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CHAPTER I

INTRODUCTION TO ACOUSTICS

In this discussion of noise and vibration, it is intended to move quickly into the use of acoustical terms and to become acquainted with some of the elementary acoustical procedures without necessarily knowing or comprehending all the acoustics background that goes into the development of this material. Textbooks or reference books in acoustics* may be studied for a more detailed discussion and technical understanding of this material.

1. DECIBELS

Just as "inches" are used to measure distance and "degrees" are used to measure temperature, "decibels" are used to measure sound intensity. As in electrical engineering, decibels are used to express in logarithmic terms the ratio of two powers; i.e., if there are two electrical or acoustical powers P_1 and P_2, the ratio of those powers expressed in decibels would be

$$10 \log P_2/P_1 \quad .$$

If the power P_1 were some accepted standard reference power, such as a watt or some other basic unit of power, the decibels could be standardized to that reference value.

In acoustics, the decibel (abbreviated "dB") is used to compare both sound power and sound pressure. When describing the sound power of a sound source, the basic reference power is 10^{-12} watt (acoustic watts), and a particular sound source might be described as having a "sound power level" of, for example, 110 dB re 10^{-12} watt. When describing the sound pressure in a sound field, the basic reference pressure is 0.0002 microbar, and a particular area might be stated as having a "sound pressure level" of, say, 90 dB re 0.0002 microbar.

A microbar is equal to one dyne per sq cm or 0.1 newton per sq meter and is very nearly equal to one millionth of a standard atmosphere. It is likely that in a few years the reference pressure 0.0002 microbar will come to be known as $2\text{x}10^{-5}$ newton per sq meter. If this comes, it will be in the interest of international standardization of terminology and units.

In acoustics, the term "level" is used whenever a decibel quantity is expressed relative to a reference value, as in "sound pressure level" (referred to the reference pressure of 0.0002 microbar) and "sound power level" (referred to the reference power of 10^{-12} watt).

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*"Acoustics", Leo L. Beranek; McGraw-Hill Book Company (1954).

"Noise Reduction", Leo L. Beranek, Editor; McGraw-Hill Book Company (1960).

"NOISE AND VIBRATION ENGINEERING", Leo L. Beranek, Editor; McGraw-Hill Book Company (Planned for 1970 or 1971 Publication).

2. SOUND PRESSURE LEVEL

The ear is sensitive to sound pressure. Sound waves represent tiny oscillations of pressure just above and just below atmospheric pressure. These pressure oscillations impinge on the ear and "we hear the sound".

A "sound level meter" is also sensitive to sound pressure. When a sound level meter is properly calibrated, it relates the sound pressure of an incident sound wave to the standard reference pressure (0.0002 microbar) and it gives a reading in decibels relative to that reference pressure. "0 dB" on this scale means 0 dB above the reference pressure, which, of course, is the same as the reference pressure. This reference pressure, or 0 dB sound pressure level, represents approximately the weakest sound that can be heard by the average young human ear in the frequency region of highest sensitivity. (This definition came into existence when people were known to have maximum hearing sensitivity at youth; the truly damaging effects of too-loud music will change this for some of today's young people.)

A simple but expressive definition of "noise" is that it is "unwanted sound"; so "noise level" is often used synonymously with "sound pressure level". Both terms have the same reference pressure and are used interchangeably in this manual. The reference to 0.0002 microbar may be and frequently is omitted when it is clearly understood that the dB quantity is a sound pressure level. Hence one might say that "the noise level in the mechanical equipment room is about 85 dB".

Two abbreviations of the term "sound pressure level" are in fairly common use: "SPL" and "Lp". "SPL" is used in much of the literature and "Lp" is being used in some of the more recent literature. The abbreviation "SPL" is used throughout this manual, but it is completely interchangeable with "Lp" as found elsewhere.

3. ANALOGY BETWEEN LIGHT AND SOUND

Sound pressure and sound power can be illustrated simply with an analogy between light and sound. Suppose first that a room is illuminated with a bare 15-watt electric lamp. Even in a room with white painted walls and ceiling, this normally would be considered as a weak light source. If the room had only dark, unreflecting surfaces, the general room illumination would be very poor. Now a bare 150-watt lamp would give good general illumination if the walls are white, or light-colored, or highly reflecting (and depending, of course, on the size of the room and the distance to the lamp). However, the same 150-watt lamp might not give adequate room illumination if the walls and ceiling were black, or dark-colored or non-reflective. Thus, it is reasonably obvious that the intensity of the general room illumination depends not only on the power rating of the lamp, but also on the light-reflecting (or absorbing) properties of the room surfaces, on the size of the room, and on the distance to the light source. Further, if the lamp had a lamp-shade or if it were recessed in a flush-mounted ceiling receptacle, the light would be brighter in some directions than in others.

All the same factors apply to sound in a room! "Sound pressure level" is somewhat analogous to room illumination; "sound power level" is somewhat analogous to the power rating of the lamp. A "weak" sound source would produce low sound levels while a "stronger" sound source would produce higher sound levels. A constant sound source that would produce one sound level in a hard-walled bare room would produce a different sound level in the same room surfaced with a large amount of soft, fluffy acoustic absorption material.

The sound source would produce a higher sound level a few inches away than it would several feet away. It might radiate higher sound levels from one side than from another side. It would produce different sound levels in a large room than it would in a small room. Thus, the sound level in the room depends not only on the sound source (actually its "sound power"), but also on the sound absorption properties of the room surfaces, on the size of the room, the distance to the sound source, and also the directional characteristics of the sound source. In effect, the sound pressure levels heard by a person in the room are determined both by the sound power radiated by the source and by the "acoustic characteristics" of the room. All of this is merely leading up to the fact that (1) there is need for a way of rating a sound source that is independent of the environmental surroundings, and (2) there is need for a way of describing the "acoustic characteristics" of the room that is independent of the sound source. Then, with these two independently determined bits of information, any known, definable room or space and the sound field or "sound pressure level" ("SPL") about the room can be determined, remembering that it is the sound pressure level to which people respond in their living and working environments. Just as the 150-watt lamp may produce relatively poor to good illumination in a given room, so also will a sound source produce relatively low or high sound pressure levels in a given room. Further, just as electric lamps are rated by a power rating, so also sound sources are rated by a power rating.

4. SOUND POWER LEVEL

The quantity "sound power level" expresses, in decibels relative to the reference power of 10^{-12} watt, the total amount of sound power radiated by a sound source, regardless of the space into which the source is placed. As suggested above, if the power level of a sound source is known and if the "acoustic characteristics" of a space are known, it will then be possible to estimate or calculate the sound pressure level in that space. Ultimately it is the SPL that usually must be determined because it is on that basis that people judge an acoustic environment.

Two abbreviations of "sound power level" are in common use: "PWL" and "L_W". "PWL" is used throughout this manual, but it is completely interchangeable with "L_W".

The need for sound power level data has grown rapidly in recent years. Consider ventilation system diffusers as an example. In earlier years, one manufacturer might have published sound pressure levels for his diffusers measured at some named distance in his highly reverberant test room. Another manufacturer might have published sound <u>pressure</u> levels of his diffusers at another distance in his test room (undoubtedly of different size and acoustic characteristics than that of his competitor). Still another manufacturer might have published noise data that he measured in a particular mock-up of a room that was intended to represent a typical office of a large building. Well, each manufacturer might have felt justified in his procedure, but the same identical diffuser in all those different test conditions could have yielded variations as high as 5 to 10 dB in sound pressure level. To provide a more realistic rating of <u>diffusers</u> (and other noise sources as well) rather than the <u>test rooms</u> in which they were measured, the importance of sound <u>power</u> level has come to be realized as a true indicator of the quantity of noise radiated by a source regardless of the surroundings. This fact has been recognized by most manufacturers of equipment to be rated or selected in terms of noise output; and they are in the process of obtaining and providing sound <u>power</u> level data for their equipment. The test facilities are quite expensive and the tests are not always simple, but steps are being taken to provide PWL data to the designer and customer.

5. SOUND POWER REFERENCE 10^{-12} WATT

The reference power for sound power level data in present U.S. and international usage is 10^{-12} watt as stated above. This reference should <u>always</u> be quoted (as in "110 dB re 10^{-12} watt") so as not to be confused with SPL data or with earlier PWL data that used 10^{-13} watt as the reference power.

Most of the U.S. acoustics literature before about 1963-1966 (and even some current literature) uses 10^{-13} watt as the reference power. The 10^{-12} watt value is now accepted as the U.S. and international standard. A 10 dB error can result by not using the correct reference. If it is desired to use PWL data given in "dB relative to 10^{-13} watt", reduce those numerical values of PWL by 10 dB to convert them to PWL values in "dB relative to 10^{-12} watt". Conversely, if it is desired to convert from 10^{-12} watt reference to 10^{-13} watt reference, add 10 dB to the 10^{-12} watt PWL values to get "dB re 10^{-13} watt". In this manual and in most current literature, 10^{-12} watt is the reference power for PWL data.

6. FREQUENCY, HZ AND CPS

With the recent trend in U.S. and international standards to recognize the early men of science, many new names for old units are being adopted. The traditional unit for frequency in the U.S. has been "cycles per second", abbreviated "cps". The new international unit for frequency, recently adopted by U.S. standards groups, is "Hertz", abbreviated "Hz". Throughout this manual the new unit "Hz" will be used; it has the same meaning as "cycles per second".

7. "OVERALL" FREQUENCY RANGE AND OCTAVE BANDS OF FREQUENCY

In order to represent properly the total noise of a noise source, it is usually desirable or necessary to break the total noise down into its various frequency components; that is, how much of the noise is low frequency, how much high frequency and how much is in the middle frequency range. This is essential for any comprehensive study of a noise problem for two reasons: (1) people react differently to low frequency and high frequency noise (for the same sound pressure level, high frequency noise is much more disturbing and is more capable of producing hearing loss than is the case for low frequency noise); and (2) the engineering solutions to reduce or control noise are different for low frequency and high frequency noise (low frequency noise is more difficult to control, in general).

It is conventional practice in acoustics to determine the frequency distribution of a noise by passing that noise successively through several different filters that separate the noise into 8 or 9 "octaves" on a frequency scale. Just as with an "octave" on a piano keyboard, an "octave" in sound analysis represents the frequency interval between a given frequency (such as 300 Hz) and twice that frequency (600 Hz in this illustration).

The normal frequency range of hearing for most people extends from a low frequency of about 20 Hz up to a high frequency of 10,000 to 15,000 Hz, or even higher for some people. By virtue of U.S. adoption of a recent international frequency standard in acoustics, most octave-band noise analyzing filters now cover the audio range of about 22 Hz to about 11,200 Hz in nine octave frequency bands. These filters are identified by their geometric mean frequencies; hence 1000 Hz is the label given to the octave frequency band of 700-1400 Hz. The nine octave bands of the "new" international standard are as follows (the numbers are frequently rounded off):

Octave Frequency Range (Hz)	Geometric Mean Frequency of Band (Hz)
22-44	31½
44-88	62½
88-175	125
175-350	250
350-700	500
700-1400	1000
1400-2800	2000
2800-5600	4000
5600-11,200	8000

The term "overall" designates the full frequency coverage of all the octave bands, hence 22-11,200 Hz, or in some cases, 44-11,200 Hz when the 31 Hz band is omitted.

The frequency bands in use in the U.S. before adoption of the bands listed above are as follows: 20-75, 75-150, 150-300, 300-600, 600-1200, 1200-2400, 2400-4800 and 4800-10,000 Hz. Most of the literature in acoustics before about 1963 will refer to these "old" frequency bands. The "new" international standard frequencies (sometimes called "preferred frequencies" in current literature) are used in this manual. Essentially the "old" and "new" frequency bands may be considered as being equivalent, with a few exceptions that will not be significant to the material in this manual. A set of filters used to separate a complex sound into octave bands is commonly referred to as an "octave band analyzer".

When a sound pressure level or a sound power level includes all the audio range of frequency, the resulting value is called the "overall" level. When the level refers to the sound in just one specific octave frequency band, it is called an "octave band level" and the frequency band is either stated or clearly implied.

For some special situations, a noise spectrum may be studied in finer detail than is possible with octave frequency bands. In such cases one-third octave bands might be used or even narrower filter bands might be used, for example to separate one particular frequency from another one if it is desired to separate the causes of a particular complex noise. The bandwidth and the identifying frequency of the band should always be specified.

8. WEIGHTING NETWORKS: A-, B- AND C-SCALES

Sound level meters are usually equipped with "weighting circuits" that tend to represent the frequency characteristics of the average human ear for various sound intensities. Hence, "overall" readings are sometimes taken with "A-scale" or "B-scale" or "C-scale" settings on the meter. The "A-scale" setting of a sound level meter filters out as much as 20 to 40 dB of the sound below 100 Hz, while the "B-scale" setting filters out as much as 5 to 20 dB of the sound below 100 Hz. The "C-scale" setting is reasonably "flat" with frequency, i.e. it retains essentially all the sound signal over the full "overall" frequency range. A plot of the frequency response of the electrical system of a sound level meter meeting USASI (U.S.A. Standards Institute, formerly American Standards Association) standards for the A-, B- and C-scale weighting networks is shown in Figure 1 at the end of this chapter. For several years the A-scale and B-scale readings were held in disfavor because they do not provide any knowledge of the frequency distribution of the noise, but there is a revival in the use of A-scale readings as a single-number indicator of the relative loudness of a sound as heard by the human ear. It is very important, when reading A-, B- or C-scale sound levels, to positively identify the scale setting used. The resulting values are called "sound levels" and are frequently identified as dB(A), or dB(B) or dB(C) readings. Note that these readings do not represent true "sound pressure levels" because some of the actual signal has been removed by the weighting filters.

For most acoustic applications the octave frequency band readings are the most useful. It is always possible to construct A-, B- or C-scale readings from all the octave band readings, but it is never possible to exactly construct the octave band readings from the weighting scale readings.

9. ADDITION OF DECIBELS

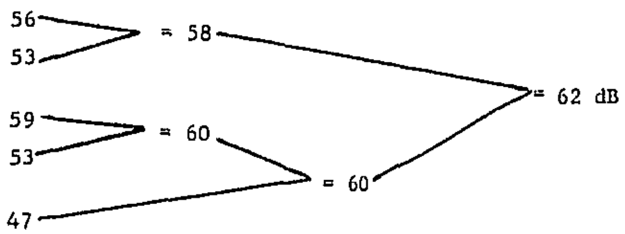
Since decibels are logarithmic values it is not proper to add them by normal algebraic addition. For example, 63 dB plus 63 dB <u>does not</u> equal 126 dB but only 66 dB.

A very simple, but adequate schedule for adding decibels is as follows:

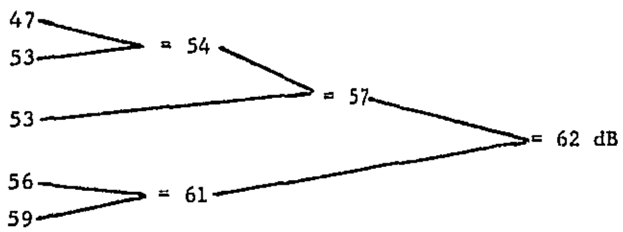
When two decibel values differ by:	Add the following amount to the higher value:
0 or 1 dB	3 dB
2 or 3 dB	2 dB
4 - 8 dB	1 dB
9 dB or more	0 dB

When several decibel values are to be added, perform the above operation on any two numbers at a time; the order does not matter. Continue the process until only a single value remains. A table repeating these rules is included in the section on noise sources.

As an illustration, add the following five noise levels:



Or, suppose the same numbers are arranged in a different order, as in



Sometimes, using different orders or adding may yield sums that might differ by 1 dB, but this is not too significant a difference in acoustics. In general, the above simplified summation procedure will yield accurate sums to the nearest 1 dB. This degree of accuracy is considered acceptable for the material given in these notes.

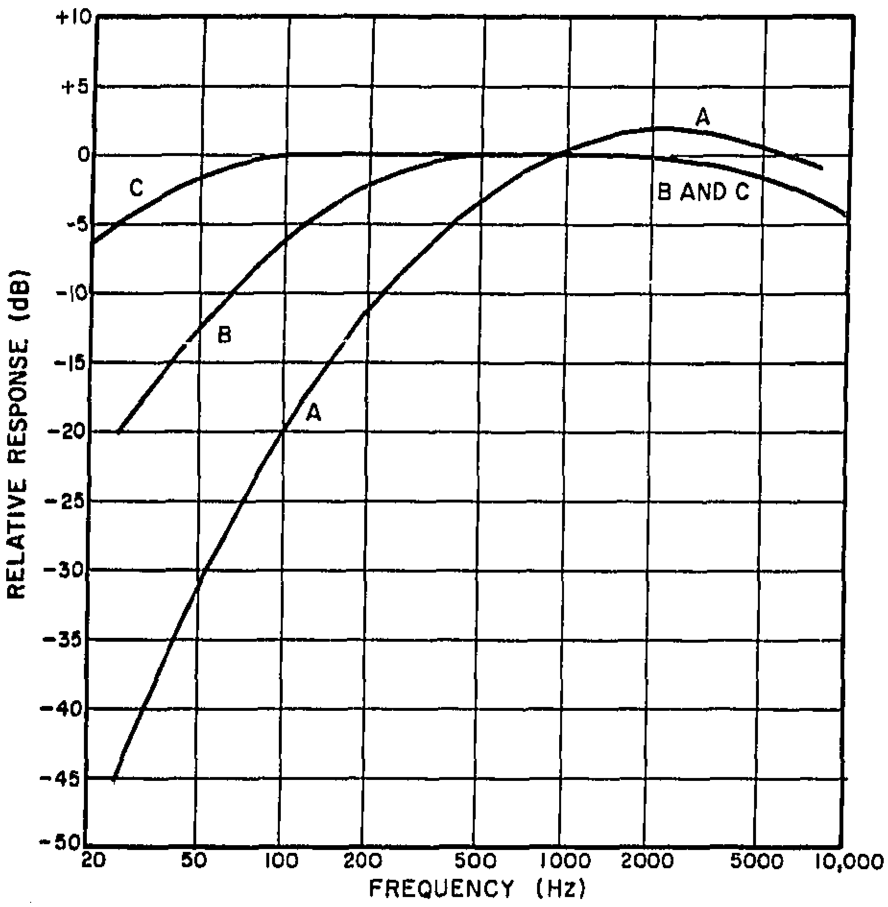


FIG. 1 APPROXIMATE ELECTRICAL FREQUENCY RESPONSE OF THE A-, B-, AND C-SCALE WEIGHTING NETWORKS OF USASI-APPROVED SOUND LEVEL METERS. (Taken from General Radio Company "Handbook of Noise Measurement")

CHAPTER 2

NOISE CRITERIA

The degree of disturbance or annoyance of an intruding unwanted noise depends essentially on three things: (1) the amount and nature of the intruding noise, (2) the amount of background noise already present before the intruding noise occurred and (3) the nature of the working or living activity of the people occupying the area in which the noise is heard. People trying to sleep in their quiet suburban homes would not tolerate very much intruding noise; while office workers in a busy mid-city office could have greater amounts of noise without even noticing it; and factory workers in a continuously noisy manufacturing space might not even hear a noisy nearby equipment installation.

It is common practice in acoustical engineering to rate various environments by "noise criteria" and to describe these criteria by fairly specific noise level values. Detailed discussions of noise criteria can be found in other literature*, and only a brief useful summary of that material is introduced here. In the interest of brevity, many important details and qualifications are omitted. Thus, in a complex problem, additional reading or acoustical assistance may be necessary.

1. NOISE CRITERION CURVES

From earlier studies of many types of noise environments that people have found either "acceptable" or "unacceptable" for various <u>indoor</u> working or living activities, a family of "Noise Criterion Curves" ("NC" curves) has been evolved. Figure 1 presents these curves. Each curve represents a reasonably acceptable balance of low frequency to high frequency noises for particular situations. These curves are also keyed-in to the "speech communication" conditions permitted by the noise. Thus, the lower NC curves prescribe noise levels that are quiet enough for resting and sleeping or for excellent listening conditions, while the upper NC curves describe rather noisy work areas where even speech communication becomes difficult and restricted. The curves within this total range may be used to set desired noise level goals for almost all typical <u>indoor</u> functional areas where some acoustic need must be served. For convenience in using the NC curves, the octave band sound pressure levels of Figure 1 are enumerated in Table 1.

In Table 2, a number of typical indoor living, working and listening spaces are grouped together into "categories" and each category is assigned a

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*For a quantitative discussion of noise criteria and noise levels, refer to a textbook or reference book on acoustics, such as "Noise Reduction", Leo L. Beranek, Editor; McGraw-Hill Book Company (1960) or "Handbook of Noise Control", C. M. Harris, Editor; McGraw-Hill Book Company (1954), or to the latest issue of the ASHRAE "Guide and Data Book", American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 346 East 47th Street, New York 10017 or to selected topics of the Journal of the Acoustical Society of America representative range of noise criterion values. Low Category Numbers indicate areas in which relatively low noise levels are desired; higher Category Numbers indicate areas in which relatively higher noise levels are permissible. Any occupied or habitable area not specifically named in Table 2 can be added under any appropriate Category Number as long as the acoustic requirements of the new area are reasonably similar to those of the areas already named under that category. A 5-10 dB range of NC values is given in Table 2 for each of the first five categories. In general, the lower limit of each range should be used for the more critical spaces or the more sensitive or criticaloccupants of an area, while the upper limit of each range may be used for the less critical spaces or occupants of an area. An exception to this generalization may occur when it is clearly known that the background noise of an area is so quiet and the walls between adjoining rooms have such low "transmission loss" that speech sounds or other clearly identifiable sounds may intrude from one office to another and be disturbing to occupants of either area. In this type of situation, "masking noise" may have to be introduced into the rooms in order to reduce some of the intelligibility of the intruding sounds, and the higher range of noise criterion values may actually be useful, as long as the mechanical equipment noise itself is relatively unobtrusive and not too identifiable. When properly controlled as to spectrum shape and sound level, ventilation system noise (the gentle "hissing" of diffusers, under-window induction units, dampers or air valves) sometimes provides some of this "masking noise". In more critical cases, where spectrum and level must be held under close control, electronic noise sources may be used.

A special note of concern is given for the Category 1 and 2 areas of Table 2. For a very quiet community area or for a quiet building with no internal ventilation system noise, the NC-20 noise criterion should be applied for indoor conditions. For a noisy city environment outdoors or for a building with a ventilation system known to fall in the NC-30 noise range, an NC-30 noise criterion can be applied to rooms other than bedrooms or auditoriums. For bedrooms or auditoriums or for situations that do not clearly fall at the NC-20 lower limit or NC-30 upper limit, NC-25 indoor noise criterion levels should be applied.

The reader may refer to the most recent issue of the ASHRAE Guide and Data Book for a listing of other typical situations and the associated range of NC values. The ASHRAE Guide usually lists a 10 dB range of NC values for each space, leaving it to the option of the user to select the specific NC value for his own particular situation.

For music or performing arts centers or concert halls, there is increasing evidence that a complete absence of noise is required in order to provide a full appreciation of the very low level sounds sometimes coming from the stage area. Thus, an NC-15 to NC-20 criterion should be applied as the goal for high quality concert halls. Acoustical assistance may be required to achieve these goals.

It is noted here that much of the known data on criteria do not extend down to the very low frequency band of 31 Hz. Some of the noise source data, however, include 31 Hz levels. For most ordinary noise problems, there will be no serious concern for the 31 Hz band so it can be ignored for most calculations. If it is known that a serious problem involves decision-making at 31 Hz, acoustical assistance should be obtained.

2. SPEECH INTERFERENCE LEVELS

A reasonably steady broad-band noise with moderate to high noise levels in the frequency bands of 500 to 2000 Hz will produce some degree of interference with speech, since most of the intelligibility of the human voice falls in this frequency range. The term "speech interference level" of a noise is now defined as the arithmetic average of the sound pressure levels of the noise in the three octave bands centered at 500, 1000 and 2000 Hz.* Table 3 gives the average "speech interference level" of a noise that will just barely permit reliable speech communication for a range of voice levels and distances. The data are based on tests performed out-of-doors where there are no reflecting surfaces to help reinforce the speech sounds, but the values can be used as approximations for indoor conditions as well. Also, to a first approximation (but not exactly), if a noise follows the shape of an NC curve, the "PSIL" value of the noise will nearly equal the NC curve number.

As a simple example of the use of Table 3, if the noise levels in a Mechanical Equipment Room average 62 dB in the 500, 1000 and 2000 Hz bands, barely reliable speech conversations could be carried on in that room by shouting at a 16-ft distance, by using a loud voice level at a distance of 8 ft, by using a raised voice at a distance of 4 ft or by using a normal voice level at a distance of 2 ft.

3. OUTDOOR BACKGROUND NOISE

People tend to compare an intruding noise with the background noise that was present before the new noise came into existence. If the new noise has distinctive sounds that make it readily identifiable or if its noise levels are considerably higher than the background or "ambient" levels, it will be noticeable to the residents and it might be considered objectionable. On the other hand, if the new noise has a rather unidentifiable, unobtrusive sound and its noise levels blend into the ambient levels, it will hardly be noticed by the neighbors and it probably will not be considered objectionable.

Thus, in trying to estimate the effect of a new noise on a neighbor, it is necessary to know or to estimate the background noise levels in the absence of the new noise. Since the equipment is probably planned for continuous day and night operation, and since people are less tolerant of an intruding noise at night, the nighttime ambient noise levels are important to the evaluation of the problem.

Where possible (and especially if a sensitive neighborhood is located nearby), the average minimum nighttime noise levels should be measured several times

*"SIL" was originally defined in terms of the three formerly-used octave bands 600-1200, 1200-2400 and 2400-4800 Hz. With the acceptance of the new international frequency bands in the U.S., an adjustment of values has been made and the new values are being identified by the notation "PSIL" in order to designate that they are based on the now "preferred" frequencies.

during several typically quiet nights. Readings should be taken in octave bands and readings should be taken when there is no nearby truck or auto traffic that would give falsely high values.

If background measurements cannot be made, the ambient noise levels can be estimated approximately with the use of Tables 4 and 5. In Table 4, the condition should be determined that most nearly describes the community or residential area or the nearby traffic activity (which frequently helps set the ambient levels in an otherwise quiet neighborhood) that would exist during the quietest time that the equipment would be in operation. For the condition that is selected, there is an appropriate "Noise Code No." at the right-hand side of Table 4 that is used to enter Table 5. For that particular Noise Code No., Table 5 then gives an estimate of the approximate average minimum background noise levels for that area and traffic condition. This is not an infallible estimate but it will serve in the absence of actual measurements.

It is cautioned that these estimates should be used only as rough approximations of background noise and that local conditions can give rise to a wide range of actual noise levels.* It is, nevertheless, realistic to utilize a method such as this to help determine the amount of noise that a new noise can make without becoming noticeably louder than the general background.

4. NOISE REDUCTION PROVIDED BY A BUILDING

An intruding noise coming from an outdoor noise source or by an outdoor noise path may be heard by a neighbor who is either indoors in his own building or outdoors on his property. If he is outdoors he may judge the intruding noise against the more-or-less steady background noise due to other noises in the area. If he is indoors, he may tend to judge the noise by whether it is audible or identifiable or intrusive into his surroundings. If the noise, when heard indoors in the neighbor building, can be made to be no greater than the appropriate NC values that would normally apply there, it is quite likely that there will be no complaint against the noise.

When outdoor noise passes into a building it suffers some noise reduction, even if the building has open windows. The actual amount of noise reduction depends on building construction, orientation, wall area, window area, open window area, interior acoustic absorption, etc. For practical purposes, however, the approximate noise reduction values provided by a few typical building constructions are given in Table 6. If these amounts of noise reduction are added to the indoor NC values, one would obtain the <u>outdoor</u> sound pressure levels that would yield the <u>indoor</u> NC values, applicable when outdoor noise passes through the building wall and comes indoors. For convenience and identification, the listed wall constructions are

*A procedure similar to this is given in the ASHRAE Guide. It is cautioned, however, that the actual curves and sound levels used in the ASHRAE Guide are not identical to those used in this manual, even though the ASHRAE material originally was developed from data first presented in an earlier Baltimore Aircoil Company Bulletin. The data presented here are recommended as being slightly more conservative and somewhat more specific than the equivalent data offered in the ASHRAE Guide.

labeled with letters A through G, and are described in the notes under Table 6. Note that Wall A represents no wall at all, hence no noise reduction; and the use of Wall A indicates that the selected NC curve would actually apply in this special case to an outdoor activity (such as for a screened-in sleeping porch, a drive-in theater, an outdoor restaurant, an outdoor terrace, and the like).

5. OUTDOOR NOISE CRITERION

From the foregoing material it is possible to estimate an approximate outdoor noise criterion for almost any type of neighbor situation. Two somewhat independent approaches should be tried, and the decision based on the results of those two approaches.

The first approach provides an "outdoor noise level criterion" that will essentially produce the desired indoor noise levels after the noise passes through the wall of the neighbor building. These outdoor noise levels are merely the arithmetic sum of the appropriate indoor noise criterion levels from Table 1 and the noise reduction values of the neighbor's building as taken from Table 6.

The second approach provides another "outdoor noise level criterion" that is essentially based on the possible "intrusion" of the new noise into the existing outdoor background noise, as determined from Tables 4 and 5. To be completely inconspicuous, the new noise, when extrapolated to the neighbor's location, should be kept at or below the outdoor background noise levels in all octave bands. (If a noticeable pure tone signal is present in the intruding noise, its octave band level must be 5-10 dB lower than the background level in that octave band in order not to be noticeable. It may be difficult or economically impractical to reduce the noise to such low levels that they are essentially undetectable in the background. In this case it may be necessary to permit a small amount of intrusion; this may be done at a risk of generating complaints against the noise. A noise excess of about 5 dB above the background (at night) may produce some annoyance but it probably will not lead to legal action. An excess of about 10 dB above background noise will generally produce mild to strong complaints, and an excess of 15 dB or more is almost certain to generate serious complaints and ultimately legal action.

When the outdoor noise criterion levels are obtained by these two approaches, a decision should be reached on the final levels to be used. The lower octave band levels from each approach will certainly yield a non-intrusive noise; the upper octave band levels from each approach may be acceptable if they do not produce the high noise level excesses mentioned above.

6. PROTECTION OF HEARING

When people are exposed repeatedly to high noise levels for long periods of time, hearing loss may result. The noise levels in mechanical equipment rooms ("MERs") or power plants in buildings are frequently high enough to constitute hazardous exposures for essentially continuous occupancy in those work areas.

Table 7 lists the maximum sound pressure levels recommended by two groups for protection of hearing for personnel exposed to these levels for essentially 8 hours per day for many years. Even these levels will produce some hearing loss to some individuals. For details, the reader should refer to the original sources of data.*

Table 8 lists the noise levels considered acceptable for single part-time exposures on a daily basis. Part A of Tables 7 and 8 applies for broad-band noise (no pure tones present), while Part B of each table applies for narrow-band or pure-tone noise.

The CHABA Report emphasizes the value of rest or recovery periods of relative quietness intermixed with periods of high noise levels. During these periods of "quietness" (which must be at least 10 dB quieter in all bands than the levels given in Table 7), the ears begin a recovery process from the previous noise exposure that somewhat helps prepare the listener for the next noise exposure. In effect, for situations where the steady-state noise levels are just marginally above the recommended noise levels of Tables 7 and 8, it is possible to reduce the effect of the higher noise levels by intentionally providing some scheduled periods of "quiet". Or, if the nature of the operator's work in the machinery room is somewhat intermittent, it would be possible to permit these higher noise level exposures, provided that intermittent periods of relative quiet are also assured. Certain generalizations can be given for the intermittent sequences of noise and quiet:

(1) for long intervals of noise exposure, relatively long periods of recovery are required;

(2) for short intervals of noise exposure, relatively short periods of recovery are required;

(3) the higher the noise level, the more beneficial is the short-term removal from the noise.

The CHABA Report provides data on various amounts of intermittent exposures to noise to show the value of these recovery periods. A representative condition is shown in Figure 2. This plot shows the noise levels considered acceptable for certain intervals of noise "on" when they are followed by 10-minute intervals of noise "off". For use of these plots, the operator should be exposed to noise levels at least 10 dB below the Table 7 values during the 10-minute recovery periods.

It is strongly recommended that a separate control room be provided for each MER that must be attended, so that operating personnel can be provided a relatively quiet environment that does not involve hearing-loss noise levels.

*"Noise and Conservation of Hearing," Department of the Army Technical Bulletin TB MED 251, 25 January 1965.

"Hazardous Exposure to Intermittent and Steady-State Noise," National Academy of Science and National Research Council, Committee on Hearing, Bioacoustics, and Biomechanics ("CHABA"), January 1965. (Also published in the Journal of the Acoustical Society of America, Vol. 39, No. 3, pp. 451-464, March 1966).

If the conditions of Tables 7 and 8 and Figure 2 cannot be met, ear protectors or a medically-supervised hearing conservation program are advised.

7. WALSH-HEALEY REGULATION

The following excerpts are taken from the Federal Register, Volume 34, No. 96, May 20, 1969 regarding U.S. Department of Labor Safety and Health Standards, and include corrections issued in July 1969:

Para. 50-204.1 Scope and Application

(a) The Walsh-Healey Public Contracts Act requires that contracts entered into by any agency of the United States for the manufacture or furnishing of materials, supplies, articles, and equipment in any amount exceeding $10,000 must contain, among other provisions, a stipulation that "no part of such contract will be performed nor will any of the materials, supplies, articles, or equipment to be manufactured or furnished under said contract be manufactured or fabricated in any plants, factories, buildings, or surroundings or under working conditions which are unsanitary or hazardous or dangerous to the health and safety of employees engaged in the performance of said contract. Compliance with the safety, sanitary, and factory inspection laws of the State in which the work or part thereof is to be performed shall be prima-facie evidence of compliance with this subsection.

Para. 50-204.10 Occupational Noise Exposure

(a) Protection against the effects of noise exposure shall be provided when the sound levels exceed those shown in Table I of this section when measured on the A scale of a standard sound level meter at slow response. When noise levels are determined by octave band analysis, the equivalent A-weighted sound level may be determined from the accompanying chart.

(b) When employees are subjected to sound exceeding those listed in Table I of this section, feasible administrative or engineering controls shall be utilized. If such controls fail to reduce sound levels within the levels of the table, personal protective equipment shall be provided and used to reduce sound levels within the levels of the table.

(c) If the variations in noise level involve maxima at intervals of 1 second or less, it is to be considered continuous.

(d) In all cases where the sound levels exceed the values shown herein, a continuing, effective hearing conservation program shall be administered.

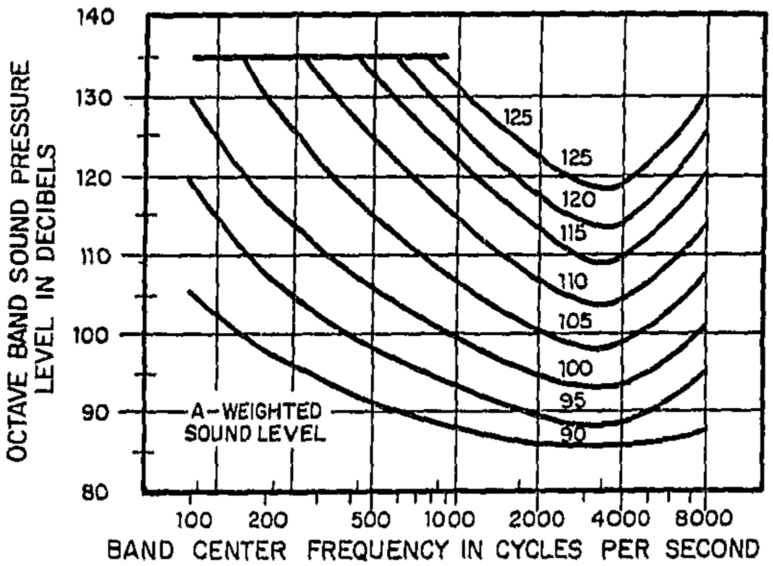
Exposure to impulsive or impact noise should not exceed 140 dBC peak sound pressure level.

TABLE I

PERMISSIBLE NOISE EXPOSURES[1]

Duration per day, hours	Sound level dBA
8	90
6	92
4	95
3	97
2	100
1½	102
1	105
½	110
¼ or less	115

[1]When the daily noise exposure is composed of two or more periods of noise exposure of different levels, their combined effect should be considered, rather than the individual effect of each. If the sum of the following fractions: C1/T1+C2/T2. . . . Cn/Tn exceeds unity, then, the mixed exposure should be considered to exceed the limit value. Cn indicates the total time of exposure at a specified/"actual"/noise level, and Tn indicates the total time of exposure permitted at that level.



Equivalent sound level contours. Octave band sound pressure levels may be converted to the equivalent A-weighted sound level by plotting them on this graph and noting the A-weighted sound level corresponding to the point of highest penetration into the sound level contours. This equivalent A-weighted sound level, which may differ from the actual A-weighted sound level of the noise, is used to determine exposure limits from Table I.

Simple examples of mixed exposures (refer to Table 1 and to footnote under Table I):

a. Suppose 1 hour at 100 dBA $C_1 = 1$ $T_1 = 2$
1 hour at 95 dBA $C_2 = 1$ $T_2 = 4$
2 hours at 90 dBA $C_3 = 2$ $T_3 = 8$
4 hours at 85 dBA $C_4 = 4$ $T_4 = 00$

$$C1/_{T1} + C2/_{T2} + C3/_{T3} + C4/_{T4}$$
$$= 1/2 + 1/4 + 1/4 + 0 = 1 \text{ (acceptable)}$$

b. Suppose 2 hours at 100 dBA $C_1 = 2$ $T_1 = 2$
2 hours at 95 dBA $C_2 = 2$ $T_2 = 4$
4 hours at 90 dBA $C_3 = 4$ $T_3 = 8$

$$C1/_{T1} + C2/_{T2} + C3/_{T3}$$
$$= 1 + 1/2 + 1/2 = 2 \text{ (not acceptable)}$$

c. Suppose repeated peak impact levels of 112 dBA for 8 hours each day (but not exceeding 140 dBC). Suppose noise peaks last 50 milliseconds (1/20 sec) at operator position and occur every 4 seconds. Total of 7200 impacts in 8 hours at 1/20 sec each. Total peak noise exposure time is

7200 x 1/20 = 360 seconds
= 6 minutes @ 112 dBA (acceptable)

8. EAR PROTECTORS

Table 9 presents the approximate minimum attenuation of a good, fitted ear plug (Air Force Type V-51R) and a reasonably comfortable softly-sealing ear muff (Air Force Type PRU-1/P) used singly or in combination. Other current models of well-fitted molded ear plugs and ear muffs will approximate or exceed these values, although poorly fitting protectors will have leakage and will fall short of these values by as much as 5 to 10 dB. In practice, ear plugs are more likely to be poor-fitting because they work loose with time.

The details of fitting, maintaining and the need for persistent use of ear protection are not discussed here as that must fall to the Medical Director and ultimately to the user. It is merely emphasized that ear protectors have no equal for certain specific noise situations.

In the words of Dr. Aram Glorig, leading otologist in this field, "the best ear protector is the one that is worn!"

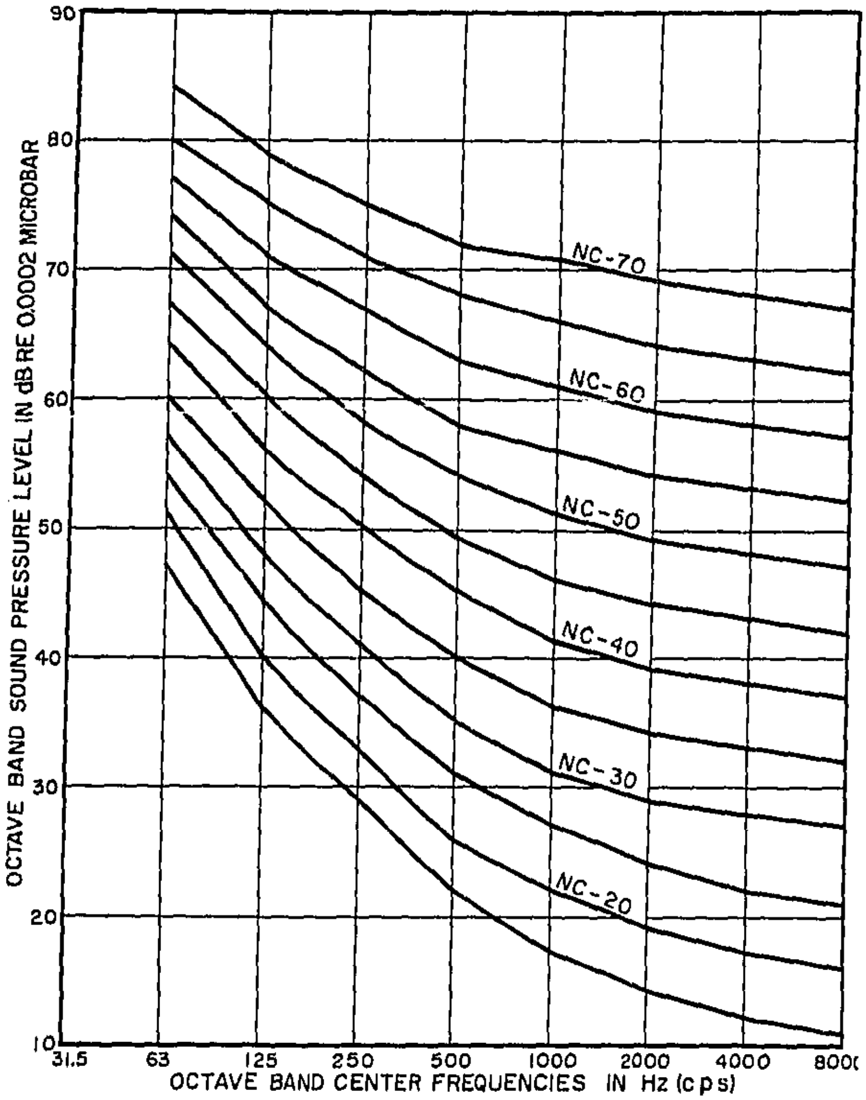


FIG.1 INDOOR NOISE CRITERION "NC" CURVES. REFER TO TABLE 1 FOR NUMERICAL VALUES OF SOUND PRESSURE LEVELS OF NC CURVES. REFER TO TABLE 2 FOR APPLICABLE AREAS.

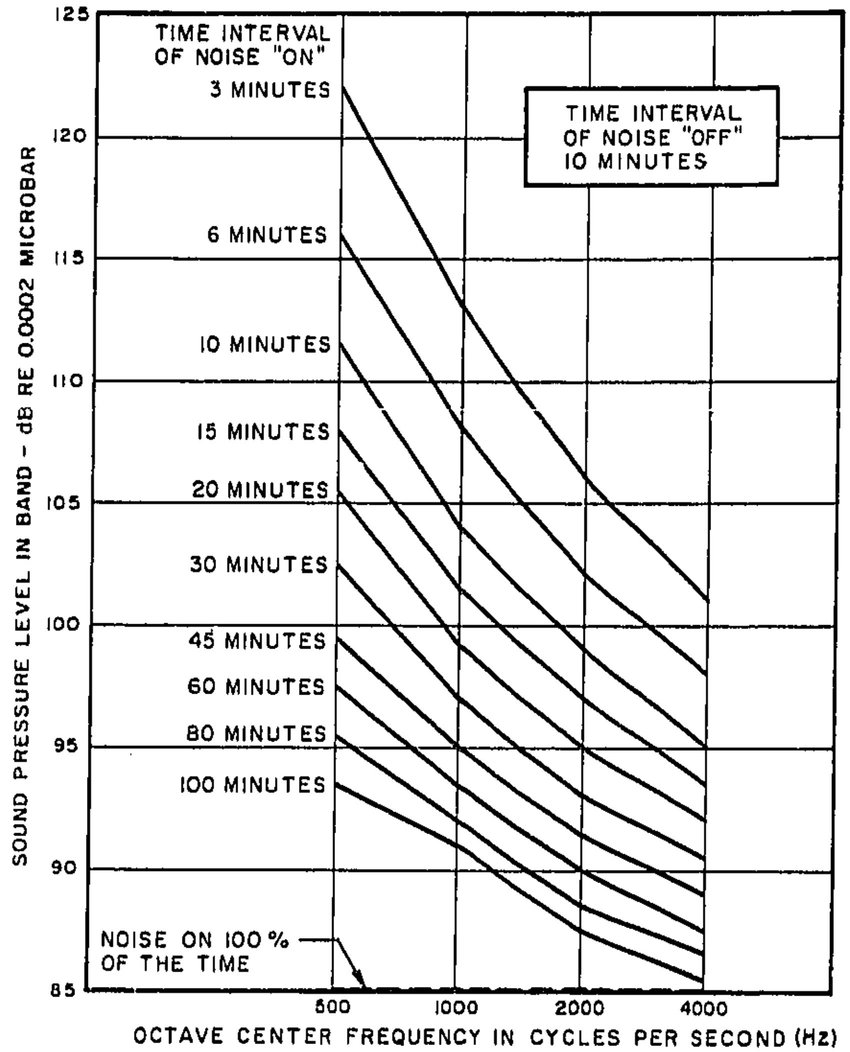


FIG. 2 HEARING PROTECTION CONTOURS FOR LONG-DURATION BROAD-BAND NOISE, WITH SYSTEMATICALLY SCHEDULED 10-MIN. ABSENCES FROM THE NOISE.

TABLE 1

OCTAVE BAND SOUND PRESSURE LEVEL (SPL) VALUES
ASSOCIATED WITH THE NOISE CRITERION
CURVES OF FIGURE 1 AND TABLE 2

NOISE CRITERION CURVES	63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
NC-15	47	36	29	22	17	14	12	11
NC-20	51	40	33	26	22	19	17	16
NC-25	54	44	37	31	27	24	22	21
NC-30	57	48	41	35	31	29	28	27
NC-35	60	52	45	40	36	34	33	32
NC-40	64	56	50	45	41	39	38	37
NC-45	67	60	54	49	46	44	43	42
NC-50	71	64	58	54	51	49	48	47
NC-55	74	67	62	58	56	54	53	52
NC-60	77	71	67	63	61	59	58	57
NC-65	80	75	71	68	66	64	63	62

TABLE 2

CATEGORY CLASSIFICATION AND SUGGESTED NOISE CRITERION RANGE FOR INTRUDING MECHANICAL EQUIPMENT NOISE AS HEARD IN VARIOUS INDOOR FUNCTIONAL ACTIVITY AREAS

CATEGORY	AREA (AND ACOUSTIC REQUIREMENTS)	NOISE CRITERION
1	Bedrooms, sleeping quarters, hospitals, residences, apartments, hotels, motels, etc. (for sleeping, resting, relaxing).	NC-20 to NC-30
2	Auditoriums, theaters, large meeting rooms, large conference rooms, churches, chapels, etc. (for very good listening conditions).	NC-20 to NC-30
3	Private offices, small conference rooms, classrooms, libraries, etc. (for good listening conditions).	NC-30 to NC-35
4	Large offices, reception areas, retail shops and stores, cafeterias, restaurants, etc. (for fair listening conditions).	NC-35 to NC-40
5	Lobbies, laboratory work spaces, drafting and engineering rooms, maintenance shops such as for electrical equipment, etc. (for moderately fair listening conditions).	NC-40 to NC-50
6	Kitchens, laundries, shops, garages, machinery spaces, power plant control rooms, etc. (for minimum acceptable speech communication, no risk of hearing damage).	NC-45 to NC-65

TABLE 3

SPEECH INTERFERENCE LEVELS ("PSIL"):
AVERAGE NOISE LEVELS* (IN DB) THAT PERMIT
BARELY ACCEPTABLE SPEECH INTELLIGIBILITY
AT THE DISTANCES AND VOICE LEVELS SHOWN

Distance (ft)	Voice Level: Normal	Raised	Very Loud	Shouting
½	74	80	86	92
1	68	74	80	86
2	62	68	74	80
4	56	62	68	74
6	53	59	65	71
8	50	56	62	68
10	48	54	60	66
12	46	52	58	64
16	44	50	56	62

*PSIL (Speech Interference Level in "Preferred" Octave Bands) is arithmetic average of noise levels in the 500, 1000 and 2000 Hz octave frequency bands. PSIL values apply for average male voices (reduce values 5 dB for female voice), with speaker and listener facing each other, using unexpected word material. PSIL values may be increased 5 dB when familiar material is spoken. Distances assume no nearby reflecting surface to aid the speech sounds.

TABLE 4

ESTIMATE OF OUTDOOR BACKGROUND NOISE BASED ON GENERAL TYPE OF COMMUNITY AREA AND NEARBY AUTOMOTIVE TRAFFIC ACTIVITY

(Determine the appropriate conditions that seem to best describe the area in question during the time interval that is most critical; i.e., day or night, probably night if for sleeping. Then refer to corresponding Noise Code No. in Table 5 for average minimum background noise levels to be used in noise analysis. Use lowest Code No. where several conditions are found to be reasonably appropriate.)

	CONDITION	NOISE CODE NO.
1.	Nighttime, rural; no nearby traffic of concern	1
2.	Daytime, rural; no nearby traffic of concern	2
3.	Nighttime, suburban; no nearby traffic of concern	2
4.	Daytime, suburban; no nearby traffic of concern	3
5.	Nighttime, urban; no nearby traffic of concern	3
6.	Daytime, urban; no nearby traffic of concern	4
7.	Nighttime, business or commerical area	4
8.	Daytime, business or commercial area	5
9.	Nighttime, industrial or manufacturing area	5
10.	Daytime, industrial or manufacturing area	6
11.	Within 300 ft of intermittent light traffic route	4
12.	Within 300 ft of continuous light traffic route	5
13.	Within 300 ft of continuous medium-density traffic	6
14.	Within 300 ft of continuous heavy-density traffic	7
15.	300 to 1000 ft from intermittent light traffic route	3
16.	300 to 1000 ft from continuous light traffic route	4
17.	300 to 1000 ft from continuous medium-density traffic	5
18.	300 to 1000 ft from continuous heavy-density traffic	6
19.	1000 to 2000 ft from intermittent light traffic	2
20.	1000 to 2000 ft from continuous light traffic	3
21.	1000 to 2000 ft from continuous medium-density traffic	4
22.	1000 to 2000 ft from continuous heavy-density traffic	5
23.	2000 to 4000 ft from intermittent light traffic	1
24.	2000 to 4000 ft from continuous light traffic	2
25.	2000 to 4000 ft from continuous medium-density traffic	3
26.	2000 to 4000 ft from continuous heavy-density traffic	4

TABLE 5

OCTAVE BAND SOUND PRESSURE LEVELS OF
OUTDOOR BACKGROUND NOISE CODE NUMBERS OF TABLE 4

NOISE CODE NO. IN TABLE 4	OCTAVE BAND CENTER FREQUENCY IN HZ							
	63	125	250	500	1000	2000	4000	8000
1	40	37	32	27	22	18	14	12
2	45	42	37	32	27	23	19	17
3	50	47	42	37	32	28	24	22
4	55	52	47	42	37	33	29	27
5	60	57	52	47	42	38	34	32
6	65	62	57	52	47	43	39	37
7	70	67	62	57	52	48	44	42

TABLE 6

APPROXIMATE NOISE REDUCTION OF OUTSIDE NOISE PROVIDED BY TYPICAL EXTERIOR WALL CONSTRUCTION

OCTAVE FREQUENCY BAND (HZ)	A	B	C	D	E	F	G
63	0	9	13	19	14	24	32
125	0	10	14	20	20	25	34
250	0	11	15	22	26	27	36
500	0	12	16	24	28	30	38
1000	0	13	17	26	29	33	42
2000	0	14	18	28	30	38	48
4000	0	15	19	30	31	43	53
8000	0	16	20	30	33	48	58

A: No wall; outside conditions.

B: Any typical wall construction, with open windows covering about 5% of exterior wall area.

C: Any typical wall construction, with small open air vents of about 1% of exterior wall area, all windows closed.

D: Any typical wall construction, with closed but operable windows covering about 10-20% of exterior wall area.

E: Sealed glass wall construction, 1/4 in. glass thickness over approximately 50% of exterior wall area.

F: Approximately 20 lb/sq ft solid wall construction with no windows and no cracks or openings.

G: Approximately 50 lb/sq ft solid wall construction with no windows and no cracks or openings.

TABLE 7

A. MAXIMUM SOUND PRESSURE LEVELS
RECOMMENDED FOR HEARING CONSERVATION FOR
FULL-TIME EXPOSURE TO BROAD-BAND NOISE

Octave Frequency Band (Hz)	Sound Pressure Level in Band in dB re 0.0002 microbar	
	Recommended by TB MED 251	Recommended by CHABA
125	--	97
250	92	92
500	85	89
1000	85	86
2000	85	85
4000	85	85
8000	85	86

B. MAXIMUM SOUND PRESSURE LEVELS
RECOMMENDED FOR HEARING CONSERVATION FOR
FULL-TIME EXPOSURE TO NARROW-BAND NOISE OR PURE TONES

Octave Frequency Band (Hz)	Sound Pressure Level in Band in dB re 0.0002 microbar	
	Recommended by TB MED 251	Recommended by CHABA
125	--	92
250	87	87
500	80	84
1000	80	81
2000	80	80
4000	80	80
8000	80	81

TABLE 8

A. MAXIMUM SOUND PRESSURE LEVELS RECOMMENDED FOR HEARING CONSERVATION FOR PART-TIME EXPOSURE TO BROAD-BAND NOISE

Octave Frequency Band (Hz)	Sound Pressure Level (in dB) in Band for Single Exposure of Duration: 4 Hrs.	2 Hrs.	1 Hr.	½ Hr.	¼ Hr.
125	103	111	119	127	135
250	96	101	107	115	123
500	90	94	99	105	112
1000	88	91	95	100	106
2000	86	88	91	95	100
4000	85	87	90	93	98
8000	87	90	95	100	105

B. MAXIMUM SOUND PRESSURE LEVELS RECOMMENDED FOR HEARING CONSERVATION FOR PART-TIME EXPOSURE TO NARROW-BAND NOISE OR PURE TONES

Octave Frequency Band (Hz)	Sound Pressure Level (in dB) in Band for Single Exposure of Duration: 4 Hrs.	2 Hrs.	1 Hr.	½ Hr.	¼ Hr.
125	93	95	98	105	112
250	88	90	94	100	106
500	83	86	91	96	101
1000	82	85	89	93	97
2000	81	83	86	90	94
4000	80	82	85	88	92
8000	82	85	90	94	99

TABLE 9

APPROXIMATE ATTENUATION (IN DB) OF
WELL-FITTED EAR PLUGS AND EAR MUFFS
(Poor fitting reduces attenuation significantly)

OCTAVE FREQUENCY BAND (HZ)	EAR PLUGS	EAR MUFFS	COMBINED
31	16	12	20
63	18	14	22
125	20	16	24
250	22	19	27
500	24	24	30
1000	27	30	34
2000	30	30	40
4000	33	35	45
8000	35	30	40

Note: Some ear protectors may achieve higher values of attenuation. Refer to manufacturers' literature.

CHAPTER 3

NOISE DATA FOR MECHANICAL AND ELECTRICAL EQUIPMENT

Noise data have been collected and studied for almost all the different types of electrical and mechanical equipment that might be encountered in the Mechanical Equipment Room (MER) of a building. The noise data have been evaluated in an attempt to correlate noise levels with some of the more obvious noise-influencing parameters, such as type, speed, power rating, etc.* The noise data are summarized here, as a function of some of those parameters. It is believed that the noise levels given represent approximately the 80 to 90 percentile values; in other words about 80 to 90% of the equipment might be expected to have no higher noise levels than the estimates given, but possibly 10 to 20% of the equipment may have higher noise levels. The sample-size of the data would not seem to justify any finer resolution.

1. SOUND POWER LEVEL AND SOUND PRESSURE LEVEL DATA

Noise data in some of the tables at the end of this section are given in terms of sound power level (PWL). This is done wherever the original measurements included enough data regarding the acoustic characteristics of the room in which the measurements were made to be able to separate the effect of the room from the noise characteristics of the source alone. Where the measurements or measurement conditions did not provide adequate data on the influence of the room, it has been decided to use only the resulting sound pressure level (SPL) values.

In order to "standardize" all sound pressure levels to a common condition, a distance of 3 ft has been selected. This decision is based on at least three considerations: (1) because of crowded conditions in mechanical spaces most measurements are taken at close distances, (2) much of the quoted data in the literature refers to a 3-ft distance, although this is not a universally used distance, and (3) when considering the various building elements that provide noise control (walls, ceilings, floors, etc.), the floor is always a near-by element and it is not unreasonable to consider that the noise at a 3-ft distance will approximate the noise levels impinging on the floor at the base of the equipment. Thus, it appears that the 3-ft SPL values would be the <u>highest</u> SPLs necessary in a noise evaluation (specifically applicable to the floor) and that the levels would decrease for greater distances within the room. Also, for most applications, at 3-ft distances the noise levels are essentially in the near field of large pieces of equipment and are reasonably independent of the acoustic characteristics of the room. Thus, these close-in levels can be taken for any room and/or equipment configuration with only a small amount of uncertainty due to room acoustics. Later in the notes, the SPL reduction for greater distances from the equipment will be shown.

2. EQUIPMENT NOISE SUMMARIES

Brief discussions of the noise of the various types of equipment included in the notes are given in the paragraphs that follow. The equipment is

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*A large part of the current study of these data has been performed under Contract No. DACA-73-68-C-0017 with the Office of the Chief of Engineers of the Department of the Army.

subdivided into the following general groups:

Refrigeration System Equipment
- Packaged Chillers with reciprocating compressors
- Packaged Chillers with rotary-screw compressors
- Packaged Chillers with centrifugal compressors
- Absorption machines

Heating System Equipment
- Boilers
- Steam Valves

Air Handling Equipment
- Fans

Liquid Circulation System Equipment
- Cooling Towers
- Pumps

Prime-Mover Equipment
- Reciprocating Engines
- Turbine Engines
- Electric Motors
- Steam Turbines
- Gears

Electric Equipment
- Transformers

Air Compressors

3. PACKAGED CHILLERS WITH RECIPROCATING COMPRESSORS

Noise data for 24 reciprocating compressors or packaged chillers with reciprocating compressors have been collected and studied. These units range in size from 15 tons to 150 tons cooling capacity. The noise levels have been reduced to a common 3-ft distance from the front of the compressor.

In terms of noise production, it appears that the measured compressors can be divided into two groups: 15-50 tons and 51-150 tons. The two ends of the total range have been extended slightly to cover compressors from 10 to 175 tons. When cooling requirements exceed about 100 to 150 tons, centrifugal compressors become more economical so there are few reciprocating units rated above about 150 tons.* Even in this collection of data, several of the larger units are actually made up of assemblies of two to four smaller compressors.

The suggested near-maximum noise level estimates are given in the upper portion of Table 1. Apparently,there is not a large enough range in speed of these machines to justify a noise adjustment for speed. Although major interest is concentrated here on the compressor component of a refrigeration

*A ton of cooling capacity is defined as the amount of heat removal required to produce one ton of ice per 24 hour period.

machine, an electric motor is usually the drive unit for the compressor. The noise levels attributed here to the compressor will encompass the drive motor most of the time, so these values are taken to be applicable to either a reciprocating compressor alone or a motor-driven packaged chiller containing a reciprocating compressor.

4. PACKAGED CHILLERS WITH ROTARY-SCREW COMPRESSORS

Based on data for only three units, the octave band sound pressure levels (at 3-ft distance) believed to represent essentially maximum noise levels for rotary-screw compressors are listed in the middle portion of Table 1. These data apply for the size range of 100-300 tons cooling capacity, operating at or near 3600 RPM.

5. PACKAGED CHILLERS WITH CENTRIFUGAL COMPRESSORS

Noise levels have been measured for 20 centrifugal type compressors. These measured compressors range in size from 140 tons to 1500 tons and represent several leading manufacturers. The noise levels may be influenced by motors, gears or steam turbines used to drive the compressors, but the measurement positions are generally selected to emphasize the compressor noise. The noise levels given in Table 1 represent essentially the maximum values found for the twenty units when divided into the two size groups: under 500 tons and 500 or more tons.

6. ABSORPTION MACHINES

Noise data have been acquired for only a few steam absorption machines. More data would be sought if these were seriously noisy devices, but they are quiet enough that they are usually ignored in any noise survey of a mechanical equipment room. The machine usually includes one or two small pumps. Steam flow noise or steam valve noise may also be present.

It is believed that the noise levels given at the bottom of Table 1 will give adequate coverage of most absorption machines used in refrigeration systems for buildings.

7. BOILERS

Noise data have been measured or collected for at least 36 boilers, ranging in size from 50 to 2000 boiler horsepower ("BHP"). It has not been possible to correlate noise with heating capacity alone or with any other known design parameter. Noise levels at the normalized 3-ft distance may be as high for the smallest as for the largest units. Hence, the estimated noise levels given in Table 2 are believed applicable for all boilers, although some units will exceed these values. These 3-ft noise levels are applicable to the front of the boiler; so when other distances are of concern, the distance should always be taken from the front surface of the boiler. Noise levels are much lower off the side and rear of the typical boiler. When one sees the wide variety of blower assemblies, burners and combustion chambers found on various boilers, it is no wonder that the noise output cannot be simply associated with heating capacity.

Heating capacity of boilers may be expressed in four different ways, as follows:

(a) sq. ft of heating surface
(b) BTU/hour
(c) lb of steam/hour
(d) BHP (boiler horsepower).

To a first approximation, these terms are interrelated as follows:

10 sq. ft of heating surface = 1 BHP
33,500 BTU/hour = 1 BHP
33 lb of steam/hour = 1 BHP

All ratings have been reduced here to equivalent BHP.

8. STEAM VALVES

Based on the noise level data of three groups of steam valves, mostly on high pressure steam lines, estimated near-maximum noise levels are given in Table 2 for what is considered to be a typical thermally-insulated steam pipe and valve. Even though the noise is generated at and near the orifice of the valve, the pipes on either side of the valve radiate a large part of the total noise energy that is radiated. Hence, the pipe is considered, along with the valve, as a part of the noise source.

9. FANS

The recent issues of the ASHRAE Guide have summarized the most detailed available data on ventilating fans. The ASHRAE Guide should be used for estimating the PWL of the fan noise transmitted along the inlet and discharge ducts for the specific type, size and operating condition of the fan. Approximate MER noise levels due to an enclosed fan can be estimated by deducting the approximate "transmission loss" of the enclosure from the PWL of the fan and then estimating the SPL radiated into the room by that reduced PWL. This will become more apparent later in the notes.

In the event that a suitable PWL estimate cannot be obtained from the fan manufacturer or from the ASHRAE Guide, a rough estimate of the in-duct fan PWL can be obtained from Table 3.

10. COOLING TOWERS

It must be realized that the generalizations drawn here may not apply exactly to all cooling towers, but it is believed that these generalizations are one step closer toward the useful data frequently required by the architect or engineer in laying out cooling towers and cooling tower noise control treatments in any given acoustic environment. It is still desirable to try to obtain from the manufacturer actual measured noise levels for all directions of interest, but if these data are not forthcoming, it is essential to be able to construct approximately the directional pattern of the cooling tower noise.

For aid in identification, four general types of cooling towers are sketched in Figure 1:

A. the centrifugal-fan blow-through type,
B. the axial-flow blow-through type (with the fan or fans located on a side wall),
C: the induced-draft propeller-type, and
D. the "underflow" forced-draft propeller-type (with the fan located under the assembly).

a. Noise Levels at a Distance. Part A of Table 4 gives estimated sound power levels (PWLs) for propeller-type cooling towers. In the absence of more accurate measured or estimated data from the tower manufacturer, the Table 4A data may be used for estimating the total noise output of types B, C and D cooling towers listed immediately above and shown in Figure 1, provided they are driven by propeller-type fans.

Part B of Table 4 gives the estimated sound power levels of cooling towers driven by centrifugal-type fans, such as shown in type A of Figure 1.

To obtain the average outdoor sound pressure levels at any distance from an unobstructed cooling tower, a "distance term" is applied to the sound power levels of Table 4. This "distance term" is given later in the notes. When the distance term is applied, the calculation yields the average SPL all around the cooling tower as though there were no directionality variations of the noise.

b. Directional Corrections. It is obvious, of course, that the noise differs for different radiating surfaces of a typical tower, and it is valuable to know, at least approximately, the amount of the directional variations. Table 5 gives some approximate corrections for the directional effects of the four types of towers considered here. These corrections are to be added to the average SPLs calculated for the particular distance involved. Please note the qualifications to the use of Table 5, as given under the caption of the table. These corrections apply to the five principal directions from a cooling tower, i.e. in a direction perpendicular to each of the four sides and to the top of the tower. If it is necessary to estimate the SPL at some direction other than the principal directions, one should feel free to interpolate between the values given for the principal directions.

c. Close-in Noise Levels. The noise data given in the preceding discussion are most useful in estimating the noise levels of cooling towers as heard at some distance away. Although the sound power level data can be used to estimate approximately the close-in noise levels, considerable close-in data have been collected and are suggested for use in determining, for example, the type of wall or floor required to separate the cooling towers from quiet parts of the building. Based on the data, Table 6 gives the estimated close-in noise level summaries for the four types of towers. Although very little data were studied for the "underflow" tower, it would seem reasonable to expect the close-in noise levels of the axial-flow blow-through type to be comparable to those of the "underflow" type. Thus, these two types are combined in Table 6; functionally the fans perform similar operations, the only significant difference is their location relative to the tower assembly.

d. Half-speed Operation. When it is practical to do so, the cooling tower fan can be reduced to half-speed in order to reduce noise; it also reduces cooling capacity. Half-speed produces approximately two-thirds cooling capacity and approximately 8-10 dB noise reduction in the octave bands that contain most of the fan-induced noise. The noise reduction for half-speed operation is approximately as follows: Reduce the octave band SPLs or PWLs of full-speed cooling tower noise by the following amounts for half-speed operation, where f_B is the blade passage frequency and is calculated from the relation

$$f_B = \frac{\text{No. of fan blades x shaft RPM}}{60}$$

Octave band that contains:	Noise reduction due to half-speed:
1/8 f_B	3 dB
1/4 f_B	6 dB
1/2 f_B	9 dB
f_B	9 dB
2 f_B	9 dB
4 f_B	6 dB
8 f_B	3 dB

If the blade passage frequency is not known, assume that it falls in the 125 Hz band for propeller-type cooling towers and in the 250 Hz band for centrifugal-fan cooling towers. Water fall noise usually dominates in the upper octave bands and it would not change significantly with reduced fan speed.

e. Limitations. The data given here represent the most complete survey to date on cooling tower noise, but it must still be expected that noise levels may vary from manufacturer to manufacturer and from model to model as specific design changes take place. Whenever possible, request the manufacturer to supply the specific noise levels for the specific needs.

Most of the preceding discussion assumes that cooling towers will be used in outdoor locations. If they are located inside enclosed mechanical equipment rooms or within courts formed by several solid walls, the sound patterns will be distorted. In such instances, the PWL of the tower (or appropriate portions of the total PWL) can be placed in that setting, and the enclosed or partially enclosed space can be likened to a room having certain estimated amounts of reflecting and absorbing surfaces. Because of the limitless number of possible arrangements, this is not simply handled in a general way, so the problem of partially enclosed cooling towers is not treated here in detail. In the absence of a detailed analysis of cooling tower noise levels inside enclosed spaces, it is suggested that the close-in noise levels of Table 6 be used as general approximations.

f. Evaporative Condensers. Evaporative condensers are somewhat similar to cooling towers in terms of noise generation. A few evaporative condensers have been included with the cooling towers, but not enough units have been measured to justify a separate study of evaporative condensers alone. In the absence of noise data on specific evaporative condensers, it is suggested that noise data be used for the most nearly similar type and size cooling tower.

g. Air-cooled Condensers. For some installations, an outdoor air-cooled condenser may serve as a substitute for a cooling tower or evaporative condenser. The noise of an air-cooled condenser is made up almost entirely of fan noise and possibly air-flow noise through the condenser coil decks. In general, the low frequency fan noise dominates. Since most of the low frequency noise of a typical cooling tower is due to the fan system, in the absence of specific data on air-cooled condensers, it would not be unreasonable to use noise data for the most nearly similar type and size cooling tower.

11. PUMPS

Noise data have been collected and studied for a large number of pumps ranging in size from 3 HP to approximately 2000 HP. The various pumps covered a speed range of 450 to 3600 RPM. All pumps were loaded but not necessarily at full rated load. The name-plate horsepower of the drive motor or turbine has usually been used to rate the pump power. All noise data have been normalized to the reference distance of 3 ft in an indoor situation. Based on the measured noise data, a schedule of noise levels for various speed and power ranges is given in Table 7.

12. NATURAL-GAS AND DIESEL RECIPROCATING ENGINES

A fairly thorough study was carried out in 1966-67 for the Office of the Chief of Engineers of the Department of the Army and for the American Gas Association to collect and study the noise of reciprocating and turbine engines. The estimated PWLs for the casing noise of reciprocating engines, driven by natural gas or liquid fuel, are given in Table 8. The estimated PWLs of unmuffled exhaust noise are given in Table 9, and the estimated PWLs for untreated turbocharger inlet noise are given in Table 10. Various corrections are also listed in these tables.

13. GAS TURBINE ENGINES

Sound power levels are given in Table 11 for the casing noise of unenclosed gas turbine engines. The footnotes indicate approximate amounts of noise reduction that can be applied if the turbine engine is fitted with some sort of enclosure. The approximate PWLs of unmuffled exhaust and intake noise of turbine engines are given in Table 12, Parts A and B.

The turbine engines listed here cover a power range of 200-5000 KW. Although the original noise study included turbine engines up to 19000 KW, there are special noise problems associated with the very large outdoor installations that are considered beyond the scope of the present material. To convert between HP and KW, use the following relationship:

$$HP = 1.5\ KW$$

14. ELECTRIC MOTORS

The noise data of more than 90 electric motors (or groups of motors) have been accumulated and summarized. The data include tables of noise levels listed in the IEEE Publication No. 85* and in a paper by Heitner#, as well as noise levels collected or measured for many motors in field installations. The data study included large numbers of both "drip-proof" (or "splash-proof" or "weather-protected") motors and "totally-enclosed fan-cooled" (TEFC) motors, but no significant noise difference was found for these two groups. Noise levels were found to increase with HP rating and to decrease with speed approximately in accordance with the estimated values given in Table 13. The total range of motor power covered was 1 to 4000 HP, and the total range of motor speed was 450 to 3600 RPM. Some motors range 10 to 30 dB below the noise level curves of Table 13, but a few motors exceed the noise estimates throughout the speed and power ranges.

15. STEAM TURBINES

Noise data for eight steam turbines have been collected, covering a power range of 500 to 11,000 HP. The noise levels are found generally to increase with increasing power rating, as shown in Table 14.

16. GEARS

Noise data have been measured or collected for nine large gears in the power handling range of 300 to 23,200 HP. It is generally true that the noise output increases with increasing speed and power, but it is not possible to predict in which frequency band the gear tooth contacts or the "ringing frequencies" will occur for any unknown gear. Thus, the noise level estimate of Table 15 assumes a flat spectrum in all octave bands at and above 125 Hz. Although the spectrum is known not to be flat, this estimate permits peak frequencies to fall into any octave band. The estimate given here is not claimed to be highly accurate but it will provide a reasonable engineering evaluation of the gear noise.

17. TRANSFORMERS

Transformers typically are covered by NEMA sound level ratings, and transformer manufacturers usually quote the NEMA ratings when asked to specify the noise output of their products. Some manufacturers, however, produce and market transformers having sound levels below the applicable NEMA ratings. These quieter transformers may be sold at somewhat higher prices.

The current NEMA Standards Publication No. TR 1-1968 specifies the method for measuring and calculating the sound level rating for a transformer. In effect, the procedure consists of averaging a large number of A-scale sound level meter readings taken all around the transformer (at suitably specified

*"Test Procedure for Airborne Noise Measurements on Rotating Electric Machinery", February 1965. This table of noise data is repeated in the NEMA Publication MC1-1967 on Motors and Generators.

#"How to Estimate Plant Noises", Irving Heitner, Hydrocarbon Processing, December 1968, Vol. 47, No. 12, pages 67-74.

positions) at distances of 1 ft from various surfaces of the transformer (or at 6-ft distances from fan-cooled radiating surfaces). The reader is referred to the various applicable NEMA publications for more detailed discussions of the procedure.

It is important to understand the significance of the NEMA "audible sound level," as it is called in the specification. Interest here is limited to 60-Hz (cycle) power. Due to the magneto-strictive action of the transformer core material, the core goes through a complete cycle of oscillation for each half-cycle of voltage change. Thus, for 60-Hz operation, maximum sound output from the core occurs at 120 Hz and its harmonics (240, 360, 480 Hz and so on).

The A-scale weighting network of the sound level meter intentionally discriminates against low-frequency sound; it somewhat simulates the response of the human ear for low-level sounds at low frequency. To be specific, the A-scale network reduces the signal levels of the transformer frequencies, of interest here, by the following amounts (in accordance with USASI standards for sound level meters):

60 Hz	-27 dB
120 Hz	-16 dB
240 Hz	-9 dB
360 Hz	-5 dB
480 Hz	-4 dB

This means, simply, that if a transformer produced at the 1-ft position a true sound pressure level of 66 dB at 120 Hz (and assuming no other components present), the A-scale reading would be 66-16 = 50 dBA. Note the designation "dBA" to indicate an A-scale reading in decibels, and note also that this value is called a "sound level," not a "sound pressure level".

Based on data and experience with a few noisy transformers, an estimating procedure has been derived which, it is believed, will provide a maximum reasonable sound pressure level in a transformer room based on the NEMA sound level rating for that transformer. Thus, it is necessary for the electrical engineer on the job to determine the electrical power handling requirements of the planned transformer and to estimate or obtain from a manufacturer the probable NEMA sound level rating for that transformer. The NEMA rating number (in dBA) should then be added algebraically to the values listed in the right-hand column of Table 16 to obtain the estimated maximum SPLs near the transformer.

In this development, assumptions have been made regarding "harmonic content" of the transformer noise and the possibility of standing waves in the transformer room. For most transformers and transformer rooms, the SPLs will not be as high as the estimated values. Many transformers are quieter than the NEMA standard, many transformers do not produce unusually high 240, 360 and 480 Hz noise components, and for many installations there will be no strong standing wave build-up, so this procedure will appear to yield high sound pressure levels when tested against many existing situations. However, this procedure is designed to protect a room against the marginally "noisy" transformer in which each of these effects may be somewhat pronounced.

There are a few points to keep in mind in the application of this procedure.

1. Where a manufacturer is willing to guarantee that his product will produce a lower sound level rating than the otherwise-applicable NEMA rating, the manufacturer's sound level value (the average dBA reading taken at 1-ft distance in accordance with the NEMA method) may be used when entering Table 16 for obtaining the octave band SPLs.

2. The purchase specification should state that the sound level of the purchased transformer shall not exceed the applicable NEMA sound level rating, and that the transformer shall be removed if it does not comply.

3. Although the procedure developed here is based on transformer noise rather than cooling fan noise, it is believed that the noise estimate will protect against a reasonable amount of fan noise for any large forced-air cooled transformer.

18. AIR COMPRESSORS

Two types of air compressors are frequently found in buildings: one is a relatively small compressor (usually under 5 HP) used to provide a high pressure air supply for operating the controls of the ventilation system, and the other is a medium size compressor (possibly up to 100 HP) used to provide "shop air" to maintenance shops, machine shops or some laboratory spaces, or to provide ventilation system control pressure for large buildings. Larger compressors are used for special industrial processes or special facilities, but these are not considered within the scope of this survey.

From the measured data on nine compressors, it has been possible to arrive at noise estimates that will encompass both reciprocating and centrifugal compressors in the range of 1 - 100 HP. These are shown in Table 17.

19. MULTIPLE NOISE SOURCES (DECIBEL ADDITION)

Since a mechanical equipment room generally contains several pieces of equipment, it will be necessary frequently to add together the noise levels at a particular location in the room due to a number of sources. When noise levels are combined "by decibel addition", the four simple steps of Table 18 should be followed.

20. VENTILATION OPENINGS IN MER WALL

When room ventilation air is brought into an MER through a hole in the exterior wall, that hole will allow noise to escape to the outside. The escaping noise may be disturbing to nearby neighbors. The power level of sound that passes through an opening into or out of a room is approximately

$$\text{PWL (in dB re } 10^{-12}\text{ watts)}$$
$$= \text{SPL} + 10 \log A\text{-}10$$

where SPL is the sound pressure level at or near the opening and A is the cross-section area in sq ft of the opening. A new term "Area Factor" ("AF") is defined as follows:

$$\text{"AF"} = 10 \log A - 10.$$

Then

$$\text{PWL} = \text{SPL} + \text{"AF"}.$$

Table 19 gives a range of values of "AF" for a representative group of areas.

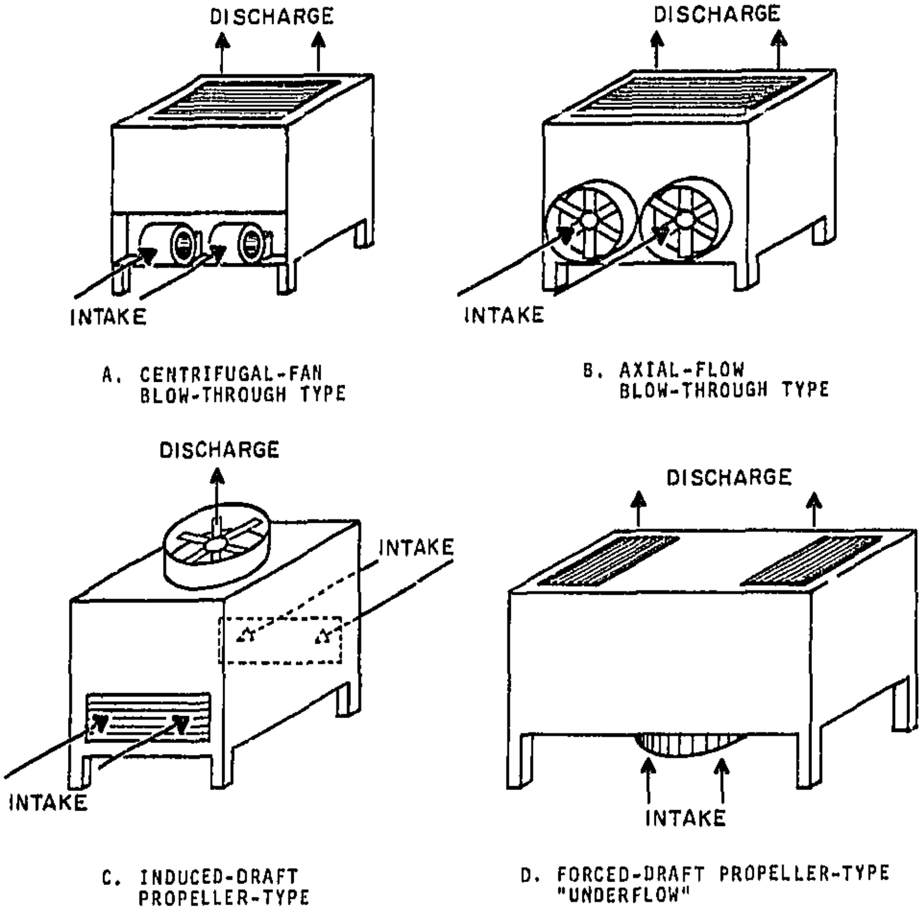

FIG. 1 PRINCIPAL TYPES OF COOLING TOWERS

TABLE 1

ESTIMATED SOUND PRESSURE LEVELS (IN DB) AT 3-FT DISTANCE
DUE TO VARIOUS TYPES OF REFRIGERATION MACHINES

OCTAVE BAND (HZ)	31	63	125	250	500	1000	2000	4000	8000
MACHINE TYPE AND COOLING CAPACITY (TONS):									
PACKAGED CHILLERS WITH RECIPROCATING COMPRESSORS									
10-50	82	86	84	86	87	86	84	80	75
51-175	85	90	89	92	93	92	90	86	81
PACKAGED CHILLERS WITH ROTARY-SCREW COMPRESSORS									
100-300	70	76	80	92	89	85	80	75	73
PACKAGED CHILLERS WITH CENTRIFUGAL COMPRESSORS									
Under 500	87	88	89	90	90	91	92	87	80
500 and more	89	90	91	92	93	97	99	94	87
ABSORPTION MACHINES									
All sizes	88	91	86	86	86	83	80	77	72

TABLE 2

ESTIMATED SOUND PRESSURE LEVELS (IN DB) AT 3-FT DISTANCE
DUE TO BOILERS AND STEAM VALVES

OCTAVE BAND (HZ)	31	63	125	250	500	1000	2000	4000	8000
BOILERS (50-2000 BHP)*									
	92	92	92	89	86	83	80	77	74
STEAM VALVES (WITH THERMAL INSULATION)									
	70	70	70	70	75	80	85	90	95

* Distance should be measured from the front surface of the boiler.

TABLE 3

APPROXIMATE OCTAVE BAND SOUND POWER LEVELS*
OF VENTILATING FAN NOISE (DB RE 10^{-12} WATT)

	NAME-PLATE RATING OF FAN DRIVE MOTOR							
OCTAVE FREQUENCY BAND (HZ)	1 to 2 HP	2.1 to 4 HP	4.1 to 8 HP	9 to 16 HP	17 to 32 HP	33 to 64 HP	65 to 128 HP	129 to 256 HP
31	84	87	90	93	96	99	102	105
63	86	89	92	95	98	101	104	107
125	86	89	92	95	98	101	104	107
250	84	87	90	93	96	99	102	105
500	80	83	86	89	92	95	98	101
1000	75	78	81	84	87	90	93	96
2000	70	73	76	79	82	85	88	91
4000	65	68	71	74	77	80	83	86
8000	60	63	66	69	72	75	78	81

NOTES:

1. Add 3 dB to all values if static pressure produced by fan is 1.5 to 3 in. water gauge.
2. Add 6 dB to all values if static pressure produced by fan is 3.1 to 6 in. water gauge.
3. Add 9 dB to all values if static pressure produced by fan is 6.1 to 12 in. water gauge.
4. The values given here apply to the noise radiated from the discharge end of the fan. Subtract 2 dB to obtain an estimate of the noise radiated from the intake end of the fan. These values do not represent the noise radiated by the fan casing and ducts.

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* These approximations may be used in the absence of more definite fan data from the manufacturer or from the ASHRAE Guide and Data Book. See Appendix B for excerpts on fan noise from ASHRAE Guide.

TABLE 4

A. APPROXIMATE OCTAVE BAND SOUND POWER LEVELS OF PROPELLER TYPE COOLING TOWER IN dB re 10^{-12} WATT

OCTAVE FREQUENCY BAND (HZ)	4 to 8 HP	9 to 16 HP	17 to 32 HP	33 to 64 HP	65 to 128 HP	129 to 256 HP
31	96	99	102	105	108	111
63	101	104	107	110	113	116
125	101	104	107	110	113	116
250	96	99	102	105	108	111
500	93	96	99	102	105	108
1000	89	92	95	98	101	104
2000	86	89	92	95	98	101
4000	82	86	89	92	95	98
8000	78	81	84	87	90	93

B. APPROXIMATE OCTAVE BAND SOUND POWER LEVELS OF CENTRIFUGAL TYPE COOLING TOWER IN dB re 10^{-12} WATT

OCTAVE FREQUENCY BAND (HZ)	4 to 8 HP	9 to 16 HP	17 to 32 HP	33 to 64 HP	65 to 128 HP	129 to 256 HP
31	85	88	91	94	97	100
63	86	89	92	95	98	101
125	86	89	92	95	98	101
250	84	87	90	93	96	99
500	83	86	89	92	95	98
1000	81	84	87	90	93	96
2000	82	85	88	91	94	97
4000	76	79	82	85	88	91
8000	69	72	75	78	81	84

TABLE 5

APPROXIMATE CORRECTIONS TO AVERAGE SPLs FOR DIRECTIONAL EFFECTS OF COOLING TOWERS

(Add these decibel corrections to the average SPL calculated for a given distance from the tower. Do not apply these corrections for close-in positions, such as less than 10 ft. Also, these corrections apply when there are no reflecting or obstructing surfaces that would modify the normal radiation of sound from the tower.)

OCTAVE BAND (Hz)	31	63	125	250	500	1000	2000	4000	8000
CENTRIFUGAL-FAN BLOW-THROUGH TYPE									
Front	+3	+3	+2	+3	+4	+3	+3	+4	+4
Side	0	0	0	-2	-3	-4	-5	-5	-5
Rear	0	0	-1	-2	-3	-4	-5	-6	-6
Top	-3	-3	-2	0	+1	+2	+3	+4	+5
AXIAL-FLOW BLOW-THROUGH TYPE									
Front	+2	+2	+4	+6	+6	+5	+5	+5	+5
Side	+1	+1	+1	-2	-5	-5	-5	-5	-4
Rear	-3	-3	-4	-7	-7	-7	-8	-11	-8
Top	-5	-5	-5	-5	-2	0	0	+2	+1
INDUCED-DRAFT PROPELLER-TYPE									
Front	0	0	0	+1	+2	+2	+2	+3	+3
Side	-2	-2	-2	-3	-4	-4	-5	-6	-6
Top	+3	+3	+3	+3	+2	+2	+2	+1	+1
"UNDERFLOW" FORCED-DRAFT PROPELLER-TYPE									
Any side	-1	-1	-1	-2	-2	-3	-3	-4	-4
Top	+2	+2	+2	+3	+3	+4	+4	+5	+5

TABLE 6

ESTIMATED CLOSE-IN SOUND PRESSURE LEVELS (IN DB) FOR THE INTAKE AND DISCHARGE OPENINGS OF VARIOUS COOLING TOWERS

(3-ft to 5-ft Distance)

OCTAVE BAND (Hz)	31	63	125	250	500	1000	2000	4000	8000
CENTRIFUGAL-FAN BLOW-THROUGH TYPE									
Intake	85	85	85	83	81	79	76	73	68
Discharge	80	80	80	79	78	77	76	75	74
AXIAL-FLOW BLOW-THROUGH TYPE (INCLUDING "UNDERFLOW" TYPE)									
Intake	97	100	98	95	91	86	81	76	71
Discharge	88	88	88	86	84	82	80	78	76
PROPELLER-FAN INDUCED DRAFT TYPE									
Intake	97	98	97	94	90	85	80	75	70
Discharge	102	107	103	98	93	88	83	78	73

TABLE 7

ESTIMATED SOUND PRESSURE LEVELS (IN DB) OF
PUMPS (AT 3-FT DISTANCE INDOORS) AS A
FUNCTION OF POWER AND SPEED

PUMP RPM	RATED HP	31	63	125	250	500	1000	2000	4000	8000
		OCTAVE BAND FREQUENCY - HZ								
1600-3600	Under 12	77	77	80	82	82	80	77	74	69
	12-24	80	80	83	85	85	83	80	77	72
	25-49	83	83	86	88	88	86	83	80	75
	50-99	86	86	89	91	91	89	86	83	78
	100-199	89	89	92	94	94	92	89	86	81
	200-400	92	92	95	97	97	95	92	89	84
	Over 400	95	95	98	100	100	98	95	92	87
900-1599	Under 12	72	72	75	77	77	75	72	69	64
	12-24	75	75	78	80	80	78	75	72	67
	25-49	78	78	81	83	83	81	78	75	70
	50-99	81	81	84	86	86	84	81	78	73
	100-199	84	84	87	89	89	87	84	81	76
	200-400	87	87	90	92	92	90	87	84	79
	Over 400	90	90	93	95	95	93	90	87	82
450-899	Under 12	70	70	73	75	75	73	70	67	62
	12-24	73	73	76	78	78	76	73	70	65
	25-49	76	76	79	81	81	79	76	73	68
	50=99	79	79	82	84	84	82	79	76	71
	100-199	82	82	85	87	87	85	82	79	74
	200-400	85	85	88	90	90	88	85	82	77
	Over 400	88	88	91	93	93	91	88	85	80

TABLE 8

ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR CASING NOISE OF UNENCLOSED GAS AND DIESEL RECIPROCATING ENGINES*

ESTIMATED PWL = "BASE PWL" (from table below)
+ SPEED CORRECTION
+ FUEL CORRECTION (in dB re 10^{-12} watt)

CONTINUOUS RATING OF ENGINE HP	"BASE PWL" IN OCTAVE FREQUENCY BAND							
	63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
15-23	95	99	99	98	98	97	91	84
24-37	97	101	101	100	100	99	93	86
38-59	99	103	103	102	102	101	95	88
60-94	101	105	105	104	104	103	97	90
95-149	103	107	107	106	106	105	99	92
150-239	105	109	109	108	108	107	101	94
240-379	107	111	111	110	110	109	103	96
380-599	109	113	113	112	112	111	105	98
600-949	111	115	115	114	114	113	107	100
950-1499	113	117	117	116	116	115	109	102
1500-2399	115	119	119	118	118	117	111	104
2400-3800	117	121	121	120	120	119	113	106

For Engine Speed:	SPEED CORRECTION (in all bands)
Under 600 rpm	- 5 dB
600-1500 rpm	- 2 dB
Above 1500 rpm	0 dB

For Engine Fuel:	FUEL CORRECTION (in all bands)
Natural Gas Only**	- 3 dB
Liquid Fuel Only	0 dB
Gas and/or Liquid Fuel	0 dB

* This table is generally applicable for determining noise control designs of the casing noise of all engines, even though the actual PWL values would not hold for some large engines with unducted turbochargers or unmuffled Roots blowers opening directly into the room.

**With or without a small amount of "pilot oil".

TABLE 9

ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR UNMUFFLED EXHAUST NOISE OF GAS AND DIESEL RECIPROCATING ENGINES

ESTIMATED PWL = "BASE PWL" (from table below)
+ TURBOCHARGER CORRECTION
+ EXHAUST PIPE LENGTH CORRECTION
(in dB re 10^{-12} watt)

CONTINUOUS RATING OF ENGINE HP	"BASE PWL" IN OCTAVE FREQUENCY BAND 63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
15-23	122	128	124	116	112	106	96	88
24-37	124	130	126	118	114	108	98	90
38-59	126	132	128	120	116	110	100	92
60-94	128	134	130	122	118	112	102	94
95-149	130	136	132	124	120	114	104	96
150-239	132	138	134	126	122	116	106	98
240-379	134	140	136	128	124	118	108	100
380-599	136	142	138	130	126	120	110	102
600-949	138	144	140	132	128	122	112	104
950-1499	140	146	142	134	130	124	114	106
1500-2399	142	148	144	136	132	126	116	108
2400-3800	144	150	146	138	134	128	118	110

For Air Intake to Engine:	TURBOCHARGER CORRECTION (in all bands)
With Turbocharger	- 6 dB
Without Turbocharger	0 dB

For Exhaust Pipe Length from Engine:	EXHAUST PIPE LENGTH CORRECTION (in all bands)
0-2 ft	0 dB
3-6 ft	-1 dB
7-10 ft	-2 dB
11-14 ft	-3 dB
15-18 ft	-4 dB
19-22 ft	-5 dB
L ft	-L/4 dB

TABLE 10

ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR UNTREATED TURBOCHARGER NOISE AT AIR INLET OPENING OF GAS OR DIESEL RECIPROCATING ENGINE

ESTIMATED PWL = "BASE PWL" (from table below)
+ INLET AIR DUCT LENGTH CORRECTION
(in dB re 10^{-12} watt)

CONTINUOUS RATING OF ENGINE HP	"BASE PWL" IN OCTAVE FREQUENCY BAND							
	63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
15-23	90	88	88	89	92	93	92	84
24-37	91	89	89	90	93	94	93	85
38-59	92	90	90	91	94	95	94	86
60-94	93	91	91	92	95	96	95	87
95-149	94	92	92	93	96	97	96	88
150-239	95	93	93	94	97	98	97	89
240-379	96	94	94	95	98	99	98	90
380-599	97	95	95	96	99	100	99	91
600-949	98	96	96	97	100	101	100	92
950-1499	99	97	97	98	101	102	101	93
1500-2399	100	98	98	99	102	103	102	94
2400-3800	101	99	99	100	103	104	103	95

For Inlet Air Duct Length to Engine:	INLET AIR DUCT LENGTH CORRECTION (in all bands)
0-3 ft	0 dB
4-9 ft	-1 dB
10-15 ft	-2 dB
16-21 ft	-3 dB
22-27 ft	-4 dB
28-33 ft	-5 dB
L ft	-L/6 dB

TABLE 11

ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR CASING NOISE OF UNENCLOSED* GAS TURBINE ENGINE

CONTINUOUS RATING OF ENGINE KW	PWL (IN DB RE 10^{-12} WATT) IN OCTAVE FREQUENCY BAND 63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
200-329	111	113	114	114	114	114	114	114
330-529	112	114	115	115	115	115	115	115
530-849	113	115	116	116	116	116	116	116
850-1299	114	116	117	117	117	117	117	117
1300-1999	115	117	118	118	118	118	118	118
2000-3299	116	118	119	119	119	119	119	119
3300-5000	117	119	120	120	120	120	120	120

*If the entire engine casing is provided with a thermal insulating cover or an enclosing cabinet, the PWL values given here may be reduced by the following amounts (in dB for the octave bands indicated):

Type 1. Glass fiber or mineral wool thermal insulation with light-weight foil cover over the insulation:

2	2	3	3	3	4	5	6

Type 2. Glass fiber or mineral wool thermal insulation with minimum 20 gage aluminum or 24 gage steel or 1/2 in. thick plaster cover over the insulation:

5	5	6	6	7	8	9	10

Type 3. Enclosing metal cabinet for the entire packaged assembly, with open ventilation holes and with no acoustic absorption lining inside the cabinet:

1	1	2	2	2	2	3	3

Type 4. Enclosing metal cabinet for the entire packaged assembly, with open ventilation holes and with acoustic absorption lining inside the cabinet:

4	4	5	6	7	8	8	8

Type 5. Enclosing metal cabinet for the entire packaged assembly, with all ventilation holes into the cabinet muffled and with acoustic absorption lining inside the cabinet:

7	8	9	10	11	12	13	14

TABLE 12

A. ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR UNMUFFLED EXHAUST NOISE OF GAS TURBINE ENGINE

CONTINUOUS RATING OF ENGINE KW	PWL (IN DB RE 10^{-12} WATT) IN OCTAVE FREQUENCY BAND 63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
200-329	120	122	122	121	119	117	113	107
330-529	122	124	124	123	121	119	115	109
530-849	124	126	126	125	123	121	117	111
850-1299	126	128	128	127	125	123	119	113
1300-1999	128	130	130	129	127	125	121	115
2000-3299	130	132	132	131	129	127	123	117
3300-5000	132	134	134	133	131	129	125	119

B. ESTIMATED OCTAVE BAND SOUND POWER LEVEL (PWL) VALUES FOR UNMUFFLED AIR INTAKE NOISE OF GAS TURBINE ENGINE

CONTINUOUS RATING OF ENGINE KW	PWL (IN DB RE 10^{-12} WATT) IN OCTAVE FREQUENCY BAND 63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
200-329	102	103	103	106	112	117	117	114
330-529	105	106	106	109	115	120	120	117
530-849	108	109	109	112	118	123	123	120
850-1299	111	112	112	115	121	126	126	123
1300-1999	114	115	115	118	124	129	129	126
2000-3299	117	118	118	121	127	132	132	129
3300-5000	120	121	121	124	130	135	135	132

TABLE 13

ESTIMATED SOUND PRESSURE LEVELS (IN DB) OF ELECTRIC MOTORS
(AT 3-FT DISTANCE INDOORS) AS A FUNCTION
OF POWER AND SPEED

MOTOR RPM	RATED HP	OCTAVE BAND FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
2000-4000	Under 12	73	74	78	82	83	83	82	76	69
	12-24	78	79	83	87	88	88	87	81	74
	25-49	83	84	88	92	93	93	92	86	79
	50-99	87	88	92	96	97	97	96	90	83
	100-200	90	91	95	99	100	100	99	93	86
	Over 200	93	94	98	102	103	103	102	96	89
1000-1990	Under 12	68	69	73	77	78	78	77	71	64
	12-24	73	74	78	82	83	83	82	76	69
	25-49	78	79	83	87	88	88	87	81	74
	50-99	82	83	87	91	92	92	91	85	78
	100-200	85	86	90	94	95	95	94	88	81
	Over 200	88	89	93	97	98	98	97	91	84
450-990	Under 12	64	65	69	73	74	74	73	67	60
	12-24	69	70	74	78	79	79	78	72	65
	25-49	74	75	79	83	84	84	83	77	70
	50-99	78	79	83	87	88	88	87	81	74
	100-200	81	82	86	90	91	91	90	84	77
	Over 200	84	85	89	93	94	94	93	87	80

TABLE 14

ESTIMATED SOUND PRESSURE LEVELS (IN DB) OF STEAM TURBINES (AT 3-FT DISTANCE) AS A FUNCTION OF POWER RATING

RATED HP	RATED KW	31	63	125	250	500	1000	2000	4000	8000
500-1500	333-1000	88	93	95	91	87	87	88	85	80
1501-5000	1001-3333	90	95	97	93	89	90	92	89	85
5001-15000	3334-10000	92	97	99	95	91	93	96	93	90

TABLE 15

ESTIMATED SOUND PRESSURE LEVELS (IN DB) OF GEARS AT 3-FT DISTANCE

Values apply to 125-8000 Hz octave bands
Deduct 3 dB for 63 Hz octave band
Deduct 6 dB for 31 Hz octave band

SPEED OF SLOWER GEAR SHAFT RPM	POWER RATING OF GEAR IN HP							
	125 to 249	250 to 499	500 to 999	1000 to 1999	2000 to 3999	4000 to 7999	8000 to 15999	16000 to 32000
125-249	94	95	96	97	98	99	100	101
250-499	95	96	97	98	99	100	101	102
500-999	96	97	98	99	100	101	102	103
1000-1999	97	98	99	100	101	102	103	104
2000-3999	98	99	100	101	102	103	104	105
4000-7999	99	100	101	102	103	104	105	106
8000-16000	100	101	102	103	104	105	106	107

TABLE 16

EXTIMATED MAXIMUM SOUND PRESSURE LEVELS OF A TRANSFORMER AT 3-FT DISTANCE

First, obtain or estimate the NEMA Sound Level Rating for the Transformer. (This is an average of several A-scale readings taken at certain specified positions at a 1-ft distance from the transformer surfaces or at a 6-ft distance from the forced-air ventilated surfaces.)

OCTAVE FREQUENCY BAND (Hz)	Add the following values to the NEMA Sound Level Rating. The resulting values are sound pressure levels in dB re 0.0002 microbar
31	0
63	5
125	10
250	17
500	14
1000	9
2000	4
4000	-1
8000	-6

TABLE 17

ESTIMATED SOUND PRESSURE LEVELS (IN DB) AT 3-FT DISTANCE

DUE TO RECIPROCATING AND CENTRIFUGAL AIR COMPRESSORS

AIR COMPRESSOR POWER RANGE (HP)	OCTAVE BAND CENTER FREQUENCY - HZ								
	31	63	125	250	500	1000	2000	4000	8000
1-2	85	83	83	83	86	89	89	89	84
3-9	90	86	86	86	89	92	92	92	87
10-100	95	89	89	89	92	95	95	95	90

TABLE 18

RULES FOR ADDING SPL OR PWL CONTRIBUTIONS BY "DB ADDITION"

1. For adding any two decibel levels together-

When two decibel values differ by:	Add the following amount to the higher value:
0 or 1 dB	3 dB
2 or 3 dB	2 dB
4 to 8 dB	1 dB
9 dB or more	0 dB

2. If there are several levels of the same value, add as follows:

No. of equal levels	Add	No. of equal levels	Add
2	3 dB	6-7	8 dB
3	5 dB	8	9 dB
4	6 dB	9-10	10 dB
5	7 dB	N	10 log N dB

3. The individual components can be added in any order. The total, using this procedure, will give an answer correct to within 1 dB.

4. When combining the frequency contributions of different sources, add only noise levels from the same octave frequency band.

TABLE 19

"AREA FACTOR" ("AF") FOR USE IN DETERMINING THE PWL OF AN AREA "A" THAT TRANSMITS SOUND LEVEL SPL

$$\text{PWL (in dB re } 10^{-12}\text{w)} = \text{SPL} + (10 \log A - 10)$$
$$= \text{SPL} + \text{"AF"}$$

AREA "A" (sq ft)	"AF" (dB)	AREA "A" (sq ft)	"AF" (dB)	AREA "A" (sq ft)	"AF" (dB)
1.0	-10	6.3	-2	40	6
1.25	-9	8	-1	50	7
1.6	-8	10	0	63	8
2.0	-7	12.5	1	80	9
2.5	-6	16	2	100	10
3.2	-5	20	3	125	11
4.0	-4	25	4	160	12
5.0	-3	32	5	200	13

CHAPTER 4

CONTROL OF AIRBORNE NOISE OF MECHANICAL AND ELECTRICAL EQUIPMENT

The objective of this chapter is to provide assistance in the acoustic design of the Mechanical Equipment Room (MER), so that the airborne noise that escapes from that room is not disturbing to occupants of the rooms above, below and beside the MER nor to neighbors outside the building. In effect, it is necessary to know (1) the noise levels made by the equipment inside the MER, (2) the desired noise levels for the areas immediately adjoining the MER, and (3) the noise reduction that can be provided between the noisy MER and the quieter adjoining rooms by such structures as walls, floors, ceilings, doors, corridors and other acoustic treatments.

1. SOUND DISTRIBUTION IN A ROOM

a. SPL Variation with Distance. It is generally true that the sound pressure level (SPL) drops off as one moves away from the sound source. In an outdoor "free-field" situation (no reflecting surfaces except the ground), the SPL drops off at the rate of 6 dB for each doubling of distance from the acoustic center of the source (there are qualifications to this generalization that can be ignored for the present). In an indoor situation, all the enclosing surfaces of a room confine the sound waves so that they cannot continue spreading out indefinitely and become dissipated with distance. Instead, as the sound waves bounce around within the room, a certain amount of energy is absorbed at each reflection but, in general, there is a build-up of sound level because the sound energy is "trapped" inside the room and cannot escape (somewhat figuratively speaking). In a highly reverberant room, with walls that are hard, rigid and completely impervious, very little sound energy is absorbed at each reflection so the sound bounces around a long time before it ultimately is absorbed. In this type of room, the room becomes almost "saturated" with sound; and as one moves away from the sound source, the sound level drops off very slowly with distance (possibly only ½ to 1 dB per doubling of distance for some relatively small, but very reverberant rooms). In a highly absorptive room, however, a considerable amount of energy is absorbed at each reflection as the sound waves bounce around the room. There is less build-up of sound within the room; and as one moves away from the sound source, the sound level drops off more rapidly (possibly 2 to 4 dB per doubling of distance). Note that the walls would have to be 100% absorptive in order to have no reflected sound at all. This would then simulate the outdoor free-field condition, that requires no reflecting surfaces, and the sound level drop-off with distance would become the theoretical maximum of 6 dB per doubling of distance.

Thus, in a qualitative sense, it is seen that the reduction of sound pressure level indoors, as one moves across the room away from the sound source, is dependent on the degree of absorption and, of course, on the distance that one moves. The amount of absorption also involves surface areas of the room. All of this is expressed quantitatively by the curves of Figure 1 at the end of this chapter. As an example of the use of Figure 1, suppose a room has an amount of sound absorption that produces a "Room Constant, R" value of 1000 sq ft. At a distance of 2½ ft from the acoustic center of a non-directional sound source, the "RELATIVE SPL", as read off the left-hand side of the graph for the R=1000 curve, is -7½ dB. at a 5-ft distance, the REL SPL becomes -11 dB, indicating a reduction of 3½ dB as one doubles the distance in going from 2½ to 5-ft distance. Continuing, at a 10-ft distance, the REL SPL becomes -13 dB, indicating a reduction of 2 dB as one doubles the distance from 5 ft to 10 ft. Then, at a 20-ft distance, the REL SPL becomes -14 dB, indicating a reduction of only 1 dB as one doubles the distance from 10 ft to 20 ft. The other curves for other values of Room Constant (related to room absorption) give other variations of SPL with distance away from the source. Only if a room has an infinite Room Constant (perfect sound absorption at all the side wall and ceiling surfaces), would the sound pressure level drop off indefinitely at the outdoor rate of 6 dB per doubling of distance.

It is seen that Figure 1 offers a means of estimating the amount of noise level reduction for a piece of mechanical equipment in a room as one moves from the 3-ft distance (used as the SPL reference distance in many of the data summaries of Chapter 3) to any other distance in the room, provided one knows the Room Constant of that room. Obviously, the next step is to calculate or estimate the value of the Room Constant.

<u>b</u>. <u>Room Constant</u>. A suitable acoustics textbook will give details of a fairly accurate calculation of the Room Constant for any specific room, knowing (1) all the room dimensions, (2) the wall, floor and ceiling materials, (3) the amount and type of acoustic absorption materials, and (4) the sound absorption coefficients of the acoustic materials at various specified frequencies. For the purpose of these notes, however, such a high degree of accuracy is not considered necessary, so a simplified estimating procedure is suggested. It must be recognized that this simplification yields a less accurate estimate than does the more detailed textbook procedure, but it is nevertheless considered acceptable for use here. The basic steps of the simplified procedure are listed as follows:

1. Determine the total interior surface area of the room.
2. Determine the total area of acoustic absorption material to be applied to the walls and/or ceiling of the room.
3. From steps 1 and 2, determine the percentage of total room surface covered with absorption material.
4. From Part A of Table 1 determine the "room label" associated with the percentage figure found in step 3 above.
5. Calculate the volume of the room, in cu. ft.
6. From Figure 2 (at the end of this chapter), using the volume of step 5 and the "room label" of step 4, determine the approximate Room Constant (R in sq. ft) for the room. This value applies for octave band frequencies of 500-8000 Hz.
7. Determine the corrected values of R for 31-125 Hz as given in Part B of Table 1. The values differ depending on the type of acoustic treatment used. See the footnotes of Table 1 regarding "NRC" values normally associated with 1 in. and 2 in. thick acoustic absorption materials.

c. Example. Assume a room 40 ft long, 30 ft wide and 15 ft high. The total interior surface area is 4500 sq. ft and the volume of the room is 18,000 cu. ft. Suppose 2 in. thick acoustic panels having an NRC of 0.80 are used over the full ceiling area and in a 5-ft wide band around all four walls. The total area of acoustic treatment is 1900 sq. ft, giving 42% area coverage. In Table 1, 42% is seen to fall about midway between a "Medium-Dead Room" and a "Dead Room". In Figure 2, for a room volume of 18,000 cu. ft and a room label between "Medium-Dead" and "Dead", the value of R is found to be approximately 2000 sq. ft. This value would apply for 500-8000 Hz. At lower frequencies, the value of the corrected R would be (from Part B of Table 1):

0.2 R or 400 sq. ft at 31 Hz,
0.3 R or 600 sq. ft at 63 Hz,
0.5 R or 1000 sq. ft at 125 Hz,
0.8 R or 1600 sq. ft at 250 Hz.

Continuing this example, suppose it is desired to find the SPL reduction in this room while going from 3-ft to 20-ft distance from the noise source. In Figure 1, find the difference in REL SPL between 3 ft and 20 ft for R values of:

400, 600, 1000, 1600 and 2000 sq. ft.

These are as follows, in order:

3 dB 4 dB 5 dB 6 dB and 7 dB.

Thus, the 3-ft SPLs for the particular piece of equipment would be reduced by these amounts to obtain the 20-ft SPLs for the frequency bands, in order:

31, 63, 125, 250 and 500-8000 Hz.

d. SPL in a Room when PWL is Known. The above uses of Table 1 and Figures 1 and 2 assume that a 3-ft SPL is known for a given machine and it is desired to find the SPL of that machine at any distance (greater than 3 ft) within any room whose dimensions and acoustic absorption are known or can be estimated. That procedure was illustrated in the paragraphs above.

In the event that the sound power level (PWL) of some piece of equipment is known (rather than the 3-ft SPL), the same procedure may be used, with one small exception. In Figure 1, the ordinate of the graph, "Relative Sound Pressure Level" (abbreviated to "REL SPL") is actually related to SPL and PWL by the equation

$$\text{SPL} = \text{PWL} + \text{REL SPL}$$

for any particular Distance D and Room Constant R. In this equation, SPL is given in the standard unit "dB re 0.0002 microbar", PWL is given in the standard unit "dB re 10^{-12} watt", and REL SPL is quoted in decibels and is the conversion term that relates SPL to PWL. Sound power levels (PWLs) are given in Chapter 3 for some equipment, while SPLs are given for other equipment. In the above equation, the REL SPL is read directly off the curve of Figure 1 for a particular D and R value. Then, if the PWL is known, the SPL can be calculated.

e. Example. Suppose a hermetic centrifugal compressor is to be installed in the acoustically treated room described above and suppose it is desirable to find the SPL at a distance of 20 ft from the compressor. For this example, suppose that the compressor manufacturer submits PWL data for this unit. The

PWL values are listed in Column 2 of the accompanying table. It was learned above that the Room Constant had the values 400, 600, 1000, 1600 and 2000 sq. ft at the various frequencies. From Figure 1, REL SPL values can be determined for the particular Room Constant values at a 20-ft distance. These values are shown in Column 3 of the table below. Finally, since

$$SPL = PWL + REL\ SPL,$$

the SPLs can be calculated. These are listed in Column 4.

Col. 1 Octave Band (Hz)	Col. 2 PWL (dB re 10^{-12} w)	Col. 3 REL SPL (dB)	Col. 4 SPL at 20 ft (dB re 0.0002 microbar
31	95	-10	85
63	93	-12	81
125	94	-14	80
250	95	-16	79
500	99	-17	82
1000	102	-17	85
2000	108	-17	91
4000	105	-17	88
8000	94	-17	77

f. Qualifications. There are two points that should be kept in mind in using the data of Figure 1. These are both suggested by the caption under the abscissa of the graph: "Equivalent distance from acoustic center of a non-directional source". Strictly speaking, very few noise sources in real life are completely non-directional sources, but in this write-up and in many conventional noise problems the assumption is made that the source is non-directional, that is, that it radiates sound equally in all directions. If the true directional characteristics are known, they may be used, but for the present purpose this is not required. The second point regards the "distance from the acoustic center." The acoustic center, as the term implies, is the location that would be occupied by a "point source" of equal sound power output. The acoustic center of a noise source may be at the nearest surface of the unit being measured, or it may be located somewhere near the geometric center inside the unit. For a strictly correct use of Figure 1, the distance should be referred to the acoustic center, but in practice the location of the center is not always obvious. Hence, for practical purposes, it is suggested that distances be related to the nearest external surface that is generally considered to be the noisiest part of the unit. This will yield consistent and reasonably accurate results.

g. Simplified Table for Distance and Room Constant. The preceding paragraphs show the normal procedure for estimating the effect of SPL drop-off with distance as one moves away from a noise source in a room having an estimated Room Constant. The material given in Figure 1 is placed in a simpler form in Table 2 for the specific condition of estimating the SPL drop-off from the normalized 3-ft distance given for much of the equipment in this manual. Obviously, not all distances nor all Room Constants can be included in Table 2, but enough values of D and R are included to cover a wide range of usage.

Various intermediate values of D and R can be determined by interpolation within Table 2 or by using Figure 1. It is cautioned that Table 2 must not be used to estimate an SPL value when the PWL of the noise source is given.

To illustrate the use of Table 2, recall the example given earlier. In that example a room was found to have the following Room Constant values

400, 600, 1000, 1600 and 2000 sq. ft

for the octave bands

31, 63, 125, 250 and 500-8000 Hz.

It was desired to find the SPL reduction in going from 3 ft out to 20 ft. Using Figure 1 it was necessary to determine the REL SPL at 3 ft and the REL SPL at 20 ft, then subtract one value from the other to obtain the SPL reduction at the greater distance. In Table 2 this is simplified merely to reading the SPL reduction for the particular values of R and D involved. Again, note that this table applies only when the "starting distance" is the normalized 3-ft distance. From Table 2 for a distance of 20 ft and for the Room Constants listed, the SPL reduction is found to be

3 dB, 4 dB, 5 dB, 6 dB and 7 dB.

These values agree with those obtained by the longer procedure of reading and subtracting two values each from Figure 1. Figure 1 still must be used, however, when converting from PWL to an SPL at some specified distance.

h. Example: Effectiveness of Acoustic Absorption vs Distance. Acoustic absorption in a room will not solve all noise problems, but it can be very helpful for certain predictable situations. The reader is given the following exercise. Suppose a large mechanical room or manufacturing space is 100 ft long, 50 ft wide and 20 ft high. Calculate the Room Constant for four conditions: (1) for Condition 1 there is no acoustic absorption material in the room; (2) for Condition 2 the entire ceiling area is covered with a 1 in. thick acoustic absorption ceiling panel (NRC = 0.65 to 0.74); (3) for Condition 3 the entire ceiling area and one-half the side wall area is covered with a 1 in. thick acoustic panel (NRC = 0.65 to 0.74); and (4) for Condition 4 the entire ceiling area is covered with a 2 in. thick acoustic panel (NRC = 0.75 to 0.85). Assume that the noise level at the operator position of a machine in that room is 90 dB in all the octave bands when there is no room absorption; assume that the operator position is 3 ft from the "acoustic center" of the machine when using Figure 1. Next, determine the noise levels that would exist in that room at distances of 3 ft (the operator position), 10 ft and 50 ft from the acoustic center of the machine for the four conditions of acoustic absorption. The answers are summarized in the table on the following page.

The simplifications used in this procedure introduce possible errors of 1 or 2 dB (perhaps even 3 dB for some situations), so extreme accuracy should not be expected. However, a few obvious points from this example should be noted. First, in the high frequency region (where hearing protection is usually most important), the use of acoustic absorption on the ceiling and side walls gives relatively little protection to the operator who works only 3 ft from his own machine. Also, in a completely non-absorbent room, the SPLs do not drop off very much with distance from the machine. With acoustic absorption present, however, noise levels drop off noticeably as one moves away from the noise source. Thus, when an operator is exposed to the combined noise of several machines in the room, at least some portions of that total noise can be reduced with an application of acoustic absorption.

Octave Frequency	Condition 1	Condition 2	Condition 3	Condition 4
SPLs at 3-ft distance:				
31	90	86	85	86
63	90	86	85	85
125	90	86	86	86
250	90	87	87	87
500-8000	90	88	88	88
SPLs at 10-ft distance:				
31	89	82	80	82
63	89	82	80	81
125	89	81	80	80
250	88	81	79	80
500-8000	87	81	80	81
SPLs at 50-ft distance:				
31	89	81	78	81
63	89	81	78	79
125	89	80	77	78
250	88	78	75	77
500-8000	85	77	74	77

2. TRANSMISSION LOSS OF WALLS

Paragraph 1 above considered the distribution of sound inside a room that contains the sound source. Next it is essential to know the amount of sound that escapes from that room into adjoining spaces by way of the walls of the room containing the sound source.

a. "The Mass Law". When a sound wave strikes the "front" surface of a solid wall, there is enough energy in the tiny pressure oscillations in the air to cause the whole wall to vibrate. If the wall is relatively lightweight, it will be set into vibration more easily than if it is heavy-weight. In vibrating as a whole, this wall sets into oscillation the air particles along its "rear" or opposite surface. These vibration air particles radiate as sound energy into the space on this rear side of the wall. Thus, an incident sound wave excites the front side of the wall, and the wall re-radiates the sound wave from its rear side. (If the wall is at all porous, some sound--oscillating air particles--can actually pass through the pores of the wall.)

It is generally true that a lightweight wall will be more easily excited by an incident sound wave than will a heavyweight wall and therefore will "transmit" more radiated energy to the other side. This generalization gives rise to the effect known as "the mass law" in acoustics. To a first approximation, "the mass law" suggests that for each doubling of the surface weight of the wall there will be about 5 or 6 dB less transmitted sound. The mass law also suggests that for each doubling of the frequency of the sound there will be about 5 or 6 dB less transmitted sound. There are some qualifications to these generalities which will not be discussed here, but the "transmission loss" data given in the tables reflect these effects.

b. Transmission Loss (TL). The approximate "transmission loss" or "TL" values, expressed in dB, of a number of typical wall constructions are given in Tables 3-13.

Table No.	Construction Material
3	Solid, dense concrete or masonry
4	Hollow-core concrete or masonry
5	Stud-type partitions
6	Metal panel partition and industrial acoustic doors
7	Glass walls or windows
8	Double-glass construction
9	Wood or plywood, including 2-in. thick solid wood door
10	Plaster
11	Aluminum
12	Steel
13	Lead

The values given in these tables encompass many more materials than normally required for straight-forward noise control problems, but they are offered for the benefit of the architect or engineer who might wish to consider certain special designs or applications. They are included also to show that certain lightweight wall materials can not adequately confine the high noise levels of some mechanical equipment.

It is important to realize that the TL of a wall is merely the ratio, expressed in decibels, of the sound transmitted by a wall to the airborne sound incident upon the wall. Thus, the TL of a wall is a performance characteristic that is entirely a function of the wall weight and material, and its numerical value is not influenced by the acoustic environment on either side of the wall or the area of the wall.

c. Noise Reduction ("NR"). The total effectiveness of a wall or partition involves both the TL of the wall and certain other factors associated with the geometry and the acoustic characteristics of the "receiving room", that is the room into which the noise is transmitted. These factors are reasonably self-evident. For example, it is probably obvious that a wall with a relatively small area will transmit less total noise energy than will a wall with a relatively large area, even though each square foot of the wall has the same TL value. Also, it is probably obvious that the sound level in the "receiving room" will be influenced by the amount of acoustic absorption in the receiving room; that is, the SPL will be relatively high in a "live" receiving room having little or no acoustic absorption whereas it will be relatively low in a "dead" receiving room having large amounts of acoustic absorption.

Thus, when noise travels through a wall from one room (the "source room") to an adjoining room (the "receiving room"), three factors are involved: (1) the TL of the wall, (2) the area of the wall that is common to both rooms and that is transmitting the noise, and (3) the acoustic characteristics of the receiving room that receives the transmitted noise. The term "noise reduction" of a wall (abbreviated to "NR") is the term that includes all three of these factors. In the manual, the area of the common transmitting wall and the acoustic characteristics of the receiving room are combined into a single term, called here "the wall correction term" and designated as "C" in the equation:

$$NR = TL + C.$$

For this equation, values of TL are found in Tables 3-13 and values of C are found in Table 14. The "noise reduction" of that specific wall between the transmitting room and the receiving room is now known. The SPL in the receiving room can then be determined from

$$SPL_{\substack{\text{source}\\\text{room}}} - NR = SPL_{\substack{\text{receiving}\\\text{room}}}$$

since the SPL in the source room can be calculated from the procedures given in Paragraph 1 above.

The "wall correction term" C in table 14 depends on the ratio Sw/R_2, where Sw is the area in sq. ft of the common wall between the two rooms and R_2 is the Room Constant of the receiving room. This Room Constant can be determined from Table 1 and Figure 2.

d. Example: Control Room in MER. Suppose a glass-walled Control Room is to be located at one end of a mechanical equipment room. The MER is 80 ft long, 40 ft wide and 20 ft high and has a 2-in. thick acoustic and thermal insulation treatment applied directly to its entire ceiling area. Assume the NRC (noise reduction coefficient) of the material is 0.75. The Control Room is 20 ft long, 12 ft wide and 8½ ft high. It has an acoustic tile ceiling supported on a suspension system that provides an 18-in. air space above the ceiling. Suppose this ceiling combination has an NRC of 0.85 according to the Acoustical Materials Association Bulletin. The 20 ft dimension of the Control Room lies along the 40 ft width of the MER and a ¼-in. thick glass wall is planned as the common wall extending from the floor line to an 8-ft height. Assume that the SPL on the MER side of the glass wall to the Control Room is as follows for the nine octave frequency bands:

88	90	92	93	93	90	85	80	75 dB.

It is desired to know the SPL in the Control Room.

The volume of the MER is 80x40x20=64,000 cu. ft and the total interior surface area is 11,200 sq. ft. The area of the ceiling acoustic treatment is 3200 sq. ft, which amounts to 29% of the total room area. According to Table 1A and Figure 2, this room has a Room Constant of approximately 3000 sq. ft at 500-8000 Hz. According to Table 1B, the Room Constant at lower frequency is

600 sq. ft at 31 Hz
900 sq. ft at 63 Hz
1500 sq. ft at 125 Hz
2400 sq. ft at 250 Hz.

The area of the glass wall (Sw) separating the two rooms is 20x8=160 sq. ft. From Table 14, values of C can be determined for the various ratios of Sw/R_2.

These are summarized in Table A immediately below.

TABLE A

Octave Band (Hz)	Common Wall Area Sw	Receiving Room Constant R_2	Ratio Sw/R_2	C from Table 14 (dB)
31	160	50	3.2	-5
63	160	75	2.1	-4
125	160	125	1.3	-2
250	160	200	.80	0
500-8000	160	250	.64	+1

The TL of ¼ in. thick glass can be found in Table 7, and the NR for this glass wall can then be determined from the relationship

$$NR = TL + C.$$

This is summarized in Table B immediately below.

TABLE B

Octave Band (Hz)	TL ¼ in. glass (dB)	C (dB)	NR (dB)
31	5	-5	0
63	11	-4	7
125	17	-2	15
250	23	0	23
500	25	1	26
1000	26	1	27
2000	27	1	28
4000	28	1	29
8000	30	1	31

Now, knowing the SPL on the MER side of the glass wall and the NR of the glass wall, the SPL inside the Control Room can be estimated from

$$SPL_{\substack{\text{receiving}\\ \text{room}}} = SPL_{\substack{\text{source}\\ \text{room}}} - NR$$

This is shown in Table C below.

TABLE C

Octave Band (Hz)	SPL in MER	NR of Glass Wall	SPL in Control Room	
31	88	0	88	
63	90	7	83	
125	92	15	77	
250	93	23	70	
500	93	26	67	PSIL = 62 dB
1000	90	27	63	
2000	85	28	57	
4000	80	29	51	
8000	75	31	44	

This Control Room will have a "speech interference level" of approximately 62 dB, and according to Table 3 of Chapter 2, this would permit reliable speech communication with a normal voice at 2-ft distance, a raised voice at 4-ft distance, or a very loud voice at an 8-ft distance. There would be no hearing damage problem, as may be seen by comparing the Control Room SPLs with Table 7 in Chapter 2. If there were no glass-wall enclosure for the Control Room, the noise levels would reach 85 to 93 dB in the upper frequency bands and these could be a cause for concern for occupants stationed in the area.

It is of interest to note at this point the acoustic value of the 2-in. thick acoustic and thermal insulation material placed in the ceiling of the MER. For a brief comparison, suppose that no acoustic absorption were used in the MER. Recall that the volume of the MER in this example is 64,000 cu. ft and that the total interior surface area is 11,200 sq. ft. Now, assume there is no added acoustic absorption. From Table 1A and Figure 2, the MER is now found to have a Room Constant of approximately 500 sq. ft at 500-8000 Hz. At the lower frequencies, according to Table 1B,

0.2 R = 100 sq. ft at 31 Hz
0.2 R = 100 sq. ft at 63 Hz
0.3 R = 150 sq. ft at 125 Hz
0.5 R = 250 sq. ft at 250 Hz

In Figure 1, assuming a distance of 30 ft, it is seen that for 63 Hz (where the Room Constant changes from 100 sq. ft to 900 sq. ft with the addition of ceiling absorption), the REL SPL drops from -4 dB to -14 dB, or a reduction of 10 dB. For 500-8000 Hz, the Room Constant changes from 500 to 3000 sq. ft for a REL SPL change from -11 dB to -19 dB for a reduction of 8 dB. At shorter distances, the reduction would not be as large, due to the use of acoustic absorption.

Other examples can be worked out using specific noise sources from Chapter 3, specific distances across the room and specific wall designs to separate the noisy and quiet areas.

e. Doors and Windows. It is fairly obvious that a poorly-fitting lightweight door or a large lightweight window might constitute a weak link in an otherwise acoustically good wall. When a wall must serve an important acoustic need, then the door or window must be carefully selected to be compatible with the total need of the wall.

Because the area of a door or window is usually quite a small part of the total area of a wall, the TL of the door or window can be lower than that of the wall by certain specified amounts without seriously jeopardizing the acoustic effectiveness of the wall. In Table 15, the reduction in TL of a wall is given for a range of areas of doors and windows and for a relative TL of the door or window compared to that of the wall. As an example, suppose that a wall has a TL of 40 dB at a particular frequency and that a door has a TL of 20 dB at the same frequency. Suppose the door area is 5% of the total wall area. In Table 15, it is found for this combination of conditions that the wall TL would be reduced by 8 dB by this door. Thus, the composite wall-door combination would have an effective TL of 40-8=32 dB.

To minimize the loss of effectiveness of a wall, the door or window should be of the smallest possible area and of the largest possible TL. Doors should be gasketed and provided with a drop strip in order to minimize air leakage paths, and windows should be sealed closed. For massive single walls or for special double walls, double doors or windows should be used and large air spaces should be provided between the doors and windows. The approximate TL of a 2-in. solid wood door, gasketed around all edges, is given in Table 9 (see Footnote 2), and the approximate TL of a 4-in. thick and a 6-in. thick industrial type "acoustic door" is given in Table 6. The approximate TLs of single thicknesses of glass are given in Table 7 and the TLs of a few double glass combinations are given in Table 8.

In many situations, the structural requirements will exceed the acoustical requirements, in which case the door or window can have a TL much lower than that of the wall. A few generalizations are listed below that should aid in the selection of a door or window that will be somewhat acoustically compatible with the wall, even though the door or window TL may not meet the values suggested above as a function of their area relative to the total wall area:

(1) Where the acoustic design requires a minimum, simple, single wall construction, such as conventional stud partitions, movable metal partitions or 4-in. or 6-in. hollow-core concrete block, use ungasketed hollow-core wood doors or ungasketed metal panel doors and minimum ¼-in. thick glass windows.

(2) Where the acoustic design requires somewhat more than minimum wall construction (such as staggered stud construction, 4-in. or 6-in. solid core concrete or masonry, or acoustically filled metal panel partitions), use gasketed solid-core wood doors, or minimum 1-3/4 in. hollow metal doors packed with dense mineral or glass fiber, or special 1-3/4 in. to 2-in. thick acoustic doors with gasketing, and use windows of minimum area made up of double panes of at least 1/4-in. thick glass with at least 2-in. air space, or windows of larger but limited area made up of double panes of at least 1/4-in. thick glass with 4-in. to 6-in. air space.

(3) Where stringent acoustic requirements must be met, adhere to the door or window TL requirements given above as a function of percent area of the total wall. Use special acoustic doors or provide "sound locks" with gasketed double doors, as in Item (2) immediately above, such that doors are spaced at least 5 to 6 ft apart in an acoustically lined vestibule or corridor. Use double glass windows with maximum possible air space and glass thickness and minimum practical area. For slight improvement, the panes may be tilted relative to one another and the interior surfaces of the window framing can be given an acoustic lining.

(4) Where doors are obvious leakage paths for unwanted noise, locate them in positions that will provide minimum disturbance or maximum distance from the important work area of the room, and provide acoustic absorption in the room.

Table 15 can also be used to determine the effective TL of a wall made up of two different portions, where the two portions have different TLs, such as in a 10-in. thick poured solid concrete wall having a knock-out panel of 6-in. thick concrete block.

f. Double Walls. If MERs are bordered by work spaces where a moderate amount of noise is acceptable (such as areas of Categories 5 and 6 and possibly in some cases Category 4), the equipment noise can be adequately contained by heavy concrete walls of single thickness. Double walls of concrete can be used to achieve even greater values of TL. For example, two 8-in. thick solid-core concrete block walls separated with an 8-in. air space and structurally not connected together at any point (based on separate footings) would have TL values about 5 dB higher in the low frequency region, 10 dB higher in the middle frequency region and 15 dB higher in the high frequency region than a single 12-in. thick solid-core concrete block wall. Various intentional and unintentional structural connections between double walls have highly varying effects on the TL of double walls, however. For this reason, TL data are not quoted for double walls. In practice, double walls will give a worthwhile improvement over single walls if one of the double walls can be placed on separate footings (for an on-grade location) or on a 1-in. or 2-in. thick layer of construction cork (for upper floor locations), and if the two walls can have a minimum of structural ties. The improvement will be greatest at high frequency. The air space between the walls should be as large as possible to enhance the low frequency improvement. An obvious extension of the double wall is a wide corridor, with an acoustically treated ceiling. This is recommended as a separator between a noisy MER and a Category 2-4 area and possibly a Category 1 area. For close locations of acoustically critical areas to noisy MERs, it is essential that adequate vibration isolation be incorporated in all the machinery and piping. If a Category 1 area (NC-20 to NC-25) is to located very near a noisy MER, it would be advisable to have an acoustical engineer check the details of the designs.

It is sometimes possible to enhance the TL of a simple concrete block wall or a stud-type partition by resiliently attaching to that wall or partition additional layers of plaster skin, possibly mounted on spring clips that are installed off 1-in. or 2-in. thick furring strips, with the resulting air space filled with acoustic absorption material. These constructions become rather sophisticated and a bit expensive, but they can provide an improvement in TL of 5-10 dB in the middle frequency region and 10-15 dB in the high frequency region, when properly executed.

3. TRANSMISSION LOSS OF FLOOR-CEILING COMBINATIONS

Many mechanical equipment areas are located immediately above or below occupied floors of buildings. Airborne noise and structure-borne vibration radiated as noise may intrude into these occupied floors if adequate controls are not included in the building design. The approximate "TL" and "NR" are given in this part of the manual for five floor-ceiling combinations frequently used to control airborne machinery noise to spaces above and below the MER.

None of the data apply for equipment installations mounted in framed wood flooring or on typical lightweight metal deck with 2-3 in. thick concrete surface. These floor constructions are not stiff enough or massive enough to provide good airborne noise control or to support heavy machinery or to give an adequate base for a vibration isolation mounting system.

The five floor-ceiling combinations are discussed in the following paragraphs. All floor slabs are assumed to be of <u>dense concrete</u> (140-150 lb/cu. ft density) or of such extra thickness of less dense concrete to give the equivalent surface weight of the specified dense concrete.

<u>a</u>. <u>Type 1 Floor-Ceiling</u>. This combination is made up of a concrete floor slab with acoustic tiles or panels cemented directly to the underside of the slab. It is important to realize that the acoustic tiles add nothing to the transmission loss of the floor slab. The acoustic tiles only provide acoustic absorption in the room in which they are located and hence provide a degree of noise reduction in the room. The estimated TL of a Type 1 floor-ceiling is given in Table 16 for a few typical floor slab thicknesses.

<u>b</u>. <u>Type 2 Floor-Ceiling</u>. This floor-ceiling combination consists of a concrete floor slab below which is suspended a typical low density acoustic tile ceiling in a mechanical support system. To qualify for the Type 2 combination the acoustic tile should be not less than 3/4 in. thick, and it should have a Noise Reduction Coefficient ("NRC") of at least 0.65 (when mounted as specified by the Acoustical Materials Association). The air space between the suspended ceiling and the concrete slab above should be at least 15 in., but the TL improves if the air space is larger than this. The estimated TL of a Type 2 floor-ceiling is given in Table 17 for a few typical dimensions of concrete floor slab thickness and air space.

<u>c</u>. <u>Type 3 Floor-Ceiling</u>. This floor-ceiling combination is very similar to the Type 2 combination, except that the acoustic tile material is of the "high TL" variety. This means that the material is of high density and usually has a foil backing to decrease the porosity of the back surface of the material. (Ask the acoustic tile representative to identify his "high TL" material.) An alternate version of the Type 3 combination includes the suspended ceiling system that consists of a lightweight metal panel sandwich construction consisting of a perforated panel on the lower surface and a solid panel on the upper surface, with acoustic absorption material in between. The minimum "NRC" for the Type 3 acoustic material must be 0.65. The estimated TL of a Type 3 floor-ceiling is given in Table 18 for a few typical dimensions of concrete floor slab thickness and air space.

d. Type 4 Floor-Ceiling. The Type 4 floor-ceiling combination consists of a concrete floor slab, an air space, and a resiliently supported plaster ceiling. This combination is for use in critical situations where a high TL is required. The plaster ceiling should have at least 1 in. thickness of high density plaster (minimum 12 lb/sq. ft surface weight) and the air space should be at least 18 in. thick. The ceiling should be supported on resilient ceiling hangers that provide at least 1/10 in. static deflection under load. Neoprene-in-shear or compressed glass fiber hangers can be used, or steel springs can be used if they include a pad or disc of neoprene or glass fiber in the mount. A thick felt pad hanger arrangement can be used if it meets the static deflection requirement. The hanger system must not have metal-to-metal short-circuit paths around the isolation material of the hanger.

Where the plaster ceiling meets the vertical wall surface, the perimeter edge of the ceiling must not make rigid contact with the wall member. A 1/4-in. open joint should be provided at this edge, which is filled with a non-hardening caulking or mastic or fibrous packing after the ceiling plaster is set.

The estimated TL of a Type 4 floor-ceiling combination is given in Table 19 for a few typical dimensions of floor slab, air space and ceiling thicknesses. It is cautioned that this combination is for use in critical situations, and special care must be exercised to produce a good, resiliently supported, non-porous, dense ceiling. Acoustic tile can be added to the underside of the plaster ceiling but it will not change the transmission loss of the combination; it will only add to the acoustic absorption of the room.

e. Type 5 Floor-Ceiling. The Type 5 floor-ceiling combination is the same as the Type 4 combination, except that a "floating concrete floor" is mounted on top of the structural floor slab. The floating concrete floor should not support any large operating equipment. It should extend over that part of the mechanical room floor area within 20 ft of, but not under, any vibration isolated concrete inertia bases carrying specific pieces of operating machinery. The floating concrete floor should be supported off the structure floor at a height of at least 2 in. with the use of properly spaced blocks of compressed glass fiber or multiple-layers of ribbed or waffle-pattern neoprene pads or steel springs (in series with two layers of ribbed or waffle-pattern neoprene pads). The density and loading of the compressed glass fiber or neoprene pads should follow the manufacturers' recommendations. If steel springs are used, their static deflection should not be less than 1/4 in. The 2-in. space between the floating slab and the structure slab should be covered with a 1-in. thickness of low-cost glass fiber or mineral wool blanket of 3 to 4 lb/cu. ft density. Around all the perimeter edges of the floating floor (around the walls and around all concrete inertia bases within the floating floor area) there should be 1-in. gaps that are later packed with mastic or fibrous filling and then sealed with a waterproof non-hardening caulking or sealing material. If a curb is provided around the perimeter of the floated slab to help discourage water leakage into the sealed perimeter joints, several floor drains should be set in the structure slab under the floating slab to provide run-off of any water leakage into this cavity space.

As with the Type 4 combination, the Type 5 combination includes a resiliently supported plaster ceiling under the structure slab. The estimated TL of a Type 5 floor-ceiling combination is given in Table 20 for a few typical dimensions of floating floor slab in combination with the Type 4 structures of Table 19. It is to be noted that the floating slab is intended to improve the airborne TL of a floor; it is not suggested here as a vibration isolation mounting base for large equipment, although it will provide certain benefits to some structure-borne noise of pipe supports, duct supports, drainage lines, electrical conduit and the like.

As a general rule, to be reinforced later in the section under vibration isolation, the MER structural floor slab for an upper floor in a multi-floor building should not be less than 6 in. thick for completely rotary-action equipment, nor less than 8 in. thick for reciprocating-action equipment. These suggestions are based on acoustic considerations only and are not intended to represent structural requirements of the building. Even thicker floor slabs will be more beneficial acoustically. Where possible, large equipment should be located over principal or secondary beams in the flooring layout.

In the upper frequency bands of Tables 16-20, extremely high TL values (say, anything above 60 or 65 dB) are indicated as possible. In practice, these values cannot be achieved without making a real concentrated effort to stop all escape paths of airborne and structure-borne noise.

f. Noise Reduction of Floor-Ceiling Combinations. Paragraph 2.*c*. discussed the conversion of transmission loss of a wall into the noise reduction of a wall by use of the "wall correction term", designated by the letter "C" in Table 14. The same type of correction must be applied to convert the TL of a floor-ceiling combination to its NR value. This applies, of course, to the situation in which the MER is immediately above or below an adjoining area of concern. For identification purposes, the term is called "floor correction term" here, but it is represented by the same letter "C" and it is also obtained from Table 14, based on (1) Room Constant, and (2) common floor-ceiling area of the receiving room. The value of "C" will differ, of course, from room to room, so it must be redetermined for each room of interest above, below or beside a machine room.

In the equipment noise summary tables, SPLs are given at 3-ft distances from the equipment. These values should be used as the source room SPLs in the relationship

$$SPL_{\substack{\text{source}\\ \text{room}}} - NR = SPL_{\substack{\text{receiver}\\ \text{room}}}$$

when estimating noise levels in areas on the floor immediately below the equipment. However, it should be cautioned that the 3-ft SPLs are fairly localized values and they drop off with distance from the unit. Thus, the SPLs calculated for the receiving room below would apply immediately beneath the equipment and would drop off with distance away from that location. In fact, it can be expected that the SPLs below will be generally somewhat lower than the calculations would indicate, depending on receiving room size, the value of Sw used in the calculation and possibly the floor area occupied by the equipment in the MER.

4. SOUND DISTRIBUTION OUT-OF-DOORS

a. Effect of Distance. As a general rule, sound from an essentially localized source spreads out as it travels away from the source, and the sound pressure level (SPL) due to that source decreases at the rate of 6 dB per doubling of distance (referred to as "the inverse square law"). This effect is due to spreading only, and this is an effect common to all types of energy propagation originating from an essentially point source and free of any special focussing or beam-controlling devices. In addition, the air absorbs a certain amount of sound energy due to "molecular absorption". For short distances (less than a few hundred feet) this energy absorption can be ignored, but for sound propagation over a reasonably large distance it should be considered. Further, the "molecular absorption" effect is greater at high frequencies than it is at low frequencies.

If a noise source is placed out-of-doors (with no nearby reflecting surfaces or objects except the ground) and if it is assumed to be an "omni-directional" or "non-directional" source (i.e., it radiates equally in all directions), the SPL at a distance "D" is given by:

$$\text{SPL} = \text{PWL} - 10 \log (2\pi D^2) + 10 \text{ dB}.$$

This relation merely states that a given PWL, expressed in dB re 10^{-12} watts, will produce a given value of SPL over an entire hemispherical surface of radius "D" surrounding that source, where "D" is expressed in feet and SPL is expressed in the standard form of "dB re 0.0002 microbar." For short distances the effect of absorption can be ignored. The quantity $[10 \log (2\pi D^2) - 10 \text{ dB}]$ is called a "Distance Term" in Table 21 such that

$$\text{SPL} = \text{PWL} - \text{DISTANCE TERM}$$

and dB values for the Distance Term are given for values of D out to 100 ft. Since "molecular absorption" is negligible at short distances, the values given in Table 21 apply to all octave bands.

For distances beyond 100 ft, molecular absorption becomes effective at the higher frequencies, and the Distance Term should be obtained from Table 22.

b. Effect of Atmospherics. Precipitation, wind, wind gradients (with altitude), temperature, temperature gradients (with altitude), and relative humidity are possible atmospheric factors in sound transmission.

Rain, mist, fog, hail, sleet and snow are the various forms of precipitation to consider. These have not been studied extensively in their natural state so there are no representative values of excess attenuation to be assigned to them. Rain, hail and sleet may change the background noise levels, and a thick blanket of snow provides an absorbent ground cover for sound traveling at grazing incidence near the ground. In practice, of course, precipitation or a blanket of snow are intermittent, temporary and of relatively short total duration, and they could not be counted on for steady-state sound control, even if they should offer noticeable attenuation.

A steady, smooth flow of wind, equal at all altitudes, would have no noticeable effect on sound transmission. In practice, however, wind speeds are slightly higher above the ground than at the ground, and the resulting wind speed gradients tend to "bend" sound waves over large distances. Sound traveling with the wind is bent down to earth, while sound traveling against the wind is bent upwards above the ground. There is little or no increase in sound levels due to the sound waves being bent down; in fact, there is additional loss at the higher frequencies and at the greater distances. There can be some reduction of sound levels at relatively long distances (beyond a few hundred yards) when the sound waves are bent upward, for sound traveling against the wind.

Irregular, turbulent or gusty wind provides fluctuations in sound transmission over large distances. The net effect of these fluctuations may be an average reduction of a few decibels per 100 yards for gusty wind with speeds of 15 to 30 mph. However, gusty wind or wind direction cannot be counted on for noise control over the lifetime of an installation.

Constant temperature with altitude produces no effect on sound transmission, but temperature gradients can produce bending in much the same way as wind gradients do. Air temperature above the ground is normally cooler than at the ground, and the denser air above tends to bend sound waves upward. With "temperature inversions" the warm air above the surface bends the sound waves down to earth. These effects are negligible at short distances but they may amount to several dB at very large distances (say, over a half-mile). Again, there is little or no increase, but there may be a decrease in sound levels. Thus, temperature gradients cannot be relied on as a noise control aid.

Very low relative humidity (10 to 20%) increases the effect of "molecular absorption" of sound energy. These low values of relative humidity are seldom found in most inhabited areas. Average values of relative humidity found in most populated areas are used in arriving at the data given in Table 22.

In summary, there are atmospheric effects which would seldom increase but could decrease sound levels at large distances from a source. These decreases are usually of an intermittend, short-time duration and they are usually beneficial to the receiver (in giving temporary noise reduction) when they occur, but it is best not to rely on them for long-time benefits in terms of noise control design.

c. Attenuation Provided by Barriers. A wall, a building, a large mound of earth, a hill or some other type of solid structure, if large enough, can serve as a partial "barrier" to sound and can provide a moderate amount of sound reduction for a receiver located within the "shadow" provided by the barrier.

Table 23 gives a sketch of a barrier and the excess attenuation that might be expected from the barrier as a function of certain dimensions. This attenuation is in addition to the distance effect that might be obtained from Tables 21 or 22. For a barrier to be effective, its lateral width should extend beyond the line-of-sight between the source and receiver by at least as much as the height of the barrier extends above the line-of-sight. Also there must be no nearby large reflecting surfaces that can reflect sound around the barrier into the shadow zone. The distance D in the sketch of Table 23 must be very large compared to the distance R and the height H. The attenuation values given in

Table 23 will apply equally for the two conditions:

(1) Sound source at Point A and receiver at Point B, or
(2) Sound source at Point B and receiver at Point A.

The barrier loses effectiveness at very large distances because sound that passes over the top of the barrier may be bent back down to the ground by wind and temperature gradients. If D is greater than 1 mile, the attenuation values used should be only about one-half the values given in Table 23.

If a barrier wall is to be built or used as a noise control device, the TL of the wall (or building) should exceed by at least 10 dB in all frequency bands the excess attenuation to be expected from the wall.

If the barrier is a large "thick" building, the distance R should be taken from Point A to the near wall of the building and the height H should be the height of the building at that near wall. There should be no large openings entirely through the building that would destroy the effectiveness of the building as a barrier. A few small open windows in the near and far walls would probably be acceptable, provided the interior rooms are large.

Caution: Note that a large reflecting surface, such as the barrier wall, may reflect more sound in the opposite direction than there would have been with no wall at all present. If there is no special focussing effect, the wall may produce at most only about 2 or 3 dB higher levels in the direction of the reflected sound.

d. Attenuation Provided by Trees. Heavy dense growths of woods provide a small amount of sound attenuation. To be effective both winter and summer, there should be a reasonable mixture of both deciduous and evergreen trees. Also the ground cover should be sufficiently dense that sound cannot pass under the absorbent upper portion of the trees. For dense woods of several hundred feet depth, the sound may pass over the tops of the trees, in which case the attenuation through the trees should never be considered greater than the excess attenuation over the trees, as determined from the application of Table 23.

Table 24 gives the approximate excess attenuation of sound through dense woods, where dense woods are taken as having an average "visibility penetration" of about 70 to 100 ft. Occasional trees and hedges give no significant attenuation. "Visibility penetration" is the average maximum distance in the woods at which some small portions of a large (3-ft square) white cloth can still be seen.

e. Noise Reduction of a House or Building. Outdoor noise normally suffers some noise reduction when it passes indoors into a house or building, even when the building has open windows. The amount of noise reduction (NR) varies with the building construction, orientation, wall area, window area, open window area, etc. Some estimated NR values for building constructions were given in Table 6 of Chapter 2.

5. MUFFLERS

Without going into the details of design, construction, use and limitations of mufflers, some representative muffler data are given in Tables 25-28.

Table 25 lists the approximate noise reduction values claimed by several manufacturers for their various lines of reactive mufflers for use with reciprocating engine exhausts.

Table 26 lists the noise reduction (or "insertion loss") for several typical commercial duct mufflers, as used in air-conditioning duct systems and for other noise control applications.

Table 27 lists the approximate noise reduction of parallel baffles of various dimensions made up from 4-in. thick absorbent baffles.

Table 28 gives approximate noise reduction values for 8-ft lengths of various combinations of thick parallel baffles.

It is cautioned that muffler design and construction is a specialized field. These data are offered so that an architect and engineer may have some feeling for the amount of muffling required to meet certain kinds of problems; it is not expected that the architect or engineer would design his own mufflers for specific problems.

Table 29 gives the approximate attenuation of unlined and lined ducts for use with inlet and outlet ducts of various engines, if ASHRAE Guide duct data are not available. Please note and observe the comments given with the table.

Table 30 gives the approximate attenuation provided by lined and unlined duct turns for use with inlet and outlet ducts of various engines, if ASHRAE Guide data are not available. Please note and observe the comments given with the table.

6. DATA FORMS

In the Appendix at the end of these notes, a group of blank Data Forms is given. Each Data Form is designed to simplify the calculations involved in each particular type of analysis. The Data Forms may be duplicated for later uses on other problems.

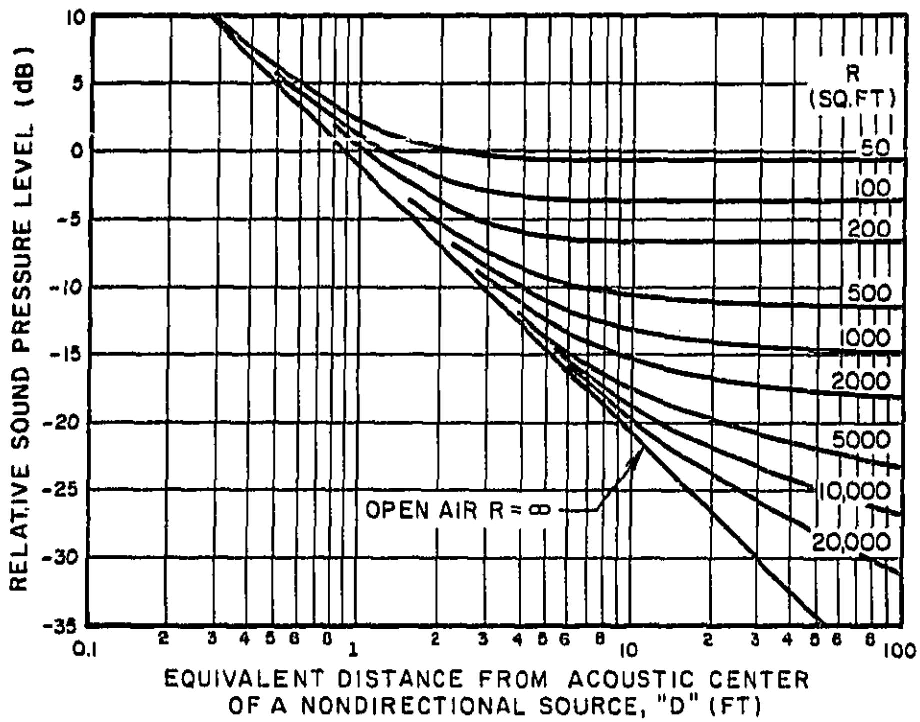


FIG.1 APPROXIMATE RELATIONSHIP BETWEEN "RELATIVE SOUND PRESSURE LEVEL" AND DISTANCE TO A NOISE SOURCE, FOR VARIOUS ROOM CONSTANT VALUES

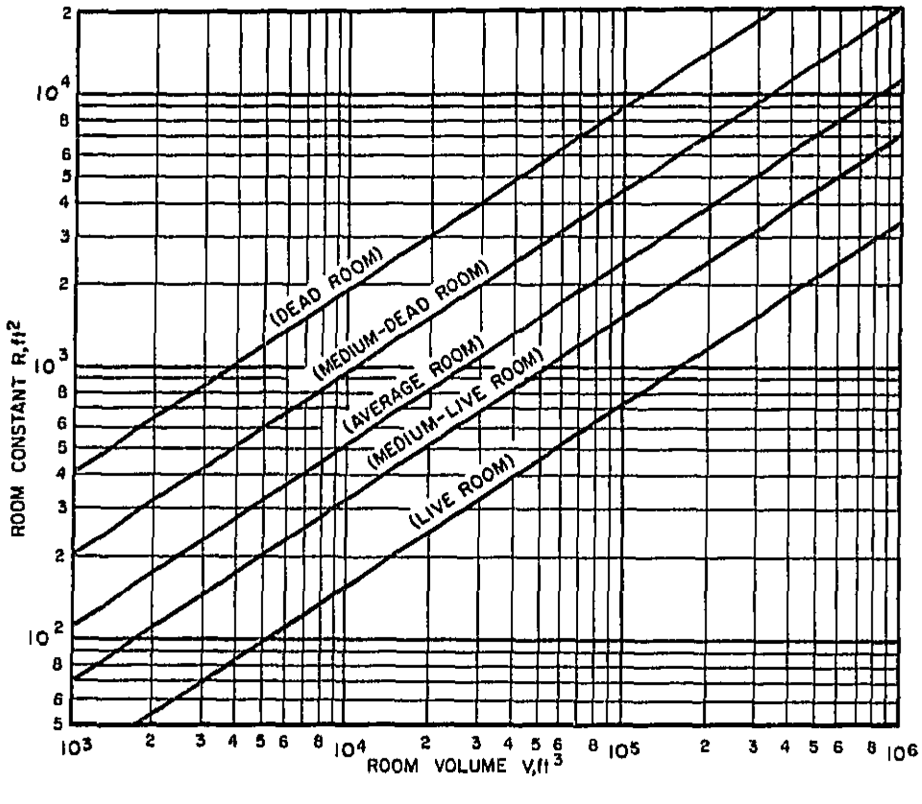


FIG.2 APPROXIMATE RELATIONSHIP BETWEEN ROOM VOLUME AND ROOM CONSTANT FOR SPACES OF VARIOUS AVERAGE ACOUSTIC ABSORPTION (AT MID-FREQUENCY REGION OF 500-1000 Hz)

TABLE 1

ACOUSTIC TREATMENT DETAILS FOR USE WITH

FIGURES 1 AND 2 IN ESTIMATING ROOM CONSTANT

PART A. SURFACE COVERAGE OF ACOUSTIC MATERIAL

Percentage of Total Room Surface Area Covered with Absorption Material	Room Label on Figure 2 Curves
0%	"Live Room"
10%	"Medium-Live Room"
15-20%	"Average Room"
30-35%	"Medium-Dead Room"
50-60%	"Dead Room"

PART B. LOW FREQUENCY CORRECTION TO "R"

Octave Band (Hz)	Corrected R to be used in Figure 1 for NRC = 0.65 - 0.74 and if there is no acoustic absorption	NCR = 0.75 - 0.85
31	0.2 R	0.2 R
63	0.2 R	0.3 R
125	0.3 R	0.5 R
250	0.5 R	0.8 R

Notes:

1. "NRC" is "noise reduction coefficient." It represents the average of the acoustic absorption coefficients of the material for the four frequency bands 250, 500, 1000 and 2000 Hz. This does not necessarily mean that noise is reduced by the amount of the NRC value. NRC values are published for all acoustical materials manufactured and distributed by members of the Acoustical and Insulating Materials Association, 205 W. Touhy Avenue, Park Ridge, Illinois 60068.

2. An NRC of 0.65 to 0.74 can be met by most perforated, fissured or textured acoustic tiles or panels of 3/4-in. or 1-in. thickness or by most perforated panels containing at least 1 in. thick layers of glass fiber or mineral wool.

3. An NRC of 0.75 - 0.85 can be met by most 2-in. thick layers of acoustic absorption material or by most 3/4-in. or 1-in. thick acoustic materials spaced at least 2 in. away from the wall or 10 in. away from the ceiling from which they are supported.

TABLE 2

REDUCTION OF SPL (IN DB) IN GOING FROM NORMALIZED 3-FT DISTANCE TO A GREATER DISTANCE "D" IN A ROOM HAVING A ROOM CONSTANT "R"

ROOM CONSTANT "R"	DISTANCE "D" (IN FT) FROM EQUIPMENT							
(in sq. ft)	5	10	15	20	30	40	60	80
100	0	1	1	1	1	1	1	1
200	1	1	1	1	1	1	1	1
320	2	2	2	2	2	2	2	2
500	2	3	3	4	4	4	4	4
700	2	3	4	4	4	5	5	5
1000	2	4	5	5	6	6	6	6
2000	3	6	7	7	8	8	8	8
3200	4	7	8	8	9	10	11	11
5000	4	8	9	10	11	12	12	13
7000	4	8	10	11	12	13	14	15
10000	4	9	11	12	13	14	15	17
20000	5	10	12	14	16	17	19	20
50000	5	10	13	16	18	21	23	25
INFINITE	5	11	14	17	20	23	26	29

TABLE 3

APPROXIMATE TRANSMISSION LOSS (IN DB) OF DENSE POURED CONCRETE* OR SOLID-CORE CONCRETE BLOCK OR MASONRY

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF CONCRETE OR MASONRY (IN.)					
	4	6	8	10	12	16
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT)					
	48	72	96	120	144	192
31	29	32	33	34	35	36
63	32	33	34	35	36	37
125	34	35	36	37	38	39
250	35	36	38	40	41	43
500	37	40	43	45	47	50
1000	42	46	50	52	54	56
2000	49	53	56	58	59	61
4000	55	58	61	63	64	66
8000	60	63	66	68	69	70

TABLE 4

APPROXIMATE TRANSMISSION LOSS (IN DB) OF HOLLOW-CORE DENSE* CONCRETE BLOCK OR MASONRY

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF CONCRETE OR MASONRY (IN.)					
	4	6	8	10	12	16
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT)					
	28	36	44	52	60	76
31	24	26	28	30	31	32
63	29	30	31	32	32	33
125	32	33	33	34	34	35
250	33	34	35	36	36	37
500	34	35	36	38	39	42
1000	37	39	41	43	45	48
2000	42	46	48	50	52	55
4000	49	52	54	56	58	60
8000	55	57	59	61	63	65

* 1. "Dense" concrete = 140-150 lb/cu ft density

2. For applications involving "transmission loss" as an acoustic requirement, do not use "cinder block" or other lightweight porous block material.

TABLE 5

APPROXIMATE TRANSMISSION LOSS (IN DB)
OF CONVENTIONAL STUD-TYPE PARTITIONS[1]

OCTAVE FREQUENCY BAND (HZ)	STANDARD WOOD STUD PARTITION[2]	STAGGERED WOOD STUD PARTITION[3]	IMPROVEMENT WITH INSULATION[4]
31	10	12	1
63	15	17	1
125	20	22	2
250	26	30	3
500	34	38	4
1000	40	44	4
2000	45	47	5
4000	43	45	5
8000	45	47	5

1. Partitions made with 2-1/2 in. to 3-1/2 in. wide steel studs will approximate the values given here for wood-stud construction.

2. 2x4 wood studs on 16 in. centers, nailed to 2x4 wood plates; 5/8 in. thick gypsum board nailed on both sides of studs; fill and tape joints and edges, finish as desired.

3. 2x4 wood studs staggered on 2x6 wood plates, alternate studs supporting separate walls of 5/8 in. thick gypsum board; all-nailed construction, studs for each wall on 16 in. centers; fill and tape joints and edges, finish as desired.

4. Installation of either (a) 1/2 in. thick glass fiber board or metal spring clips between studs and gypsum board, or (b) min. 1-1/2 in. thick limply supported lightweight insulation in air space between partitions will produce improvement indicated. For staggered partition, use of both types of insulation will produce twice the improvement shown in the table. Add the "improvement values" to the TL of the stud partition to which the insulation has been added.

TABLE 6

APPROXIMATE TRANSMISSION LOSS (IN DB)
OF FILLED METAL PANEL PARTITION AND
TYPICAL INDUSTRIAL ACOUSTIC DOORS

OCTAVE FREQUENCY BAND (HZ)	FILLED METAL PANEL PARTITION[1]	TYPICAL ACOUSTIC DOORS[2]	
		4" THICK	6" THICK
31	19	27	33
63	22	29	35
125	26	33	37
250	31	36	39
500	36	42	46
1000	43	47	50
2000	48	53	56
4000	50	56	61
8000	52	59	65

1. Constructed of two 18 ga. steel panels filled with 3 in. thickness of 6-8 lb/cu ft glass fiber or rock wool; joints and edges sealed air-tight.

2. Industrial type acoustic doors typically constructed of sheet steel exterior facings, 1 in. plywood under the sheet steel, densely packed filler of glass fiber or rock wool; heavy framing and hardware; double gasket seals all around door edges. "Studio-type" acoustic doors usually not as thick and heavy, with more elaborate finish details.

TABLE 7

APPROXIMATE TRANSMISSION LOSS (IN DB)
OF GLASS* WALLS OR WINDOWS

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF GLASS (IN.) 1/8	1/4	1/2	3/4
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 1½	3	6½	10
31	0	5	11	14
63	5	11	17	20
125	11	17	23	24
250	17	23	25	25
500	23	25	26	27
1000	25	26	27	28
2000	26	27	28	29
4000	27	28	30	33
8000	28	20	36	39

* Special laminated safety glass containing one or more viscoelastic layers sandwiched between glass panels will yield 3-8 dB higher values than given here for single thicknesses of glass; available in approximately 1/4 in. to 5/8 in. thicknesses.

TABLE 8

APPROXIMATE TRANSMISSION LOSS (IN DB) OF
A FEW TYPICAL DOUBLE-GLASS WINDOWS*

OCTAVE FREQUENCY BAND (HZ)	GLASS-AIR SPACE-GLASS THICKNESSES (inches) ¼ - ¼ - ¼	¼ - 1½ - ¼	¼ - 6 - ¼
31	13	14	15
63	18	19	20
125	23	23	24
250	24	25	28
500	24	27	31
1000	26	31	37
2000	28	34	40
4000	30	37	43
8000	36	42	46

* Thermal-insulation double-glass windows typically have 1/4" to 1" sealed air space between 1/4" to 3/8" glass panels. For larger air spaces, individual glass panels should be mounted separately in rubber or neoprene gaskets. For large temperature differences across the window, provide desiccant or small ventilation ports in the inner space to eliminate condensation on the cold glass.

TABLE 9

APPROXIMATE TRANSMISSION LOSS (IN DB)

OF WOOD[1] OR PLYWOOD

(4 LB/SQ FT/IN. SURFACE DENSITY)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF WOOD OR PLYWOOD (IN.) 1/4	1/2	1	2[2]	4
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 1	2	4	8	16
31	0	0	5	10	16
63	0	4	11	15	18
125	5	10	16	17	19
250	11	15	18	19	20
500	16	17	19	20	26
1000	18	19	20	26	32
2000	19	20	26	32	37
4000	20	26	32	37	41
8000	26	32	37	41	45

1. Wood construction requires tongue-and-groove joints, overlapping joints, or sealing of joints against air leakage. For intermediate thicknesses, interpolate between thicknesses given in table.

2. For 2 in. solid wood doors that are well-gasketed all around, these values of TL may be used.

TABLE 10

APPROXIMATE TRANSMISSION LOSS (IN DB)

OF DENSE* PLASTER

(9 LB/SQ FT/IN. SURFACE DENSITY)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF PLASTER (IN.) 1/2	3/4	1	1-1/2	2
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 4-1/2	7	9	13	18
31	9	12	15	18	21
63	15	18	21	24	26
125	21	24	26	27	27
250	26	27	27	28	28
500	27	28	28	29	29
1000	28	29	29	30	33
2000	29	30	33	37	40
4000	33	37	40	44	47
8000	40	44	47	50	53

*If light-weight non-porous plaster is used, these TL values may be used for equal values of surface weight. These data <u>must not be used</u> for porous or so-called "acoustic plaster".

If plaster is to be used on typical stud wall construction, estimate the total thickness or weight of the plaster and use the TL values given here for that thickness, but increase the TL values where appropriate so that they are not less than those given in Table 23 for the applicable stud construction.

TABLE 11

APPROXIMATE TRANSMISSION LOSS (IN DB)

OF SHEET ALUMINUM

(14 LB/SQ FT/IN. SURFACE DENSITY)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF ALUMINUM (IN.) 1/16	1/8	1/4
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 1	2	3-1/2
31	0	1	7
63	1	7	13
125	7	13	19
250	13	19	23
500	19	23	25
1000	23	25	26
2000	25	26	27
4000	26	27	28
8000	27	28	32

TABLE 12

APPROXIMATE TRANSMISSION LOSS (IN DB)

OF SHEET STEEL

(40 LB/SQ FT/IN. SURFACE DENSITY)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF STEEL (IN.) 1/16	1/8	1/4
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 2-1/2	5	10
31	3	9	15
63	9	15	21
125	15	21	27
250	21	27	33
500	27	33	38
1000	33	38	39
2000	38	39	39
4000	39	39	37
8000	39	37	40

TABLE 13

APPROXIMATE TRANSMISSION LOSS (IN DB) OF SHEET LEAD
(60 LB/SQ FT/IN. SURFACE DENSITY)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF LEAD (IN.) 1/16	1/8	3/16	1/4
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 4	7½	11	15
31	7	13	16	19
63	13	19	22	25
125	19	25	28	31
250	25	31	34	37
500	31	37	40	43
1000	37	43	46	49
2000	43	49	51	53
4000	49	53	54	55
8000	53	55	55	55

TABLE 14

APPROXIMATE WALL OR FLOOR CORRECTION TERM "C"
FOR USE IN THE EQUATION NR = TL + "C"

(Select nearest integral value of C)

RATIO S_W/R_2	"C" (dB)	RATIO S_W/R_2	"C" (dB)	RATIO S_W/R_2	"C" (dB)
0.00	+6	1.7	-3	15	-12
0.07	+5	2.2	-4	20	-13
0.15	+4	2.9	-5	25	-14
0.25	+3	3.7	-6	31	-15
0.38	+2	4.7	-7	40	-16
0.54	+1	6.1	-8	50	-17
0.75	0	7.7	-9	63	-18
1.0	-1	9.7	-10	80	-19
1.3	-2	12	-11	100	-20

S_W is the area of the wall or floor (in sq ft) common to the "transmitting" and "receiving" rooms.

R_2 is the Room Constant of the "receiving" room; include low frequency values of R_2.

TABLE 15

APPROXIMATE TRANSMISSION LOSS OF A
WALL CONTAINING DOORS OR WINDOWS

DOOR OR WINDOW AREA AS PERCENT OF TOTAL WALL AREA	IF TL OF DOOR OR WINDOW IS LESS THAN TL OF WALL BY	THEN, EFFECTIVE TL OF COMPOSITE WALL IS LESS THAN TL OF ORIGINAL WALL BY
40%	3 dB	1 dB
	6	4
	10	7
	15	11
	20	16
20%	3	1
	6	2
	10	4
	15	9
	20	13
10%	3	0
	6	1
	10	3
	15	6
	20	10
5%	3	0
	6	0
	10	1
	15	4
	20	8
2%	3	0
	6	0
	10	1
	15	2
	20	5
1%	3	0
	6	0
	10	0
	15	1
	20	3

TABLE 16

APPROXIMATE TRANSMISSION LOSS (IN DB)

OF TYPE 1 FLOOR-CEILING COMBINATION

(SEE TEXT FOR DESCRIPTION OF TYPE 1)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF DENSE CONCRETE SLAB (IN.) 6	8	10	12
	APPROXIMATE SURFACE WEIGHT (LB/SQ FT) 72	96	120	144
31	32	33	34	35
63	33	34	35	36
125	35	36	37	38
250	36	38	40	41
500	40	43	45	47
1000	46	50	52	54
2000	53	56	58	59
4000	58	61	63	64
8000	63	66	68	69

TABLE 17

APPROXIMATE TRANSMISSION LOSS (IN DB) OF

SOME TYPE 2 FLOOR-CEILING COMBINATIONS

(SEE TEXT FOR DESCRIPTION OF TYPE 2)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF DENSE CONCRETE SLAB (IN.) 6	8	10	12
	AIR SPACE BETWEEN SLAB AND SUSPENDED ACOUSTIC CEILING (IN.) 15	18	24	24
31	33	35	37	38
63	35	37	39	40
125	38	40	42	43
250	40	43	46	47
500	45	49	52	54
1000	52	57	60	62
2000	59	63	66	67
4000	64	68	71	72
8000	69	73	76	77

TABLE 18

APPROXIMATE TRANSMISSION LOSS (IN DB) OR SOME TYPE 3 FLOOR-CEILING COMBINATIONS (SEE TEXT FOR DESCRIPTION OF TYPE 3)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF DENSE CONCRETE SLAB (IN.) 6	8	10	12
	AIR SPACE BETWEEN SLAB AND SUSPENDED "HIGH TL" ACOUSTIC CEILING (IN.) 15	18	24	24
31	36	38	40	41
63	38	40	42	43
125	41	43	45	46
250	43	46	49	50
500	48	52	55	57
1000	55	60	63	65
2000	62	66	69	70
4000	67	71	74	75
8000	72	76	79	80

TABLE 19

APPROXIMATE TRANSMISSION LOSS (IN DB) OF SOME TYPE 4 FLOOR-CEILING COMBINATIONS (SEE TEXT FOR DESCRIPTION OF TYPE 4)

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF DENSE CONCRETE SLAB (IN.) 6	8	10	12
	AIR SPACE BETWEEN SLAB AND RESILIENTLY SUSPENDED PLASTER CEILING (IN.) 18	24	30	30
	THICKNESS OF DENSE PLASTER CEILING (IN.) 1	1	1-1/2	2
31	39	41	44	45
63	41	43	46	48
125	45	47	50	53
250	47	50	54	57
500	52	56	60	64
1000	59	64	68	72
2000	66	70	74	77
4000	71	75	78	82
8000	76	80	84	87

TABLE 20

APPROXIMATE TRANSMISSION LOSS (IN DB) OF
SOME TYPE 5 FLOOR-CEILING COMBINATIONS
(SEE TEXT FOR DESCRIPTION OF TYPE 5)

FOR FLOATING FLOOR SLAB OF 3 IN. THICKNESS SUPPORTED RESILIENTLY 2 IN. ABOVE STRUCTURE SLAB:

ADD 3 DB TO TABLE 19 VALUES

FOR FLOATING FLOOR SLAB OF 4 IN. THICKNESS SUPPORTED RESILIENTLY 2 IN. ABOVE STRUCTURE SLAB:

ADD 4 DB TO TABLE 19 VALUES

FOR FLOATING FLOOR SLAB OF 5 IN. THICKNESS SUPPORTED RESILIENTLY 2 IN. ABOVE STRUCTURE SLAB:

ADD 5 DB TO TABLE 19 VALUES

NOTE: THE 3, 4 AND 5 DB INCREMENTS GIVEN HERE FOR 3, 4 AND 5 IN. THICK FLOATING SLABS MAY ALSO BE USED WHEN A FLOATING SLAB IS ADDED TO ANY OTHER FLOOR-CEILING COMBINATION SHOWN IN TABLES 16-18.

TABLE 21

DISTANCE TERM [$10 \log (2\pi D^2) - 10$ dB]
FOR CALCULATING SPL OUT TO A DISTANCE OF 100 FT
FROM A NOISE SOURCE OF POWER PWL
SPL = PWL - DISTANCE TERM
where PWL is in dB re 10^{-12} watts

DISTANCE D (ft)	DISTANCE TERM (dB)	DISTANCE D (ft)	DISTANCE TERM (dB)
$1\frac{1}{4}$	0	19-21	24
2	4	22-23	25
3	8	24-26	26
4	10	27-29	27
5	12	30-33	28
6	14	34-37	29
7	15	38-42	30
8	16	43-47	31
9	17	48-53	32
10	18	54-59	33
11	19	60-67	34
12-13	20	68-75	35
14	21	76-84	36
15-16	22	85-94	37
17-18	23	95-100	38

TABLE 22

DISTANCE TERM, INCLUDING ABSORPTION LOSSES,
FOR CALCULATING SPL FOR DISTANCES OF 100 FT TO 10,000 FT
FROM A NOISE SOURCE OF POWER PWL
SPL = PWL - DISTANCE TERM
where PWL is in dB re 10^{-12} watts

DISTANCE D (ft)	DISTANCE TERM (TO NEAREST dB) FOR OCTAVE FREQUENCY BAND (Hz)					
	31-250	500	1000	2000	4000	8000
100	38	38	38	38	39	39
112	39	39	39	39	40	41
126	40	40	40	40	41	42
141	41	41	41	41	42	43
158	42	42	42	42	43	44
178	43	43	43	44	44	46
200	44	44	44	45	46	47
224	45	45	45	46	47	48
252	46	46	46	47	48	50
282	47	47	47	48	49	51
316	48	48	48	49	50	53
356	49	49	49	50	52	54
400	50	50	51	51	53	56
448	51	51	52	52	54	57
504	52	52	53	54	56	59
564	53	53	54	55	57	61
632	54	54	55	56	59	63
712	55	56	56	57	60	65
800	56	57	57	58	62	67
900	57	58	58	60	64	70

TABLE 22 (continued)

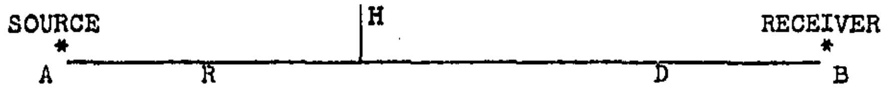
DISTANCE D (ft)	DISTANCE TERM (TO NEAREST dB) FOR OCTAVE FREQUENCY BAND (Hz) 31-250	500	1000	2000	4000	8000
1000	58	59	59	61	66	72
1120	59	60	61	62	68	75
1260	60	61	62	64	70	78
1410	61	62	63	65	73	81
1580	62	63	64	67	75	85
1780	63	64	66	68	77	89
2000	64	65	67	70	79	93
2240	65	67	68	72	82	97
2520	66	68	70	74	85	102
2820	67	69	71	75	89	108
3160	68	70	72	77	92	114
3560	69	72	74	80	96	120
4000	70	73	76	82	101	128
4480	71	74	77	84	105	136
5040	72	76	79	87	111	145
5640	73	77	81	90	116	154
6320	74	78	83	93	123	165
7120	75	80	85	96	130	178
8000	76	82	87	100	138	191
9000	77	83	90	104	146	207
10000	78	85	92	108	155	222

TABLE 23

APPROXIMATE NOISE REDUCTION (IN DB)

PROVIDED BY A SOLID BARRIER

(Do not go above 24 dB or below 0 dB attenuation in any bands. See text for discussion. Use one-half of attenuation for D greater than 1 mile.)



RATIO H^2/R (ft)	NOISE REDUCTION IN FREQUENCY BAND 63 HZ	125 HZ	250 HZ	500 HZ	1000 HZ	2000 HZ	4000 HZ	8000 HZ
0.3-0.4	0	0	3	6	9	12	15	18
0.5-0.8	0	2	5	8	11	14	17	20
0.9-1.2	1	4	7	10	13	16	19	22
1.3-1.9	3	6	9	12	15	18	21	24
2.0-3.1	5	8	11	14	17	20	23	24
3.2-4.9	7	10	13	16	19	22	24	24
5-8	9	12	15	18	21	24	24	24
9-12	11	14	17	20	23	24	24	24
13-20	13	16	19	22	24	24	24	24
over 20	15	18	21	24	24	24	24	24

TABLE 24

APPROXIMATE NOISE REDUCTION (IN DB)

PROVIDED BY DENSE WOODS

(Mixed Deciduous and Evergreen trees; 20-40 ft height, visibility penetration of 70 to 100 ft)

OCTAVE FREQUENCY BAND (HZ)	EXCESS ATTENUATION (in dB per 100 ft. of woods)
63	1/2
125	1
250	1-1/2
500	2
1000	3
2000	4
4000	4-1/2
8000	5

Notes:

1. For average 10-20 ft height, use one-half the rate given in the table.

2. For sparse woods of 200-300 ft visibility penetration, use one-half the rate given in the table.

TABLE 25

APPROXIMATE NOISE REDUCTION (IN DB) OF

TYPICAL REACTIVE MUFFLERS USED WITH RECIPROCATING ENGINES

(See text for discussion)

OCTAVE FREQUENCY BAND (HZ)	MUFFLER SERIES BY RELATIVE SIZE		
	SMALL SIZE	MEDIUM SIZE	LARGE SIZE
	HIGH PRESSURE-DROP LINE		
63	16	20	25
125	21	25	29
250	21	24	29
500	19	22	27
1000	17	20	25
2000	15	19	24
4000	14	18	23
8000	14	17	23
	LOW PRESSURE-DROP LINE		
63	10	15	20
125	15	20	25
250	13	18	23
500	11	16	21
1000	10	15	20
2000	9	14	19
4000	8	13	18
8000	8	13	18

Refer to manufacturers' literature for more specific data.

TABLE 26

APPROXIMATE NOISE REDUCTION (IN DB) OF

VARIOUS LENGTHS OF COMMERCIAL DUCT MUFFLERS

OCTAVE FREQUENCY BAND (HZ)	MUFFLER LENGTH 3 FT.	5 FT.	7 FT.
	"LOW PRESSURE-DROP CLASS"		
63	4	8	10
125	7	12	15
250	9	14	19
500	12	16	20
1000	15	19	22
2000	16	20	24
4000	14	18	22
8000	9	14	18
	"HIGH PRESSURE-DROP CLASS"		
63	8	11	13
125	10	14	18
250	15	23	30
500	23	32	40
1000	30	38	44
2000	35	42	48
4000	28	36	42
8000	23	30	36

Refer to manufacturers' literature for more specific data.

TABLE 27

APPROXIMATE NOISE REDUCTION (IN DB) OF
8-FT LONG, 4-IN. THICK PARALLEL BAFFLES
SEPARATED BY VARIOUS WIDTH AIR SPACES

OCTAVE FREQUENCY BAND (HZ)	WIDTH OF AIR SPACE 4 IN.	8 IN.	12 IN.	16 IN.	24 IN.
	PERCENT OPEN AREA 50%	67%	75%	80%	86%
63	3	2	1	1	0
125	6	5	3	2	2
250	16	13	8	6	4
500	32	25	16	13	10
1000	56	38	19	16	12
2000	48	35	13	11	8
4000	40	26	10	8	6
8000	20	18	7	6	4

TABLE 28

APPROXIMATE NOISE REDUCTION (IN DB) OF
8-FT LONG PARALLEL BAFFLES OF
TYPICAL THICKNESSES AND SEPARATIONS

OCTAVE FREQUENCY BAND (HZ)	THICKNESS OF BAFFLE (IN.) 8	8	12	16
	WIDTH OF AIR SPACE (IN.) 8	12	12	10
63	5	4	5	11
125	11	8	12	16
250	20	18	20	23
500	30	27	30	32
1000	27	23	24	25
2000	22	19	19	22
4000	18	14	14	17
8000	15	10	10	14

Refer to manufacturers' literature for more specific data.

TABLE 29

APPROXIMATE ATTENUATION OF VARIOUS STRAIGHT DUCTS WITH AND WITHOUT LINING (MAY BE USED IN ABSENCE OF MORE COMPLETE ANALYSIS OF DUCT ATTENUATION BY METHODS OF ASHRAE GUIDE)

These values are offered only for use in approximating the duct losses of a typical gas turbine engine inlet or exhaust duct before mufflers are added. Do not use these values to design noise control treatments for ducts.

Values given are in dB attenuation per ft of duct length, for the eight octave bands from 63 Hz to 8000 Hz. These values assume duct cross dimensions in the region of 3 to 5 ft. These are very rough estimates and should not be used to analyze the duct losses of a ventilation duct layout. If at all possible use the duct analysis methods of the ASHRAE Guide.

For elevated temperatures, as in gas turbine exhaust ducts, duct dimensions appear shorter in terms of sound wavelengths and therefore usually less effective in attenuating sound. In absence of more thorough analysis at specific exhaust temperature, take only two-thirds the length of the exhaust duct when estimating the attenuation of a hot exhaust duct.

Type 1. For ducting with no internal or external duct lining (dB per ft)

Round Duct:	.10	.07	.03	.02	.02	.02	.02	.02
Rectangular:	.20	.14	.07	.05	.04	.04	.04	.04

Type 2. For ducting with an external thermal duct lining (dB per ft)

Round Duct:	.15	.09	.04	.02	.02	.02	.02	.02
Rectangular:	.30	.20	.09	.05	.04	.04	.04	.04

Type 3. For ducting with 1-in. thick internal acoustic absorption duct lining, material and construction to withstand temperature and flow speed (dB per ft)

Round Duct:	.15	.16	.22	.6	.8	.8	.6	.4
Rectangular:	.30	.25	.25	.7	.8	.8	.6	.4

Type 4. For ducting with 2-in. thick internal acoustic absorption duct lining, material and construction to withstand temperature and flow speed (dB per ft)

Round Duct:	.20	.20	.3	.9	1.1	1.1	.9	.6
Rectangular:	.30	.28	.4	.9	1.1	1.1	.9	.6

TABLE 30

APPROXIMATE ATTENUATION OF VARIOUS DUCT TURNS WITH AND WITHOUT LINING (MAY BE USED IN ABSENCE OF MORE COMPLETE ANALYSIS OF DUCT ATTENUATION BY METHODS OF ASHRAE GUIDE)

These values are offered only for use in approximating the duct losses of a typical gas turbine engine inlet or exhaust duct before mufflers are added. Do not use these values to design noise control treatments for ducts.

Values given are in dB attenuation (or insertion loss) for the eight octave bands from 63 Hz to 8000 Hz. These values assume duct cross dimensions in the region of 3 to 5 ft. These are very rough estimates and should not be used to analyze the duct losses of a ventilation duct layout. If at all possible use the duct analysis methods of the ASHRAE Guide.

For elevated temperatures, as in gas turbine exhaust ducts, duct dimensions appear shorter in terms of sound wavelengths and therefore usually less effective in attenuating sound. In absence of more thorough analysis at specific exhaust temperature, use only two-thirds the attenuation given below when applying these data to a hot exhaust duct.

If there is more than one turn in the duct configuration, the attenuation of successive turns can be added together only if the turns are located at least 20 ft apart.

Type 1. For ducting with no internal duct lining, 90° rounded turn of round duct or 90° rounded turn of rectangular duct, with or without turning vanes (dB per turn)

0	1	2	3	3	3	3	3

Type 2. For ducting with no internal duct lining, 90° square turn of rectangular duct, with turning vanes of chord length less than 6 in. (dB per turn)

1	4	5	4	3	3	3	3

Type 3. For ducting with continuous 1-in. or 2-in. thick internal acoustic absorption duct lining, 90° rounded turn of round duct or 90° rounded turn of rectangular duct, with or without turning vanes (dB per turn)

1	2	3	4	5	6	6	6

Type 4. For ducting with continuous 1-in. or 2-in. thick internal acoustic absorption duct lining, 90° square turn of rectangular duct, with turning vanes of chord length less than 6 in. (dB per turn)

1-in. lining:

1	5	6	6	7	8	9	10

2-in. lining:

1	6	7	8	9	10	11	12

CHAPTER 5

VIBRATION ISOLATION OF MECHANICAL AND ELECTRICAL EQUIPMENT

1. VIBRATION CRITERIA

An approximation of the "threshold of sensitivity" of individuals to feelable low frequency vibration is shown in Figure 1 (at the end of this section). Obviously, this may vary over a relatively wide range for different individuals and for different ways in which a person might be subjected to vibration (standing, seated, through the finger tips, etc.). The exact threshold values are not particularly important, for the purpose of this discussion, but it is considered important to realize that vibration can evidence itself in two different ways to an individual: (1) as "feelable" vibration, and (2) as "audible" sound radiated from a vibrating surface. Along the right-hand side of Figure 1, the low frequency ends of some "NC-equivalent" curves are shown. The complete "NC-equivalent" curves for this sequence are shown in Figure 2. To illustrate, using either Fig. 1 or Fig. 2, if a wall or floor of a building has a vibration acceleration level of -60 dB re 1 G at a frequency of 63 Hz, the sound level radiated by that wall or floor would be equivalent to the noise level of an NC-30 curve at 63 Hz (about 57 dB re 0.0002 microbar). In a quiet room, a 57 dB sound pressure level at 63 Hz would be faintly audible; yet Figure 1 suggests that the vibration itself would be imperceptible for -60 dB acceleration level at 63 Hz. In fact, Figure 1 reveals that the acceleration level would have to be about 20 to 30 dB higher than this value in order to be barely perceptible as feelable vibration.

The objective of this discussion is to emphasize that the <u>sound</u> radiated by a vibrating wall or floor may be more important in noise control than the <u>feelable vibration</u> of that wall or floor, especially in the frequency region above 30 to 60 Hz. In other words, if a piece of equipment is vibration-isolated well enough to produce no audible noise radiation from a wall or floor, then it is quite probable that there will be no feelable vibration associated with that machinery (as observed in some part of the building outside the actual MER). This is especially true for equipment operating at speeds in the vicinity of 1800 RPM (or higher), which would give rise to 30 Hz (or higher) driving frequency. For slow speed machinery, the vibration isolation treatment should provide lower frequency isolation.

Actual vibration levels of mechanical and electrical equipment are not given here, but the vibration isolation recommendations that are given are aimed at achieving acceptable radiated noise levels (as "NC-equivalents") and essentially imperceptible "feelable" vibration levels in occupied spaces of the building.

2. VIBRATION IN BUILDINGS

Almost every structure has many natural frequencies of vibration depending on its mass, stiffness, dimensions, method of mounting, etc. In buildings, many of the natural frequencies of floors, beams, walls, columns, doors, windows, ceilings, etc., frequently fall in the range of 10 to 60 Hz. Typically, much of the mechanical equipment used in buildings operates at speeds that produce noise and vibration in this same frequency range: for example,

600 RPM = 10 Hz
1800 RPM = 30 Hz
3600 RPM = 60 Hz

Thus, it is important to separate or "isolate" these "driving" frequencies of the equipment from the "natural" frequencies of the building, as well as to reduce at all frequencies the structure-borne noise that is so often disturbing.

For a vibration isolation mount to be effective, the driving frequency of the source (that is to be isolated) should be higher than the natural frequency of the isolation mount by a factor of about 3 to 10, or even more. For many conditions, the higher this ratio, the more effective the isolation. If the natural frequency of the isolation mount just equals the driving frequency of the vibrating source, the system may go into violent oscillation at that frequency, limited only by the damping within the system. Some magnification of the vibration will occur, in fact, as long as the driving frequency is within the range of about 0.3 to 1.4 times the natural frequency of the mount. Below its natural frequency, an isolator provides no isolation. All of this is shown somewhat quantitatively in Figure 3, where a value of greater than unity in "transmissibility" means an increase in vibration (magnification) and a value of less than unity means a decrease in vibration (isolation).

In view of these frequency characteristics of an isolator, the first important step in the selection of a vibration mount is to be sure that the natural frequency of the isolator is lower (by a factor of at least 3 to 10, if possible) than the driving frequency to be protected. As an aid in taking this first step, the natural frequency (in Hz and in cycles per minute) of a vibration isolator is given in Table 1 (at the end of this section) as a function of the static deflection (in inches) of the isolator under load.

In most vibration isolator catalogues, there is usually a curve that purports to show the vibration isolation efficiency of a mount. According to that curve, isolation efficiencies of 80%, 90%, 95%, even 98% would appear to be quite commonplace and simple to achieve. The curve usually fails to state that these efficiencies can be achieved only when the isolated system is mounted on an infinitely massive and rigid base. An upper floor slab that deflects up to 1 in. when it is completely loaded hardly qualifies as infinitely rigid. In fact, the actual isolation efficiency of a mount decreases from the idealized maximum as the actual floor deflection increases. As an example, if the isolator static deflection just equals the floor static deflection, the practical limit of isolation efficiency for the mount approaches approximately 50% whereas it would have approached 100% for a completely rigid base. In effect, this means that high deflection floors (usually comparatively lightweight floors or large-span floors) require larger deflection isolators in order to achieve the desired degree of isolation.

Although there is no simple rule-of-thumb, a few suggestions are offered on estimating the desired static deflections of vibration isolators.

(1) For a highly critical installation, the natural frequency of the isolator should be about 1/6th to 1/10th the driving frequency that is to be controlled (or even lower), and the static deflection of the isolator should

not be less than about 6 to 10 times the static deflection of the floor when the equipment load is added.

(2) For a less critical situation, the natural frequency of the isolator should be about 1/3rd to 1/6th the driving frequency that is to be controlled (or even lower), and the static deflection of the isolator should not be less than about 3 to 6 times the static deflection of the floor when the equipment load is added.

Of course, when practical or economic limitations prohibit application of these suggestions, compromises have to be made. The need for compromise is illustrated by the vibration isolation of a cooling tower. Suppose a 75-HP cooling tower is mounted directly above an executive office that qualifies for an NC-25 criterion (in other words, it is a critical installation). Suppose the propeller fan runs at 240 RPM (a reasonable speed for a 10-ft to 14-ft blade diameter), and suppose the roof deck deflects an additional 1 in. when the fully loaded cooling tower is installed. According to one of the suggestions, the natural frequency of an isolator should be about 1/8th the driving frequency, or 30 RPM which is approximately 0.5 Hz. In Table 1, a spring with a 30-in. static deflection would meet this suggestion. Such a spring would stand about 6 ft tall when uncompressed--hardly practical! According to the second suggestion, the static deflection of the isolator should be about 6 to 10 times the floor deflection, or 6 to 10 in. This, at least, is possible although it still represents a spring about 2 ft tall when uncompressed. This illustrates one of the reasons why large cooling towers are usually installed on dunnage that is supported directly off the tops of the columns of the building instead of on roof decks. The compression of the columns is fairly negligible when the cooling tower load is added. A resulting 5-in. or 6-in. static deflection of the springs is a reasonable compromise decision. Note (from Table 1) that these springs would have a natural frequency of approximately 80 CPM or 1.3 Hz which is removed by a factor of 3 from the driving frequency of 240 CPM or 4 Hz. The cooling tower represents an extreme example because of the low shaft speed.

Before leaving this example, however, it should be pointed out that the next higher "driving" frequency of the cooling tower fan is the blade passage frequency, which would be about 40 Hz for a 10-bladed fan (240 RPM x 10 blades/60 = 40 Hz). A 5-in. to 6-in. static deflection spring with a natural frequency of 1.3 Hz can provide good isolation at 40 Hz (when properly used). Thus, noise or vibration at the blade passage frequency would be controlled by the spring. As an incidental point, people would not "hear" the noise of the cooling tower at the shaft speed of 240 RPM or 4 Hz, but they might feel the vibration caused by an unisolated and unbalanced fan. Hence, an isolator whose natural frequency is only 1/3rd that of the driving frequency would be worthwhile in this case.

For most other equipment, more reasonable values occur. For example, suppose a reciprocating-compressor operates at 1200 RPM and it is to be isolated over a critical area, and the floor deflection might be 1/4 in. when the machine is installed. According to one of the approaches, the static deflection of the isolator should be such that its natural frequency is 1/6th to 1/10th the driving frequency; in other words, a static deflection of between 0.9 and 2.4 in. By the second approach, the static deflection of the isolator should be about 6 to 10 times the floor deflection; in other words, about 1.5 to 2.5 in. Choosing the upper end of the range, a value of 2 to 2.5 in. static deflection would represent a practical isolator for this installation.

In the material that follows, a fairly complete collection of vibration isolation mounting details are given. These are based on experience with many actual installations. Paragraph 3 names some general conditions applicable to all mounting systems, Paragraphs 4-8 describe in general terms five "types of mounting systems" to be called upon specifically for use with certain equipment, and Paragraphs 9-22 give detailed recommendations for the vibration isolation of specific equipment, some of which require the mounting assemblies described in Paragraphs 4-8. Paragraphs 23-25 refer to piping, connections and auxiliary equipment.

3. GENERAL CONDITIONS

In the vibration isolation recommendations that follow, several general conditions are assumed. These are summarized here.

a. Building Uses. Isolation recommendations are given for three general equipment locations: (1) "on grade slab", (2) "on upper floor above non-critical area", and (3) "on upper floor above critical area". In spite of these categories of locations, it is assumed that the building under consideration is an occupied building involving many spaces that would require or deserve the low noise and vibration environments of such buildings as hotels, hospitals, office buildings and the like, as characterized by Categories 1-4 of the Noise Criterion table. Hence, the recommendations are aimed at providing the required low vibration levels throughout the building. Since an on-grade slab usually represents a more rigid base than is provided by a framed upper floor, the vibration isolation recommendations can generally be somewhat relaxed. Of course, vibration isolation treatments must be the very best when a high quality occupied area is located immediately under the MER, as compared with the case where a "buffer zone" or non-critical area is located between the MER and the critical area.

If a building is intended to serve entirely for such uses as those of Categories 5 and 6 of the Noise Criterion tables, the recommendations given here are too severe and can be simplified at the user's discretion.

b. Floor Slab Thickness. It is assumed that MER upper floor slabs will be constructed of dense concrete of minimum 140-150 lb/cu. ft density, or if lighter concrete is used, the thickness will be increased to provide the equivalent total mass of the specified floor. For large MERs containing arrays of large and heavy equipment, it is assumed that the floor slab thickness will be in the range of 8 to 12 in., with the greater thicknesses required by the greater floor loads. For smaller MERs containing smaller collections of lighter-weight but typical equipment, floor slab thicknesses of 6 to 10 in. are assumed. For occasional locations of one or a very few pieces of small high speed equipment (say 1800 RPM or higher) having no reciprocating action, floor slabs of 4 to 6 in. may be used with reasonable expectation of satisfactory results. However, for reciprocating action machines operating at the lower speeds (say, under 1200 RPM), any reduced floor slab thicknesses from those listed above begin to invite problems. There is no clear cross-over from "acceptable" to "unacceptable" in terms of floor slab thickness, but each reduction in thickness increases the probability of later difficulties due to vibration.

The thicknesses mentioned here are based on experience with the "acoustics" of equipment installations. These statements on thicknesses are in no way intended to represent structural specifications for a building.

"Housekeeping pads" under the equipment are assumed, but the height of these pads is not to be used in calculating the thickness of the floor slab.

c. Steel Spring Isolators. As a general rule, unhoused free-standing stable steel springs are preferred over housed spring assemblies. The housed spring is frequently an "unstable spring"; that is, it will tilt sidewise as it is loaded, and the housing is required to keep it in a somewhat upright orientation. In so doing, however, the housing (or its internal lining of neoprene strips) tends to short-circuit the coils of the spring or even bind the spring when it is badly "tilted" inside the housing. All of this reduces the effectiveness of the spring. Further, the housing frequently so encloses the spring that it is hidden from view, and inspection is made difficult.

On the other hand, the stable steel spring has a larger diameter and requires more space (the diameter is comparable to the compressed height), but it is clearly in view for critical inspection (if not recessed inside an enclosing pocket of a concrete inertia block).

Visual inspection is an important step in providing a satisfactory installation, and the engineer can perform a useful service by designing the mounting system to be "easy to inspect".

The horizontal stiffness of the spring should be approximately equal to its vertical stiffness.

d. Steel Springs Plus Neoprene Pads. It is a specific recommendation that whenever a steel spring is used, at least one and preferably two pads of ribbed or waffle-pattern neoprene be used in series with the spring (either under the base of the spring or on top of the spring). It is suggested that this pad be considered in addition to the anti-skid ribbed pad that is frequently supplied already cemented to the bottom of the spring base. Grout or building dirt frequently fills up the cavities or grooves of the anti-skid pad and it loses the effectiveness it might once have had.

From a simple point of view, a steel spring is a coiled-up rod of steel that connects a piece of noisy equipment to a floor. The coiled rod is very effective in isolating low frequency vibration from the floor, but, like any rod, it will transmit high frequency noise from the machine to the floor. A rubber or neoprene isolator, however, is most effective at high frequency. Thus, if a steel spring is used in series with a neoprene pad, both the low frequency vibration and the high frequency structure-borne noise are reduced.

In the material that follows, whenever a steel spring is specified, at least one and preferably two ribbed or waffle-pattern neoprene pads should be used with that spring (even though this statement is not repeated with every spring specification)! Although there are times when this is not necessary, the exceptions are too few to discuss, so they will be ignored.

For many equipment installations, there is no need to bolt down the isolation mounts to the floor because the smooth operation of the machine and the weight of the complete assembly keep the system from moving. For some systems, however, it may be necessary to restrain the equipment from "creeping" across the floor. In these situations, it is imperative that the hold-down bolts not short-circuit the neoprene pads.

e. Structural Ties, Rigid Connections. Each piece of isolated equipment must be free of any structural tie or rigid connection that can short-circuit the isolation joint. Electrical conduit should be long and "floppy" so that it does not offer any resistance or constraint to the free movement of the equipment. Piping should be resiliently supported or contain flexible connections as discussed later. Limit stops on the spring isolators, shipping bolts on the isolators and leveling bolts on the isolators should be set and inspected to insure that they are not inadvertently short-circuiting the spring mounts.

All building trash should be removed from under the isolated base of the equipment. This seemingly innocent supplication becomes more meaningful when a wastebasket full of loose grout, 2x4s, nuts, bolts, soft drink bottles, beer cans, welding rods, pipes and pipe couplings are removed from beneath a single base, after the contractor has left the job but could not understand why the isolated equipment was still noisy on the floor below.

It is recommended that large 2-in. to 4-in. clearances be provided under all isolated equipment bases in order to facilitate inspection and removal of trash from under the base.

4. TYPE I MOUNTING ASSEMBLY

The specified equipment should be mounted rigidly on a large integral concrete inertia block. (Unless specified otherwise, all concrete referred to in this manual should have a density of at least 140-150 lb/cu ft.) The length and the width of the inertia block should be at least 50% greater than the length and width of the supported equipment. Mounting brackets for stable steel springs should be located off the sides of the inertia block at or near the height of the vertical center-of-gravity of the combined completely assembled equipment and concrete block. If necessary, curbs or pedestals should be used under the base of the steel springs in order to bring the top of the loaded springs up to the center-of-gravity position. As an alternate, the lower portion of the concrete inertia block can be lowered into a pit or cavity in the floor so that the steel springs will not have to be mounted on curbs or pedestals. In any event, the clearance between the floor (or all the surfaces of the pit) and the concrete inertia block shall be at least 4 in. and provision should be allowed to check this clearance at all points under the block.

The ratio of the weight of the concrete inertia block to the total weight of all the supported equipment (including the weight of any attached filled piping up to the point of the first pipe hanger) shall be in accordance with the recommendations given in the paragraph and table for the particular equipment requiring this mounting assembly. The inertia block adds stability to the system and reduces motion of the system in the vicinity of the driving frequency. For reciprocating machines or for units involving large starting torques, the inertia block provides much-needed stability.

The static deflection of the free-standing stable steel springs shall be in accordance with the recommendations given in the paragraph and table for this particular equipment. There shall be adequate clearance all around the springs to assure no contact between any spring and any part of the mounted assembly for any possible alignment or position of the installed inertia block.

5. TYPE II MOUNTING ASSEMBLY

This mount is the same as the Type I Mount in all respects except (1) the mounting brackets and the top of steel springs shall be located as high as practical on the concrete inertia block but not necessarily as high as the vertical center-of-gravity position of the assembly, and (2) the clearance between the floor and the concrete inertia block shall be at least 2 in.

If necessary, the steel springs can be recessed into pockets in the concrete block, but clearances around the springs should be large enough to assure no contact between any spring and any part of the mounted assembly for any possible aligiment or position of the installed inertia block. Provision must be made to allow positive visual inspection of the spring clearance in its recessed mounting.

When this type mounting is used for a pump, the concrete inertia block can be given a T-shape in plan and the pipes to and from the pump can be supported rigidly with the pump onto the "wings" of the T. In this way the pipe elbows will not be placed under undue stress.

The weight of the inertia block and the static deflection of the mounts shall be in accordance with the recommendations given in the table for the particular equipment.

6. TYPE III MOUNTING ASSEMBLY

The equipment or the assembly of equipment should be mounted on a sufficiently stiff steel frame that the entire assembly can be supported on flexible point supports without fear of distortion of the frame or mis-alignment of the equipment. The frame is then mounted on resilient mounts, either steel springs or neoprene-in-shear mounts or isolation pads, as the static deflection would require. If the equipment frame itself already has adequate stiffness, no additional framing is required and the isolation mounts may be applied directly to the base of the equipment.

The vibration-isolated assembly should have enough clearance under and all around the equipment to prohibit contact with any structural part of the building during operation. If the equipment has large starting and stopping torques and the isolation mounts have large static deflections, consideration should be given to providing limit stops on the mounts. Limit stops might also be desired for large deflection isolators if the filled and unfilled weights of the equipment are very different.

7. TYPE IV MOUNTING ASSEMBLY

The equipment should be mounted on an array of "pad mounts". The pads may be of compressed glass fiber or of multiple layers of ribbed-neoprene or waffle-pattern neoprene of sufficient height and of proper stiffness to support the load while meeting the static deflection recommended in the applicable accompanying tables. Cork, cork-neoprene or felt pad materials may be used if their stiffness characteristics are known and providing they can be replaced periodically whenever they have become sufficiently compacted that they no longer provide adequate isolation.

The floor should be grouted or shimmed to assure a level base for the equipment and therefore a predictable uniform loading on the isolation pads. The pads should be loaded in accordance with the loading rates recommended by the pad manufacturer for the particular densities or durometers involved. In general, most of these pads are intended for load rates or 30-60 psi, and if they are underloaded (for example, at less than about 10 psi) they will not be performing at their maximum effectiveness.

8. TYPE V MOUNTING ASSEMBLY (FOR PROPELLER-TYPE COOLING TOWERS)

Large, low-speed propeller-type cooling towers located on roof decks of large buildings may produce serious vibration in their buildings if adequate vibration isolation is not provided. In extreme cases, the vibration may be evident two or three floors below the cooling towers.

It is recommended that the motor, drive shaft, gear reducer and propeller be mounted as rigidly as possible on a "unitized" structural support and that this entire assembly be isolated from the remainder of the tower with stable steel springs in accordance with Table 8. Adequate clearance between the propeller tips and the cooling tower shroud should be provided to allow for starting and stopping vibrations of the propeller assembly. Several of the cooling tower manufacturers provide isolated assemblies as described here.

In addition, where the cooling tower is located on a roof deck directly over an acoustically critical area, the waterfall noise may be objectionable and this can be reduced with the use of 4 layers of ribbed or waffle-pattern neoprene located between the base of the cooling tower and the supporting structure of the building. This latter treatment is usually not necessary if there is a non-critical area immediately under the cooling tower.

A single treatment alternate to the combined two treatments mentioned above is the isolation of the entire cooling tower assembly on stable steel springs, also in accordance with Table 8. The springs should be in series with at least two layers of ribbed or waffle-pattern neoprene if there is an acoustically critical area immediately below the cooling tower (or within about 25 ft horizontally on the floor immediately under the tower). It may be desirable to provide limit stops on these springs to limit movement of the tower when it is emptied.

Pad materials, when used, should not be short-circuited by bolts or rigid connections. Cooling tower piping should be vibration isolated in accordance with suggestions given for piping.

9. TABLES OF RECOMMENDED VIBRATION ISOLATION DETAILS

A common format is used for all the tables that summarize the recommended vibration isolation details for the various types of equipment. A brief description of the format is given here. The reader may refer to any one of the typical tables included as Tables 2-13.

The three columns on the left of the table define the equipment conditions covered by the recommendations: location, "rating" and speed of the equipment. The "rating" is given by a "power" range for some equipment, "cooling capacity" for some and "heating capacity" for some. The rating and speed ranges generally cover the range of equipment that might be encountered in a typical building. Subdivisions in rating and speed are made to accomodate variations in the isolation details.

The three columns on the right of the table summarize three basic groups of recommendations: Column 1, the type of mounting (from Paragraphs 4-8); Column 2, the suggested minimum ratio of the weight of the inertia block (when required) to the total weight of all the equipment mounted on the inertia block; and Column 3, the suggested minimum static deflection of the isolator to be used.

Regarding the weight of the inertia block, the larger weight of the range given should be applied: (1) where the nearby critical area is very critical--such as Category 1 or 2 areas, (2) where the speed of the equipment is near the lower limit of the speed range given, or (3) where the rating of the equipment is near the upper limit of the rating range. Conversely, the lower end of the weight range may be applied: (1) where the nearby critical area is less critical--such as Category 3 or 4, (2) where the speed is near the upper limit of the speed range, or (3) where the rating is near the lower limit of the rating range.

Regarding the static deflection of the isolators, these minimum values are keyed to the approximate span of the floor beams; that is, as the floor span increases, the floor deflection increases; and therefore the isolator deflection must increase. The specified minimum deflection in effect specifies the type of isolator that can be used:

Deflection Range	Isolator
1/2 in. and over	Steel Spring or Air Spring*
0.3 to 0.5 in.	Double deflection neoprene-in-shear
0.10 to 0.25 in.	Neoprene-in-shear, or 1-in. to 2-in. thick compressed glass fiber pads, or 2 to 4 layers ribbed or waffle-pattern neoprene pads

The three categories of equipment location are probably obvious. Two other categories should be mentioned, although they are not specifically listed. If vibrating equipment is to be supported or hung from the overhead floor slab, immediately beneath an acoustically critical area, provide the same degree of vibration isolation as recommended for the location designated as "on upper floor above critical area". Similarly, if the vibrating equipment is hung from an overhead floor slab beneath a non-critical area, provide the same vibration

*Air Springs are excellent as low-frequency isolators for special problems. They are not yet considered practical, however, for low-maintenance equipment mounts since they require a pressure-controlled air supply and occasional inspection for proper operation.

isolation as recommended for the location designated as "on upper floor above non-critical area".

10. CENTRIFUGAL AND AXIAL-FLOW FANS

The recommended vibration isolation mounting details for fans are given in Table 2, at the end of this section. Ducts should contain flexible connections at both the inlet and discharge of the fans, and all connections to the fan assembly should be clearly flexible. The entire assembly should bounce easily and with little restraint when one jumps up and down on the unit.

Where supply fan assemblies are located over critical areas, it is desirable to install the entire inlet casing and all auxiliary equipment (coil decks and filter sections) on "floated concrete slabs". The floated slab may also serve to reduce airborne noise from the fan inlet area into the floor area below.

Large ducts (cross-section area over 15 sq. ft) that are located within about 30 ft of the inlet or discharge of a large fan (over 20 HP) should be supported from the floor or ceiling with resilient mounts having a static deflection of at least 1/4 in.

11. RECIPROCATING-COMPRESSOR REFRIGERATION EQUIPMENT

The recommended vibration isolation details for this equipment are summarized in Table 3. These recommendations apply also to the drive unit used with the reciprocating compressor.

Pipe connections from this assembly to other equipment should contain flexible connections (see Paragraph 24) and piping should be given resilient support (see Paragraph 23).

12. ROTARY-SCREW-COMPRESSOR REFRIGERATION EQUIPMENT

The recommended vibration isolation details for this equipment are summarized in Table 4. Piping to and from this equipment should be given resilient support (Paragraph 23).

13. CENTRIFUGAL-COMPRESSOR REFRIGERATION EQUIPMENT

The recommended vibration isolation details for this equipment, including the drive unit and the condenser and chiller tanks, are summarized in Table 5. Piping to and from this assembly should be given resilient support (Paragraph 23).

14. ABSORPTION-TYPE REFRIGERATION EQUIPMENT

The recommended vibration isolation details for this equipment are summarized in Table 6. Piping should be given resilient support (Paragraph 23).

15. BOILERS

The recommended vibration isolation details for boilers are summarized in Table 7. These apply for boilers with integrally attached blowers, but these do not necessarily apply to blowers on separate mounts.

Piping should be given resilient support in accordance with Paragraph 23. A flexible connection or a thermal expansion joint should be installed in the exhaust breaching between the boiler and the exhaust stack.

16. STEAM VALVES

Steam valves are usually supported entirely on their pipes; refer to Paragraph 23 for the resilient support of piping, including steam valves.

17. COOLING TOWERS

The recommended vibration isolation details for propeller-type cooling towers are summarized in Table 8. Additional details for the installation are given in Paragraph 8 which describes the Type V mounting assembly.

The recommended vibration isolation details for centrifugal-fan cooling towers are summarized in Table 9. Cooling tower piping should be isolated in accordance with Paragraph 23.

18. MOTOR-PUMP ASSEMBLIES

Recommended vibration isolation details for motor-pump units are summarized in Table 10. Electrical connections to the motors should be made with long "floppy" lengths of flexible armored cable, and piping should be resiliently supported as in Paragraph 23. For most situations, a good isolation mounting of the piping will overcome the need for flexible connections in the pipe.

An important function of the concrete inertia block (Type II mounting) is its stabilizing effect against undue "bouncing" of the pump assembly at the instant of starting. This gives better long-time protection to the associated piping.

These same recommendations may be applied to other motor-driven rotary devices such as centrifugal-type air compressors and motor-generator sets in the power range up to a few hundred horsepower.

19. STEAM TURBINES

Table 11 provides a set of general vibration isolation recommendations for steam-turbine-driven rotary equipment, such as gears, generators or centrifugal-type gas compressors. When a steam turbine is used to drive centrifugal-compressor refrigeration equipment, refer to the material given in Table 5; and when it is used to drive reciprocating-compressor refrigeration equipment or reciprocating-type gas compressors, refer to the recommendations given in Table 3.

Piping associated with the steam turbine and the remainder of the assembly should be vibration isolated according to Paragraph 23.

20. GEARS

When a gear is involved in a drive system, vibration isolation should be provided in accordance with recommendations given for either the main power drive unit or the driven unit, whichever imposes the more stringent isolation conditions. The more stringent conditions are usually those requiring the largest inertia block or the largest static deflection for the spring mounts. Tables 3, 5 and 11 may possibly be involved in the comparison.

21. TRANSFORMERS

Recommended vibration isolation details for indoor transformers are given in Table 12. In addition, power leads to and from the transformers should be as flexible as possible.

In outdoor locations, earth-borne vibration to nearby neighbors is usually not a problem, so no vibration isolation is suggested. If vibration should become a problem, the transformer could be installed on neoprene or compressed glass fiber pads having 1/4 in. static deflection.

22. AIR COMPRESSORS

Recommended mounting details for centrifugal-type air compressors of less than 100 HP are the same as those given for motor-pump units in Table 10. The same recommendations would apply for small (under 10 HP) reciprocating-type air compressors. For reciprocating-type air compressors (with more than two cylinders) in the 10-50 HP range, follow the recommendations given in Table 3 for the particular conditions.

For 10-100 HP one- or two-cylinder reciprocating-type air compressors, the recommendations of Table 13 apply. This equipment is a potentially serious source of low frequency vibration in a building if it is not isolated. In fact, the compressor should not be located in certain parts of the building, even if it is vibration isolated. The "forbidden" locations are indicated in Table 13.

When these compressors are used, all piping should contain flexible connections and the electrical connections should be made with flexible armored cable. Refer to Paragraph 24 for use of flexible connections and to Paragraph 23 for resilient pipe supports.

23. RESILIENT PIPE SUPPORTS

All piping in the MER that is connected to vibrating equipment should be supported from resilient ceiling hangers or from floor-mounted resilient supports. As a general rule, the first three pipe supports nearest the vibrating equipment should have a static deflection of at least one-half the static deflection of the mounting system used with that equipment. Beyond the third pipe support, the static deflection can be reduced to approximately 1/2 in. for the remainder of the pipe run in the MER.

When a pipe passes through the MER wall, a minimum 1 in. clearance should be provided between the pipe and the hole in the wall. The pipe should be supported on either side of the hole, so that the pipe does not rest on the wall. The clearance space should then be stuffed with fibrous filler material and sealed with a non-hardening caulking compound.

Vertical pipe chases through a building should not be located beside acoustically critical areas (Categories 1-3). If they are located beside critical areas, pipes should be resiliently mounted from the walls of the pipe chase for a distance of at least 10 ft beyond each such area.

Pipes to and from the cooling tower should be resiliently supported for their full length between the cooling tower and the associated MER. Steam pipes should be resiliently supported for their entire length of run inside the building. Resilient mounts should have a static deflection of at least 1/2 in.

In highly critical areas, domestic water pipes and waste lines can be isolated with the use of 1/4 in. to 1/2 in. thick wrappings of felt pads under the pipe strap or pipe clamp.

As mentioned earlier, whenever a steel spring isolator is used, it should be in series with a neoprene isolator. For ceiling hangers, a neoprene washer or grommet should always be included; and if the pipe hangers are near very critical areas, the hanger should be a combination hanger that contains both a steel spring and a neoprene-in-shear mount.

During inspection, check that the hanger rods are not touching the sides of the isolator housing and thereby shorting-out the spring.

24. FLEXIBLE PIPE CONNECTIONS

To be at all effective, a flexible pipe connection should have a length that is approximately 6 to 10 times its diameter. Tie rods should not be used to bolt the two end flanges of a flexible connection together. Flexible connections are either of the bellows-type or are made up of wire-reinforced neoprene piping, sometimes fitted with an exterior braided jacket to confine the neoprene. These connections are useful when the equipment is subject to fairly high-amplitude vibration, such as for reciprocating-type compressors. Flexible connections generally are not necessary when the piping and its equipment are given thorough and compatible vibration isolation.

25. NON-VIBRATING EQUIPMENT

When an MER is located directly over or near a critical area, it is usually desirable to isolate most of the "non-vibrating equipment" with a simple mount made up of one or two pads of neoprene or a 1 in. or 2 in. layer of compressed glass fiber. Heat exchangers, hot water heaters, water storage tanks, large ducts and some large pipe stands may not themselves be noise sources, yet their pipes or their connections to vibrating sources transmit small amounts of vibrational energy that they then may transmit into the floor. A simple minimum isolation pad will usually prevent this noise transfer.

26. SUMMARY

In this section, fairly complete vibration isolation mounting details are laid out for most of the equipment included in an MER. Most of these details have been developed and proven over many years of use. Although all the entries of the accompanying tables probably have not been tested in actual equipment installations, the schedules are fairly self-consistent in terms of various locations and degrees of required isolation. Hence, the mounting details are considered quite realistic and fairly reliable. They are not extravagant in their make-up when considered in the light of the extremely low vibration levels required to achieve near-inaudibility.

The noise and vibration control methods given here are designed to be simple to follow and to put into use. If these methods and recommendations are carried out, with appropriate attention to detail, most equipment installations will be tailored to the specific needs of the building and will give very satisfactory results acoustically.

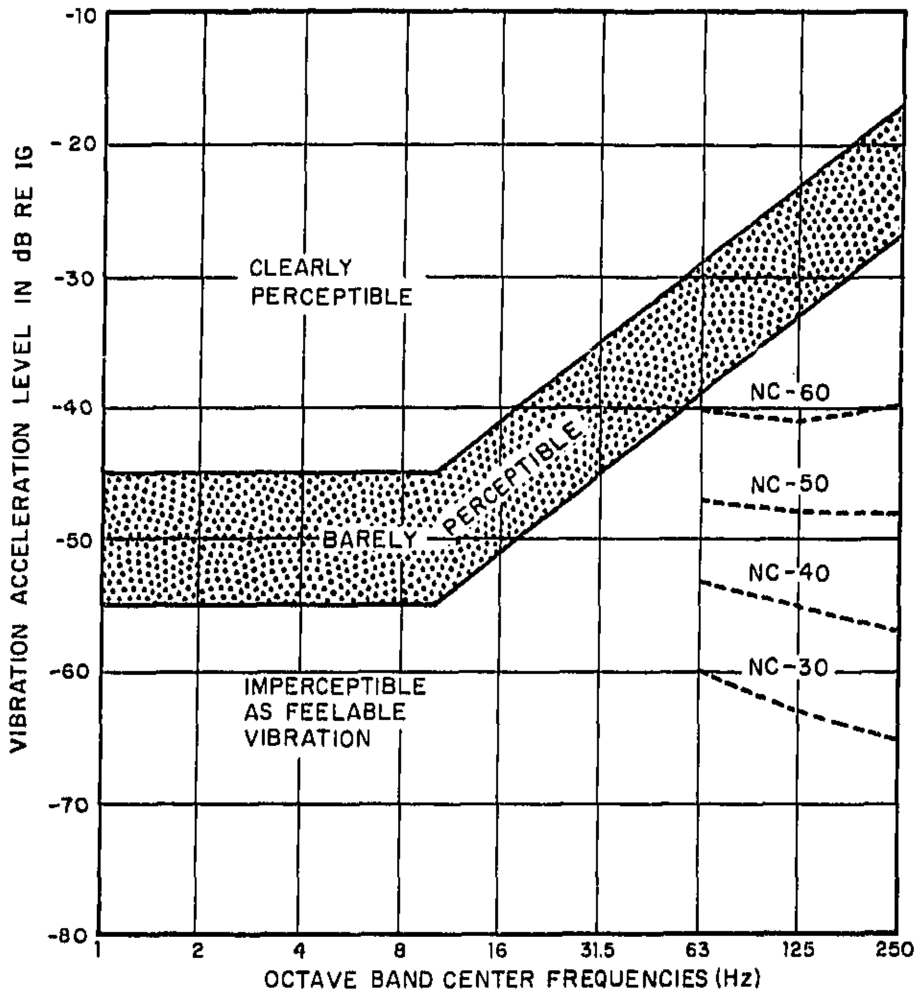


FIG. 1 VIBRATION ACCELERATION LEVELS NEAR THE ONSET OF "FEELABILITY"

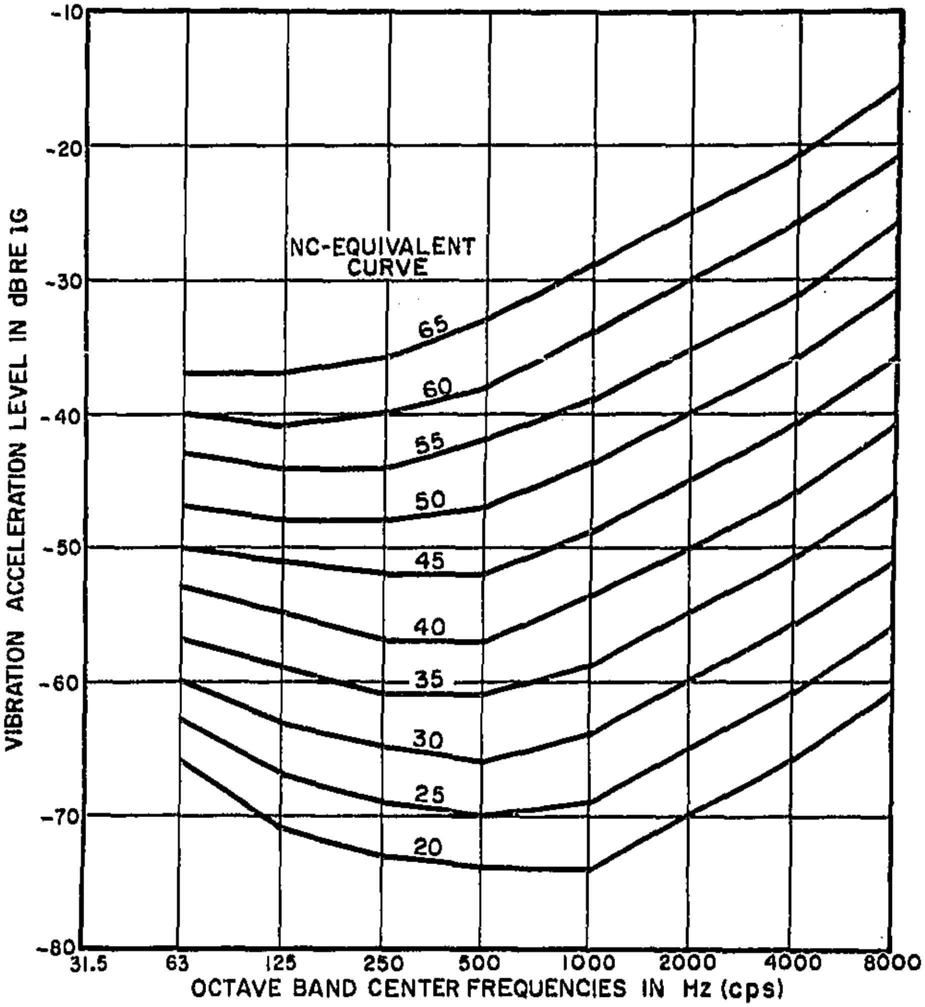


FIG. 2 ACCELERATION LEVELS OF A LARGE VIBRATING SURFACE THAT WILL PRODUCE RADIATED SOUND LEVELS INTO A ROOM APPROXIMATING THE SOUND LEVELS OF THE NC CURVES.

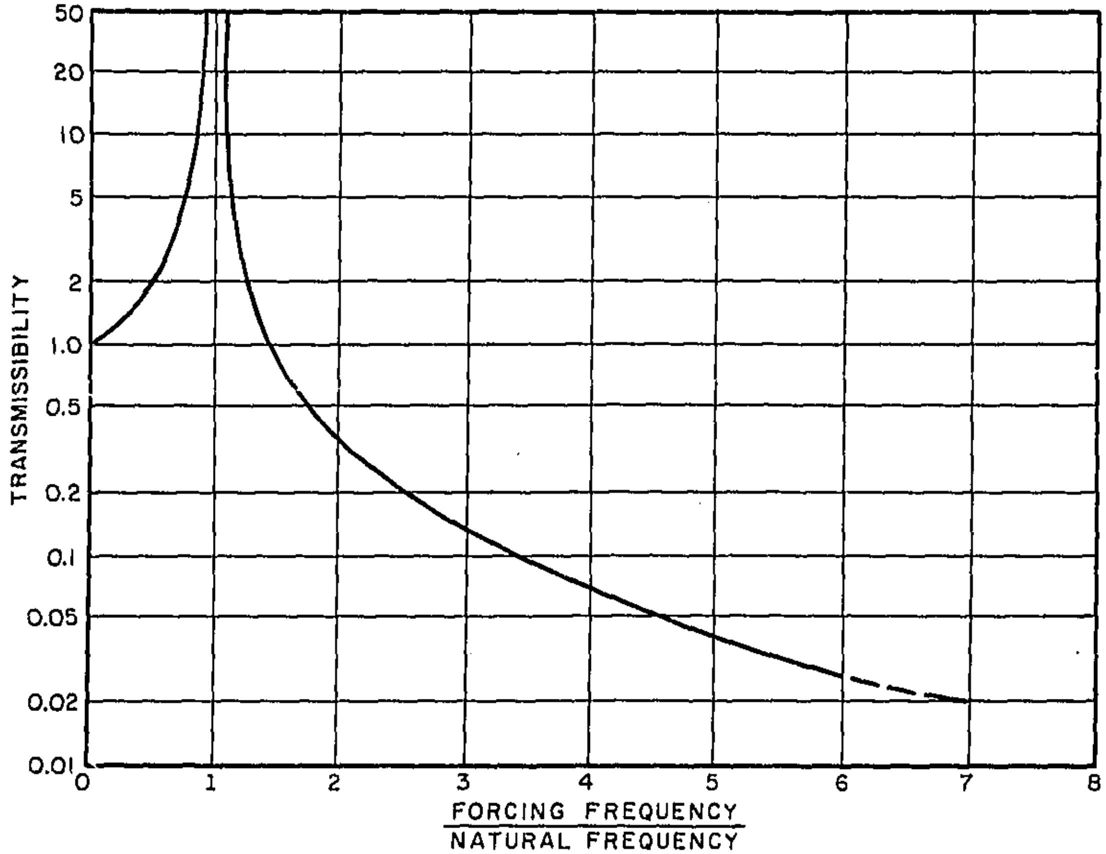


FIG. 3 TRANSMISSIBILITY OF UNDAMPED SINGLE-DEGREE-OF-FREEDOM SYSTEM.

TABLE 1

THE NATURAL FREQUENCY OF AN ISOLATOR
AS A FUNCTION OF ITS STATIC DEFLECTION
UNDER LOAD

$$F_n = 3.13\sqrt{\frac{1}{S.D.}}\ \text{Hz} = 188\sqrt{\frac{1}{S.D.}}\ \text{CPM}$$

STATIC DEFLECTION (IN.)	NATURAL FREQUENCY (Hz)	NATURAL FREQUENCY (CPM)	STATIC DEFLECTION (IN.)	NATURAL FREQUENCY (Hz)	NATURAL FREQUENCY (CPM)
0.02	22	1320	2.7	1.9	114
0.04	15.7	940	3.0	1.8	108
0.06	12.8	770	3.4	1.7	102
0.1	10	600	3.8	1.6	96
0.2	7	420	4.4	1.5	90
0.3	6.0	360	5	1.4	84
0.4	5.0	300	6	1.3	78
0.6	4.0	240	7	1.2	72
0.8	3.5	210	8	1.1	66
1.1	3.0	180	10	1.0	60
1.2	2.8	168	12	0.9	54
1.4	2.6	156	15	0.8	48
1.7	2.4	144	20	0.7	42
2.0	2.2	132	27	0.6	36
2.4	2.0	120	39	0.5	30

TABLE 2

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR

CENTRIFUGAL AND AXIAL-FLOW FANS

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (HP)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3 30'	COLUMN 3 40'	COLUMN 3 50'
ON GRADE SLAB	UNDER 3	UNDER 600	NO ISOLATION REQUIRED				
		600-1200					
		OVER 1200					
	3-25	UNDER 600	III	—	1"		
		600-1200			½"		
		OVER 1200			½"		
	26-200	UNDER 600	III	—	1½"		
		600-1200			1"		
		OVER 1200			½"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 3	UNDER 600	III	—			
		600-1200	III	—			
		OVER 1200	III	—			
	3-25	UNDER 600	II	2	1"	1½"	2"
		600-1200	III	—	1½"	2"	3"
		OVER 1200	III	—	1"	1½"	2"
	26-200	UNDER 600	II	2	2"	3"	4"
		600-1200	II	2	1½"	2"	3"
		OVER 1200	II	2	1"	1½"	2"
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 3	UNDER 600	II	2	1½"	2"	3"
		600-1200	III	—	1½"	2"	3"
		OVER 1200	III	—	1"	1½"	2"
	3-25	UNDER 600	II	3	2"	3"	4"
		600-1200	II	2	1½"	2"	3"
		OVER 1200	II	2	1"	1½"	2"
	26-200	UNDER 600	II	3	3"	4"	5"
		600-1200	II	2	2"	2½"	3"
		OVER 1200	II	2	1"	1½"	2"

COL 1: MOUNTING TYPE (SEE TEXT)

COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD

COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 3

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR RECIPROCATING-COMPRESSOR REFRIGERATION EQUIPMENT ASSEMBLY (INCLUDING MOTOR, GEAR OR STEAM TURBINE DRIVE UNIT)

<table>
<tr><th colspan="3">EQUIPMENT CONDITIONS</th><th colspan="5">MOUNTING RECOMMENDATIONS</th></tr>
<tr><th rowspan="2">EQUIPMENT LOCATION</th><th rowspan="2">COOLING CAPACITY (TONS)</th><th rowspan="2">SPEED RANGE (RPM)</th><th rowspan="2">COLUMN 1</th><th rowspan="2">COLUMN 2</th><th colspan="3">COLUMN 3</th></tr>
<tr><th>30'</th><th>40'</th><th>50'</th></tr>
<tr><td rowspan="6">ON GRADE SLAB</td><td rowspan="3">10-50</td><td>600-900</td><td>III</td><td></td><td colspan="3">2"</td></tr>
<tr><td>901-1200</td><td>III</td><td></td><td colspan="3">1½"</td></tr>
<tr><td>1201-2400</td><td>III</td><td></td><td colspan="3">1"</td></tr>
<tr><td rowspan="3">51-175</td><td>600-900</td><td>II</td><td>2-3</td><td colspan="3">2"</td></tr>
<tr><td>901-1200</td><td>III</td><td></td><td colspan="3">2"</td></tr>
<tr><td>1201-2400</td><td>III</td><td></td><td colspan="3">1½"</td></tr>
<tr><td rowspan="6">ON UPPER FLOOR ABOVE NON-CRITICAL AREA</td><td rowspan="3">10-50</td><td>600-900</td><td>II</td><td>2-3</td><td>2"</td><td>3"</td><td>4"</td></tr>
<tr><td>901-1200</td><td>II</td><td>2-3</td><td>1½"</td><td>2"</td><td>3"</td></tr>
<tr><td>1201-2400</td><td>II</td><td>2-3</td><td>1½"</td><td>1½"</td><td>2"</td></tr>
<tr><td rowspan="3">51-175</td><td>600-900</td><td>II</td><td>3-4</td><td>3"</td><td>4"</td><td>5"</td></tr>
<tr><td>901-1200</td><td>II</td><td>3-4</td><td>2"</td><td>3"</td><td>4"</td></tr>
<tr><td>1201-2400</td><td>II</td><td>2-3</td><td>2"</td><td>2"</td><td>3"</td></tr>
<tr><td rowspan="6">ON UPPER FLOOR ABOVE CRITICAL AREA</td><td rowspan="3">10-50</td><td>600-900</td><td>II</td><td>3-4</td><td>3"</td><td>4"</td><td>5"</td></tr>
<tr><td>901-1200</td><td>II</td><td>3-4</td><td>2"</td><td>3"</td><td>4"</td></tr>
<tr><td>1201-2400</td><td>II</td><td>2-3</td><td>2"</td><td>2"</td><td>3"</td></tr>
<tr><td rowspan="3">51-175</td><td>600-900</td><td>I</td><td>4-6</td><td>3"</td><td>4"</td><td>5"</td></tr>
<tr><td>901-1200</td><td>II</td><td>3-5</td><td>2"</td><td>3"</td><td>4"</td></tr>
<tr><td>1201-2400</td><td>II</td><td>3-4</td><td>2"</td><td>2"</td><td>3"</td></tr>
</table>

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 4

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR
ROTARY-SCREW-COMPRESSOR REFRIGERATION EQUIPMENT ASSEMBLY
(INCLUDING MOTOR DRIVE UNIT)

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	COOLING CAPACITY (TONS)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	100-500	2400-4800	III		1"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	100-500	2400-4800	III		1"	$1\frac{1}{2}$"	2"
ON UPPER FLOOR ABOVE CRITICAL AREA	100-500	2400-4800	II	2-3	1"	$1\frac{1}{2}$"	2"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 5

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR
CENTRIFUGAL-COMPRESSOR REFRIGERATION EQUIPMENT ASSEMBLY
(INCLUDING CONDENSER AND CHILLER TANKS AND
MOTOR, GEAR OR STEAM TURBINE DRIVE UNIT)

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	COOLING CAPACITY (TONS)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	100-500	OVER 3000	III		3/4"		
	501-2000	OVER 3000	III		1"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	100-500	OVER 3000	III		1"	$1\frac{1}{2}$"	2"
	501-2000 PACKAGED	OVER 3000	III		$1\frac{1}{2}$"	2"	3"
	501-2000 BUILT-UP	OVER 3000	II	2-3	$1\frac{1}{2}$"	2"	3"
ON UPPER FLOOR ABOVE CRITICAL AREA	100-500	OVER 3000	II	2-3	$1\frac{1}{2}$"	2"	3"
	501-2000	OVER 3000	II	3-5	$1\frac{1}{2}$"	2"	3"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 6

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR
ABSORPTION-TYPE REFRIGERATION EQUIPMENT ASSEMBLY

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	COOLING CAPACITY (TONS)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	ALL SIZES		IV		$\frac{1}{4}$"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	ALL SIZES		III		$\frac{1}{2}$"	3/4"	1"
ON UPPER FLOOR ABOVE CRITICAL AREA	ALL SIZES		III		1"	$1\frac{1}{2}$"	2"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF ISOLATOR IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 7

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR BOILERS

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	HEATING CAPACITY (BHP)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	UNDER 200		---		NOT REQUIRED		
	200-1000		IV		1/8"		
	OVER 1000		IV		1/4"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 200		III		1/8"	¼"	½"
	200-1000		III		¼"	½"	1"
	OVER 1000		III		¼"	½"	1"
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 200		III		½"	1"	1½"
	200-1000		III		1"	1½"	2"
	OVER 1000		III		1"	1½"	2"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF ISOLATOR IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 8

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR PROPELLER-TYPE COOLING TOWERS (WHERE SEVERAL TOWERS ARE PLACED AT THE SAME GENERAL LOCATION, USE POWER RANGE FOR TOTAL POWER OF ALL TOWERS)

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (HP)	FAN SPEED (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB OUTSIDE THE BUILDING	VIBRATION ISOLATION USUALLY NOT REQUIRED						
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 25	150-300 301-600 OVER 600	V	INSTALL ON DUNNAGE ATTACHED TO BUILDING COLUMNS ONLY	5" 3" 3"	SPRINGS MAY BE LOCATED UNDER DRIVE ASSEMBLY OR UNDER TOWER BASE	
	25-150	150-300 301-600 OVER 600	V		6" 4" 3"		
	OVER 150	150-300 301-600 OVER 600	V		6" 5" 4"		
ON UPPER FLOOR ABOVE CRITICAL AREA	SAME AS FOR LOCATION ABOVE NON-CRITICAL AREA, EXCEPT INSTALL RIBBED OR WAFFLE-PATTERN NEOPRENE BETWEEN TOWER AND BUILDING.						

COL 1: MOUNTING TYPE (SEE TEXT).
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 9

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR CENTRIFUGAL-TYPE COOLING TOWERS
(POWER IS TOTAL OF ALL FANS AT THE SAME GENERAL LOCATION)

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (HP)	FAN SPEED (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB OUTSIDE THE BUILDING	VIBRATION ISOLATION USUALLY NOT REQUIRED						
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 25	450-900	III		1"	1½"	2"
		901-1800			3/4"	1"	1½"
		OVER 1800			3/4"	1"	1½"
	25-150	450-900	III		1½"	2"	3"
		901-1800			1"	1½"	2"
		OVER 1800			3/4"	1"	1½"
	OVER 150	450-900	III		2"	3"	4"
		901-1800			1½"	2"	3"
		OVER 1800			1"	1½"	2"
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 25	450-900	III		1½"	2"	3"
		901-1800			1"	1½"	2"
		OVER 1800			3/4"	1"	1½"
	25-150	450-900	III		2"	3"	4"
		901-1800			1½"	2"	3"
		OVER 1800			1"	1½"	2"
	OVER 150	450-900	III		3"	4"	6"
		901-1800			1½"	2"	3"
		OVER 1800			1"	1½"	2"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 10

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR MOTOR-PUMP ASSEMBLIES

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (HP)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3 30'	COLUMN 3 40'	COLUMN 3 50'
ON GRADE SLAB	UNDER 20	450-900	II	$1\frac{1}{2}$-$2\frac{1}{2}$	$1\frac{1}{2}$"		
		901-1800	II OR III	2±	$1\frac{1}{2}$"		
		1801-3600	II OR III	2±	1"		
	20-100	450-900	II	2-3	$1\frac{1}{2}$"		
		901-1800	II	$1\frac{1}{2}$-$2\frac{1}{2}$	1"		
		1801-3600	II	$1\frac{1}{2}$-$2\frac{1}{2}$	3/4"		
	OVER 100	450-900	II	2-3	2"		
		901-1800	II	2-3	$1\frac{1}{2}$"		
		1801-3600	II	$1\frac{1}{2}$-$2\frac{1}{2}$	1"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 20	450-900	II	2-3	$1\frac{1}{2}$"	2"	3"
		901-1800	II	$1\frac{1}{2}$-$2\frac{1}{2}$	1"	$1\frac{1}{2}$"	2"
		1801-3600	II	$1\frac{1}{2}$-$2\frac{1}{2}$	3/4"	1"	$1\frac{1}{2}$"
	20-100	450-900	II	2-3	$1\frac{1}{2}$"	2"	3"
		901-1800	II	2-3	1"	$1\frac{1}{2}$"	2"
		1801-3600	II	$1\frac{1}{2}$-$2\frac{1}{2}$	1"	$1\frac{1}{2}$"	2"
	OVER 100	450-900	II	3-4	2"	3"	4"
		901-1800	II	2-3	$1\frac{1}{2}$"	2"	3"
		1801-3600	II	2-3	1"	$1\frac{1}{2}$"	2"
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 20	450-900	II	3-4	$1\frac{1}{2}$"	2"	3"
		901-1800	II	2-3	1"	$1\frac{1}{2}$"	2"
		1801-3600	II	2-3	3/4"	1"	$1\frac{1}{2}$"
	20-100	450-900	II	3-4	2"	3"	4"
		901-1800	II	2-3	$1\frac{1}{2}$"	2"	3"
		1801-3600	II	2-3	1"	$1\frac{1}{2}$"	2"
	OVER 100	450-900	II	3-4	3"	4"	5"
		901-1800	II	2-3	2"	3"	4"
		1801-3600	II	2-3	$1\frac{1}{2}$"	2"	3"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 11

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR
STEAM-TURBINE-DRIVEN ROTARY EQUIPMENT,
SUCH AS GEAR, GENERATOR OR GAS COMPRESSOR
USE TABLE 3 FOR RECIP. COMPR. DRIVEN BY STEAM TURBINE;
USE TABLE 5 FOR CENTR. COMPR. DRIVEN BY STEAM TURBINE)

<table>
<tr><th colspan="3">EQUIPMENT CONDITIONS</th><th colspan="5">MOUNTING RECOMMENDATIONS</th></tr>
<tr><th rowspan="2">EQUIPMENT LOCATION</th><th rowspan="2">POWER RANGE (HP)</th><th rowspan="2">SPEED RANGE (RPM)</th><th rowspan="2">COLUMN 1</th><th rowspan="2">COLUMN 2</th><th colspan="3">COLUMN 3</th></tr>
<tr><th>30'</th><th>40'</th><th>50'</th></tr>
<tr><td rowspan="3">ON GRADE SLAB</td><td>500-1500</td><td>OVER 3000</td><td>III</td><td></td><td colspan="3">1"</td></tr>
<tr><td>1501-5000</td><td>OVER 3000</td><td>III</td><td></td><td colspan="3">1½"</td></tr>
<tr><td>5001-15000</td><td>OVER 3000</td><td>III</td><td></td><td colspan="3">2"</td></tr>
<tr><td rowspan="3">ON UPPER FLOOR ABOVE NON-CRITICAL AREA</td><td>500-1500</td><td>OVER 3000</td><td>II</td><td>2-3</td><td>1"</td><td>1½"</td><td>2"</td></tr>
<tr><td>1501-5000</td><td>OVER 3000</td><td>II</td><td>2-3</td><td>1½"</td><td>2"</td><td>3"</td></tr>
<tr><td>5001-15000</td><td>OVER 3000</td><td>II</td><td>2-3</td><td>2"</td><td>3"</td><td>4"</td></tr>
<tr><td rowspan="3">ON UPPER FLOOR ABOVE CRITICAL AREA</td><td>500-1500</td><td>OVER 3000</td><td>II</td><td>3-5</td><td>1"</td><td>1½"</td><td>2"</td></tr>
<tr><td>1501-5000</td><td>OVER 3000</td><td>II</td><td>3-5</td><td>1½"</td><td>2"</td><td>3"</td></tr>
<tr><td>5001-15000</td><td>OVER 3000</td><td>II</td><td>3-5</td><td>2"</td><td>3"</td><td>4"</td></tr>
</table>

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 12

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR TRANSFORMERS

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (KVA)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	UNDER 10		IV		1/8"		
	10-100		IV		1/8"		
	OVER 100		IV		1/4"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 10		IV		1/8"	¼"	¼"
	10-100		III		¼"	½"	½"
	OVER 100		III		¼"	½"	1"
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 10		III		¼"	½"	3/4"
	10-100		III		½"	3/4"	1"
	OVER 100		III		½"	1"	1½"

COL 1: MOUNTING TYPE (SEE TEXT)
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF ISOLATORS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

TABLE 13

RECOMMENDED VIBRATION ISOLATION MOUNTING DETAILS FOR ONE- OR TWO-CYLINDER RECIPROCATING-TYPE AIR COMPRESSORS IN THE 10-100 HP SIZE RANGE

EQUIPMENT CONDITIONS			MOUNTING RECOMMENDATIONS				
EQUIPMENT LOCATION	POWER RANGE (HP)	SPEED RANGE (RPM)	COLUMN 1	COLUMN 2	COLUMN 3		
					30'	40'	50'
ON GRADE SLAB	UNDER 20	300-600	I	4-8	4"		
		601-1200	I	2-4	2"		
		1201-2400	I	1-2	1"		
	20-100	300-600	I	6-10	5"		
		601-1200	I	3-6	3"		
		1201-2400	I	2-3	1½"		
ON UPPER FLOOR ABOVE NON-CRITICAL AREA	UNDER 20	300-600	NOT RECOMMENDED				
		601-1200	I	3-6	4"	NO*	NO*
		1201-2400	I	2-3	2"	4"	NO*
	20-100	300-600	NOT RECOMMENDED				
		601-1200	NOT RECOMMENDED				
		1201-2400	I	3-6	3"	6"	NO*
ON UPPER FLOOR ABOVE CRITICAL AREA	UNDER 20	300-600	NOT RECOMMENDED				
		601-1200	NOT RECOMMENDED				
		1201-2400	I	3-6	4"	NO*	NO*
	20-100	300-2400	NOT RECOMMENDED				

COL 1: MOUNTING TYPE (SEE TEXT).
COL 2: MINIMUM RATIO OF WEIGHT OF INERTIA BLOCK TO TOTAL WEIGHT OF SUPPORTED LOAD
COL 3: MINIMUM STATIC DEFLECTION OF STABLE STEEL SPRINGS IN INCHES FOR INDICATED FLOOR SPAN IN FEET

* "NO" INDICATES "NOT RECOMMENDED" FOR THIS COMBINATION OF CONDITIONS

APPENDIX A

DATA FORMS

The procedures offered in this manual are summarized in a series of Data Forms, which, when filled in, provide simple and convenient forms for calculation and documentation of most of the acoustical aspects of the mechanical system design for airborne noise control. Blank copies of the Data Forms are given in this Appendix, and they can be reproduced and used for any particular analysis.

DATA FORM 1

ROOM CONSTANT OF SOURCE ROOM OR RECEIVER ROOM

ROOM NO. OR DESIGNATION ______________________________

1. AVERAGE ROOM DIMENSIONS (IN FT.)
   LENGTH __________ WIDTH ______________ HEIGHT ______________
2. VOLUME OF ROOM ________________ CU. FT.
3. TOTAL INTERIOR SURFACE AREA OF ROOM ____________________ SQ. FT.
4. AREA OF PLANNED ACOUSTIC TREATMENT* ____________________ SQ. FT.
5. PERCENT AREA COVERED BY ACOUSTIC TREATMENT _____________ %
   (100 x Item 4/Item 3)
6. "ROOM LABEL" FOR ITEM 5 FROM TABLE 1A, CHAPTER 4

   ____
7. FOR ITEMS 2 AND 6, ROOM CONSTANT FROM FIG. 2, CHAPTER 4
   R = _______________ SQ. FT. FOR 500 - 8000 Hz
8. CHECK ACOUSTIC ABSORPTION TREATMENT:

	☐ NONE OR NRC = 0.65 - 0.74	☐ NRC = 0.75 - 0.85
THEN, FOR 31 Hz	0.2 R = __________	0.2 R = __________
63 Hz	0.2 R = __________	0.3 R = __________
125 Hz	0.3 R = __________	0.5 R = __________
250 Hz	0.5 R = __________	0.8 R = __________

9. ROOM CONSTANT FOR ALL OCTAVE BANDS, IN SQ. FT.#
   (Repeat appropriate values from Items 7 and 8)

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

*Add 50% of floor area to Item 4 if floor is carpeted or has drapes or upholstered furniture. Treat this as NRC = 0.65 material.

#Add to all bands any area always open to the outside, i.e., having 100% absorption.

DATA FORM 2 SHEET 1

MECHANICAL EQUIPMENT ROOM SPL DUE TO EQUIPMENT

EQUIPMENT IDENTIFICATION

TYPE ________________________ HP ________ RPM ________

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. SPL OF EQUIPMENT AT 3-FT DISTANCE, FROM CHAPTER 3 TABLES OR OTHER SOURCE

2. SPECIAL ADJUSTMENTS, IF ANY, TO ITEM 1 DATA.

   EXPLAIN: ________________________

3. RESULTING SPL AFTER ADJUSTMENTS

4. DISTANCE FROM EQUIPMENT TO VARIOUS WALLS AND SURFACES OF INTEREST (ALL DISTANCES IN FT).

   NORTH WALL ________ SOUTH WALL ________ EAST WALL ________ WEST WALL ________

   CEILING ________ FLOOR* ________ (*Assume 3 ft unless different)

   SURFACE "A" (IDENTIFY): ________________ DISTANCE ________

   SURFACE "B" (IDENTIFY): ________________ DISTANCE ________

   SURFACE "C" (IDENTIFY): ________________ DISTANCE ________

   SURFACE "D" (IDENTIFY): ________________ DISTANCE ________

5. ROOM CONSTANT FOR THIS ROOM (FROM ITEM 9 OF DATA FORM 1)

MECHANICAL EQUIPMENT ROOM SPL DUE TO EQUIPMENT (CONT.)

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

6. SPL REDUCTION FOR VARIOUS DISTANCES AND ROOM CONSTANTS, FROM TABLE 2 OR FIGURE 1 OF CHAPTER 4 (FILL IN SPACES ONLY FOR SURFACES OF INTEREST)

NORTH									
SOUTH									
EAST									
WEST									
CEIL.									
FLOOR*									
"A"									
"B"									
"C"									
"D"									

*Floor value is "0" for all bands, if distance is 3 ft

7. SPL AT SURFACES OF INTEREST FOR THIS PIECE OF EQUIPMENT ONLY (ITEM 7 = ITEM 3 - ITEM 6)

NORTH									
SOUTH									
EAST									
WEST									
CEIL.									
FLOOR									
"A"									
"B"									
"C"									
"D"									

DATA FORM 3

SUMMATION OF SPLs DUE TO ALL EQUIPMENT
IN MECHANICAL EQUIPMENT ROOM (MER)

MER NO. OR DESIGNATION ______________________

CHECK WALL OR SURFACE INVOLVED IN THIS SUMMATION

NORTH WALL ☐ SOUTH WALL ☐ EAST WALL ☐ WEST WALL ☐ CEIL-ING ☐ FLOOR ☐

OR OTHER SURFACE DESIGNATION ______________________

IN NUMBERED SPACES BELOW, IDENTIFY EQUIPMENT WHOSE NOISE LEVELS CONTRIBUTE TO TOTAL SPL AT INDICATED WALL OR SURFACE. IN SPL SPACES, INSERT SPL VALUES AT THAT SURFACE DUE TO THAT EQUIPMENT, AS TAKEN FROM ITEM 7 OF DATA FORM 2.

	OCTAVE FREQUENCY BAND IN Hz								
	31	63	125	250	500	1000	2000	4000	8000
1.									
2.									
3.									
4.									
5.									

6. TOTAL SPL AT INDICATED WALL OR SURFACES DUE TO ABOVE EQUIPMENT, USING DECIBEL SUMMATION RULES OF TABLE 18, CHAP. 3. FOR FLOOR, USE HIGHEST READING IN EACH BAND FROM ITEMS 1-5.

DATA FORM 4

SOUND TRANSMISSION FROM MER TO ADJOINING ROOM
THROUGH COMMON WALL OR FLOOR-CEILING

SOUND TRANSMITTING ROOM ______________ SOUND RECEIVING ROOM ______________

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. AREA S_W OF COMMON TRANSMITTING WALL ☐ OR FLOOR ☐

   ______________ X ______________ = ______________ SQ. FT.

2. ROOM CONSTANT R_2 OF REC. RM; ITEM 9 OF DATA FORM 1

3. RATIO S_W/R_2 (ITEM 1 / ITEM 2)

4. WALL OR FLOOR CORRECTION TERM C FOR ITEM 3 RATIOS, FROM TABLE 14, CHAPTER 4

5. PROPOSED WALL OR FLOOR CONSTRUCTION ______________

6. "TL" OF PROPOSED WALL OR FLOOR. SEE CHAPTER 4 TABLES FOR WALL TLs AND FLOOR-CEILING TLs.

7. "NR" OF PROPOSED WALL OR FLOOR. NR = TL + C
   ITEM 7 = ITEM 6 + ITEM 4

8. SPL ON MER SIDE OF WALL OR FLOOR, FROM ITEM 6 OF DATA FORM 3 FOR ALL EQUIPMENT CONTRIBUTIONS OR APPROPRIATE LINE OF ITEM 7 OF DATA FORM 2 IF FOR ONLY ONE PIECE OF EQUIPMENT

9. SPL IN RECEIVING ROOM DUE TO MER NOISE TRANSMITTED THROUGH WALL OR FLOOR OF ITEM 5 ABOVE.
   $SPL_{REC.\ RM.} = SPL_{MER} - NR.$ ITEM 9 = ITEM 8 - ITEM 7

DATA FORM 5

COMPARISON OF ROOM SPL WITH NOISE CRITERION

SOUND RECEIVING ROOM ________________________________

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. APPLICABLE ROOM CATEGORY NO. ______ FROM TABLE 2 OF CHAP. 2

2. SUGGESTED NOISE CRITERION FOR ROOM: NC-__________

3. SPL VALUES CORRESPONDING TO NC VALUE OF ITEM 2; FROM TABLE 1 OF CHAP. 2.

4. PROPOSED WALL OR FLOOR CONSTRUCTION BETWEEN MER AND REC. RM; FROM ITEM 5 OF DATA FORM 4 ________________________________

5. SPL IN RECEIVING ROOM FOR ITEM 4 WALL; FROM ITEM 9 OF DATA FORM 4

6. COMPARISON OF ITEM 5 WITH ITEM 3 ABOVE. IF ITEM 5 SPL EXCEEDS ITEM 3 SPL IN ANY FREQUENCY BAND, INSERT THE AMOUNT OF THAT EXCESS IN THE APPROPRIATE SPACE BELOW

7. IF THERE IS NO NOISE EXCESS IN ANY BAND, WALL OR FLOOR DESIGN IS PREFERRED. CHECK HERE ☐

8. IF NOISE EXCESS IS NOT GREATER THAN THE FOLLOWING VALUES IN ANY BAND, WALL OR FLOOR IS ACCEPTABLE. CHECK HERE ☐

4	4	4	3	2	2	2	2	2

9. IF NOISE EXCESS IS WITHIN FOLLOWING VALUES IN ANY BAND, WALL OR FLOOR IS MARGINAL. CHECK HERE ☐

5-7	5-7	5-7	4-6	3-5	3-5	3-5	3-5	3-5

10. IF NOISE EXCESS IS GREATER THAN ITEM 9 VALUES IN ANY BAND, WALL OR FLOOR IS UNACCEPTABLE. CHECK HERE ☐

DATA FORM 6

ESTIMATED SPL AT 3 FT DISTANCE INDOORS WHEN
SOURCE PWL IS KNOWN

EQUIPMENT IDENTIFICATION

TYPE ______________________________ HP ______ RPM ______

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. PWL FOR EQUIPMENT IN DB RE 10^{-12} WATT. IF PWL IS REFERRED TO 10^{-13} WATT, SUBTRACT 10 DB FROM ALL VALUES.

2. APPROXIMATE OVERALL DIMENSIONS OF EQUIPMENT, IN FEET. IGNORE ANY PORTION OBVIOUSLY NOT PRODUCING OR RADIATING NOISE.

LENGTH__________ WIDTH__________ HEIGHT__________

3. ADD THE THREE DIMENSIONS OF ITEM 2; DIVIDE THE SUM BY 6. THE RESULT IS APPROXIMATELY THE "RADIUS" OF THE EQUIPMENT.

SUM __________ DIVIDED BY 6 = __________ FT.

4. ADD 3 FT TO THE ITEM 3 VALUE. THIS GIVES APPROXIMATELY A DISTANCE OF 3 FT FROM THE ASSUMED "ACOUSTIC CENTER" OF THE UNIT.

ITEM 3 __________ + 3 = __________ FT.

5. ROOM CONSTANT OF MECHANICAL EQUIPMENT ROOM, FROM ITEM 9 OF DATA FORM 1

6. DETERMINE "RELATIVE SPL" FROM FIG. 1 OF CHAP. 4 FOR THE DISTANCE DIMENSIONS OF ITEM 4 ABOVE, FOR EACH OF THE ROOM CONSTANT VALUES OF ITEM 5 ABOVE. INSERT "REL SPL" AND PROPER SIGN (+ OR -) IN SPACES BELOW.

7. APPROXIMATE SPL AT 3-FT DISTANCE. SPL = PWL + REL SPL. ITEM 7 = ITEM 1 + ITEM 6 (CAUTION: KEEP CORRECT SIGNS!) THIS SPL AT THE NORMALIZED 3-FT DISTANCE MAY BE INSERTED INTO ITEM 1 OF DATA FORM 2 AND THIS EQUIPMENT GIVEN THE SAME TREATMENT AS ALL OTHER INDOOR EQUIPMENT IN THE ANALYSIS PROCEDURE.

DATA FORM 7

NOISE ESCAPE THROUGH OPENINGS IN SOURCE ROOM

ROOM IDENTIFICATION ______________________________

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. SPL AT OPENING

2. AREA OF NOISE ESCAPE OPENING: ______________________________ SQ. FT

3. "AREA FACTOR" FROM TABLE 19 OF CHAP. 3 FOR ITEM 2 AREA; SAME VALUE FOR ALL FREQUENCY BANDS.

4. PWL OF NOISE AT OPENING (IN DB RE 10^{-12} WATT). ITEM 4 = ITEM 1 + ITEM 3

5. ATTENUATION INSERTED IN PATH TO OUTSIDE (MUFFLER, LINED DUCT, LINED BEND,ETC.---FROM MUFFLER MFGR, ASHRAE GUIDE, ETC.)

6. PWL OF ESCAPING NOISE (IN DB RE 10^{-12} WATT). ITEM 6 = ITEM 4 - ITEM 5

DATA FORM 8

ESTIMATED OUTDOOR SOUND PRESSURE LEVEL (SPL)
DUE TO AN OUTDOOR SOUND SOURCE PWL

1. DISTANCE FROM NOISE SOURCE TO CRITICAL NEIGHBOR:______________FT.

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

2. TOTAL PWL OF ALL OUTDOOR NOISE SOURCES AT SOURCE POSITION

3. OUTDOOR DISTANCE TERM FROM TABLE 21 OR 22 OF CHAPTER 4 FOR ITEM 1 DISTANCE

4. TENTATIVE OUTDOOR SPL AT DISTANCE OF ITEM 1:
(ITEM 4 = ITEM 2 - ITEM 3)

5. ATTENUATION OF BARRIER, IF ANY (TABLE 23, CHAPTER 4)

6. ATTENUATION OF WOODS, IF ANY (TABLE 24, CHAPTER 4)

7. OTHER ATTENUATION, IF ANY

8. ESTIMATED OUTDOOR SPL AT NEIGHBOR POSITION (ITEM 8 = ITEM 4 - ITEM 5 - ITEM 6 - ITEM 7)

DATA FORM 9

CRITERION SPL FOR CRITICAL NEIGHBOR

CRITICAL NEIGHBOR ______________________________ CRITICAL TIME: DAY NIGHT

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. OUTDOOR BACKGROUND SPL AT NEIGHBOR (FROM BACKGROUND MEASUREMENTS OR FROM TABLES 4 AND 5, CHAPTER 2)

2. LET EQUIPMENT NOISE EXCEED BACKGROUND BY ________ dB (SEE PARAGRAPH 5 OF CHAPTER 2 FOR DISCUSSION)

3. TENTATIVE OUTDOOR SPL CRITERION (ITEM 1 + ITEM 2)

4. RECOMMENDED INDOOR SPL CRITERION FOR NEIGHBOR FROM TABLE 2 OF CHAPTER 2: "NC-_______"

5. OCTAVE BAND SPL FOR ITEM 4 "NC" CURVE FROM TABLE 1, CHAP. 2.

6. APPROXIMATE NOISE REDUCTION PROVIDED BY NEIGHBOR'S BUILDING, FROM TABLE 6 OF CHAPTER 2.

7. TENTATIVE OUTDOOR SPL CRITERION (ITEM 5 + ITEM 6)

8. FINAL OUTDOOR SPL CRITERION. LOWER SPL IN EACH OCTAVE BAND FROM ITEMS 3 AND 7 IS CONSERVATIVE CRITERION.

DATA FORM 10

COMPARISON OF OUTDOOR SPL WITH NEIGHBOR CRITERION SPL

CRITICAL NEIGHBOR ______________________________

OCTAVE FREQUENCY BAND IN Hz								
31	63	125	250	500	1000	2000	4000	8000

1. OUTDOOR CRITERION SPL FOR CRITICAL NEIGHBOR (FROM ITEM 8 OF DATA FORM 9)

2. ESTIMATED OUTDOOR SPL AT NEIGHBOR POSITION (FROM ITEM 8 OF DATA FORM 8)

3. IF ITEM 2 SPL EXCEEDS ITEM 1 SPL IN ANY BAND, INSERT THE AMOUNT OF THAT <u>EXCESS</u> IN THE APPROPRIATE SPACE BELOW

4. IF THERE IS NO NOISE EXCESS IN ANY BAND, NOISE CONDITION IS <u>PREFERRED</u>. CHECK HERE ☐

5. IF NOISE EXCESS IS NOT GREATER THAN THE FOLLOWING VALUES IN ANY BAND, CONDITION IS <u>ACCEPTABLE</u>. CHECK HERE ☐

4	4	4	3	2	2	2	2	2

6. IF NOISE EXCESS IS WITHIN FOLLOWING VALUES IN ANY BAND, CONDITION IS <u>MARGINAL</u>. CHECK HERE ☐

5-7	5-7	5-7	4-6	3-5	3-5	3-5	3-5	3-5

7. IF NOISE EXCESS IS GREATER THAN ITEM 6 VALUES IN ANY BAND, CONDITION IS <u>UNACCEPTABLE</u>. CHECK HERE ☐
CONSIDER ALTERNATE APPROACHES OR POSSIBLE NOISE REDUCTION MEASURES.

DATA FORM 11

ESTIMATED SOUND POWER LEVEL (PWL) OF
RECIPROCATING ENGINE CASING NOISE

1. CONTINUOUS RATING OF ENGINE: ___________ HP OR _____________ KW

2. ENGINE SPEED: ______________ RPM

3. FUEL: GAS ONLY /__/ LIQUID ONLY /__/ GAS AND LIQUID /__/

FREQUENCY BAND IN Hz							
63	125	250	500	1000	2000	4000	8000

4. BASE PWL FROM TABLE 8, CHAPTER 3, FOR ITEM 1 RATING:

5. SPEED CORRECTION FROM TABLE 8, CHAPTER 3, FOR ITEM 2 SPEED:

6. FUEL CORRECTION FROM TABLE 8, CHAPTER 3, FOR ITEM 3 FUEL:

7. ESTIMATED PWL (in dB re 10^{-12} watt) OF CASING NOISE:
(ITEM 7 = ITEM 4 + ITEM 5 + ITEM 6)
CAUTION: KEEP SIGNS CORRECT!

DATA FORM 12

ESTIMATED SOUND POWER LEVEL (PWL) OF
UNMUFFLED RECIPROCATING ENGINE EXHAUST NOISE

1. CONTINUOUS RATING OF ENGINE: ____________HP, OR ____________KW

2. AIR INTAKE: TURBOCHARGER /__/, NO TURBOCHARGER /__/

3. EXHAUST PIPE LENGTH: ____________ FT

FREQUENCY BAND IN Hz							
63	125	250	500	1000	2000	4000	8000

4. BASE PWL FROM TABLE 9, CHAPTER 3 FOR ITEM 1 RATING:

5. TURBOCHARGER CORRECTION FROM TABLE 9 FOR ITEM 2 AIR INTAKE:

6. EXHAUST PIPE LENGTH CORRECTION FROM TABLE 9 FOR ITEM 3 LENGTH:

7. ESTIMATED PWL (in dB re 10^{-12} watt) OF UNMUFFLED EXHAUST NOISE
(ITEM 7 = ITEM 4 + ITEM 5 + ITEM 6)
CAUTION: KEEP SIGNS CORRECT!

DATA FORM 13

ESTIMATED SOUND POWER LEVEL (PWL)
OF UNTREATED TURBOCHARGER NOISE AT
AIR INLET OPENING OF RECIPROCATING ENGINE

1. CONTINUOUS RATING OF ENGINE: ____________HP OR ____________KW

2. INLET AIR DUCT LENGTH: ____________ FT

FREQUENCY BAND IN Hz							
63	125	250	500	1000	2000	4000	8000

3. BASE PWL FROM TABLE 10, CHAPTER 3 FOR ITEM 1 RATING:

4. INLET AIR DUCT LENGTH CORRECTION FROM TABLE 10, CHAPTER 3 FOR ITEM 2 LENGTH:

5. ESTIMATED PWL (in dB re 10^{-12} watt) OF UNTREATED TURBOCHARGER NOISE
(ITEM 5 = ITEM 3 + ITEM 4)
CAUTION: KEEP SIGNS CORRECT!

DATA FORM 14 SHEET 1

ESTIMATED SOUND POWER LEVEL (PWL) OF CASING,
EXHAUST AND INTAKE NOISE OF GAS TURBINE ENGINE

1. CONTINUOUS RATING OF ENGINE: ____________HP OR ______________KW

2. ENGINE CASING COVER: NONE /_/ OR TYPE _____ FROM TABLE 11, CHAP. 3

3. EXHAUST DUCT: ROUND /_/, RECTANGULAR /_/ , LENGTH __________ FT.
   DUCT LINING: TYPE ________ FROM TABLE 29, CHAP. 4
   DUCT TURNS: NO.__________ TYPE_________ FROM TABLE 30, CHAP. 4

4. INTAKE DUCT: ROUND /_/, RECTANGULAR /_/ , LENGTH __________ FT.
   DUCT LINING: TYPE _________ FROM TABLE 29, CHAP. 4
   DUCT TURNS: NO._________ TYPE_________ FROM TABLE 30, CHAP. 4

5. EXHAUST MUFFLER SUPPLIED: YES /_/ NO /_/

6. INTAKE MUFFLER SUPPLIED: YES /_/ NO /_/

FREQUENCY BAND IN Hz							
63	125	250	500	1000	2000	4000	8000

7. PWL (in dB re 10^{-12} watt) OF CASING NOISE FROM TABLE 11 OF CHAPTER 3 FOR ITEM 1 RATING:

8. NOISE REDUCTION PROVIDED BY CASING COVER (IF ANY) OF ITEM 2 AS ESTIMATED IN TABLE 11, CHAP. 3 FOOTNOTES:

-	-	-	-	-	-	-	-

9. PWL OF CASING NOISE WITH COVER (IF ANY)
   ITEM 9 = ITEM 7 + ITEM 8
   CAUTION: KEEP SIGNS CORRECT!

10. PWL (in dB re 10^{-12} watt) OF EXHAUST NOISE FROM TABLE 12A, OF CHAPTER 3 FOR ITEM 1 RATING:

(continued on Sheet 2)

FREQUENCY BAND IN Hz							
63	125	250	500	1000	2000	4000	8000

11. ATTENUATION PROVIDED BY EXHAUST DUCT AND TURNS OF ITEM 3, FROM TABLES 29 AND 30 OF CHAPTER 4 OR FROM ASHRAE GUIDE (Include hot temperature correction):

-	-	-	-	-	-	-	-

12. INSERTION LOSS OF EXHAUST MUFFLER OF ITEM 5, IF PROVIDED (use muffler manufacturer's data for appropriate bands, corrected for exhaust temperature):

-	-	-	-	-	-	-	-

13. PWL OF EXHAUST NOISE OUT OF DUCT AND MUFFLER (AS APPLICABLE)
ITEM 13 = ITEM 10 + ITEM 11 + ITEM 12
CAUTION: KEEP SIGNS CORRECT!

14. PWL (in dB re 10^{-12} watt) OF AIR INTAKE NOISE FROM TABLE 12B OF CHAPTER 3 FOR ITEM 1 RATING

15. ATTENUATION PROVIDED BY INTAKE DUCT AND TURNS OF ITEM 4, FROM TABLES 29 AND 30, CHAP. 4, OR FROM ASHRAE GUIDE:

-	-	-	-	-	-	-	-

16. INSERTION LOSS OF INTAKE MUFFLER OF ITEM 6, IF PROVIDED (use muffler manufacturer's data for appropriate bands):

-	-	-	-	-	-	-	-

17. PWL OF INTAKE NOISE OUT OF DUCT AND MUFFLER (AS APPLICABLE)
ITEM 17 + ITEM 14 + ITEM 15 + ITEM 16
CAUTION: KEEP SIGNS CORRECT!

APPENDIX B

EXCERPTS FROM CHAPTER 31 SOUND AND VIBRATION CONTROL

ASHRAE GUIDE AND DATA BOOK: SYSTEMS AND EQUIPMENT 1967

Sound Generated by Fans

Selection of as quiet a fan as practicable will often reduce the required duct attenuation. For centrifugal fans in central systems, the following items should be considered:

1. Sound level, as well as the usual selection factors should be considered in evaluating various fan types.
2. The fan should be selected to operate near its efficiency peak when handling the required air quantity and static pressure.
3. The system should be designed for smooth air flow, and resistance should be as low as economically feasible.[41]
4. Duct connections to the fan inlet should be designed to provide a smooth approach of the air.[42]

To evaluate the effect of fan design, it is necessary to consider the entire sound spectrum.[43,44] The sound power spectra[45] shown in Fig. 10 illustrate the following facts about ventilating fans of the centrifugal type:

1. When fans of *different* design are considered for a given application, neither the rpm nor the tip speed indicates which fan will be quieter. Backward-curved fans, if properly selected, are usually somewhat more efficient and a little quieter than forward-curved fans, although their tip speed is higher. On the other hand, forward-curved fans are more compact and lighter. Their lower speed may have other advantages, such as less bearing noise and more tolerance in regard to wheel unbalance.
2. The sound power spectra of centrifugal ventilating fans fall off from low to high frequency at an average rate of 4 to 6 dB per octave. The overall power level, being a *logarithmic* sum, is primarily a measure of the noise at frequencies below 180 cps. Experience points to the 250 cps band as being the most critical one in silencing centrifugal central station fans, with the 125 and 500 cps bands being next. Noise comparisons of different fan types can therefore be based on the sound power level in these bands, but not on overall levels.
3. The sound spectra of centrifugal fans for air-conditioning systems should be relatively smooth, without prominent pure tone components. The spectrum for the backward-curved fan in Fig. 10 shows a peak at the blade passage frequency (number of blades times revolutions per second). This is typical for fans having a relatively small number of blades because of the distinct pressure field around each blade. The peak is more pronounced for high flow rates at relatively low static; it may be aggravated by a cut-off located too close to the wheel periphery. Because of the added annoyance of pure tones, the sound output at blade frequency should be closely examined when selecting fans for critical applications.

The sound power spectrum of a vaneaxial fan,[44] as illustrated in Fig. 11, is quite different from that of a centrifugal fan. Relatively little sound is generated at low frequencies. The spectrum usually has a very strong peak at the blade passage frequency and often has secondary peaks at multiples of this frequency.

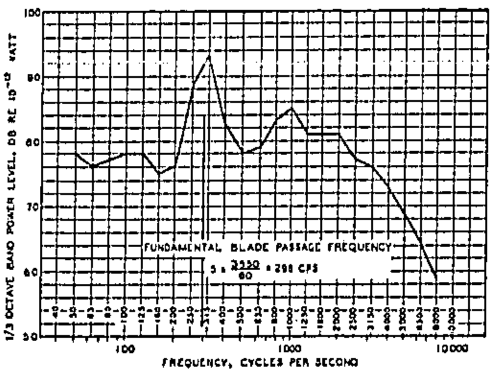


Fig. 11 .... Sound Power Spectrum of a Vaneaxial Fan with 5 blades at 3550 rpm

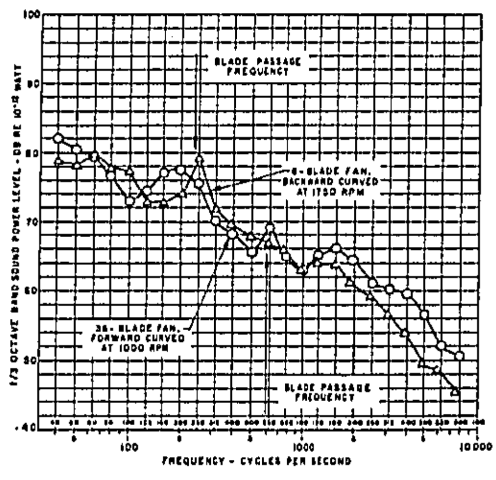


Fig. 10 .... Sound Power Spectra of Two Types of 12½ in. Diameter Centrifugal Ventilating Fans Delivering 1120 cfm Against 1.4 in. Static

Proper sizing of the fan is the most important factor in assuring a minimum of fan noise for any given type of fan. The noise will be lowest if the fan is sized to operate near the peak of its efficiency curve when delivering the required air volume (cfm) against a required static pressure.[47] Noise and efficiency of any fan are not related on an energy basis because of the exceedingly small amount of energy required to produce noise. However, the same factors which produce noise in a fan also tend to reduce the fan efficiency. As Fig. 12 shows, the noise energy may easily be doubled (increased by 3 dB) for a drop in efficiency of only a few percent.

An undersized fan will operate under the right-hand part of the curve. Its efficiency will be lower and the noise higher than that of the optimum sized fan. The primary cause of both lower efficiency and higher noise appears to be the higher air flow velocities through the fan.[48] To avoid excessive noise, the fan should be large enough so that the velocity pressure corresponding to the fan outlet velocity is between 15 and 25 percent of the static pressure for forward-curved fans, and between 7 and 13 percent for backward-curved fans.

Oversized fans are not only uneconomical, but are also noisier than optimum sized fans. This is due to separation of air flow over the fan blades, which occurs when a fan operates to the left of the peak of the static efficiency curve shown in Fig. 12. Flow separation has less effect on efficiency than it has on noise. The efficiency of an oversized fan is, therefore, not a good measure of its quietness.

Since fan design and application details have such an important effect, fan sound ratings from the manufacturer should be consulted. It must be borne in mind, however, that the environment and technique used to obtain these ratings are of the utmost importance. These should always be clearly specified, and their significance in any selection or comparison clearly understood.[1,49]

If actual test data on the power levels of the fan are not available, the approximate sound power level can be estimated using Table 10 and Fig. 13.[50] Table 10 shows the base sound power levels for nine types of fan wheels with a diameter of 36 in., operating at a speed of 1000 rpm. This table cannot be used to compare the sound power output of the various types of fans since, for the base conditions given, each fan will be producing a different volume of air and pressure. The comparisons between the fans can only be made on the basis of equivalent capacities and pressures.

Fig. 13, which was developed using the fan laws relating to noise levels, provides the corrections to the base sound power levels for the specific fan size and speed of interest. In using this method of estimating sound power level of a fan, it is essential that the fan be selected at or very near its peak efficiency. If the fan is not operating at peak efficiency, the sound power levels can increase as much as 15 decibels if the static efficiency is about 30 percent.

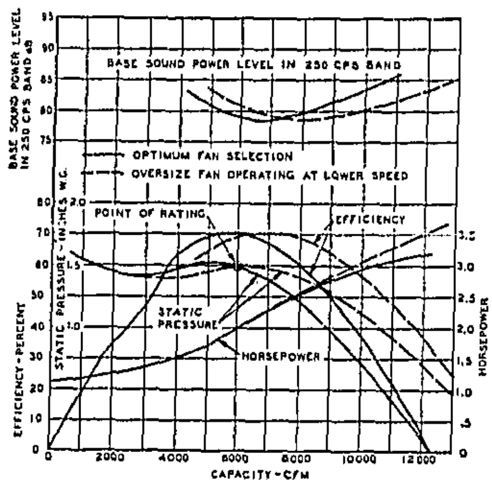


Fig. 12 .... Comparison of Two Similar Centrifugal Fans of Different Size and Speed, Selected to Deliver 6000 cfm Against 1.5 in. Static Pressure

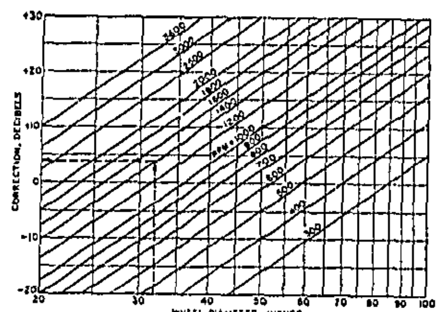
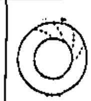
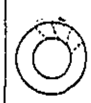
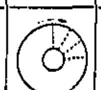
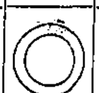
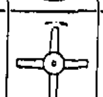
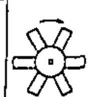
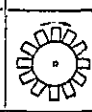


Fig. 13 .... Correction in Decibels to be Applied to Base Sound Power Level Values from Table 10 to Compensate for Actual Wheel Diameter and Actual RPM

Table 10 . . . . Base Sound Power Levels for Various Types of Fans in Decibels re 10^{-13} Watt

(Special Note: This table cannot be used for direct comparison of the noise generated by any one fan with the noise generated by any other fan in this list. The table has been prepared to serve a completely different purpose. Please refer to accompanying text for further explanation.)

Type	Diagram	Description	Applications	Base Sound Power Levels, dB re 10^{-13} Watt — Octave Band Center Frequencies—cps							
				63	125	250	500	1000	2000	4000	8000
C-1		Centrifugal fan with backwardly curved airfoil blades.	1. General ventilation and air conditioning. 2. Industrial applications where corrosion, erosion, or dirt is not a major problem.	97	94	92	90	88	85	82	80
C-2		Centrifugal fan with backwardly curved or sloped, single thickness blades.	1. General ventilation and air conditioning. 2. Industrial applications where corrosion, erosion, or dirt is not a major problem.	97	95	93	94	93	88	81	79
C-3		Centrifugal fan with single thickness blades with forward curved heel and radial or nearly radial tip.	1. Used principally for industrial applications where medium to high pressure requirements must be met. May be used in moderately dirty applications.	108	104	94	92	90	88	89	87
C-4		Centrifugal fan with single thickness radial blades. Blades are relatively short in direction of air flow.	1. Industrial applications where corrosion or erosion is a problem, or dust loading is very heavy. Also used in conveying systems where material passes through the fan wheel.	103	103	96	96	93	88	85	84
C-5		Centrifugal fan with single thickness radial blades. Blades are relatively long in direction of air flow.	1. Industrial applications where relatively small volumes at high pressure are required.	114	111	104	104	100	97	94	91
C-6		Centrifugal fan with single thickness blades curved forward at both heel and tip.	1. General ventilation and air conditioning for low pressure, high capacity requirements.	116	112	106	101	100	98	93	88
A-1		Axial fan with relatively long blades and small hub.	1. Designed to meet requirements of high capacity at very low pressures.	92	93	92	91	91	88	84	80
A-2		Axial fan where hub is about 50 percent of fan tip diameter.	1. General ventilation or air conditioning. 2. Industrial applications where corrosion, erosion, or dirt is not a major problem.	96	93	97	96	94	90	86	85
A-3		Axial flow fan with relatively short blades and large hub.	1. Industrial applications where requirement is for high pressure at medium capacity.	90	86	90	93	93	89	83	81

To estimate the sound power levels of a fan:

1. Select the proper fan using standard fan selection procedures. Note the wheel diameter and the rpm.
2. Identify the type of fan in Table 10 and record the base sound power level in each octave band.
3. Enter Fig. 13 with the wheel diameter and rpm to obtain the correction factor.
4. Add the correction factor algebraically to each of the octave band base sound power levels.
5. For critical applications add five decibels to the sound power level in the octave band where the blade passage frequency occurs when calculated using Equation 14.

$$f_B = \frac{N \times \text{rpm}}{60} \qquad (14)$$

where

f_B = blade passage frequency, cycles per second.
N = number of fan blades.

6. The result is the estimated sound power level of the selected fan in each of the octave bands.

Example The optimum selection of a pressure blower for some specified duty has been found to be a fan with a 32 in. dia wheel operating at 1400 rpm. Estimate the resulting sound power level.

Solution: According to Table 10, a pressure blower falls into the classification of fan type No. C-5 and the base sound power level values for this type of fan are listed in the right hand part of Table 10 (see line 1).

By referring to Fig. 13 and using the parameters of 32 in. dia and 1400 rpm, a correction factor of + 4 dB is obtained, as shown by the dashed line (see line 2).

According to the catalog data, this specific fan is made with 10 blades. Therefore the blade frequency is:

$$f_B = \frac{10 \times 1400}{60} = 233 \text{ cps}$$

which falls in the 250 cps octave band (see line 3). The sum of lines 1, 2 and 3 is the sound power level of the fan shown in line 4.

Item	Octave Band Center Frequencies—cps							
	63	125	250	500	1000	2000	4000	8000
1. Base sound power level, dB (Table 10)	114	111	104	104	100	97	94	91
2. Correction, dB (Fig. 13)	+4	+4	+4	+4	+4	+4	+4	+4
3. Blade frequency correction, dB			+5					
4. Estimated power level at operating conditions, dB	118	115	113	108	104	101	98	95

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CHAPTER 3

NOISE LEVEL DATA

Noise levels measured at several plants or equipment installations have been collected and are summarized in the enclosed Tables 1-3. These are divided roughly into various types of industries, and the levels given represent the approximate upper and lower limits found at various operator positions. This does not represent an exhaustive survey of plants or plant noise; the data merely indicate that hearing damage noise levels exist in many plant areas.

TABLE 1

SOME REPRESENTATIVE NOISE LEVEL RANGES
AT VARIOUS OPERATOR POSITIONS IN
VARIOUS INDUSTRIES

(MANY ACTUAL SITUATIONS INCLUDED,
BUT BY NO MEANS A COMPLETE LISTING)

	OCTAVE FREQUENCY BAND IN Hz								
	31	63	125	250	500	1000	2000	4000	8000
WOOD AND PULP PROCESSING									
	88	102	108	114	114	112	111	106	97
	72	79	81	90	91	86	81	76	67
POWER SAWS, MOLDERS, PLANERS									
	89	95	101	106	109	109	106	102	101
	60	65	69	71	73	74	73	72	70
PRINTING (INCL. NEWSPAPERS), BOOKBINDING									
	85	95	102	98	96	92	89	88	90
	68	73	73	72	73	73	70	68	64
ROCK CRUSHING AND GRINDING									
	92	97	96	98	100	96	96	94	90
	80	88	86	85	84	82	80	74	70
ROCK DRILLS AND AIR COMPRESSORS									
	80	88	98	102	103	98	95	90	88
	70	80	88	88	84	85	80	80	75
COAL CAR SHAKE-OUT									
	100	119	115	111	108	105	104	103	98
	90	111	105	101	100	95	94	92	82

TABLE 2

SOME REPRESENTATIVE NOISE LEVEL RANGES
AT VARIOUS OPERATOR POSITIONS IN
VARIOUS INDUSTRIES

(MANY ACTUAL SITUATIONS INCLUDED,
BUT BY NO MEANS A COMPLETE LISTING)

	OCTAVE FREQUENCY BAND IN Hz								
	31	63	125	250	500	1000	2000	4000	8000
PETROLEUM PLANT									
	95	102	107	111	105	98	91	90	85
	75	80	78	75	73	70	66	61	54
PLASTICS PROCESSING									
	90	94	103	105	108	103	102	99	97
	72	77	77	84	82	81	80	74	64
TEXTILES									
	83	88	90	94	97	99	100	97	100
	58	60	62	67	66	71	71	65	56
LEATHER PROCESSING, SHOE MANUFACTURING									
	80	87	88	91	93	95	96	95	94
	70	75	75	72	76	78	75	74	72
MACHINE SHOPS (GRINDING, PUNCHING, RIVETING)									
	88	98	104	108	102	106	108	110	109
	70	76	74	78	78	74	70	71	66
BOTTLING AND CANNING PLANTS									
	88	95	101	102	98	95	91	90	92
	65	72	75	70	68	65	63	60	57

TABLE 3

SOME REPRESENTATIVE NOISE LEVEL RANGES
AT VARIOUS OPERATOR POSITIONS IN
VARIOUS INDUSTRIES

(MANY ACTUAL SITUATIONS INCLUDED,
BUT BY NO MEANS A COMPLETE LISTING)

	OCTAVE FREQUENCY BAND IN Hz								
	31	63	125	250	500	1000	2000	4000	8000
ELECTRIC GENERATING STATIONS									
	106	104	108	107	105	103	100	94	84
	82	86	89	82	81	80	84	72	62
GAS COMPRESSOR STATIONS									
	126	109	103	99	96	96	95	99	108
	85	83	85	90	84	76	76	77	73
MECHANICAL EQUIPMENT ROOMS									
	90	94	93	90	88	89	89	86	80
	70	72	75	76	73	68	65	62	53
ROAD MACHINERY, FARM TRACTORS									
	85	95	106	104	102	102	98	95	92
	68	72	78	79	75	72	70	63	58

CHAPTER 5

PRINCIPLES, METHODS AND EXAMPLES OF NOISE CONTROL IN MACHINE DESIGN

Although there still exist many questions on the psychological and physiological effects of noise on people, there is no question that too many people are currently exposed to too much noise. With this premise as an accepted fact, we wish to consider here briefly some of the basic methods of noise control that are available and that are in practical use in many places where people have agreed that some noise must be stopped.

Of course, it is highly desirable at the time of the original design to reduce the noise generated and radiated by a machine. Usually, however, a complex machine represents an evolutionary growth of one or more simpler machines, and as the size, speed, complexity and performance increase, concern for noise is lost along the way, if indeed there was ever any such concern in the first place. As a result, the completed machine may be noisy and it is probably so uniquely put together that it is virtually impossible to go back into the machine and simply insert a few noise-reduction treatments. As a result we rather seldom have the opportunity to change the "internal workings" of a complex machine; instead, we are usually restricted to working around the perimeter of the problem. This imposes rather serious limitations on the noise control that can be achieved.

Nevertheless, whether we can work on the inside or the outside of the machine, there are certain basic approaches to noise control. First, actual noise level goals or criteria are established for the work space in question. In most factory spaces, the goal is to achieve "safe" noise levels for the protection of hearing or to achieve low enough noise levels to carry on some degree of reliable speech communication. Next, we almost always include measurements of the noise and vibration of the machine that is to be quieted, in order to determine and to quantify the principal components and paths of noise. Then, we are in a position to design noise control treatments for the machine.

1. NOISE PRODUCING MECHANISMS

Let us first look briefly at a few of the typical mechanisms that produce noise. This is not a complete list; but it perhaps will begin to remind one of the basic noise sources of various types of machines.

Figure 1 illustrates some of the basic movements in machines that can give rise to noise or vibration. Incidentally, we can treat vibration almost synonymously with noise, because usually a vibration source either produces noise itself or causes something else to which it is attached to produce or radiate noise. Hence, the term "structure-borne noise" frequently describes this mixture of noise and vibration.

Figure 2 illustrates the mechanisms whereby high speed air movement can generate turbulence; and turbulence is almost synonymous with noise. Remember that sound is caused by the vibration of air particles, and turbulent air flow produces vibration of the air particles in the airflow. The noise radiated from the rear of a jet engine is a dynamic example of how turbulence produces noise.

Figure 3 illustrates one of the possible but usually less serious producers of noise. Motors and transformers are relatively simple examples of noise caused by electro-magnetic induction, but there are some industrial applications that involve tremendous amounts of noise and vibration.

Figure 4 may suggest "musical acoustics" but it is intended to highlight two mechanisms whereby a small amount of energy may produce an exaggerated amount of sound. A small amount of energy at the resonant frequency of a particular structure can produce large amounts of sound; the structure may be a gear, a subway wheel, a steel linkage in a machine, a panel of a cabinet enclosing a machine or even a special size and shape of an air space. The "sounding board" represents almost any structure to which a vibrating device is rigidly attached. The floor is a sounding board for a motor and pump, if you live on the floor under that motor and pump and if they are not properly vibration-isolated. The steel framing of a large machine may be the "sounding board" for a relatively small vibrator inside the machine.

The sources and paths of sound shown by Figures 1-4 are only fragmentary but they suggest the noise complexity of a machine that may be made up of many of these mechanisms simultaneously in operation, each performing its small but necessary function. A more complete, but still brief, discussion of noise sources and general approaches to noise reduction is given in the paper reproduced at the end of these notes: "Guidelines for Designing Quieter Equipment" by Clayton H. Allen. A reprint is also included that gives some general information on several aspects of the noise problem: "The Anatomy of Noise" by Leo L. Beranek and Laymon N. Miller (from Machine Design, September 14, 1967).

2. NOISE CONTROL APPROACHES

Some of the most vital basic steps to noise control are included in the following list. These steps must be taken, where applicable, if any noise source is to be quieted.

a. reduction of certain impact or acceleration effects,

b. reduction of unbalanced forces,

c. reduction of large radiating areas,

d. elimination of noise leakage paths,

e. use of acoustic enclosures to contain the noise or acoustic barriers to shield or deflect the noise,

f. use of acoustic absorption material to absorb sound energy inside confined spaces and in sound-control passageways,

g. use of mufflers or attenuators to reduce noise in gas flow paths,

h. use of vibration isolation mounts to isolate a vibration source from a noise radiator,

i. use of flexible connections between the isolated source and its base structure

j. use of vibration damping materials to reduce noise radiation from thin surfaces, and

k. use of alternate less-noisy methods for performing the same function.

3. EXAMPLES OF NOISE CONTROL

We can demonstrate the use of some of these noise control methods with actual examples from industry.

a. Quieted Stock Tubes for Automatic Screw Machines. One of the well-publicized noise control treatments of a few years ago was a quieted stock tube for automatic screw machines*. A layer of fabric webbing placed between the outer solid-wall tubing and the inner helically wound steel liner serves partially as vibration isolation and partially as vibration damping. Figure 5 gives measured noise levels in an aisle position about 5 ft from a six-spindle stock tube array for four different combinations of stock and stock tubes. The two lowest curves represent the noise levels for an operation involving round stock. The upper curve of this pair (shown by the letter "C" inside the circle) is for conventional stock tubes, and the lower curve of this pair (shown by the letter "S" inside the circle) is for the "silent" stock tubes. The more dramatic evidence of the effectiveness of the "silent" stock tube is shown by the upper two curves of Figure 5 where hexagonal stock is rotating, rattling and thrashing around inside conventional ("C" inside the hexagonal data points) and "silent" ("S" inside the hexagonal data points) stock tubes. In this comparison, the "silent" stock tubes range 10 to 20 dB quieter than the conventional stock tubes. This is not intended to represent a thorough evaluation of stock tubes, for we have not studied the effect of spindle speed, stock lengths, stock size or stock tube size; but this comparison does show a significant reduction of noise for the special quieted stock tubes, using vibration isolation and vibration damping techniques. (In the oral presentation, magnetic tape recordings are played for these four conditions.)

b. Vibration Damping Materials. Strategic use of vibration damping material on thin metal surfaces is used extensively on aircraft fuselage skins and frames. The actual reduction of radiated or shell transmitted noise may be as little as only 2 or 3 dB or as much as 5 to 10 dB, but there are situations where every decibel is vital. Damping materials or damping tape are frequently applied to thin structural members inside some machines to reduce the structure-borne transmission of sound from gears, bearings, cams, ratchets, relays, etc. Damping materials are also used on large thin panels that form the cabinet-like enclosures of some machines, notably on household appliances such as dishwashers, automatic washing machines, and refrigerators, on many of the office type duplicating or copying machines and on the interior surfaces of automobile doors, hoods, trunk lids and other large surfaces. Sometimes, sound absorption blankets pressed and held against a metal surface can provide this vibration damping action while also serving to reduce build-up of noise levels inside a machine cover.

*Schweitzer, B. J.: "A Silent Stock Tube for Automatic Screw Machines", Noise Control, Vol. 2, No. 2, p. 14, March 1956.

c. High-Pressure Air Exhaust Muffling. Release of high pressure air is a typical noise in many plants. Each single brief spurt of escaping air may not be so troublesome all by itself, but in plants having many automatically controlled air-operated devices or systems there is an almost continuous chatter of air releases around the work area. In one shop recently we found over twenty air escape ports, each giving off a short blast every 5 to 30 seconds. The shop manager was amazed to hear and comprehend all these air discharges when it was brought to his attention. The high frequency pitch of the air escape noise contributes to speech masking and when an operator works near a few of these they may contribute to long-range hearing damage. Small inexpensive mufflers are commercially available or 6-12 in. lengths of piping filled on the inside with loosely packed glass or mineral fiber can reduce much of the air escape noise.

Figure 6 illustrates the noise levels generated by a blast of air released from an ordinary shop air nozzle when fed by a 130-160 PSI air supply. The middle solid curve represents the noise levels for normal discharge of the nozzle. The high frequency end of this noise spectrum is capable of masking speech. When the air blast is directed against an obstacle, the noise made by the disturbed air stream usually results in even higher noise levels, as shown by the upper dashed curve of Figure 6. In this example, the air discharge was merely directed against a finger at 6-in. distance. Where air is used to remove stock parts, such as laminations or stampings, from an automatic punch press, these noise levels could be produced. Such noise levels are potentially high enough to contribute to the hearing damage problem.

A simple homemade muffler produced the noise levels shown by the lower dotted curve of Figure 6. This muffler was produced by wrapping the discharge end of the air nozzle with a 3-in. layer of porous flexible plastic foam and recessing the wrapping into a large fruit-juice can. In the high frequency region, this simple arrangement yielded a noise reduction of 30 to 40 dB. (In the oral presentation, magnetic tape recordings are played to illustrate the noise levels of Figure 6.)

d. Plastic Pelletizing Machine. Several plants use a high speed, multiple-blade cutting drum to pelletize extruded plastic materials. Schematically the cutting operation may be illustrated simply by the sketch in Figure 7. Continuous length, spaghetti-like strands of extruded material are fed into the rotating cutting drum and are cut into small pellets of any desired dimension. The high speed rotation of the cutting blades past the cut-off edge of the anvil produces a siren-like sound of very high intensity, possibly reaching sound pressure levels of 110 to 120 dB a few inches from the cutting edge. The fundamental frequency of the sound is the "blade passage frequency" of the cutting blades and this can typically fall in the range of several hundred to a few thousand cycles per second. Higher harmonics of this fundamental frequency are also present.

In one particular noise reduction program, a special acoustic enclosure was devised for this type of cut-off machine. A thick-walled, acoustically-lined form-fitting housing was designed to enclose the cutter and its drive mechanism, and acoustically-lined openings were provided for the entry of the plastic strands and for the exit of the pelletized stock.

The approximate noise levels in the aisle beside one machine are shown in Figure 8, for the case of no enclosure and for the case of the acoustic enclosure. The overall effectiveness of such an enclosure is usually limited by the sound leakage paths through the openings by which stock material is fed and removed and by the air and sound leakage paths in the various joints around the enclosure and in some of the gasketed covers that give access into the machine. Where ventilation of a drive motor is required, acoustically lined ducts or passageways must be provided for cooling air. Also, for maximum noise reduction it is necessary that the enclosure make no physical contact with any part of the cutting assembly or its drive mechanism.

It should be pointed out that the machine was never used in normal production runs without a protective enclosure. The original enclosure, however, did not provide adequate noise control and it was for this reason that the special acoustic enclosure was designed and added. (In the oral presentation, a magnetic tape recording is played to illustrate the cutter noise.)

e. Motor Room. In one plant a bank of electric motors and gears produced high noise levels in an adjoining work area. The heat radiated by the motors also added to the discomfort of the area. A light weight enclosure having its own ventilation arrangement reduced both the noise and the heat in the shop area.

Figure 9 shows the noise levels in the work space "before" and "after" the enclosure was provided. The enclosure wall was made up of 1/2-in. thick gypsum board mounted on metal studs, with all air cracks sealed. When the enclosure was installed, the reduced noise levels in the shop space (the lower dashed curve in Figure 9) were actually due to the machines in the shop rather than the motors and gears inside the enclosure.

f. Automatic Punch Press. The average noise levels are shown in Figure 10 for a typical operator position of a punch press at one manufacturing plant. The upper curve shows the noise levels for the original machine and the lower curve shows the noise levels following completion of an initial noise reduction treatment. The shaded area shows the design goal range desired for the final total shop noise reduction program. The upper limit of this range is the CHABA criterion for hearing preservation in the presence of steady-state narrow-band noise and the lower limit of the range is the NC-75 curve.

The initial treatment to this first punch press consisted of placing acoustic covers of metal or safety glass over all openings from the impact area of the punch press. The machine still has the same accessibility as before this acoustic treatment was added, since in the original version several expanded metal guards were already used to protect the operator. The expanded metal guards have been replaced by sliding solid safety glass panels fitted with gasketed seals. Additional noise control work is still to be undertaken, but this example illustrates that even a punch press can be quieted.

g. Stamping Machine. The average noise levels for a typical operator position of a large impact-type machine are shown in Figure 11. This is a high-speed automatic stamping machine that is very massive and includes several thick large-area steel panels that radiate the noise of each impact blow. It would be desirable to reduce the noise levels at the operator position to achieve approximately those shown by the lower dashed curve.

Extensive sound measurements have been made all around the machine in order to estimate the approximate sound power contributions made by each panel, each exposed piece of massive framing, each opening near the actual die set and each ventilation opening into the interior of the machine. In addition, vibration measurements have been made on all important structural components of the machine in order to calculate the sound levels expected to be radiated by the structure. A comparison of the measured sound levels directly in front of a large structural member with the expected sound levels based on vibration data for that structure is an important step in the diagnosis of a complex machine. Suppose that the vibration measurements indicate that a heavy, stiff framing member will not radiate very much noise. On the other hand, suppose that high sound levels are measured directly in front of that framing member. This paradox suggests that the high sound levels are probably due to some other nearby sound source and attention should be focussed on locating and identifying the sound source. If both the measured sound levels and the sound levels that are calculated from the vibration data tend to support each other, then there is reasonably good assurance that the structural member is correctly diagnosed and that an appropriate noise control treatment might be applied.

This particular machine is so complex that it has not yet been fully treated acoustically by the manufacturer. Several steps of a complete treatment have been carried out and a few compromises have been considered, but it is not expected that the design goal can be reached with partial or compromise treatments.

h. Horizontal Punch Press. A few years ago, a horizontal-acting punch press was producing excessive noise levels in an IBM shop area*. The acoustics group at IBM produced a cover for this machine that produced a noise reduction of approximately 15 dB in the middle frequency bands and up to 20-25 dB in the high frequency bands. The acoustic features of the enclosure included:

(1) gasketed safety glass viewing windows,

(2) snugly fitting access ports,

(3) muffled inlet ports for feeding stock into the machine,

(4) muffled ventilation openings into the enclosure to provide cooling air,

(5) adequate thickness of steel stock,

(6) internal surface damping, and

(7) internal absorption to contain the noise.

Note that there is a build-up of noise levels inside an enclosure, compared to the close-in noise levels if there were no enclosure, so the enclosure wall material and weight must be adequate. The use of absorption material helps reduce the inside build-up.

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*Engstrom, J. R.: "Noise Reduction by Covers", Noise Control, Vol. 1, No. 2, March 1955.

i. Pencil Shaping Machine. The first step in making a batch of pencils is to take two thin strips of cedar, groove them, insert leads in the grooves, and then glue the strips together. Stacks of the glued strips are then fed into the hopper of the molding machine that shapes the pencils. The feed mechanism provides a continuous flow of these strips. The upper cutter assembly cuts out the upper profile of a line of 8 pencils, and the lower cutter assembly cuts away the remaining unwanted material. The cut pencils then drop onto a conveyor or into a bin.

The cutter blades of this particular molding machine rotate at 14,400 RPM. The noise levels at the operator position reach and sometimes exceed 110 dB. Although the machine is quite compact, there are many openings into the cutter area and the siren-like sound is free to escape to the room. An experimental program was carried out to determine how much noise reduction could be achieved by closing up many of the openings through which sound escapes. An experimental sealed enclosure produced nearly 30 dB noise reduction at the peak frequency of the cutter.

Much of this noise reduction could be achieved with simple add-on pieces to the existing machine; but to achieve all of this noise reduction (and even more, if desired), some design changes would have to be made. To our knowledge, no follow-up work was ever done by the manufacturer because, at that time, there was no incentive. No one was asking for quieter machines and he could sell all the noisy machines that he could produce. So, why change!

An inspection of many other molding machines would show that a small effort toward closing up the noise escape paths could easily achieve a large amount of noise reduction.

j. Sonic Pile Driver. One of the dramatic devices introduced into the building construction industry in 1961 was the sonic pile driver. The sonic pile driver consists of a mechanical arrangement that converts the energy of two 500-HP diesel engines into an alternating up and down force which is coupled to the top of the pile. The speed of the engine is locked onto the longitudinal resonance of the pile casing. As the casing compresses and elongates, not over one-fourth inch at the lower end for the resonant frequency of about 100 cps, the weight of the casing and the engine load clamped at the top serve to "push" the piling into the ground.

In some actual pile driving on one job, conventional steam pile driving required 30 minutes to sink a pile 40-ft deep and the sonic pile driver "pushed" a similar pile into the ground in 45 seconds. The sonic pile driver is less noisy and the noise is of much shorter duration than that for the impact type pile driver with its repeated blows at from one to two blows per second. The vibration in the earth is less severe as well, and static loading tests on two piles driven by each method on the job described here showed a more stable setting of the sonically driven piles.

In addition to demonstrating a positive use of resonance in a mechanical system, this example illustrates the use of an unusual and imaginative way to do a job by a new and possibly quieter method.

k. Barriers and Partial Enclosures. Almost any machine or area can receive some benefit from a barrier or partial enclosure that may deflect or reflect sound to less critical spaces or that may provide "shielded" areas of lower sound levels or that may actually absorb some of the sound energy. Slides of a few representative forms of partial enclosures are shown in the oral presentation. See Figure 12 for some examples.

Depending on the noise source, the dimensions, construction and geometry of the barrier, the layout of the room, the operator position, etc., these barriers may produce localized or general noise reduction ranging from 2 to 10 dB in the low frequency region up to 5 to 20 dB in the high frequency region. For any larger amounts of noise reduction, one would have to set out to provide a total enclosure rather than a partial enclosure .

One example shows the 20 dB noise reduction achieved between two adjoining rooms containing power hammers. Each room has acoustic absorption lining and a large front opening for easy access of large parts. The enclosure provides little benefit to the operator exposed to his own noise, but noticeable reduction for all other noises to which he might be exposed.

When estimating time exposures, it is sometimes the nearly steady-state condition of all the noises of other equipment that may be a major or controlling part of the exposure of one operator and his intermittent or marginal noise-producing equipment. Thus, it may be important to reduce the "other" noise to an operator when it is difficult or impossible to reduce the noise of his own machine.

4. "DO'S AND DON'TS" IN NOISE CONTROL

The following outline is offered as a starting point for pursuing a noise reduction program on a noise source. Be aware of good noise design; use good acoustic principles whenever possible. Build one unit; check the noise output, using appropriate noise and vibration equipment. Re-design and modify as required. Follow the outline below as a checklist both to establish good acoustic design in the first place and to guide remedial steps later if necessary.

A. Airborne vs. Structure-borne Noise and Vibration

1. Have to identify which type and which paths.
2. Both finally radiated to ear by air paths.
3. In general, sound from a machine can be "heard" at lower levels than vibration can be "felt". Therefore, reduce vibration till it can't be "felt", maybe even more, depending upon environment.

B. Air Sources and Solid Sources of Sound

1. Air Sources (pressure fluctuations in air due to air movement).
   (a) Jet action of air stream produces turbulence (air cleaning, air conveying, ventilation, etc.)

(b) Periodic interrupted flow produces discrete frequencies (fans, motor vents).

(c) Air movement around obstacles (turbulence).

(d) Secondary air sources:
cracks, openings in covers, open ends of ducts, close-coupling by air of structural parts.

2. Solid sources of sound

(a) Any solid member of a system that moves or is contacted by any other moving solid member.

(b) Solid member may oscillate, expand and contract, deform, bend, slide, rotate, hit or be hit, accelerate or decelerate from uniform motion.

(c) Solid member can be set into vibration by air coupling and then transmit vibration to other members or re-radiate it as sound energy.

C. Reduction of Noise and Vibration

1. Reduction of air source noise at the source.
2. Reduction of airborne noise and vibration.
3. Reduction of solid source noise at the source.
4. Reduction of structure-borne noise and vibration.

D. Reduction of Air Source Noise

1. Reduce air flow velocity.
2. Diffuse air exhaust stream to reduce turbulence at edge and in surroundings.
3. Reduce or eliminate periodic interrupted air flow (cooling vanes on motors, less air flow through rotating part of motor; vary fan blade cutoff).
4. Smooth flow in ducts or in necessary air streams; streamline obstacles in air streams.
5. Secondary air sources:
cover holes, treat necessary open holes or ducts, break up close air coupling.

E. Reduction of Airborne Noise and Vibration

1. Reduce vibration amplitude of radiating member.
2. Reduce area of radiating member.
3. Reduce air coupling of radiating member (even drill holes to allow free flow of air to reduce pressure build-up).
4. Remove moving parts from large radiating surfaces (actual separation or by use of vibration isolation mountings).

5. Shift frequency of noise to lower frequency region (lower frequency noise less efficiently radiated from small sources, and people more tolerant at low frequencies).

6. Control the direction of radiation of sound away from the listener (good for high frequency only; barriers, baffles).

7. Provide mufflers for all required openings that can radiate noise.

8. Enclose or partially enclose the noise or noise radiator (as massive as necessary consistent with rest of system); cover all holes or cracks for air escape; gasket access doors; must have no rigid connections between noise source or radiator and the enclosure structure; consider relative stiffness of isolation mount for frequency to be controlled. For undamped enclosures, apply surface damping to reduce resonances.

9. Use acoustic absorption (glass fiber, etc.) to absorb contained sound (on inside surface of a wall or box, not on outside; porous material absorbs bouncing sound waves, does not take out much energy transmitted through the material).

10. Effectively increase distance from noise radiator to listerner, give chance for sound to spread out; same energy in the room but less intense if further away (baffles, lined ducts, directivity).

F. Reduction of Solid Source Noise

1. Change mode of operation to produce less force on the system; look for and avoid basic designs that serve as sound amplifiers.

2. Seek other ways of accomplishing the end objective or movement (electric vs. magnetic, mechanical vs. hydraulic, etc.).

3. Provide smooth finishes for sliding contacts and rolling parts (includes cams and cam followers, linkages on common shafts, gears, sliding parts); adequate lubrication to reduce stick-slip motion.

4. For rotating parts, provide maximum balance to insure uniform speed and minimum acceleration and deceleration; provide minimum clearances in shafts and mating bearings to prevent vibration.

5. For non-uniform motion, provide minimum acceleration to do the job properly but have uniform acceleration (avoid "jerk": rate of change of acceleration); use maximum available time to produce the necessary velocity change, avoid peak acceleration.

6. Reduce weight of accelerated parts, including rotating unbalanced parts; surface dampen remaining light weight surfaces.

7. Reduce accumulated backlash or clearances in a string of linkages to reduce "jerkiness" to the final action; provide spring loading to final action to give constant force to resist jerkiness.

8. Apply acceleration forces only as rapidly as parts can follow to reduce impact, overshoot, undue flexing or deformation.

9. Reduce impact force to minimum necessary; look for other ways to transmit force or information to a system than by impact.

10. Use "soft" surfaces where possible to reduce impact (cams, cam followers, hammers, gears); use soft inserts under impact surfaces when possible; reduce mass and area of impact parts; use damping material on impact parts.

11. Apply damping materials to eliminate resonances of rods, panels, linkages, gears.

G. <u>Reduction of Structure-borne Noise and Vibration</u>

1. Provide vibration isolation for mounting of a noise or vibration source to its base; have no short-circuiting rigid connections (flexible connection in pipes or wiring, free coil turn in wiring connection). If rigid connections required, isolate next larger assembly that includes the rigid connections. Isolation mounting stiff enough to transmit performance requirement or provide functional operation, soft enough to prevent transmission of high frequency vibrational forces. Caution that springs, as steel bars, transmit some high frequency noise. Use rubber pads with springs. Use rubber-in-shear or felt, cork or rubber pads for high frequency isolation.

2. Reduce the weight of a vibration assembly, attach it to heavy-weight base with isolation mounts. Always try to support a vibration source from a massive "inertia block" (with use of isolation mounts; design curves on transmissibility assume infinite mass and rigidity for base). Avoid vibration isolation of a heavy source on a light-weight flexible base; base may be as flexible as isolation mount.

3. Avoid resonance of isolation mount with driving frequency. Design mount resonance frequency at least 2 to 4 times above lowest driving frequency.

4. Reduce the radiating area of structural paths (drill holes).

5. Add vibration damping materials to structure paths that will transmit vibration to another point in the system or to parts that can vibrate at various resonances. Surface damping, "spaced damping". Effect of temperature and frequency. Effectiveness somewhat proportional to thickness. Most effective on thin stock and at regions of maximum bending.

6. Provide area, weight, and impedance "mismatches" at junctions of different parts. (Impedance mismatch: materials with large differences in values of density x velocity of sound)

7. Avoid close air coupling between large surfaces (air a good spring connection for large areas at close spacing; example Thermopane glass not much better acoustically than single glass of same total weight).

8. Avoid structural connections that amplify force (illustrate with "T" connection; slight flexing of base will amplify to large motion of top).

5. CONCLUSION

It would be dishonest to imply that noise reduction comes simply and at no cost. The treatment may not be simple to execute even though it may be simple in concept. The cost may involve changes in attitude by the user, the operator, or the shop foreman. Some compromises in machine speed, performance, accessibility, or convenience may be required. If noise reduction is a controlling requirement, some of these compromises may have to be made.

Most of the examples described here relate to noise reduction steps added to a machine without actually changing the "internal workings" of the machine. If design engineers can adapt some of these guide lines into their original designs, possibly some noise reduction can be built into a machine without having to add it on later. This, of course, is the real objective. There is no "magic" in acoustics. If noise reduction is wanted, noise reduction must be designed into a machine, not added onto it as an after-thought. Some of the basic noise reduction principles have been given in this discussion, but the real need facing all of us is the motivation to do something about it. Many of the methods, materials and knowledge are available.

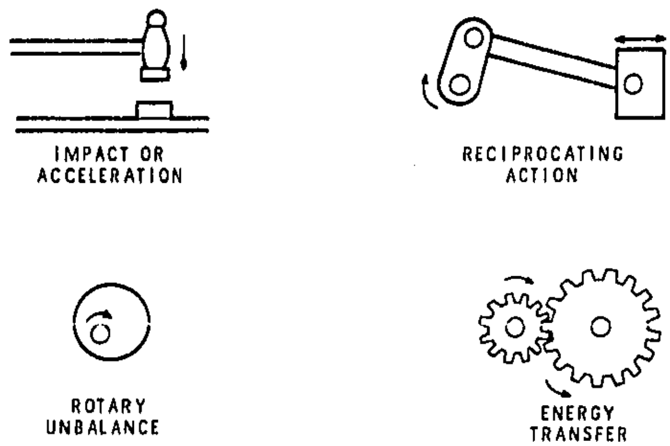

FIGURE 1
MECHANICAL NOISE EXCITATION

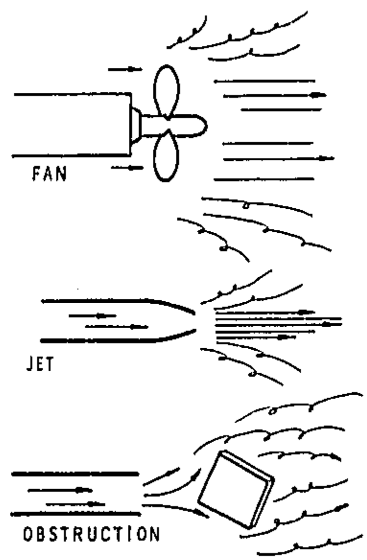

FIGURE 2
TURBULENCE
ASSOCIATED
WITH
MOVING
AIR

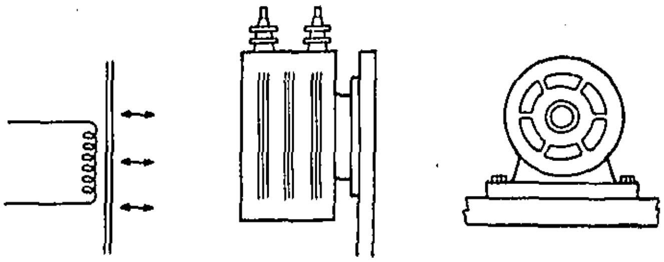

FIGURE 3
ELECTRO-MAGNETIC
INDUCTION

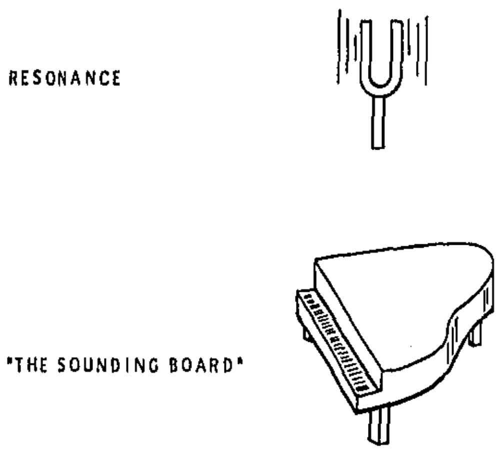

FIGURE 4

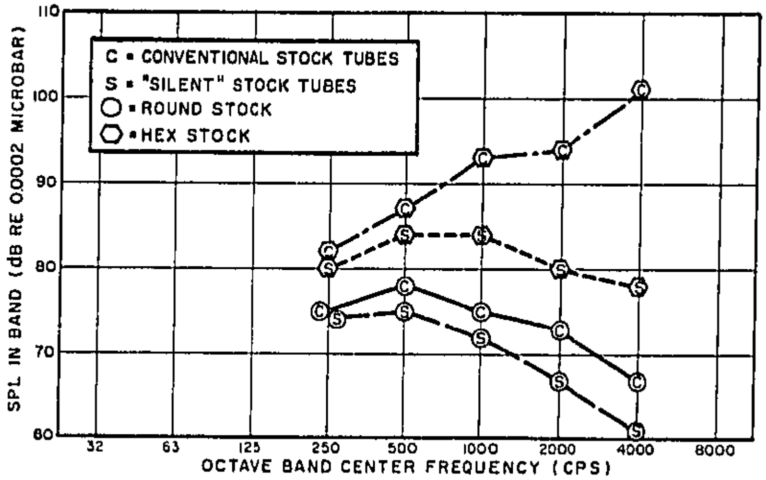

FIGURE 5
NOISE LEVELS 5 FT FROM VARIOUS STOCK TUBE ARRANGEMENTS OF AUTOMATIC SCREW MACHINE

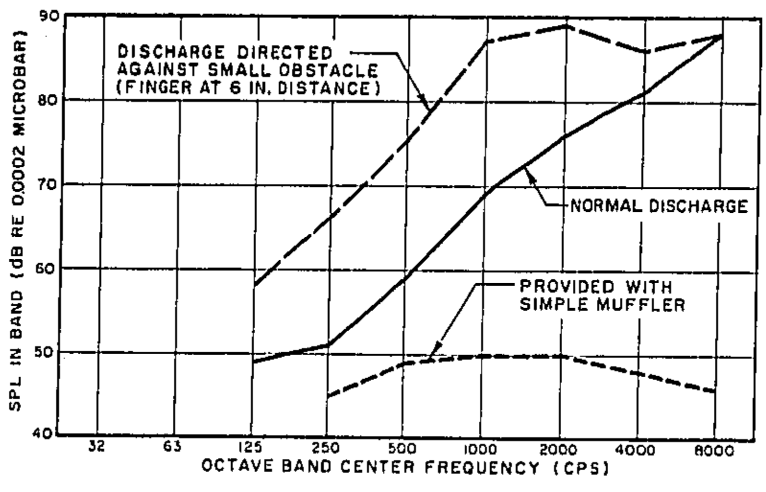

FIGURE 6
NOISE LEVELS 3 FT FROM SHOP AIR NOZZLE DISCHARGING AIR FROM 130-160 PSI LINE

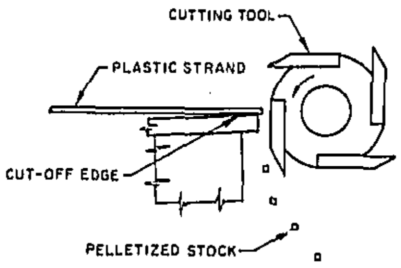

FIGURE 7
SCHEMATIC OF PELLETIZING MACHINE

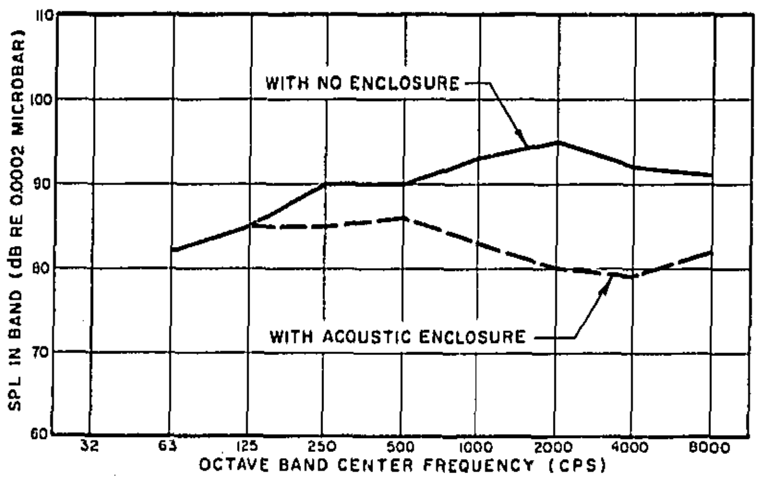

FIGURE 8
NOISE LEVELS IN AISLE POSITION
BESIDE PELLETIZING MACHINE

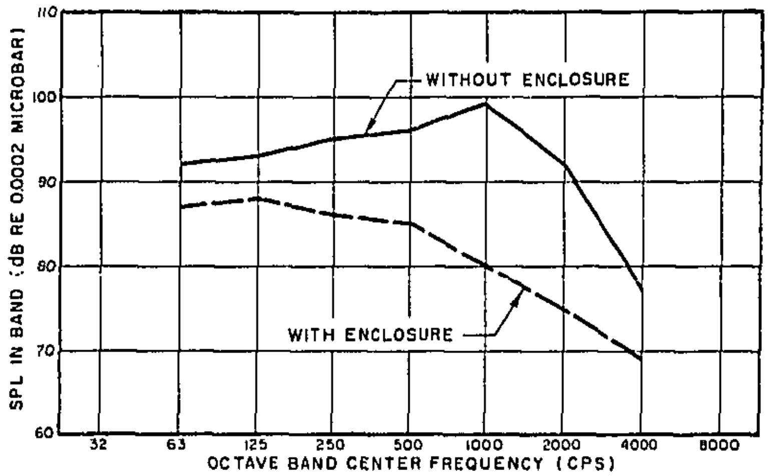

FIGURE 9
NOISE LEVELS IN FACTORY SPACE DUE TO SEVERAL NEARBY MOTOR-GEAR DRIVES

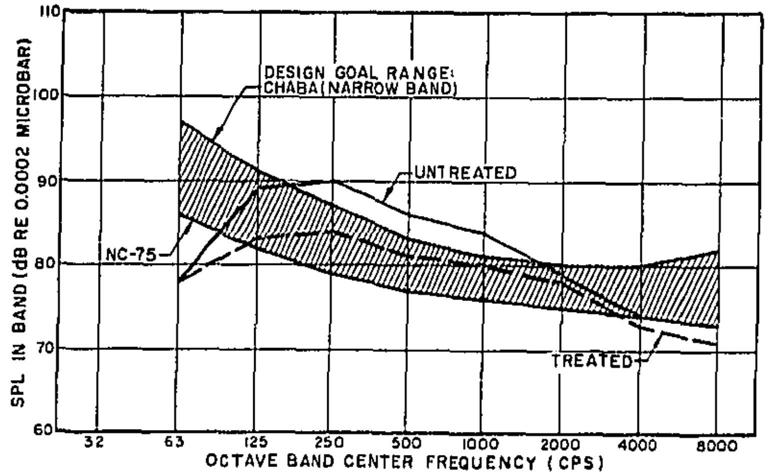

FIGURE 10
NOISE LEVELS AT OPERATOR'S CONSOLE OF PUNCH PRESS

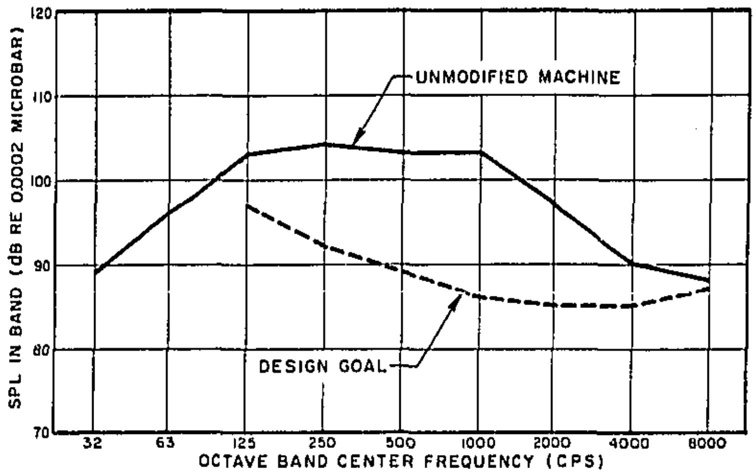

FIGURE 11
EXISTING AND DESIRED NOISE LEVELS AT OPERATOR POSITION
OF HIGH-SPEED AUTOMATIC IMPACT-TYPE MACHINE

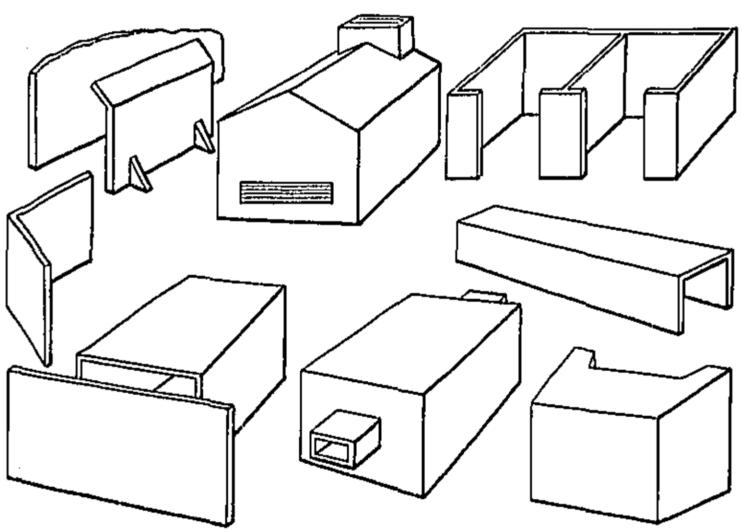

FIG. 12
PARTIAL ENCLOSURES FOR USE IN NOISY WORK AREAS

U.S. DEPARTMENT OF LABOR

WAGE AND LABOR STANDARDS ADMINISTRATION

BUREAU OF LABOR STANDARDS

WASHINGTON, D.C.

QUESTIONS AND ANSWERS
ON THE
WALSH-HEALEY SAFETY AND HEALTH REGULATIONS

These questions and answers are issued in response to inquiries concerning the revised Walsh-Healey Safety and Health Standards of May 20, 1969.

1. "Q" How many companies are effected by Walsh-Healey safety and health regulations?

   "A" It is estimated that 75,000 plant locations and up to 27 million workers are covered at one time or another.
   The spillover of influence on companies not subject to WHPC, motivated by legal and consultative actions taken by the Department, probably involves a much greater number of locations and workers.

2. "Q" What sections of the new regulations are expected to have the greatest impact on government contractors?

   "A" Since the noise regulations are new ones, and there have heretofore been no such national requirements for noise control the greatest impact might be expected there. Control of toxic gases, vapors and and dusts will also have a greater impact on contract operations.
   A great number of workers are subject to noise exposures and/or to chemicals and toxic substances. So little has been done to control the problems, in most small operations, particularly, the impact will be felt the greatest in these areas.

3. "Q" When did the noise regulations become effective?

   "A" On May 20, 1969 upon publication in the Federal Register.

4. "Q" Who will inspect plants for compliance?

   "A" Federal safety engineers and industrial hygienists. In the six States, where there are Federal-State agreements, the States will inspect.

5. "Q" Will single Federal inspectors check for compliance with all phases of the Walsh-Healey regulations or just for a single part of them?

   "A" Federal safety engineers are trained to observe and identify problem areas. If additional expertise is required in a highly technical subject area, they may request that other staff technicians come in to evaluate only that problem.

6. "Q" Is there adequate qualified manpower for inspections?

"A" About five percent of the contractor plant locations can be reached in any year. This is not wholly adequate, but it can reach some of the worst problem areas.

7. "Q" How do you select the five percent to be inspected?

"A" There are several criteria. (1) The high injury frequency rate industries; (2) the high injury severity rate industries; (3) operations with catastrophe potential; (4) industries with the size operations that would not be expected to have their own safety staffs and might need help; (5) complaints of unsafe or unhealthful conditions by employees or organized labor.

8. "Q" What is the penalty for failure to comply with the Walsh-Healey regulations?

"A" If a contractor absolutely refused to abide by the regulations, as determined by a hearing examiner on the record of a hearing, the Secretary of Labor could recommend the company be declared ineligible for government contracts for a period of three years or he could recommend that the agency contracting with them cancel an existing contract.

9. "Q" Does the U.S. Department of Labor have any special advice or recommendations as to how an individual company should go about complying with the regulations or is the individual company completely on its own?

"A" Companies hire experts in different fields based on their need for them. We would hope they would hire safety personnel full time, appoint part time safety personnel, draw on their insurance carriers, states or other sources of aid and advice. If we inspect their operations, we would expect to help them work out problems we might find if they show good faith in cooperating with us.

10. "Q" Will the Walsh-Healey provisions specifically state the degree to which (1) machine and equipment are to be guarded and (2) what constitutes adequate medical and first aid facilities?

"A" The body of the text of the regulations is somewhat detailed. In special cases, all considerations will be based upon and drawn from provisions in appropriate national standards and recognized practices. Where standards do not specifically cover the subject, techniques and concepts covered in standards or recognized good practices will be adapted to the specific case.

11. "Q" What will a company be expected to do to meet the noise regulations?

"A" When employees are subjected to sound levels exceeding those listed in the regulations, feasible administrative or engineering controls shall be initiated and utilized. If such controls fail to reduce sound levels to within the acceptable levels, prescribed personal protective equipment shall be provided and used to reduce sound levels within the prescribed levels. In all cases, where sound levels exceed acceptable levels a continuing effective hearing conservation program shall be administered.

12. "Q" Will all companies be expected to meet the regulations immediately?

"A" Good faith by an employer in attempting to meet the intent and purpose of the regulations will be the key test. In most industries and operations, technology has advanced to the point where the environment can be controlled to meet the standards. Reasonable time limits will be allowed where special costs and equipment or alterations are required. Where technology is not so advanced, good faith can be shown by initiating hearing conservation programs, providing and requiring the use of personal protective equipment and initiating discussions with technicians, designers and machine manufacturers in order to set plans and completion date, if possible, for new machine designs and new methods and processes. When new machines are purchased and new facilities built or otherwise obtained for operations, specifications should include noise level limitations and control.

13. "Q" What does the U.S. Department of Labor expect in the area of benefits in enforcing and implementing the noise control regulations?

"A" We expect the technology which has been developed over the past 15 to 20 years to now be applied at work sites to protect workers from being exposed and suffering hearing losses. We expect designers and manufacturers to start reducing the noise levels generated by their products or to provide means for operators to be protected from the excessive noise levels. We expect better and more effective personal protective equipment and devices to be developed and made available. We expect some of the old, outmoded and obsolete noisy processes to be replaced with more modern, efficient and quieter processes. We expect the increased demand for expertise in this field to result in more students studying the subject and getting trained in environmental control as a career. We expect more consultants and consulting firms to provide services in this subject, and of course, we expect fewer workers to end up with needless hearing disabilities.

14. "Q" Where there are unclear areas in the regulations, or areas subject to different interpretations, what is the best advice to industry?

"A" Clarification bulletins will be issued whenever questions arise for which they are needed. These bulletins will be made available and widely distributed and publicized. It is always impossible to cover every possible situation by regulations or standards. In such cases, good faith by the employer in applying the best available knowledge, techniques and concepts to control the situation will be acceptable. The intent and objective of the standards is to provide for the health and safety of the worker. Actions taken to meet this intent and objective will indicate good faith on the part of the employer.

15. "Q" Can a company which does not have "A" scale sound level meters but has octave band analyzers use this equipment to record their noise levels?

"A" Yes. In the regulations is a chart graph on which octave band sound pressure levels may be converted to the equivalent A-weighted sound level by plotting them on the graph and noting the A-weighted sound level corresponding to the point of highest penetration into the sound level contours.

16. "Q" How will "feasible administrative or engineering controls" be interpreted? Will high cost or economic factors of control be considered?

"A" There was much discussion over whether to state that "economically feasible" controls should be determined and implemented. It was finally decided that the word "feasible" should be interpreted in the broadest possible sense and that economic factors should certainly be one of the major considerations--but not the only factor or the controlling factor. Economics may be considered in determining the time limits allowed to an employer in which to come into full compliance with the law.

17. "Q" What is required in a "continuing effective hearing conservation program"?

"A" Where noise levels in the working environment exceed those allowable, a program is necessary to assure that the personal protective equipment provided and used is effective in preventing deterioration of a worker's hearing. The most desirable program would include pre-employment hearing examinations and periodic and regular audiometric tests and evaluations. Audiometry and the use and application of personal protective devices should be under medical supervision or be done by a nurse, audiologist or trained technician under medical direction.

18. "Q" What should be done if workers refuse to use personal protection?

"A" It is management's prerogative and duty to see that all means and measures are taken to assure that work is conducted in a safe and healthful manner. This will require good education and training techniques and effective and forceful supervision. The standards state that personal protection shall be provided and used. This puts a burden on the worker to cooperate and use what management provides. The U.S. Department of Labor will cooperate with the employers, whenever and however appropriate, to assure the cooperation of employees. However, the Federal government cannot become directly involved in labor-management relations of this sort.

19. "Q" In a multi-plant organization, if only one plant is working on a government contract are all plants subject to the WHPC safety and health standards?

"A" Although there may technically be a legal reach to all parts of the corporate operation, we are primarily concerned with operations where the contract is being performed. It would seem to be poor management practice, however, to apply double standards to the safety and health of workers in various plants merely on the basis of where a Federal contract is being performed at the moment.

20\. "Q" If only a part of a plant is working on a government contract, is the whole plant subject to the standards?

"A" If there is an interchange of workers between contract and non-contract operations, and if those working on contracts are exposed to hazards created by the other operations in the general work areas, the whole plant is subject to the standards.

21\. "Q" Are sub-contractors of a government contractor subject to the standards?

"A" Legally, it may be possible to reach sub-contractors if major portions of the contract are sub-contracted and sub-standard safety and health standards are found after complaints are filed and investigated. In such cases, the sub-contractor may be considered a substitute manufacturer. As a practical matter, it is difficult to ferret out all possible sub-contractor operations for inspection. We are primarily concerned with prime contractors but expect them to exert appropriate influence over their sub-contractors in order to keep their own eligibility for government contracts intact.

22\. "Q" If it is impossible to guard the point of operation on a machine, what action should be taken to meet the standards?

"A" The intent and objective of the standard is that all effort be taken to prevent the operator from having any part of his body in the danger zone during the operating cycle. If by positioning of the part, holding of it, using remote control or by other means, the operator is prevented from being in the danger zone the intent of the rule will be met.

23\. "Q" Does the Department expect to provide some help to small companies who do not have full time safety staffs and may not have access to all the standards adopted by reference?

"A" An inspection survey guide is being developed for use by Department field personnel to guide them as to the situations to look for while making plant surveys and which will be keyed to the standards applicable to each situation. These survey guides will be available to public contractors as a "do-it-yourself" inspection guide so that a continuing inspection program can be set up by any small contractor within his own organization to assure his meeting the requirements of the law.

24. "Q" What does "whenever this part adopts by reference standards, specifications and codes published and available elsewhere, it only serves to adopt the substantive technical portions of such standards, specifications and codes" mean?

"A" In the text of many standards and codes, there are statements that "this standard is not to be used for regulatory purposes" or other restricting statements not pertinent to the technical substance. Obviously, we do not adopt these statements as part of the standards to be met.

25. "Q" How will the standards which you have adopted be determined as applicable in any particular situation?

"A" Field inspection personnel will survey operations under a contract according to organized survey procedures, which will be available also to public contractors to let them know ahead of time what is expected of them. Hazards will be categorized and keyed to applicable standards so that fairly uniform application of the law can be achieved nationwide. The applicable standards and sections thereof will be identified so that compliance with the ordinary employment situations will be simplified.

26. "Q" What is meant in the standard under the Gases, Vapors, Fumes, Dusts and Mists section when you state that in cases where protective equipment is used, such protection must be approved for each specific application by a competent industrial hygienist or other technically qualified source?

"A" Special knowledge is needed in determining the proper control measures and protective equipment and devices to be used when hazardous substances are used or generated. Access to such knowledge must be provided either by full time staff employees, consultants or persons specifically trained in the subject area. Certain recognized agencies test and approve equipment and devices for use in specially hazardous situations. Persons competent to evaluate and prescribe appropriate approved control procedures and devices must be used to assure the meeting of the intent and purpose of the law.

27. "Q" Does the Department expect records of all injuries to employees to be kept, whether disabling or not?

"A" Yes. In small companies there are few disabling injuries in a year's time, so that keeping records of all injuries is the only way a manager will be able to analyze his safety problems. Large corporations with the best safety programs keep records of all injuries, not just disabling ones, since it is often only a matter of chance as to an injury being serious enough to be disabling or not. Good managers want all the information they can get to see if trends are being set and operations are tending out of control. Good faith in meeting the intent and purpose of the law starts with recording and using injury statistics to control cause factors.

28. "Q" What will be the criteria considered for approving a variation from the regulations?

"A" If equal or greater safety can be provided by a method which is different from that prescribed by the regulations, a request for a variation can be filed and will be evaluated.

If technology has not progressed to the point where engineering and administrative controls are feasible, a request for variation may be considered if a plan for taking all actions possible to achieve the maximum control and improvement according to prescribed time limits is submitted.

29. "Q" Why do you spell out standards for radiation?

"A" We adopt AEC standards where they apply and have jurisdiction. Where AEC does not have jurisdiction, we have spelled out the requirements to be met under WHPC.

30. "Q" Does the phrase "exposures above the TLV" mean that the USDL standards make all TLV's ceiling values?

"A" We have adopted the TLV's as time weighted averages, not ceiling values.

31. "Q" What is meant by "competent industrial hygienist or other technically qualified source"?

"A" If a company does not have a certified industrial hygienist on its payroll, they must show they have access to and utilize the services of such expertise or have persons specially trained to handle their hazard exposures.

32. "Q" What are "professionally accepted safety and health practices"?

"A" These would include practices published in Data Sheets, Manuals and Handbooks by nationally recognized and technically competent organizations.

33. "Q" What is meant by "ready availability of medical personnel"?

"A" This cannot be defined precisely, but an employer must show that plans are in effect for treating any possible injury within a reasonable and practical time limit based on the type and location of the operation.

34. "Q" What is meant by "adequately trained to render first aid"?

"A" American Red Cross certified training or the equivalent would be desirable.

35. "Q" What does "compliance with the safety, sanitary, and factory inspection laws of the State in which the work or part therefore is to be performed shall be prima facie evidence of compliance with the subsection" mean?

"A" Prima facie means at first sight or so far as can be judged from the first disclosure. Obviously, if the State has no requirements or they are so sub-standard as to not be compatible with Federal requirements, we will look beyond the first disclosure and ask that an employer meet the nationally recognized standards we have adopted. The prima facie concept is not an exclusion.

* * * * * * * *

It should be remembered that the safety and health standards were approved and concurred in by the Secretary's National Advisory Committee made up of representatives from labor, management and the public. This committee is chaired by Howard Pyle, President of the National Safety Council.

The format of the regulations is a new and unique departure from the historical form of governmental regulations development. Instead of writing out by sentence and paragraph every rule to be met under the law, nationally recognized standards were adopted by reference as the standards to be met. This drew upon all the expertise in the country which has participated in standards development and adopts the standards that enlightened and forward looking management have voluntarily devised for their own guidance.

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Guidelines for Designing Quieter Equipment

CLAYTON H. ALLEN

Bolt Beranek and Newman Inc.,
Cambridge, Mass.

Quieting of noisy equipment necessitates the consideration of both sound and vibration. Energy can feed from one to the other. Noise is best controlled at the source in the design stage. Familiarity with the principal sources of noise and the general means for its control aids in the selection of devices and mechanisms that simplify or eliminate the need for corrective measures in a finished product. Detailed and practical texts on noise generation and control are available. Although they cannot replace professional help when it is needed, they are invaluable for general guidance and for anticipating problem situations which may need study.

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Copies will be available until March 1, 1970.

Guidelines for Designing Quieter Equipment

CLAYTON H. ALLEN

NOISE

Noise is defined here as any unwanted sound, Fig.1, or vibration, Fig.2. Each may create or be created by the other.

Airborne Sound ⟷ Solidborne vibration

In controlling noise, it is necessary to distinguish between sound, which is airborne, and vibration, which is solidborne, because the means of controlling these phenomena differ.

Sound

Sound consists of small pressure variations in the air which radiate from a source at a speed of approximately 1000 fps. The frequency of sound, which is the frequency of repetition of pressure variations, is measured in cycles per second (cps) or Hertz (Hz). A wavelength is the distance between pressure maxima in a traveling wave; it is equal to the speed of sound divided by the frequency.

Vibration

Vibrations in solid material generally travel faster than airborne sound. Thus, for a given frequency, the wavelength is longer in solid materials than in air.

When the dimensions of solid structure are small compared with the vibrational wavelength, the structure may vibrate nearly as a unit. However, for high frequencies, some portions of a solid structure may vibrate almost independently from the rest and at much larger amplitudes. This is particularly true for panels and membranes and for building sections in which the vibrations travel as bending waves. Such waves may travel at various speeds, sometimes more slowly than sound in air, depending upon the frequency of the wave, the structural configuration, membrane tension, and also upon the properties of the material itself. The importance of the wavespeed as regards noise control is that it affects the efficiency with which sound radiates; when the wavespeed along a surface is less than that of sound in air, almost no sound is radiated except near edges or near stiffening members.

SOURCES OF NOISE

The sources of noise will be considered separately as sources of sound and as sources of vibration.[1]

Sources of Sound

For the purpose of noise control in machines, sound sources can be divided in three classes, as illustrated in Fig.3.

Solid Sources. A vibrating solid member al-

[1] In some machine components, liquids may be the actual source of noise; however, such sources are of limited interest, and methods for their control can generally be deduced from similarities with solidborne vibrations and airborne sound discussed here.

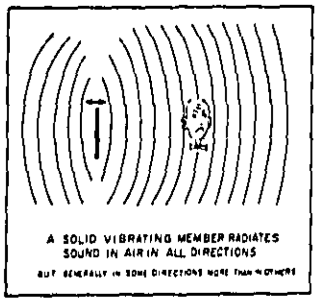

Fig.1

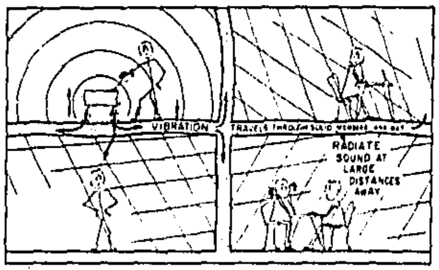

Fig.2

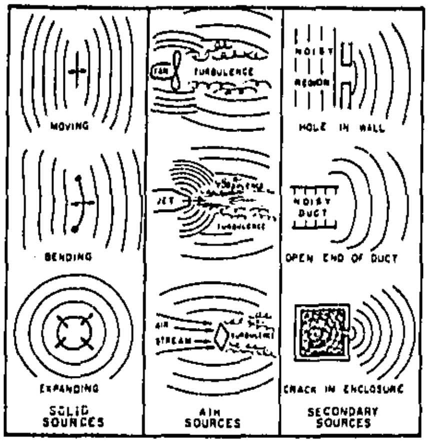

Fig.3 Sources of sound

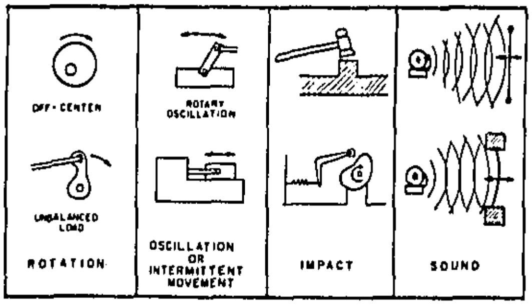

Fig.4 Sources of vibration

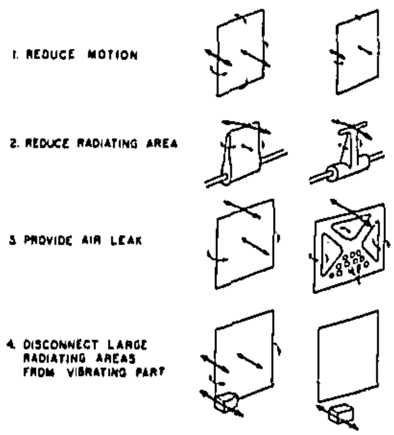

Fig.5 Reduction of airborne sound from solid sources

ternately pushes and pulls the air, creating small pressure changes that tend to radiate in all directions, generally more in one direction than in another.

Air Sources. Air that is moving creates pressure fluctuations that radiate as noise. The primary mechanism for creating the pressure fluctuations is turbulence; however, when turbulence impinges upon a solid surface, noise radiation is greatly increased.

Secondary Sources. A hole in a wall, although not a primary source of noise, may appear to be a sound source to an observer who is otherwise shielded from the true source. The hole may be treated as a sound source driven by the air on the opposite side of the barrier.

A panel or other solid surface might be considered as a secondary source of noise when driven by contact with a prime mover or through some intermediate linkage. However, we shall consider such a source as a primary solid source and the vibrating part as simply the driving mechanism; the driving part may be either solid or air.

Sources of Vibration

For the purpose of noise control in machines, we can divide sources of vibration, such as illustrated in Fig.4, into two classes, moving machine parts and moving air.

Moving Parts. Solidborne vibrations are generally produced by moving parts that cause a varying force upon the solid system as they perform one or more of the following operations:

a) Rotation with eccentric load.
b) Movement with intermittent or varying speed.
c) Impact against other parts.

Moving Air. Solidborne vibrations may also be produced by sound waves, since they cause fluctuating pressures against a solid surface. Solid surfaces can effectively convert sound pressure into vibrations when any of the following conditions hold.

a) When the surface is undamped, having one or more resonant frequencies corresponding to frequencies in the impinging airborne sound.

b) When the surface is thin and flexible, coupling well to the relatively low impedance of air.

c) When the surface is unperforated, nonporous, and large, having linear dimensions equal to or larger than half a wavelength of the sound in air.

REDUCTION OF NOISE AT THE SOURCE

The most effective means of reducing noise

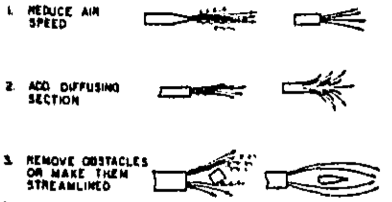

Fig.6 Reduction of airborne sound from air sources

is to alter the source so that it produces less noise.

Reduction of Airborne Noise at the Source

Airborne Noise Produced by Solid Vibrating Parts. Such noise may be reduced by the following changes, which are illustrated in Fig.5.

a) Reduction of the amount of motion, i.e., reduce the vibration amplitude.

b) Reduction of the effective area of vibrating part. This may be done by reducing the size of the vibrating part, by disconnecting it from larger radiating areas, or by providing air leaks through the part so that, as it moves, air can slip through or around it. These changes inhibit pressure build-up and thus reduce noise radiation.

c) Reduction of frequency of vibration where possible. This is very effective but is usually not practicable since the frequency of motion is governed by other operations of the machine.

Airborne Noise Produced by Air in Motion. This can be reduced by the following alterations, which are illustrated in Fig.6.

a) Reduction of the air flow velocity.

b) Addition of a diffusing section to exhaust openings.

c) Removal of obstacles from the air path (especially sharp or angular obstacles), or streamlining all objects that must remain in the air stream.

Airborne Sound from a Secondary Source. Such noise can be reduced by the following changes, which are illustrated in Fig.7.

a) Reduction of the area of opening; preferably, providing an airtight closure.

b) Directing the opening away from the listener; this helps most for high frequencies, i.e., those frequencies for which the perimeter of the opening or the dimensions of the surfaces shadowing the observer are greater than a wavelength.

c) Addition of a sound-attenuating muffler ahead of the opening; this may be an acoustically lined duct. Even a simple baffle with acoustical

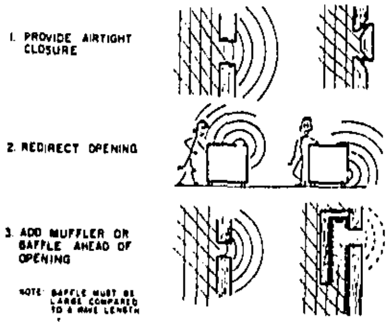

Fig.7 Reduction of airborne sound from secondary sources

lining will give some reduction, although this will vary greatly with the application.

Reduction of Solidborne Vibration at the Source

The most effective means of reducing vibration is to alter the source or change its mode of operation so that it produces less force on the system.

Vibration Caused by a Moving Part. A reduction of the accelerating (or decelerating) forces that are created between the part and the remainder of the system will reduce the radiated noise. Means for reducing accelerating forces include the following:

a) Smoothing the finishes on sliding and rolling parts to remove high spots that cause unwanted motions (chatter, lateral vibrations, and so forth).

b) Reducing the mass of accelerating parts. This includes improvement of dynamic balance for rotating parts.

c) Reducing the peak acceleration[2] of mov-

[2] Caution: The peak acceleration can be reduced by making all accelerations nearly constant over the time allotted for a given velocity change. This implies that acceleration should change abruptly from one value to another. Such an abrupt change in acceleration, known as jerk, affects the system in a complicated way and may cause serious vibrations. This subject is beyond the scope of the present discussion, except to say that accelerating forces should be applied no more quickly than the system can follow without significant transient deformation and overshoot. Otherwise, large unanticipated accelerations may result. Tolerances and clearances between parts and linkages must be considered as contributing to the flexing of the system, especially where several linkages are used in succession.

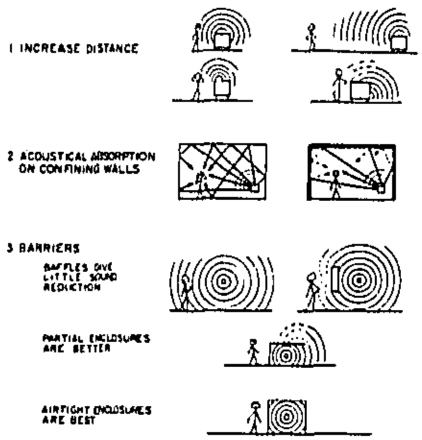

Fig.8 Airborne sound isolation

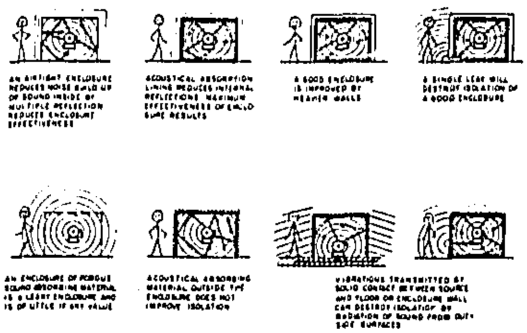

Fig.9 Airborne sound isolation by enclosures

ing parts, i.e., reduce the rate at which the velocity of a moving part is changed, by employing the maximum time available to produce the required velocity change, and by making each acceleration as nearly constant as practicable over the time available for a velocity change.

Vibration Caused by Sound. When sound waves in air strike a solid structure they may cause significant vibration in it. Some of the control procedures listed in the following are the same as illustrated in Fig.5.

a) Perforate the solid surface to admit relatively free flow of air so that the pressure on opposite sides may equalize; holes in the surface should be closer than $^1/4$ in. wavelength of the sound considered.

b) Make flexible sections more massive. This applies particularly to enclosures which must be airtight and, therefore, cannot be perforated.

c) Add damping materials to the surface of thin sections to eliminate resonances.

NOISE ISOLATION

Mechanical operations, which are unavoidably noisy, must be isolated from the listener. A satisfactory treatment usually requires interruption of all transmission paths both in the air and in the solid structure.

Airborne Sound Isolation

Sound sources can be isolated from a listener by procedures discussed below and illustrated in Fig.8.

Separation of Source and Receiver. A source and receiver may be isolated by increasing the distance between them. Due to the spreading of the sound waves, sound pressure reduces to half its value for each doubling of the distance from the source to the receiver. This means noise reduction is effective only where the sound waves can spread without confinement.

Acoustical Absorbing Treatment. Where hard walls confine the sound, as in a room or other enclosure, sound is reflected back upon itself. Then the sound pressure on the average remains high throughout the enclosure. Separating the source and receiver in such a space is not an effective isolation measure unless the source and receiver were originally very close together; in any event, the separation is effective only to the point where the total reflected sound equals the sound arriving directly from the source.

The following measures can be taken to reduce noise in an enclosure.

a) Lining a hard-wall enclosure with acoustical absorbing material reduces sound reflection and thereby lowers the pressure fluctuations throughout the enclosed space. Separation of source and receiver is effective over greater distances in a lined enclosure.

b) Where source and receiver are contained in a duct, pipe, corridor or the like, a separation between source and receiver can be made very effective by using acoustical absorbing material on the inner surfaces of the duct. The material should be thicker than $1/30$ of a wavelength and may be as much as $^1/4$ wavelength thick for maximum absorption. A bend in a duct increases absorption when the duct dimensions are larger than half a wavelength.

For some types of ducts, very efficient acoustical treatments are commercially available as packaged units that can be assembled in a wide variety of configurations.

Sound Barriers. When the distance between

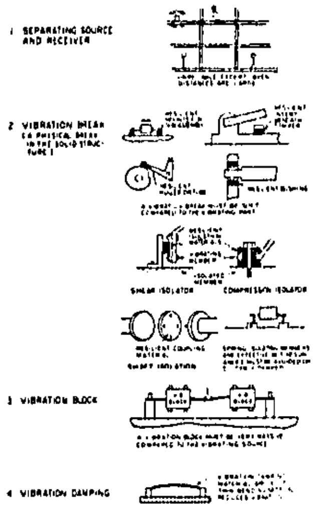

Fig.10 Vibration isolation

the source and listener cannot be increased, noise can be reduced by a suitable solid barrier between them.

If only a small amount of noise reduction is required, particularly at high frequencies, baffles and partial enclosures are effective. These devices rely upon the directionality of sound waves as indicated in Fig.8. To be most effective, such devices should:

a) Have dimensions that are large compared with a wavelength.

b) Be located near the source or the receiver to maximize the shadow effect.

Where a large amount of noise reduction is required, a total enclosure is the most effective barrier; it prevents sound radiation by confining the sound to the immediate vicinity of the source, as illustrated qualitatively in Fig.9. It is apparent that an effective enclosure must:

a) Have heavy walls that will remain substantially motionless when sound waves strike.

b) Have absorptive material inside to dissipate the sound reflected from the walls and prevent reverberant buildup of sound levels.

c) Be substantially airtight, since even a small air path through the wall will transmit a large fraction of the sound produced by the source, especially if there is little absorption inside.

d) Be vibrationally isolated from the mechanical source of sound, otherwise the enclosure (because of its larger area) may radiate more noise than the original sound source.

Vibration Isolation

Vibration generated at one point in a solid structure can be isolated from the remainder of the structure or a portion of it, as illustrated in Fig.10.

Separation of Source and Isolated Point. Generally, the physical dimensions of a mechanical device are small and will not admit any significant separation of source and observation point. In buildings, however, vibratory energy can be attenuated by spreading and by losses in the structure. Thus, separation of vibratory machinery from quiet areas is generally valuable. However, due to resonances within the structure, an increase in separation does not always reduce the vibration at a given observation position.

Vibration Break. A vibration break is generally the most effective and least expensive means of vibration isolation. Ideally, a vibration break is a physical break in the solid structure which prevent vibrational forces from being transmitted. Practically, a vibration break must be filled or bridged by some material that will maintain the location of parts with respect to each other. This bridging material is effective when it is:

a) Stiff enough to give the required alignment of parts or to transmit the required low-frequency forces for functional operation.

b) As soft as possible, consistent with condition (a) in the foregoing, in order to prevent the transmission of high-frequency vibrational forces.

c) Resistive, like felt, putty, and certain rubbers, so that it will not create a springlike system that might resonate and produce violent, perhaps destructive vibrations at some critical frequencies.. Springs may be used if sufficient resistance is added to control resonances.

(Caution: A vibration break is effective only when all solid connections between two isolated parts are broken. No solid structure may be permitted to come in contact with both sides of a vibration break.)

Vibration Block. Where a vibrating member cannot be separated physically from the remainder of the structure, a massive structure may be attached between the vibrating member and the remaining structure. This mass must be large and relatively immovable compared to the vibrating member. It serves to reflect vibrations back from the portion of the structure to be isolated.

A vibration block is particularly effective if it can be preceded or followed by a vibration break or by a relatively compliant structural section.

Vibration Damping Material. Where the source and observation point are relatively far apart, significant reduction of vibration can be obtained by application of vibration damping materials. Damping materials are particularly useful in reducing vibrations in highly resonant members such as panels, webs, springs, reeds, and so forth.

Proper use of damping materials improves the performance of vibration breaks and blocks; large vibrations tend to build up in the isolated source member unless some means of absorbing this energy is available.

The choice of vibration damping materials depends upon many factors, including type of structure considered, the frequency of vibration, the operating temperature, and the particular function of the vibrating part. Some materials like lead, sand, asphalt, impregnated felt, mastic, and so forth, are commonly used. Several specially designed damping materials and composite damping structures are commercially available. Characteristically, good damping materials should:

a) Have a stiffness that is comparable to the stiffness of the material being damped so that it will extract a maximum amount of energy during bending motion. Most damping materials are temperature-sensitive and must be carefully selected.

b) Be at least as thick as the section of metal being damped and several times thicker for maximum damping.

c) Be applied on the relatively thin sections of a vibrating member, where the maximum bending occurs.

DESIGN AND COMPROMISE

Any practical noise control problem in machine design generally involves a combination of all of the considerations discussed in the foregoing. The importance of each of the individual considerations varies according to the specific application.

In some instances, it may not be possible to reduce the total noise output, but alterations in materials or operations may permit a change in the noise spectrum, i.e. a shift of the maximum noise from high to low frequencies, thereby reducing annoyance and interference with normal activities such as speech communication, telephone, and general relaxation.

In some instances, a small amount of steady broadband noise may be used to mask another noise that is unpleasant, irregular, or distinctive. This "acoustic perfume" technique, though usually a last resort, is often an effective means for improving the acceptability of a device and may be the only practical expedient.

The problem of noise control in machine design is a continual search for compromises that will optimize the balance of mechanical performance and noise reduction. Excellent treatments of this subject are contained in the following references.

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3 Rothbart, H. A., Mechanical Design and Systems Handbook, McGraw-Hill, New York, 1964.

MACHINE DESIGN
September 14, 1967

The Anatomy of Noise

—by Leo L. Beranek
and
Laymon N. Miller

Noise is about as popular as ugliness. And like ugliness, it often seems an intangible nuisance, one difficult to deal with in precise terms. But the problem is not that complicated. Noise is a true operating characteristic of an engineered device, as real as power, strength, or speed. It can be defined, measured, contained, prevented, even legislated against. Perhaps more important, acoustics experts now have reasonably accurate ideas on just how much din you can create without annoyance, harm, or legal liability.

"In the future,

noisy products

may be illegal."

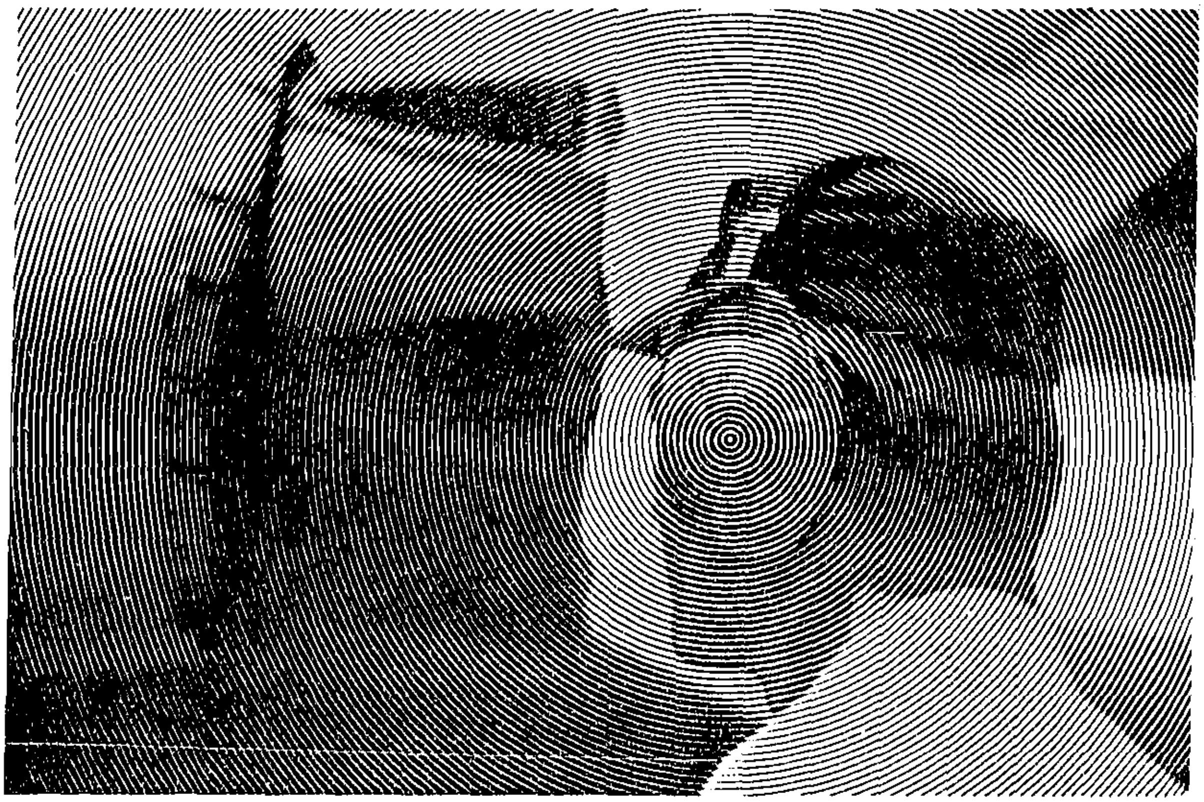
NOISE is not easily envisioned as an issue to be agonized over. Here now, gone in an instant, it seems an abstraction, one best ignored in the hope that people will learn to live with it.

In truth, noise is not so elusive. The study of noise has drawn on both psychology and physiology to become a definitive human-factors discipline. People react to noise in predictable ways. Some noise is good noise. Levels at which noise becomes annoying or harmful can be measured and defined. Standards for acceptable noise can be established as rationally as mechanical requirements.

The progress in noise control is fortunate, because noise has, in a sense, gotten out of hand. Surveys indicate that the rush into the age of technology has raised the average noise level by a decibel each year over the last 25 years.

The trend does not necessarily mean that we are on a collision course with cacophony, for society is now forcefully applying checks to prevent an unbridled increase in noise. The principal controls are social pressure, buyer resistance, and laws—truly a complex assortment of considerations.

The problem, therefore, assumes many facets. Even if the purchaser of a machine does not object to its sound output, the community may object—and place restrictions on the machine operation. Or acoustic fatigue may affect the operator in subtle ways to bring about poor mental attitudes or alleged injury.



The useful aspects of noise must also be considered. The rush of wind around a cruising automobile conceals mechanical noise and body squeaks. The hum of the turbine engine soothes the air traveler. Background noise in a busy office masks conversation in an adjacent office, and serves the same purpose as thicker walls and more expensive construction.

Noise is helpful when it signals impending failure, as with cracked bearings or pitted gears. Noise also indicates imbalance and vibration in rotating machinery. Hissing air warns of a gasket failure. Changing pitch cautions that an engine or motor is running erratically. But useful noise is rare, and annoying noise is common, so the problem usually is one of noise prevention and control.

Researchers have established recommended limits for nearly every aspect of noise. Criteria have been developed to protect hearing, to permit unstrained conversation, and even to prevent various degrees of simple annoyance. The experts know approximately how much background noise people want or expect for sleeping, for relaxing, for office work, for the classroom, for the auditorium or concert hall, for the restaurant and the retail shop, for talking on the telephone or conferring in a conference room, for working in machine shops or mechanical areas, even for feeling comfortable in an airplane, train, or automobile.

Noise-measurement instruments provide accurate descriptions of noise characteristics. And by comparing this description with the applicable noise criteria, suitable noise-control measures can be provided.

Some of the ramifications of noise and motives for its control are obvious; others are subtle. Granted, noise disturbs sleep, interrupts conversation, and generally annoys people. But the more profound aspects require penetrating insights: Does noise damage hearing or have other psychological effects? Does it reduce efficiency? What are public attitudes on noise? What noise-abatement laws must mechanical devices adhere to? In short, what are the incentives to control noise?

The case for quiet

To Prevent Deafness: Stand too close to an exploding firecracker, and you are likely to wind up deaf—at least temporarily. Everyone realizes that. Not so apparent is the fact that the effect of noise on people is cumulative; it produces an "acoustic fatigue." Repeated moderate noise builds up to inflict the same damage as a single loud noise. But even more important, repeated noise is the only type (short of a shattering explosion) that produces *permanent* hearing loss.

Some of the hearing loss from acoustic fatigue is recovered when the noise is removed. But the permanent damage cannot be determined until the injured person has been away from the injurious noise for several months. With exposure to moderately intense noise, separation of only a few weeks will allow a valid test for permanent damage. But hearing ability may not stabilize for many months after an explosion or other intense noise.

The hearing loss suffered from years of exposure to noise usually differs in various parts of the hearing range. One study (see graphs on page 178) on people who regularly worked in a 90 db noise environment showed little hearing loss in the 1000 Hz range until some 30 years of exposure. The same group, however, lost more than 50 db in the 4000 Hz range after an exposure of about 16 years.

People differ greatly in their susceptibility to hearing loss. Because some ears are more sensitive than others, it is impossible, in terms of cost, for industry to establish controls that would protect *every* worker against *all* noise-induced hearing loss.

It may be possible, however, to detect those whose hearing is most likely to deteriorate with extended noise exposure. Such individuals can then be transferred to quieter jobs. At present, many otologists believe that the best indicator of susceptibility is probably early evidence of permanent threshold shift at 4000 Hz (see graphs).

Most employers in noisy industries are giving serious attention to prevention of noise-induced hearing loss. The hearing of employees often is tested upon employment, prior to exposure to high noise levels, and at frequent times afterwards. Workers in noisy locations usually are required to wear earplugs or noise-attenuating ear muffs. A worker usually is transferred to another job if he shows a tendency toward greater-than-normal hearing loss at 4000 Hz.

Many states have laws that establish the rights of a worker to preserve his ability to understand speech, even if loss of hearing does not prevent his earning a living. A few states already have laws requiring the worker to protect his hearing and requiring the employer to provide the earplugs or ear muffs necessary for protection.

To Allow Conversation: The average sound of speech, for good verbal conversation, must be approximately 15 db above background noise in the frequency range most used in speech—300 to 5000 Hz. Ordinary conversation is conducted at a level of about 50 to 60 db in this frequency region (for talking distances of about 3 to 6 ft). Background noise, therefore, should be no higher than about 30 to 40 db in these octave bands if speech is to be understood without delays, difficulties or errors.

Addressing machines, accounting machines, data processing machines and some duplicating machines often exceed this level and are therefore intolerable in an average, quiet office. They would bring to a halt any verbal communication at normal voice levels. In a noisy plant or production area, verbal communication may be restricted to just a few shouted words. The point to remember is that even relatively low noise levels can still interfere with verbal communication—a fact to be considered during the early stages of design.

To Keep Workers Happy: No one has ever proved conclusively that noise influences work output. It does not generally affect the ability to reason, to perform mathematical calculation, or to operate equipment. But noise can cause distractions and encourage accidents. Prolonged exposure to loud noise may reduce vigilance and promote irritability, annoyance, and fatigue.

Noise also affects morale. Given the choice to work or relax in either quiet or noisy surroundings, the average person will choose quiet every time. But, *average* does not mean *every*. Surveys show that about one-fourth of the population seems to be unperturbed by any noise level. Apparently, these people could (and often do), work in noisy environments and live happily next to elevated trains, highways, and airports.

A Close Call With Chaos

BETWEEN 1951 and 1955, managers of traditionally noisy industries—such as textile mills, aircraft plants, and metalworking shops—were extremely distressed by a court decision regarding noise.

It all started in 1948 with an obscure labor-relations case in New York State. Matthew Slawinski helped operate a drop forge in the J. H. Williams Co. Continued exposure to the noisy forge had made him partially deaf. He filed a claim against his employer on the ground that noise-induced deafness is an occupational disease, and that he should receive compensation based on existing schedules for occupational disability. The New York State Court of Appeals (208 N.Y. 546) ruled in his favor, even though he was not disabled from earning full wages at his regular job. He was awarded $1,661.25.

Across the nation, hundreds of claims for occupational loss of hearing were soon filed by employees against their employers. By 1955, at least one state official estimated a total award potential on the order of billions of dollars.

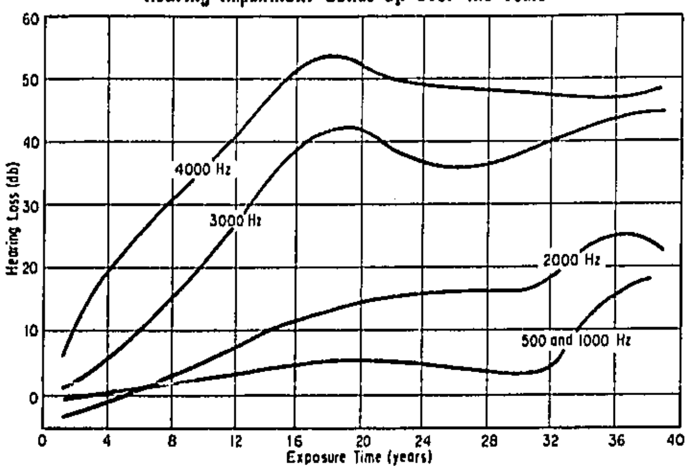
Industry was spared this agony by subsequent court rulings, first in New York, and then in other states. The courts ruled that since a part of the measured hearing loss is caused by reversible acoustic fatigue, the permanent loss cannot be determined until after the employee has been separated from the injurious noise level for six months. Most workers refuse to give up their wages for this length of time.

To Keep Neighbors Happy: The courts are swamped with suits brought by individuals and community groups seeking relief from excessive noise. In 1965, over 150 civil suits concerning jet-aircraft noise were pending before the nation's courts. Highways, factories, power plants, playgrounds, community swimming pools, sports stadiums, and police stations are opposed by local residents on the grounds that they create too much noise.

In one industrial community there is an installation of large fans that can be heard more than 8 miles away in a quiet rural town. One

Charts and Numbers

Hearing Impairment Builds Up Over the Years



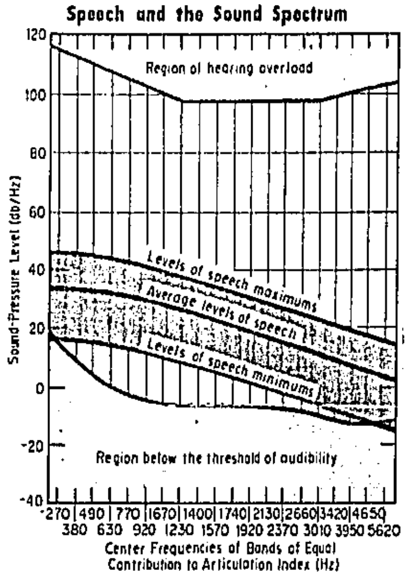
Hearing loss suffered from years of exposure to noise usually differs with the various frequencies of perceptible sound. One of the more extensive studies involves 400 men and 90 women known to have no otological impairment. The group had, for periods as long as 40 years, been regularly exposed to noise of 90 db in each of the six octave frequency bands between 150 and 9600 Hz.

The study found that appreciable hearing losses at 3,000, 4,000, and 6,000 Hz occurred in the first 15 years. At 500, 1,000, and 2,000 Hz, hearing losses increased less rapidly, essentially as linear functions of exposure time. Some of the men tested, even those as young as 30 years old, found it difficult to understand speech, after about ten years of exposure. Loss at 6,000 Hz was somewhat less than at 4,000 Hz.

Men showed greater increases in hearing losses than women. The difference may have resulted, in part, from the fact that the women had regular work breaks during each shift. The men did not.

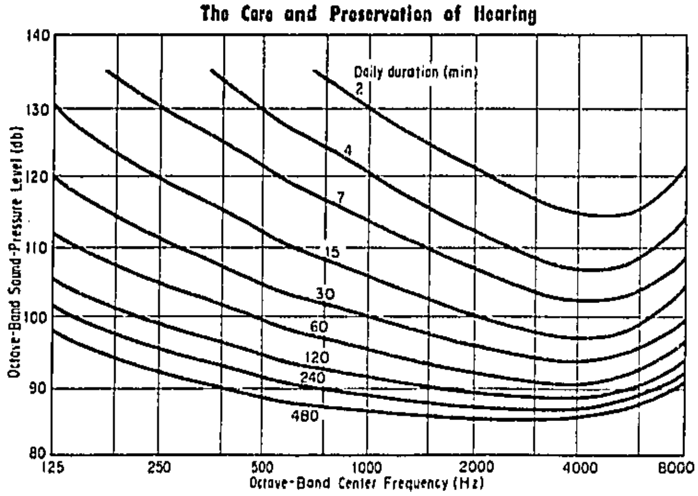
The study also showed that people differ greatly in their susceptibility to hearing loss. After 10 years exposure, the spread in hearing loss at frequencies of 3,000 Hz or more was about 15 db. After 25 years, the spread increased to about 30 db.

Speech and the Sound Spectrum



In any frequency range, speech varies in intensity by about 30 db. The peaks of speech (at the 1% level) lie about 12 db above the average level; the minimums lie about 18 db below the average. The sound-pressure levels are normalized to a bandwidth of 1 Hz. Speech levels shown are for an average young male speaking with a normal voice to a listener 3 ft away. Region below the threshhold of audibility is for continuous spectra sounds for an age group of 20 to 29 years.

The Care and Preservation of Hearing

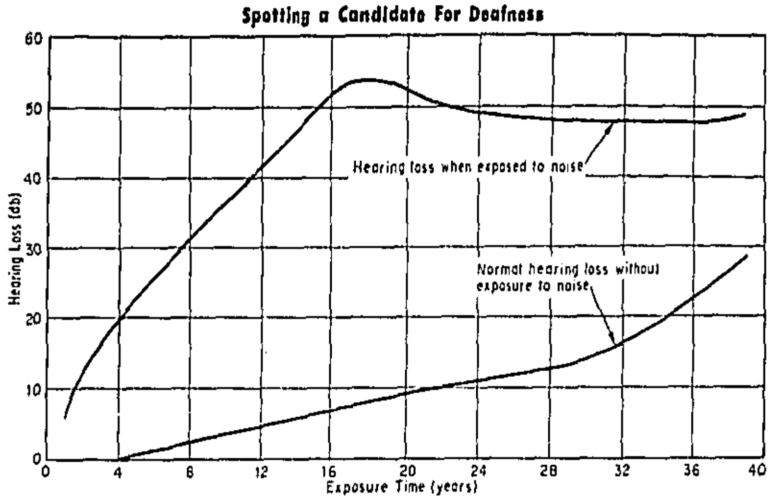


Permissible daily duration of exposure to noise can be established in terms of frequency and noise level. These particular curves are for a daily exposure for 10 years. A typical person should sustain no substantial hearing loss (less than 15 db by the American audiometric reference zero at 1,000, 2,000, and 3,000 Hz) over that period for noise conditions within the limits established by the curves. For example, a man exposed for 30 minutes daily to noise of 105 db in the 500 Hz band, 100 db in the 1000 Hz band, and 95 db in the 2,000 Hz band should suffer no impairment in ability to understand speech for at least 10 years.

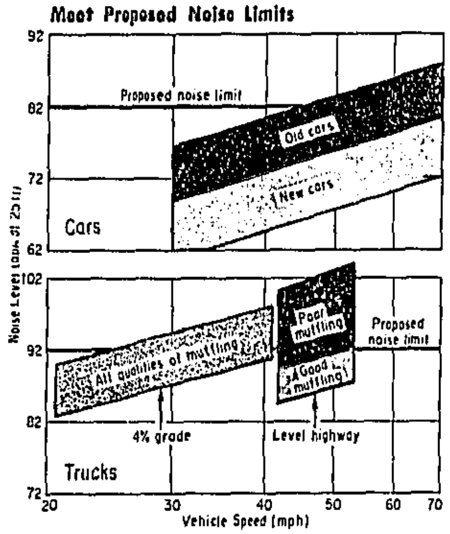
for the War on Noise

Many otologists believe that the best indication of susceptibility to loss of hearing is early evidence of permanent threshold shift at 4000 Hz. The increase in threshold at that frequency is most rapid during the early years of exposure and tends to slow down after 12 to 15 years. The curves above are adjusted to begin at a common origin, representing an age of 18.

Spotting a Candidate For Deafness



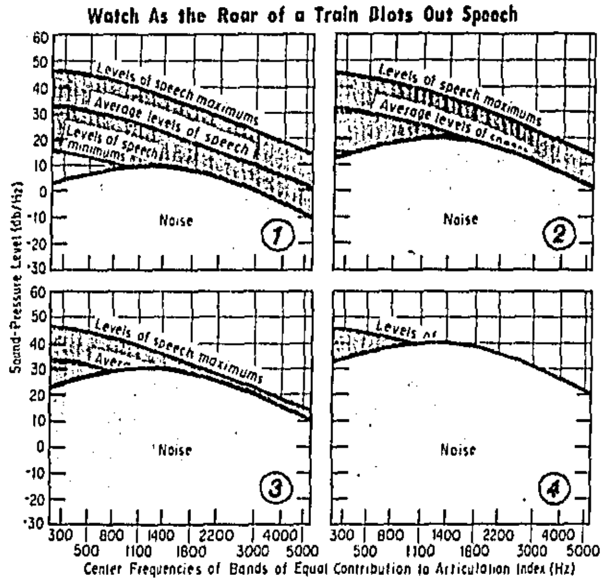
Not All Cars and Trucks Meet Proposed Noise Limits



Legislation proposed in California would limit passenger-car noise levels to 82 dbA. Measurements show that some older vehicles exceed this limit at speeds as low as 45 mph.

The proposed California legislation would limit trucks to 92 dbA. Even with good mufflers, speeds over 50 mph will cause some problems. On a level highway, a truck with poor muffling is about twice as noisy (10 dbA) as a truck with good muffling.

Watch As the Roar of a Train Blots Out Speech



Nobody needs proof that the noise of a passing locomotive interrupts conversation, but here is a graphical representation showing exactly why such loud sounds blot out speech. The noise of an approaching train, Panels 1 through 3, can be represented as a masking effect on normal speech. Panel 4 represents conditions on a station platform as the train passes.

nearby resident complained that the high-powered fans, that operate 24 hours a day, "sounded like a motorboat that would never go away." Another manufacturing plant that operates a forging hammer is threatened with a law suit if it continues nighttime work.

One community is annoyed by the nighttime noise of a truck marshalling area. Another group of neighbors are bothered by a loud-speaker system that calls outdoor workers to the telephone at a nearby power plant. Horn-honking is periodically curtailed by strict law enforcement in some noise-conscious cities. Nighttime helicopter operations at midtown heliports are restricted in several cities. To combat such nuisances, state highway patrol officers in some states are learning how to use sound-level meters.

To Obey the Law: Perhaps the significance of the concern by society is the fact that government agencies, both at the local and federal level, are beginning to show considerable interest in noise control and abatement. An example is in the aircraft industry, where the FAA has established noise criteria for the supersonic transport during take-off and landing. The agency has prepared a bill, one that the President is recommending to this session of the 90th Congress, to set maximum noise levels on all new aircraft.

Legislative proposals do not stop with aircraft. The State of California has proposed legislation which would prohibit noise levels in excess of 82 dbA for passenger cars, and 92 dbA for trucks and buses at posted freeway speeds (or the maximum speed of the vehicle).

The British government is considering a law that would require all new passenger cars and trucks to be quieter than 85 dbA. Motorcycles and other mechanically propelled two-wheeled vehicles would be limited to 90 dbA. The acoustical test for British automobiles requires measurement of the noise at a point 25 ft from the centerline of the lane in which the vehicle travels, for three different operating conditions: 1. constant speed of 30 mph and top gear; 2. starting from a steady speed of 30 mph and—begining 32 ft before passing the test microphone—accelerating as rapidly as possible over a distance of 65 ft; and 3. maintaining a constant speed of 30 mph at full throttle with brakes applied. The highest noise level obtained under these three conditions is used to rate the vehicle.

In France, maximum permissible noise levels determined under the British test regime are 83 dbA for passenger cars and small trucks, 86 dbA for motorcycles, and 90 dbA for large trucks and buses.

As technology creates even more potential for noise, and as man crowds further in on his neighbors, quiet operation may become a necessary product feature. If quietness is not essential for sales purposes, it could well be mandatory under regulatory standards. In the future, noisy products may be illegal.

To Sell More Products: Many noisy products are sold. The vacuum cleaner, automatic dishwasher, garbage disposal, air conditioner, and power lawn mower are notable examples of noisy appliances. Diesel and gas turbine generators, cooling fans, reciprocating machinery, punch presses, grinders, riveters, and many other types of machinery generate considerable noise. Transport noise, particularly from trucks, buses, trains, and airplanes, is particularly annoying.

What can be done about noise from these sources? Sometimes the product can be redesigned. For example, suitable isolation of the driving parts and possible strengthening or damping of sheet metal enclosures noticeably reduce the radiated noises from many appliances. The automatic dishwasher can be quieted but may require a 2 or 3-in. increase in size, and may cost the customer an additional $10 or $15. An industrial machine can be quieted, but the noise treatment may require operational compromise. The bus, subway, or garbage truck can be quieted, but they will cost more, and the city officials generally will not pay extra for quieter vehicles unless city dwellers complain about the noise of present vehicles.

What happens when you try to convince an appliance manufacturer that he should design a quieter but more expensive product? He immediately cites examples in which the investment did not pay off—for example, the quiet typewriter that sold no better than standard models . . . the air conditioner that was noticeably quieter than competitive models but did not sell because stores would not stock it at the higher price and because the public could not distinguish an advantage between it an other makes in the noisy department store.

However, the public is becoming much less tolerant of noisy products. Quietness has sold many products, and will sell many more. The quiet marine outboard motor is one example; the electric blanket with a quieter thermostat is another. The slogan, "The Ford rides quieter than the Rolls Royce," and the Rolls Royce slogan, "At 60 miles per hour, the loudest sound you hear is the ticking of the dashboard clock," were designed to link quietness with quality.

Three attributes are important to any discussion of noise: Intensity, time-pattern, and frequency.

How loud is loud?

Intensity is "loudness," usually measured in decibels but often measured in other units. Naturally, the higher the intensity, the more disturbing the noise. Above a certain level (and duration of exposure at that level) there is serious risk of damage to hearing.

But loudness is often more important in relative, rather than absolute, terms. If a product is used in an area where there already is appreciable background noise, the sound that it generates is less objectionable than if it is used in quiet surroundings. Strange or unfamiliar noises are more disturbing than commonly recognized noises.

Time-pattern is a factor in the relative acceptability of noise. Factors in this category include how rapidly the noise occurs, whether it occurs during the day or night, summer or winter, or whether it is steady or intermittent, whether it begins abruptly or gradually, and whether it is impulsive or continuous. A housewife, for example, will tolerate the noise from a vacuum cleaner if she uses it occasionally. But suppose she uses it every day for several hours? Or runs it when her husband is sleeping? Obviously, various standards can apply to a single device.

Frequency is usually considered in terms of frequency distribution. Most devices emit a full range of frequencies. For analysis, such complex sound is divided into frequency intervals. The relative sound energy in each interval can then be determined. Common practice is to divide sound into eight or nine ranges from about 20 to about 10,000 Hz—essentially the full audio frequency range of humans.

This type of frequency analysis is necessary for two reasons. First, the human ear is considerably more sensitive to sounds in the high frequency regions than it is to sounds in the lower regions. Thus, an analyst must know the frequency composition of a sound to judge how a human will respond to it. For example, a high-pitched sound usually must be 20 to 30 db lower than a low-pitched sound for the two to be comparable in terms of human response.

A second reason for the frequency analysis is that acoustic controls must be designed according to the frequency content of the noise. For example, a control designed to reduce high-frequency noise may not reduce low-frequency noise at all. Likewise, low-frequency controls may not silence high-frequency noise. Thus, what is acceptable noise or noise control in one product may be unacceptable in another. Interaction of all characteristics of sound must be considered.

Noise reduction should start with a study to determine the sources and paths of noise. For example, it is important to determine whether the noise is radiated directly from the source as airborne noise, or whether it is carried by structural paths to other parts of the machine where it is radiated by secondary sounding boards. In addition, it is important to determine the true cause of the noise. Is it the drive motor, gear train, bearings, hydraulic system, or exhaust?

Reducing noise

A diagnosis is easier to perform on an existing product than on a product under design. But even a completely new product usually goes through a model and field-trial stage, and a detailed acoustic study can be made at that time.

What can be done when unacceptable noise sources and paths have been identified? Several courses are possible.

Redesign the Source: Complex machines often evolve from earlier, simpler models that operate slower and perform fewer functions. The accoustic design of the later models has usually been of little concern, particularly if it would have interfered with any other functional characteristic.

Possibly some of these machines have squeezed the last ounce of function out of the basic design. It may be worthwhile to start fresh with new designs and new processes if still greater performance or noise reduction is a serious objective.

A good example is the turbojet engine. Noise from a jet engine increases according to the eighth power of the exhaust velocity. Therefore, a development such as the fan-jet, with its lower exhaust velocity, significantly reduces engine noise. From the early days of the jet engine until now, engine noise has decreasd 5 to 10 db, while engine thrust has nearly doubled. Rubber-tired subway trains and sonic pile-drivers are other examples of new design concepts that reduce operating noise.

Of course, some products are inherently noisy. The sonic boom is a phenomenon that cannot be eliminated by any known technique. The only cures available are limitations on speed, flight path, and number of flights. For other inherently noisy equipment, such as forge hammers, punch presses, and log chippers, the best solution is to isolate either the noise source or the operator.

Adjust Output Rating: Other adjustments are possible if a new design concept is not available. Power can be reduced (noise is energy converted into sound), speed can be changed, or new materials and different structures can be used.

Add Surface Damping: Damping can be helpful where lightweight metal transmits or radiates structure-borne sound. Commercially available damping tapes or spray-on deadening materials can be applied to reduce radiated sound by a few decibels.

Isolate Vibration: Vibration isolation is helpful where structural paths are major causes of noise. Isolation typically reduces noise anywhere from a few db up to 20 or 30 db.

Add Barriers: The absorbent-lined telephone booth is an example of a good acoustic barrier. Large inside surfaces of acoustic absorption and walls that reflect unwanted exterior noise combine to reduce noise as much as 5 to 15 db inside the open-front telephone booth. In a somewhat similar manner, this same type of barrier can partially contain a noise and reduce noise radiation. Barriers (and almost all noise reduction devices) are more effective in the high frequency regions than in the low frequency regions.

Line Ducts and Passages: Noise transmitted through slots or passageways can be reduced by adding absorbent linings. Ordinary lining can reduce high-frequency noise by 2 to 5 db per foot of length. Thick linings are required for substantial reduction at lower frequencies.

Use Mufflers: Special mufflers of parallel or undulated baffles may be required where large quantities of air must pass through openings. The dimensions of the mufflers can be selected to attenuate specific frequencies. Middle and high frequency sounds are more easily controlled by mufflers than are the low frequencies.

Condition the Environment: Noise reduction need not deal exclusively with the equipment generating the sound. If a machine is to be used in a fixed location, sound-conditioning the site is often helpful. Noise levels inside a room or enclosure are higher than out-of-doors where noise radiates away freely. At least some of the sound build-up can be kept under control by placing sound-absorption material inside the room.

Such techniques typically reduce the average noise level (at reasonable distances from the noise source) by about 3 to 6 db in the low frequency region, and by about 5 to 10 db in the high frequency region. At distances of only 2 to 5 ft, the direct sound is nearly always louder than the average sound in the room. Room absorption is thus not much help to the operator of a noisy machine. However, the technique lessens noise from adjacent machines.

In the outdoors, noise attenuates by 6 db for each doubling of distance away from the source. Inside a building, noise decreases at about 3 to 6 db for each doubling of distance in the vicinity of the source. But because of reflecting surfaces, the decrease may be only 0 to 2 db for each doubling of distance at locations farther from the source.

Keep People Away: It is often practical to completely house a noisy machine in its own room or enclosure. Perhaps the operator can remain in the quieter area outside the enclosure. But even if he cannot, the enclosure benefits other workers in the area.

Acoustical considerations in the design of noise enclosures include weight and composition of the walls, gasketing and silencing of doors or access ports, and lining or baffling of open passageways.

Understanding the general principles of noise aids in recognizing whether a problem exists, and helps in directing work toward solutions. But a more direct approach is to draw on experience available in the problem area at hand. Approaches found satisfactory in specific fields are detailed in articles that follow.

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