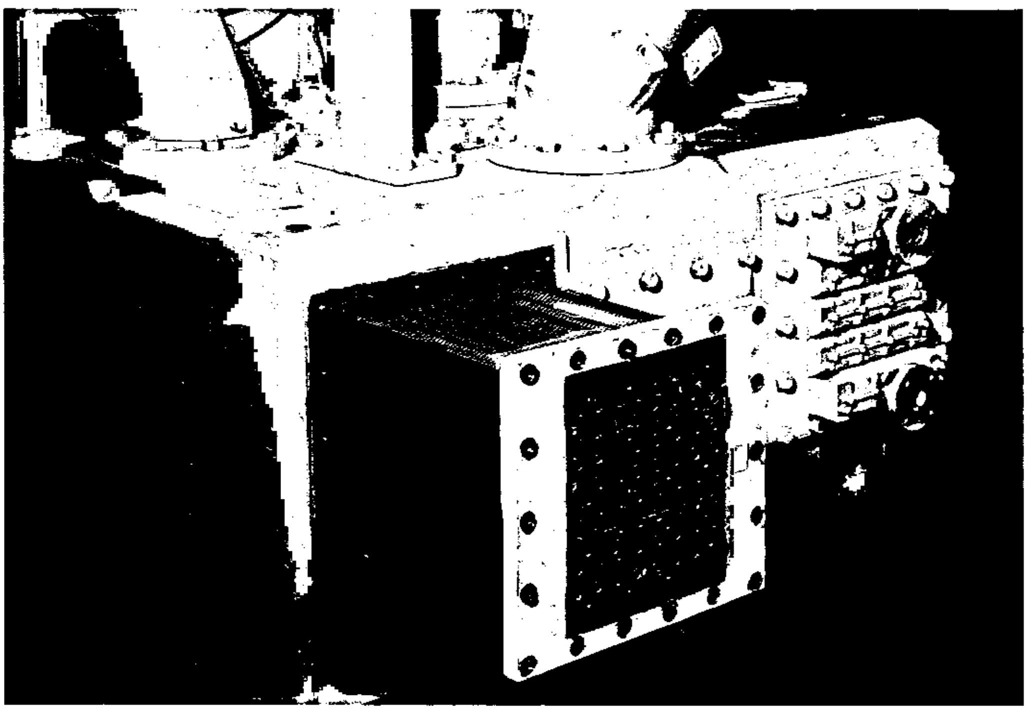


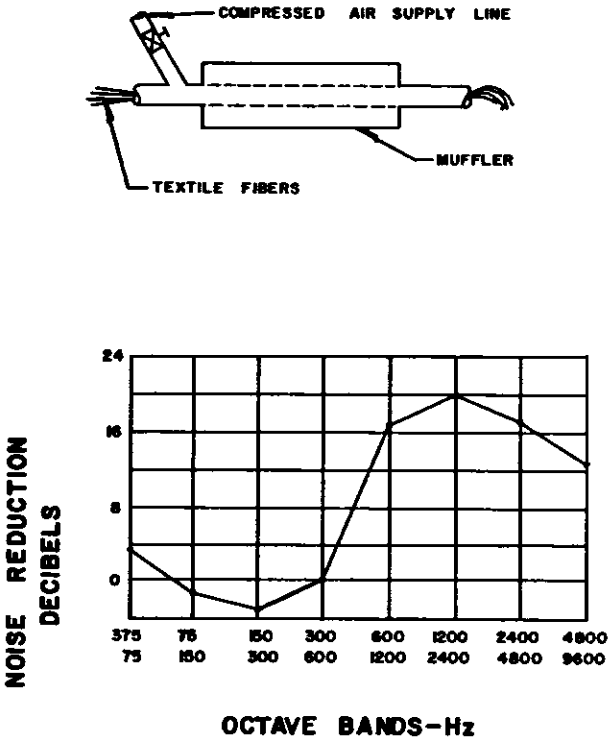
Figure 37-18. (AFTER)





Studebaker — Worthington Inc., New York, New York.

Figure 37-19. REDUCE RESPONSE OF VIBRATING SURFACE BY INCREASED STIFFNESS AND MASS (CENTRIFUGAL COMPRESSOR). These two photographs show a Worthington multi stage high speed centrifugal compressor which had noise control in mind during the design stage. Note the heavy cast construction of this machine. To meet the environmental criteria of 90 dBA, the only parts that require acoustical covering are the gear case cover and the steel interstage piping couplings. Here is another case where extra weight in the machine indicates economical noise control without the inconvenience of enclosure.



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

Figure 37-20. ABSORB THE SOUND WAVE — ALONG ITS TRANSMISSION PATH. An air ejector is used to strip waste textile fibers from perns. The curve shows the noise reduction achieved 3 feet from the ejector by means of the dissipative muffler. The air supply line is ½″, the pressure 100 psi, and a 1″ dissipative muffler was used. Notice that the noise levels in the 75 to 150 and 150 to 300 cps octave bands were increased slightly, which is characteristic of this type of muffler. The noise of this problem is due to the pressure reduction at the valve and not the velocity of air exhausting from the pipe. This is apparent since the pipe size is the same at the inlet and discharge of the muffler.

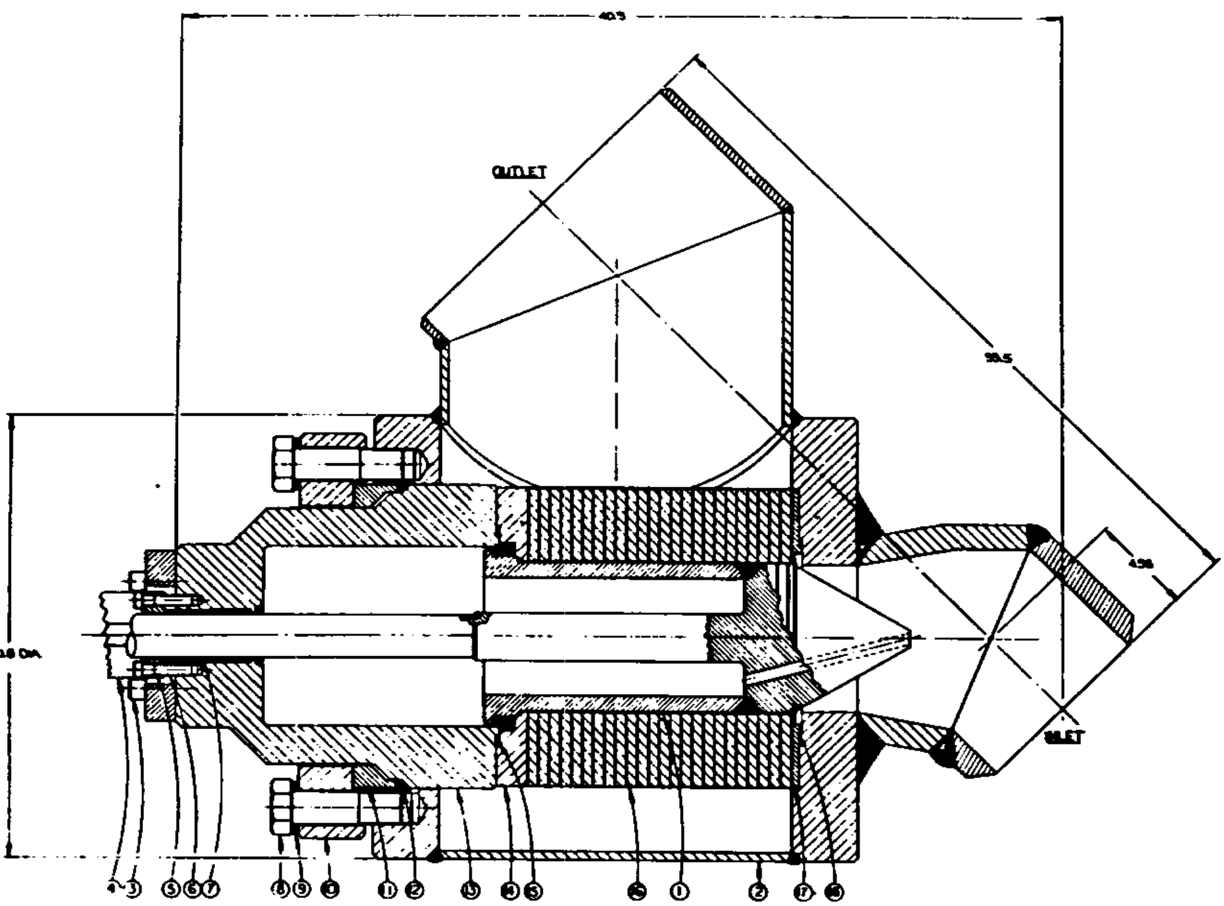
in column 6. The next step is to determine the sound pressure level (L_p) in the 3″ pipe just downstream from the valve. This can be determined by the following formula.

$$L_p = L_w - 10 \log_{10} D$$

where D = area of the pipe in sq. ft.

For a 3″ pipe $L_p = L_w - 10 \log 0.0491 = L_w + 13$ dB

This means that the octave band levels of Column



Control Components, Los Alamitos, California.

Figure 37-21. Reduce Velocity of Fluid Flow and Use Quieter Equipment — Quiet Pressure Reducing Valve.

6 would have to be increased by 13 dB to obtain the octave band sound pressure levels that would have to be contained by the pipe walls or pipe walls plus covering, or would be the basis for selecting a muffler to reduce noise transmission down the pipe.

Streamline the flow. Once the velocity of a gas stream has been reduced to a minimum, additional reduction might be achieved by streamlining any obstructions in its path. The higher the velocity the more important it is to streamline the flow path.

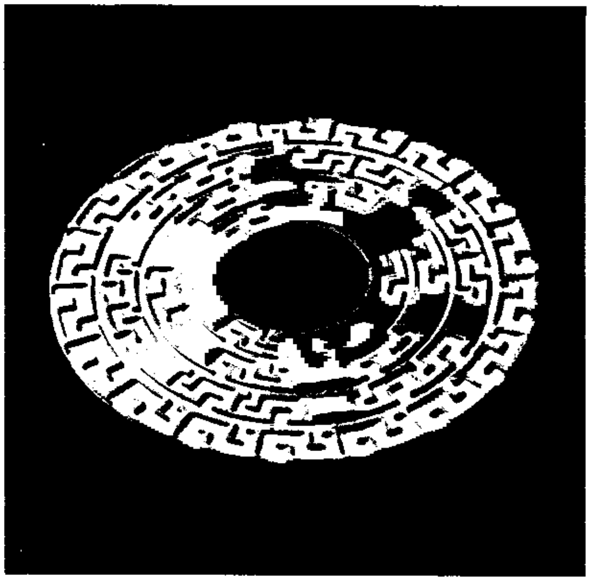
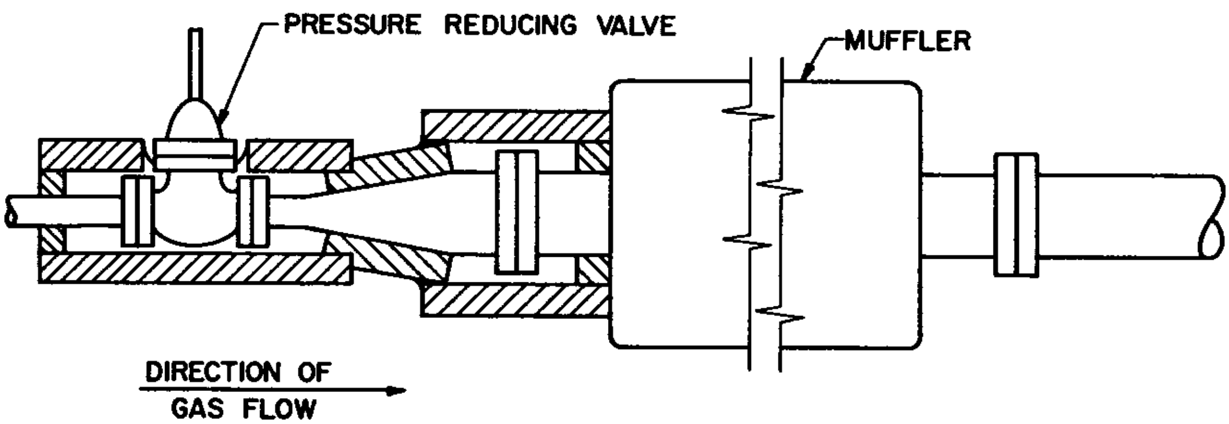


Figure 37-22. REDUCE VELOCITY OF FLUID FLOW AND USE QUIETER EQUIPMENT — QUIET PRESSURE REDUCING VALVE. The "Drag Valve" shown in the drawing at left and described on page 14 is a pressure reducing valve which can provide the desired pressure drop but at the same time limit the maximum velocity thru the value to minimize the vibration and erosion and limit the noise reduction to most any desired level. The photograph at left shows a typical plate which illustrates the tortuous path the gas must take in passing from the inside (high pressure) to the outside (low pressure). For one design where 300 PSIG air was being vented to atmosphere, the noise reduction shown below was achieved as compared to venting through the more common orifice type valve.

Frequency—Hz—octave band—center frequency of band	63	125	250	500	1000	2000	4000	8000	16000	31500
Noise reduction—in dB	24	19	28	17	18	25	34	34	29	25

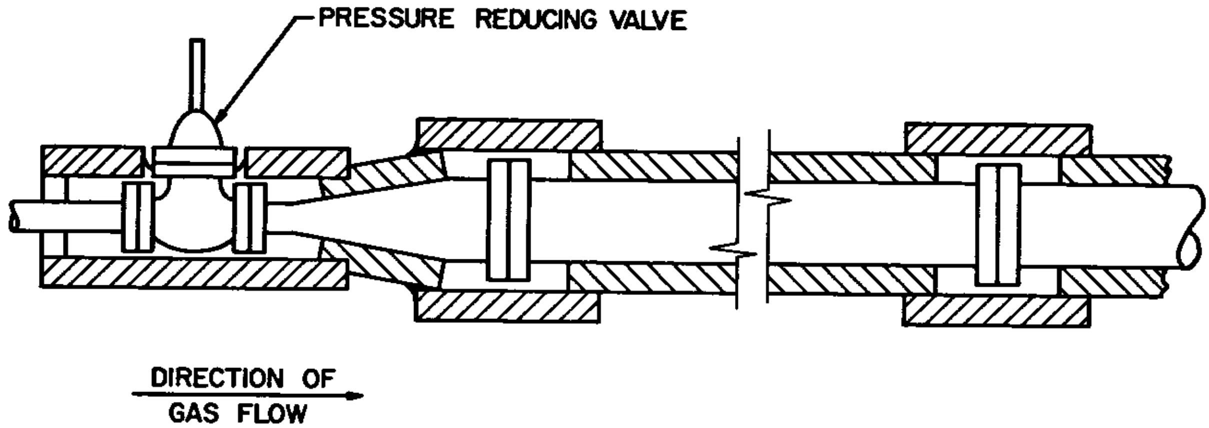
Modification of Sound Wave

Enclosures or partial enclosures. When enclosing a sound source with an unlined enclosure, the noise level inside the enclosure will build up to a considerably higher level than that measured without the enclosure. To avoid a lengthy calculation of this reverberant buildup, one can line the enclosure. If this is done so that the average sound absorption coefficient inside the enclosure is at least 0.70, the reverberant buildup will be insignificant. This allows one to select materials for the enclosure which have a transmission loss (TL) equal



duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-23. CONFINE THE SOUND WAVE — MUFFLE AND ACCOUSTICAL LAGGING: Cover valve and piping between valve and muffler with 2" fiberglas (3 to 4 lb/cu ft plus an outer covering of Coustifab 488C). For unusually high pressure drops and flows, 16-gage steel weather covering might be required.



duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-24. CONFINE THE SOUND WAVE — ACOUSTICAL LAGGING ONLY: Cover valve and downstream piping as described on Figure 23. This method might require the covering of a considerable amount of piping (100 ft or more). In this case, one must be careful not to excite equipment located downstream from the valve. Exceptions of such equipment would be thin-welled heat exchangers, separators, and spray towers.

to the difference between the noise level without the enclosure and the desired noise level with the enclosure. It is good practice to add 5 dB to this difference as a factor of safety. Table 37-2 lists the TL of the more common building materials. This method assumes that the enclosure of the noise source is complete and well sealed. If the enclosure is not complete and well sealed refer to Figure 37-27 for estimating the effect on TL of the leaks or openings. Figure 37-28 shows some common seals and fasteners used for acoustical enclosures. If openings are needed for ventilation or feeding materials into or out from the enclosure, mufflers should be installed at these openings to maintain the desired TL of the rest of the enclosure. If the desired TL is not very great, partial enclosures, that is, enclosures with fairly large openings might be sufficient. The effectiveness of a partial enclosure can be determined by calculating the percent open area as compared to a complete enclosure, and then referring to Figure 37-27.

Absorb sound wave. In general, sound absorbing materials are soft and porous. They are porous so that the sound wave can enter the material but the material must have a high enough flow resistance to reduce the amplitude of the sound wave. If the material is too dense (has too high a flow resistance) the sound will be reflected. If the material is not dense enough (flow resistance too low) the sound wave will pass through unchanged. Materials are rated in their ability to absorb sound by their sound absorption coefficient (α). This is the percent of incident sound which is absorbed in striking it. Table 37-3 lists the sound absorbing coefficients of various materials. The Acoustical Materials Association periodically publishes such data for materials manufactured by their members.

It is important to note that α varies with frequency. Figure 37-29 illustrated this for 6 lb/ft.^{3} Fiber glass. Notice that below 1000 Hz, α drops off markedly for ½″ thick material. At 1″ thickness this drop in α occurs below 500 Hz. For 3″ material the drop in α occurs when the frequency is less than 250 Hz. The point is, when using α in a noise control problem it is important to use α at the frequency of interest.

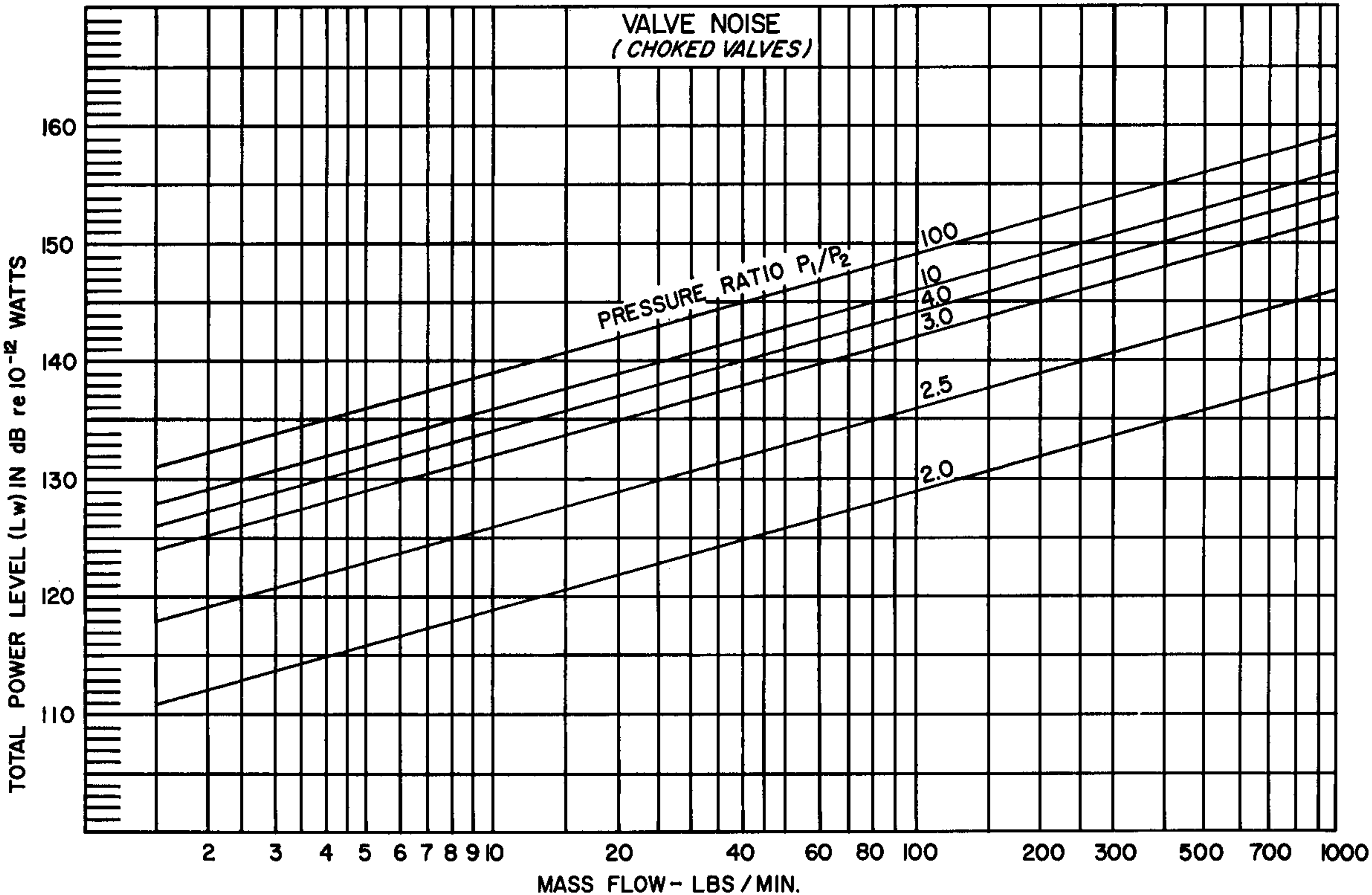
Room Absorption

Noise control by absorption in room bounding surfaces is very limited in effectiveness and relatively expensive due to the large surface areas which must be treated. Seven to 10 dB is probably the maximum reduction that can be expected, and in most cases 5 dB would be the best one could accomplish. Room absorption can only reduce the reverberant buildup in a room and therefore is not very effective for the worker if he must be close to the noise source. Room absorption is most effective where (a) the room has little or no sound absorbing material in it before treatment, (b) the room has multiple noise sources (4 or more), each producing about an equal amount of noise, and (c) the noise of any one machine alone does not exceed the criteria. If these conditions exist and room absorption appears to be the most attractive means of noise control, the reduction can be estimated as follows:

$$\text{Noise Reduction in dB} = 10 \log_{10} \frac{A_2}{A_1}$$

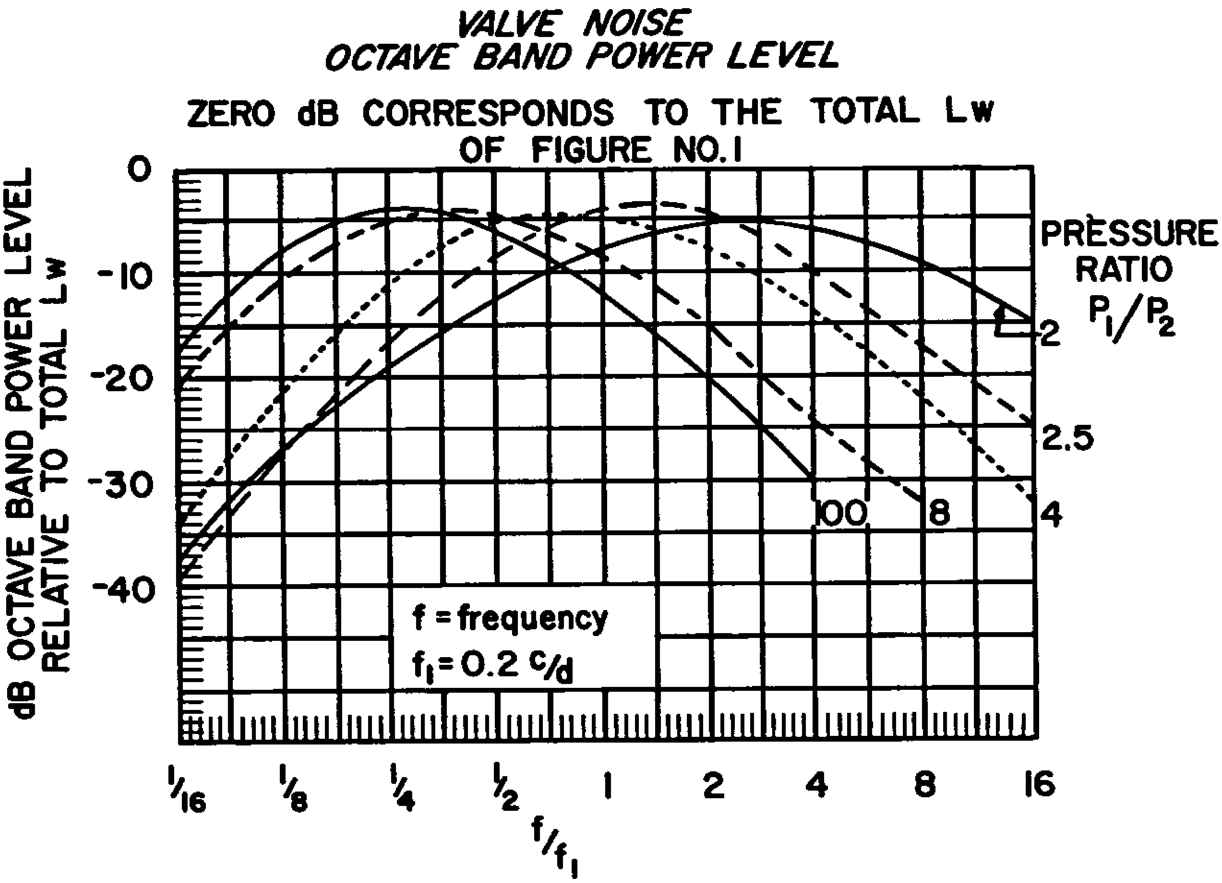
where A_1 = total number of absorption units (sabins) in the room before treatment

A_2 = total number of absorption units (sabins) in the room after treatment.



Ingersoll Rand, Allentown, Pennsylvania.

Figure 37-25. Valve Noise (Choked Values).



Ingersoll Rand, Allentown, Pennsylvania.

Figure 37-26. Valve Noise (Octave Band Power Level).

A sabin is a measure of the sound absorption of a surface and is the equivalent of one sq. ft. of a perfect absorptive surface.

This formula is presented in monograph form in Figure 37-30. The total number of absorption units mentioned above is the sum of all the room surface areas multiplied by their respective absorption coefficients. Absorption due to other materials and people should also be included in the calculation.

Absorption along Transmission Path

The most common example of noise absorption along the transmission path is the commercial muffler. Figure 37-31 illustrates the three most common types. The most common one shown by the upper sketch is the straight through lined duct type. It is very effective provided the lining is effective for the frequency of the sound involved, the length is adequate and the ID does not exceed 6″. Th performance of such a muffler can be estimated as follows:

Noise Reduction in dB per foot of length $= 12.6 \frac{P^{1.4}}{A}$

where α = absorption coefficient of the lining material at the frequency of interest

P = perimeter of the duct in inches

A = cross sectional area of the duct in sq. inches

To simplify the use of this formula, Table 37-4 shows the value of $12.6\alpha^{1.4}$ for various absorption coefficients. If the ID must be greater than 6″ or if the noise reduction required makes it necessary to use a longer muffler than desirable, then the configuration shown by the center sketch of Figure 37-31 can be used. Here the absorptive center portion makes it possible to have relatively narrow flow passages through the mufflers. This provides good performance even where muffler length must be minimized.

Where mufflers having absorption linings cannot be used, the combination resonant, reactive, and dispersive type mufflers shown by the bottom sketch of Figure 37-31 can sometimes be used. The performance of these mufflers is strongly dependent on gas flow through them, and prediction of performance in a given application is very difficult. It is recommended that this type mufflers be bought on a performance guarantee.

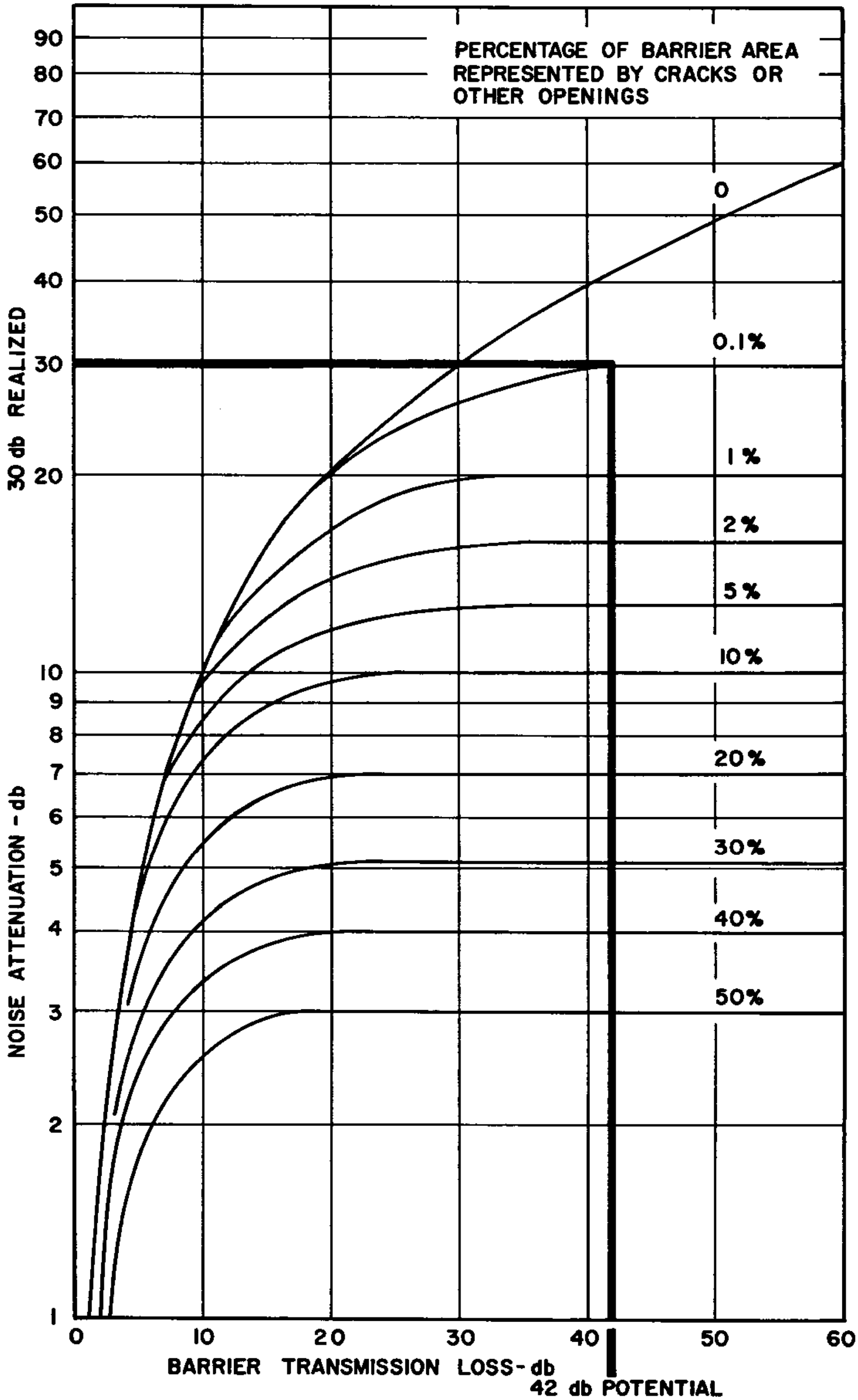
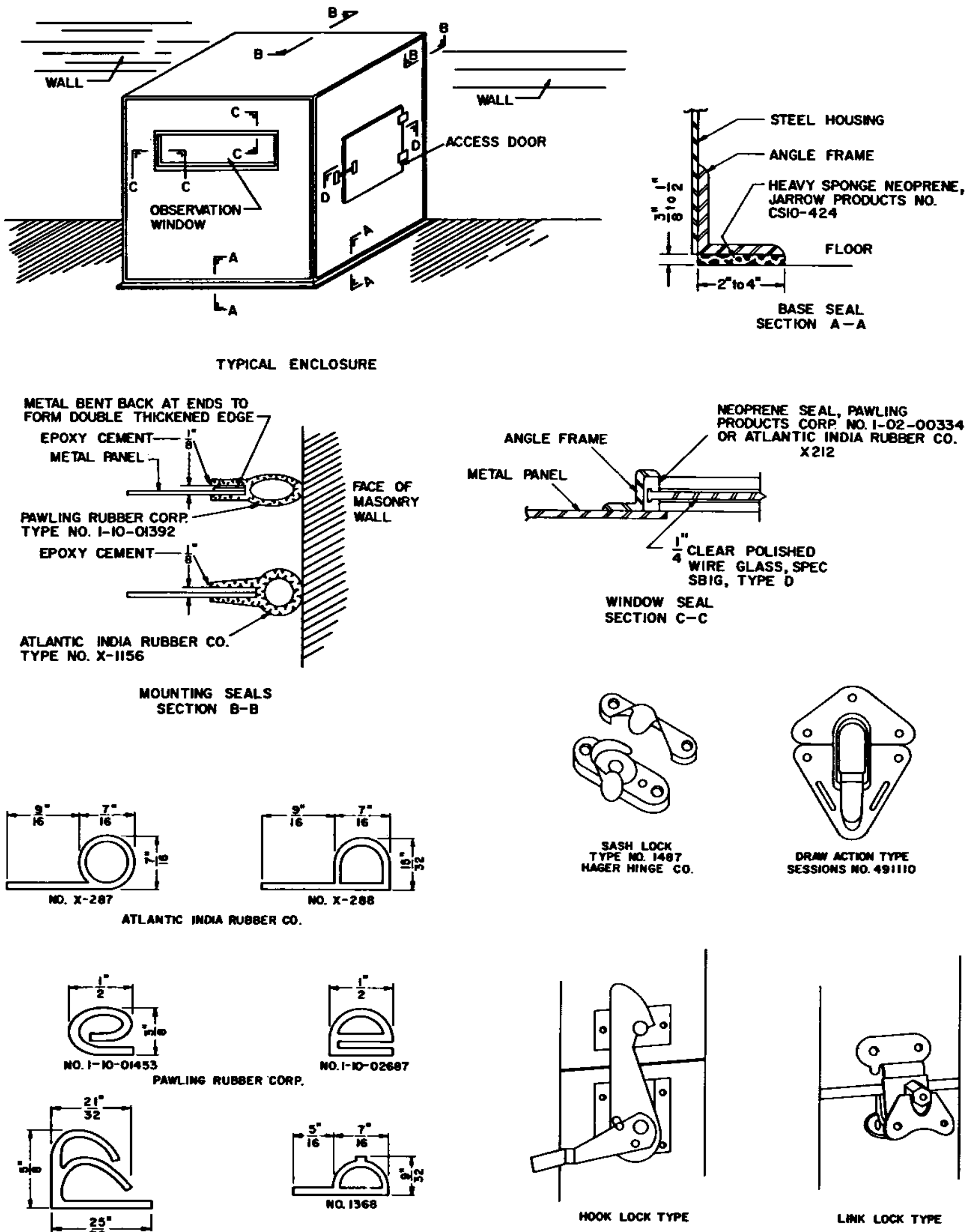
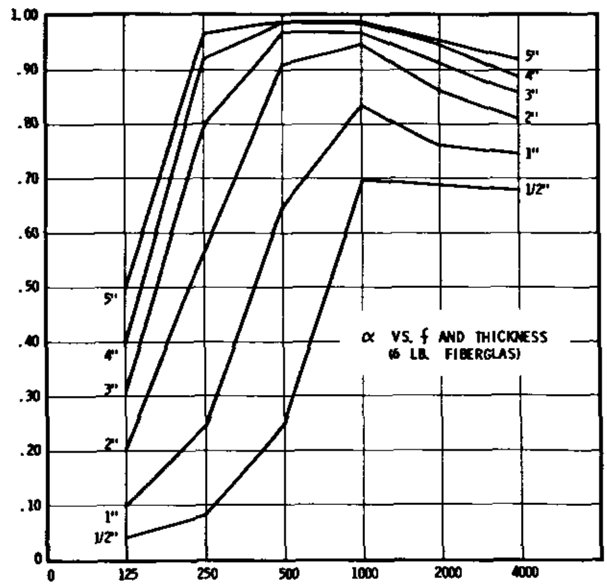


Figure 37-27. Effect of Leaks.



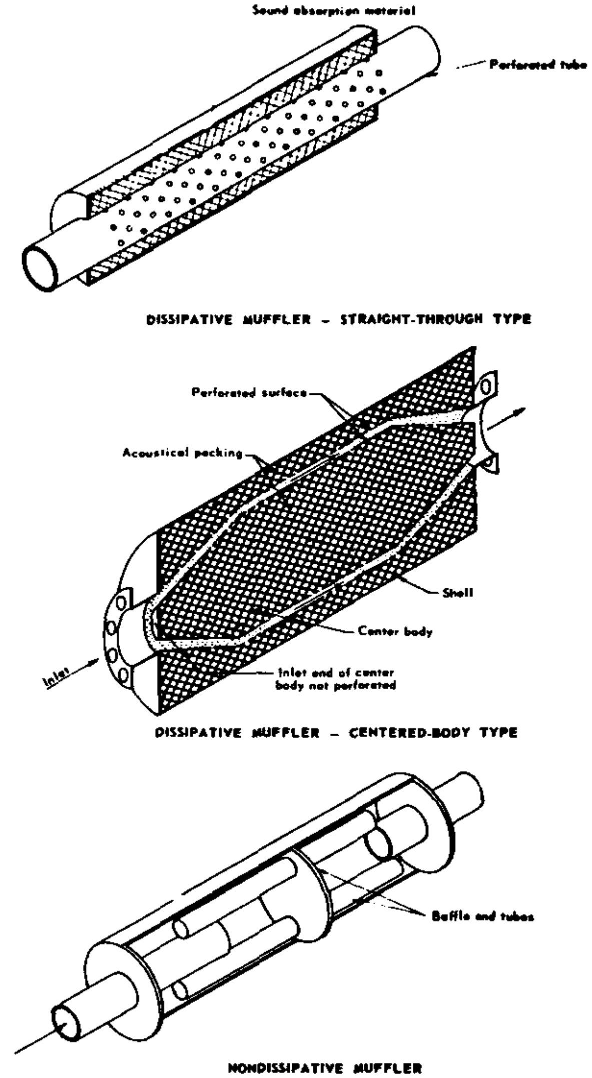
duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-28. TYPICAL ACCESS DOOR SEALS AND FASTENERS. Note: An enclosure built to house a noise source must have all cracks and openings tightly sealed in order to reduce leakage of noise. The enclosure below illustrates applicable principles of sealing and fastening base, wall, door, and observation window. It should not be considered a complete enclosure design because of possible need for ventilation, accoustical lining, etc.



Owens-Corning Fiberglas, Toledo, Ohio.

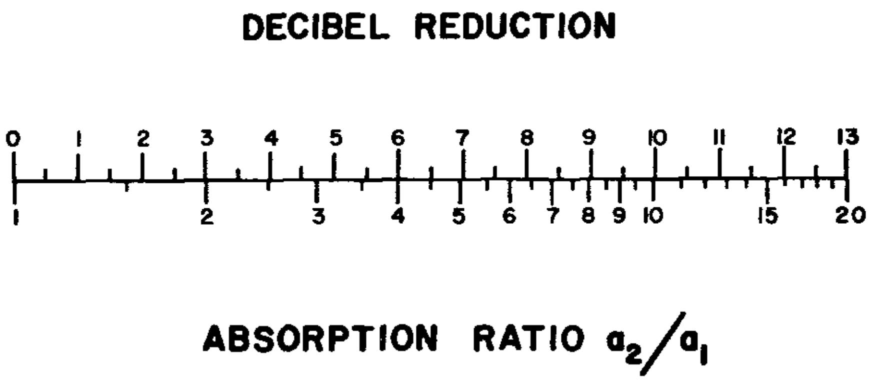
Figure 37-29. Effect of Material Thickness and Frequency on Absorption.



duPont de Nemours & Co., Wilmington, Delaware.

Figure 37-31. Mufflers.

MONOGRAPH FOR DETERMINING EFFECT OF ROOM ABSORPTION



AIHA Noise Committee: Industrial Noise Manual, 2nd Edition. Detroit, American Industrial Hygiene Association, 1966.

Figure 37-30. Monograph for Determining Effect of Room Absorption.

TABLE 37-1
OCTAVE BAND SOUND PRESSURE LEVEL DETERMINATION

Octave Band Number	f/f_1	P_1/P_2	Overall L_w	Relation of Overall to Octave Band L_w	Octave Band L_w	L_p in 3″ pipe
8	4	2.1	127 dB	−6	121	134
7	2	"	"	−5	122	135
6	1	"	"	−7	120	113
5	½	"	"	−11	116	129
4	¼	"	"	−19	108	121
3	⅛	"	"	−27	100	113
2	1/16	"	"	−37	90	103

TABLE 37-2
Sound Transmission Loss of General Building Materials and Structures

Item	Material or Structure	Thickness Inches	Wt. lb/sq ft	125	175	250	350	500	700	1000	2000	4000
A	Doors											
1	Heavy wooden door—special hardware — rubber gasket at top, sides and bottom	2½	12.5	30	30	30	29	24	25	26	37	36
2	Steel clad door — well sealed at door casing and threshold			42	47	51	48	48	45	46	48	45
3	Flush — hollow core — well sealed at door casing and threshold			14	21	27	24	25	25	26	29	31
4	Solid oak — with cracks as ordinarily hung	1¾		12		15		20		22	16	
5	Wood door (30″×84″) special soundproof constr. — well sealed at door casing and threshold.	3	7	31	27	32	30	33	31	29	37	41
B	Glass											
1		⅛	1½	27	29	30	31	33	34	34	34	42
2		¼	3	27	29	31	32	33	34	34	34	42
3		½	6	17	20	22	23	24	27	29	34	24
4		1	12	27	31	32	33	35	36	32	37	44
C	Walls — Homogeneous											
1	Steel sheet — fluted — 18 gage stiffened at edges by 2×4 wood strips — joints sealed		4.4	30	20	20	21	22	17	30	29	31
2	Asbestos board — corrugated stiffened horizontally by 2×8 in. wood beam—joints sealed		7.0	33	29	31	34	33	33	33	42	39
3	Sheet steel — 30 ga	.012	½	3	6		11		16		21	26
4	Sheet steel — 16 ga	.0598	2½	13	18		23		28		33	38

TABLE 37-2 (Cont'd.)

Sound Transmission Loss of General Building Materials and Structures

Item	Material or Structure	Thick-ness Inches	Wt. lb/sq ft	125	175	250	350	500	700	1000	2000	4000
5	Sheet steel — 10 ga	.1345	5.625	18	23		28		33		38	43
6	Sheet steel	¼	10	23	28	38	33	41	38	46	43	48
7	Sheet steel	⅜	15	26	31	39	36	42	41	47	41	51
8	Sheet steel	½	20	28	33		38		43		48	53
9	Sheet Aluminum — 16 ga	.051	.734	5	8		13		18		23	28
10	Sheet Aluminum — 10 ga	.102	1.47	8	14		19		24		29	34
11	Plywood	¼	.73		20		19		24		27	22
12	Plywood	½	1.5	8	14		19		24		29	34
13	Plywood	¾	2.25	12	17		22		27		32	37
14	Sheet Lead	1/16	3.9			32		33		32	32	32
15	Sheet Lead	⅛	8.2			31		27		37	44	33
16	Glass fiber board — 6 lb/cu ft	1	½	5	5	5	5	5	4	4	4	3
17	Laminated Glass Fiber (FRP)	⅜				26		31		38	37	38
D	Walls — Nonhomogeneous											
1	Gypsum wallboard—two ½″ sheets cemented together — joints wood battened	1	4.5	24	25	29	32	31	33	32	30	34
2	Gypsum wallboard—four ½″ sheets cemented together — fastened together with sheet metal screws — dovetail-type joints paper taped	2	8.9	28	35	32	37	34	36	40	38	49
3	¼″ plywood glued to both sides of 1 × 3 studs 16 in. O.C.	3	2.5	16	16	18	20	26	27	28	37	33
4	Same as 3 above but ½″ gypsum wallboard nailed to each face	4	6.6	26	34	33	40	39	44	46	50	50
5	¼″ Dense fiberboard on both sides of 2 × 4 wood studs 16 in. O.C. — Fiberboard joints at studs	4½	3.8	16	19	22	32	28	33	38	50	52
6	Soft type fiberboard (¾″) on both sides of 2 × 4 wood studs 16 in. O.C. — Fiberboard joints at studs	5	4.3	21	18	21	27	31	32	38	49	53
7	½″ gypsum wallboard on both sides of 2 × 4 wood studs 16 in O.C.	4½	5.9	20	22	27	35	37	39	43	48	43
8	Two ⅜″ gypsum wallboard sheets glued together and applied to each side of 2 × 4 wood studs 16 in. O.C.	5	8.2	27	24	31	35	40	42	46	53	48
9	2″ glass fiber (3 lb/cu ft) + lead — vinyl composite (0.87 lb/sq ft)					4		4		13	26	31

TABLE 37-2 (Cont'd.)

Sound Transmission Loss of General Building Materials and Structures

Item	Material or Structure	Thickness Inches	Wt. lb/sq ft	125	175	250	350	500	700	1000	2000	4000
10	⅜″ steel + 2.375 in. polyurethane foam (2 lb/cu ft) + 1/16 steel					38		52		55	64	77
11	Same as 10 above but 2½″ glass fiber (3 lb/cu ft) instead of foam					37		51		56	65	76
12	¼″ steel + 1″ polyurethane foam (2 lb/cu ft) + 0.055 in. lead vinyl composite (1.0 lb lb/sq ft)					38		45		57	56	67
E	Masonry											
1	Reinforced concrete	4	53	37	33	36	44	45	50	52	60	67
2	Brick — common	12	121	45	49	44	52	53	54	59	60	61
3	Glass Brick—3¾ × 4⅞ × 8	3¾		30	36	35	39	40	45	49	49	43

TABLE 37-2

Sound Transmission Loss of General Building Materials and Structures

Item	Material or Structure		Weight Lbs/Ft²	Loss in Decibels at Indicated Frequencies, H_z 128	192	256	384	512	768	1024	2048	4096
E	Masonry											
4	Concrete block — 4″ hollow, no surface treatment	Cinder Aggregate		27	29	32	35	37	42	45	46	48
5	Concrete block — 4″ hollow, one coat resin-emulsion paint	Cinder Aggregate		30	33	34	36	41	45	50	55	53
6	Concrete block — 4″ hollow, one coat cement base paint	Cinder Aggregate		37	40	43	45	46	49	54	56	55
7	Concrete block — 6″ hollow, no surface treatment	Cinder Aggregate		28	34	36	41	45	48	51	52	47
8	Concrete block — 8″ hollow, no surface treatment	Cinder Aggregate		18	24	28	34	37	39	40	42	40
9	Concrete block — 8″ hollow, one coat cement base paint	Cinder Aggregate		30	36	40	44	46	48	51	50	41
10	Concrete block — 8″ hollow, filled with vermiculite insulators	Cinder Aggregate		20	29	33	36	38	38	40	45	47
11	Concrete block — 4″ hollow, no surface treatment	Expanded Shale Aggregate		21	26	28	31	35	38	41	44	43
12	Concrete block — 4″ hollow, one coat resin-emulsion paint	Expanded Shale Aggregate		26	30	32	34	37	42	43	46	44
13	Concrete block — 4″ hollow, two coats resin-emulsion paint	Expanded Shale Aggregate		24	31	33	35	38	42	44	47	44
14	Concrete block — 4″ hollow, one coat cement base paint	Expanded Shale Aggregate		23	30	35	38	42	43	44	48	43
15	Concrete block — 4″ hollow, two coats cement-base paint	Expanded Shale Aggregate		34	38	40	42	45	47	49	51	46
16	Concrete block — 6″ hollow, no surface treatment	Expanded Shale Aggregate		22	27	32	36	40	43	46	45	43

TABLE 37-2 (Continued)
Sound Transmission Loss of General Building Materials and Structures

Item	Material or Structure	Weight Lbs/Ft²	Loss in Decibels at Indicated Frequencies, Hz								
			128	192	256	284	512	768	1024	2048	4096
17	Concrete block — 4″ hollow, no surface treatment		30	36	39	41	43	44	47	54	50
18	Concrete block — 4″ hollow, one coat cement base paint on face		30	36	39	41	43	44	47	54	49
19	Concrete block — 6″ hollow, no surface treatment		37	46	50	50	50	53	56	56	46
20	Concrete block — 6″ hollow, one coat resin-emulsion paint each face	Dense Aggregate	37	50	54	52	53	55	57	56	46
21	Concrete block — 8″ hollow, no surface treatment		40	47	53	54	54	56	58	58	50
22	Concrete block — 8″ hollow, two coats resin-emulsion paint each										

TABLE 37-3

Sound Absorption Coefficients of Materials

The absorption coefficient (α) of a surface which is exposed to a sound field is the ratio of the sound energy absorbed by the surface to the sound energy incident upon the surface. For instance, if 55% of the incident sound energy is absorbed when it strikes the surface of a material, the α of that material would be 0.55. Since the α of a material varies according to many factors, such as frequency of the noise, density, type of mounting, surface condition, etc., be sure to use the α for the exact conditions to be used and from performance data listings such as shown below. For a more comprehensive list of the absorption coefficients of acoustical materials, refer to the bulletin published yearly by the Acoustical Materials Association, 335 East 45th St., New York, N. Y. 10017.

Materials / Frequency	Coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Brick — glazed	0.01	0.01	0.01	0.01	0.02	0.02
Sand — dry — 4″ thick	.15	.35	.40	.50	.55	.80
Sand — dry — 12″ thick	.20	.30	.40	.50	.60	.75
Sand — wet — 14 lb water per cu ft 4″ thick	.05	.05	.05	.05	.05	.15
Water	.01	.01	.01	.01	.02	.02
Glass Fiber — mounted with impervious backing — 3 lb/cu ft, 1″ thick	.14	.55	.67	.97	.90	.85
Glass Fiber — mounted with impervious backing — 3 lb/cu ft, 2″ thick	.39	.78	.94	.96	.85	.84
Glass Fiber — mounted with impervious backing — 3 lb/cu ft, 3″ thick	.43	.91	.99	.98	.95	.93
Steel (Estimated)	.02	.02	.02	.02	.02	.02
Brick, unglazed	.03	.03	.03	.01	.05	.07
Brick, unglazed, painted	.01	.01	.02	.02	.02	.03
Carpet, heavy, on concrete	.02	.06	.14	.37	.60	.65
Same, on 40 oz hairfelt or foam rubber (carpet has porous backing)	.08	.24	.57	.69	.71	.73
Same, with impermeable latex backing on 10 oz hairfelt or foam rubber	.08	.27	.39	.34	.48	.63
Concrete Block, coarse	.36	.44	.31	.29	.39	.25
Concrete Block, painted	.10	.05	.06	.07	.09	.08
Concrete, poured	.01	.01	.02	.02	.02	.03

TABLE 37-3 (Continued)

Sound Absorption Coefficients of Materials

Materials / Frequency	Coefficients 125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Fabrics						
Light velour, 10 oz per sq yd, hung straight, in contact with wall	.03	.04	.11	.17	.24	.35
Medium velour, 14 oz per sq yd, draped to half area	.07	.31	.49	.75	.70	.60
Heavy velour, 18 oz per sq yd, draped to half area	.14	.35	.55	.72	.70	.65
Floors						
Concrete or terrazzo	.01	.01	.015	.02	.02	.02
Linoleum, asphalt, rubber or cork tile on concrete	.02	.03	.03	.03	.03	.02
Wood	.15	.11	.10	.07	.06	.07
Wood parquet in asphalt on concrete	.04	.04	.07	.06	.06	.07
Glass						
Large panes of heavy plate glass	.18	.06	.04	.03	.02	.02
Ordinary window glass	.35	.25	.18	.12	.07	.04
Gypsum Board, ½″ nailed to 2×4's 16″ o.c.	.29	.10	.05	.04	.07	.09
Marble	.01	.01	.01	.01	.02	.02
Openings						
Stage, depending on furnishings			.25 — .75			
Deep balcony, upholstered seats			.50 — 1.00			
Grills, ventilating			.15 — .50			
To outside			1.00			
Plaster, gypsum or lime, smooth finish on tile or brick	.013	.015	.02	.03	.04	.05
Plaster, gypsum or lime, rough finish on lath	.14	.10	.06	.05	.04	.03
Same, with smooth finish	.14	.10	.06	.04	.04	.03
Plywood Paneling, ⅜″ thick	.28	.22	.17	.09	.10	.11
Water Surface, as in a swimming pool	.008	.008	.013	.015	.020	.025

ABSORPTION OF SEATS AND AUDIENCE

Values given are in Sabins per square foot of seating area or per unit

Materials / Frequency	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Audience, seated in upholstered seats, per sq ft of floor area	.60	.74	.88	.96	.93	.85
Unoccupied cloth-covered upholstered seats, per sq ft of floor area	.49	.66	.80	.88	.82	.79
Unoccupied leather-covered upholstered seats, per sq ft of floor area	.44	.54	.60	.62	.58	.79
Chairs, metal or wood seats, each, unoccupied	.15	.19	.22	.39	.38	.39

TABLE 37-4

SOUND ABSORPTION COEFFICIENT VS. $12.6\alpha^{1.4}$

Sound Absorption Coefficient	$12.6\alpha^{1.4}$
0.50	4.78
0.55	5.46
0.60	6.16
0.65	6.89
0.70	7.65
0.75	8.43
0.80	9.16
0.85	10.02
0.90	10.87

Preferred Reading — Books

1. BERANEK, L. L. — *Noise and Vibration Control,* McGraw-Hill Book Co., New York, 1971.
2. HARRIS, C. M. — *Handbook of Noise Control,* McGraw-Hill Book Co., New York, 1957.
3. HARRIS, C. M. and CREDE, C. E. — *Shock and Vibration Handbook,* Volume 2, McGraw-Hill Book Co., New York, 1961.

Preferred Reading — Periodicals

Sound and Vibration — Published monthly by Acoustical Publications, Inc., 2710 E. Oviatt Road, Bay Village, Ohio 44140.

Journal of Sound and Vibration — Published bi-monthly by Academic Press, Inc. (London) Limited.