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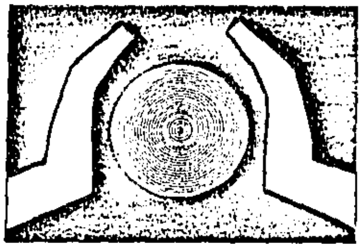
Noise Control for Reciprocating and Turbine Engines Driven by Natural Gas and Liquid Fuel



American Gas Association, Inc.

Noise Control
for
Reciprocating and Turbine Engines
driven by
Natural Gas and Liquid Fuel

December 1969

Prepared for

AMERICAN GAS ASSOCIATION, INC.
605 THIRD AVENUE
NEW YORK, NEW YORK 10016

BOLT BERANEK AND NEWMAN INC.
50 Moulton Street
Cambridge, Massachusetts 02138

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PREFACE

This manual has been prepared for the American Gas Association and its member companies by the acoustical consulting firm of Bolt Beranek and Newman Inc. This semi-technical manual is directed toward architects, mechanical engineers, structural engineers, building owners, potential natural gas customers and gas company personnel.

Noise of reciprocating and turbine engines, fueled by natural gas and liquid fuel, can be controlled, and many successful installations throughout the country attest to this. This manual provides data, guidelines and procedures for the design of quiet installations.

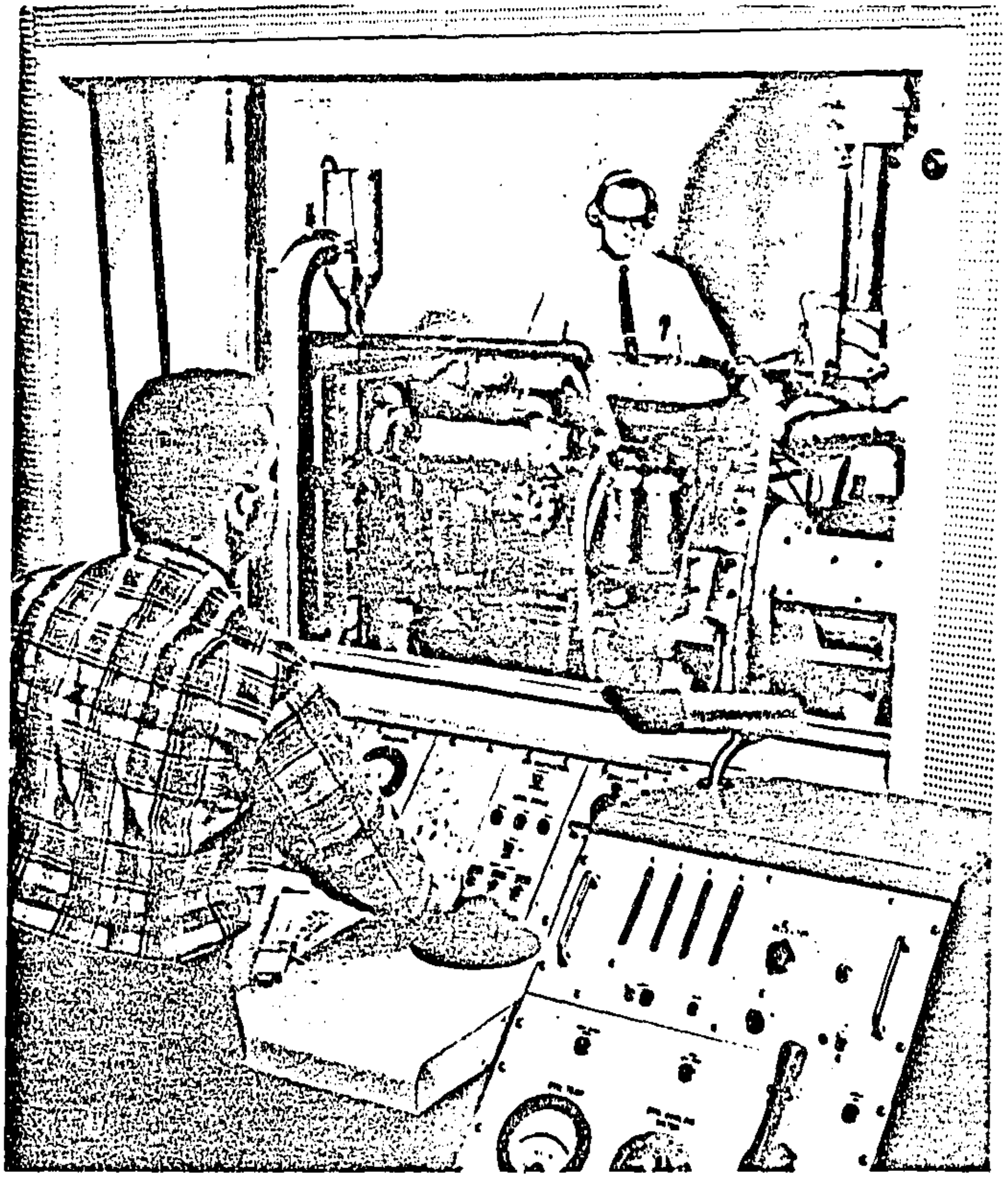
Although the engine noise data summarized in this manual represent a rather thorough study of over 120 turbine and reciprocating engines, there is no intention to suggest that these data are so complete that they could form the basis for setting up a "standard" on engine noise. This is not the objective of this manual. In fact engine noise data are offered only to permit a quantitative approach to noise control. Please do not consider this manual as any type of industrial "standard" on engine noise!

In providing this manual to the engineering public, neither the American Gas Association nor its member companies nor Bolt Beranek and Newman Inc. assumes any legal or technical responsibility for any designs based on the use of the manual.

The author wishes to thank A.G.A. and members of its Prime Mover and Large Tonnage Air Conditioning Sales Promotion Committee and James J. Kennedy, A.G.A. Manager of Prime Mover and Large Tonnage Promotion, for their interest and assistance in the preparation of this manual.

LAYMON N. MILLER
BOLT BERANEK AND NEWMAN INC.

March 1969



In Caterpillar Technical Center, engines are tested in test cell beside each control room. Both rooms have liberal use of acoustical absorption material to reduce noise levels. Double-glass viewing window and separate doors from each room into common corridor further reduce noise transmission into control room. When inside test cell, engineers and operators wear ear protection. *(Photograph is courtesy of Caterpillar Tractor Company.)*

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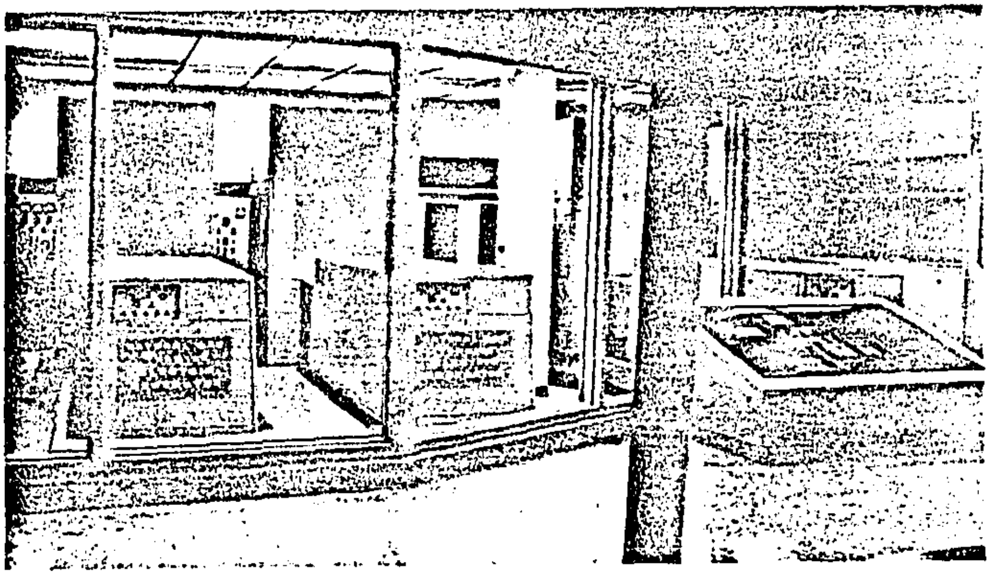
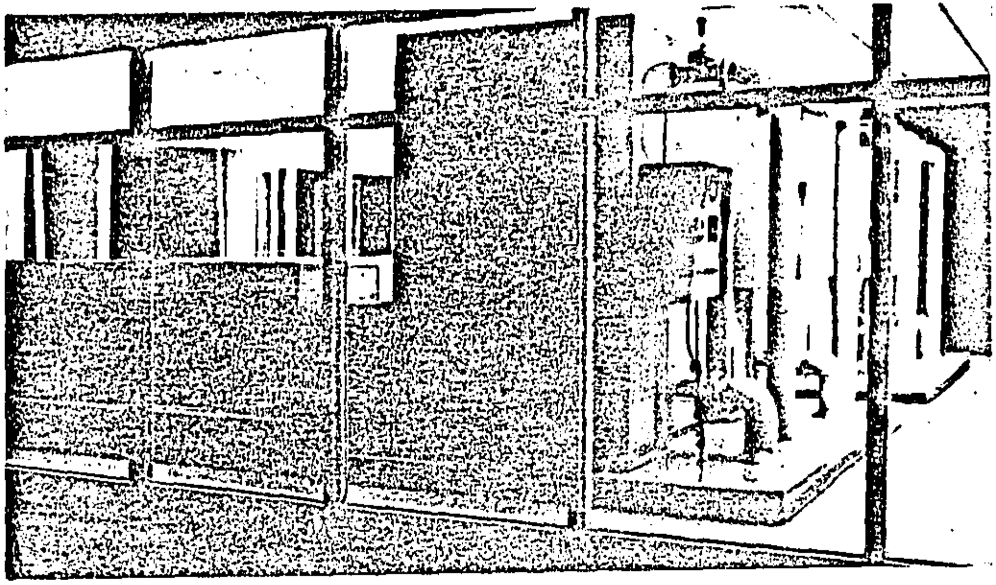


(Above) Animated flow diagram and viewing area are separated from turbine engine room noise with glass walls in total energy installation at Mountain Fuel and Supply Company's office building in Salt Lake City. *(Photograph is courtesy of AiResearch Manufacturing Division.)* (Below) Vertical boxes at rear of turbine engine cabinets are exhaust-heat water heaters which ultimately heat and cool building. Waste-heat recovery units usually provide significant amount of reduction of engine exhaust noise. *(Photograph is courtesy of AiResearch Manufacturing Division.)*



OUTLINE OF MANUAL

The introduction of the on-site total energy plant represents an important step forward in the continuing program to develop, improve and simplify the uses of natural gas energy in buildings. Other literature on the subject treats the development and uses, economics and growth, and engineering and operation of total energy systems. This manual is devoted to the acoustics—the noise and vibration—of these installations.

In a few of the first installations, enough concern was not given to their noise characteristics, and those installations have been referred to often since then by opponents of total energy. In this manual, recommendations are given for the control of the noise and vibration of these plants by proper architectural, engineering and acoustical design.

The designs cover three typical equipment locations: (1) on a basement or on-grade slab within a building, where there are no acoustically critical occupied areas beneath the engine; (2) at an upper floor location in a building, where there may be acoustically critical occupied areas above, below or beside the engine installation; and (3) in any location where concern may have to be given to the presence of neighbors near the building that houses the equipment.

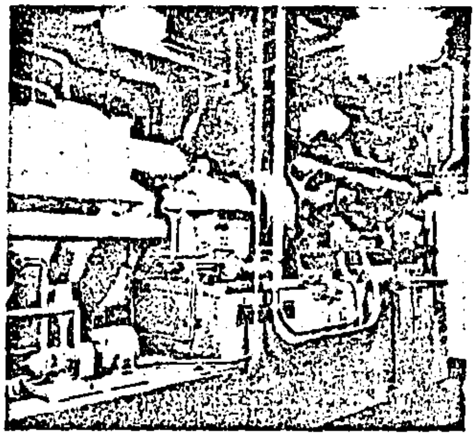
The driving mechanism for these installations may be either a reciprocating or turbine engine, and it may be fueled by either natural gas or liquid fuel. The driven mechanisms for these installations may be electric generators, refrigeration compressors (either reciprocating or centrifugal), liquid- or gas-transmitting pumps or compressors, or gears that in turn drive some of these devices. In most cases, noise control recommendations given here for the engine will take care of the noise and vibration of the driven mechanism as well.

Because of the fairly general, semi-technical coverage of this manual, many details and qualifications are omitted, and the resulting procedures may not completely solve some of the more complex noise problems associated with these power plants. However, the guidelines offered by the manual will aid in solving most of the typical noise problems that may arise in an ordinary installation.

There are no simple standard solutions to all engine noise problems; that is the reason for this manual. Although at first reading this manual may seem unduly complicated, actual use of the manual is abstracted into the filling-in of several blanks on several special data sheets that lead step-by-step to very specific design decisions. In this way, small engines and large engines, one engine or many engines, a small engine room or a large engine room, a relatively quiet or a relatively noisy adjoining area, a nearby or a far-off neighbor, a conservative or a compromise design, a big problem or a little problem, can be solved with almost equal simplicity and versatility. Simple examples along the way help illustrate the step-by-step procedures.

Because the manual is intended to solve problems rather than generate problems, the basic design procedure is aimed at providing a "conservative design" (i.e., one with reasonable assurance of achieving a satisfactory solution) but comments are offered on making some compromise decisions where they may be desired.

In the step-by-step development of the manual procedures, Section 1 outlines the basic noise problem due to engines, describing simply such necessary terms as "sound pressure level" and "sound power level" and presenting a schedule of "noise criteria" used to describe the acoustic environment in most typical living and working areas. Section 2 pertains to the vibration produced by engines and distinguishes between "feelable vibration" and "audible vibration" or "structure-borne noise." Section 3 provides quantitative data on the three principal noise sources of reciprocating engines: the casing, the exhaust and the air intake. By means of special data sheets, an example of each source is illustrated. Section 4 provides data and examples of the three principal noise sources of turbine engines. Also, the casing, the exhaust and the air intake.



Engine, gear and compressor are installed on massive concrete inertia block, which is then supported by steel springs off the floor. This type of installation is required where large reciprocating engines are located on upper floors above acoustically critical areas.

Section 5 provides a brief introduction into some basic acoustics for indoor situations, including how to estimate sound pressure levels in the engine room, then how to determine how much sound gets into a room adjoining the engine room, either on the same floor or on the floor above or below. This involves the terms "transmission loss" and "noise reduction" of a wall or a floor-ceiling combination.

Several tables of "transmission loss" data are included because the walls and floors become basic design concerns of the architect and mechanical engineer. Certain restrictions on large engines in upper-floor installations are given at the end of Section 5 under the heading "Special Situations."

Section 6 pertains to sound transmission out-of-doors and includes the noise reduction effects of distance, barriers and dense woods. Also considered are the directivity of a large exhaust opening and the noise reduction of mufflers for both reciprocating- and turbine-type engines. Section 6 also includes noise reduction involved in indoor noise escaping outdoors and outdoor noise coming indoors.

Section 7 discusses tolerances in airborne noise analysis procedure and offers a schedule for identifying a design as "preferred," "acceptable," "marginal" or "unacceptable." Sections 8 and 9 offer vibration isolation recommendations for reciprocating and turbine engines, respectively. Section 10 serves to summarize noise sources and paths, and emphasizes the need to treat each one in order to achieve a successful installation.

At the end of the text, 39 tables of data provide quantitative material on many aspects of noise control believed to be required for engine noise analysis and solution. Finally, nine data sheets provide step-by-step procedures for carrying out most of the necessary calculations in noise analysis. Copies of these blank data sheets may be used to work out design details of each job.

Section 1

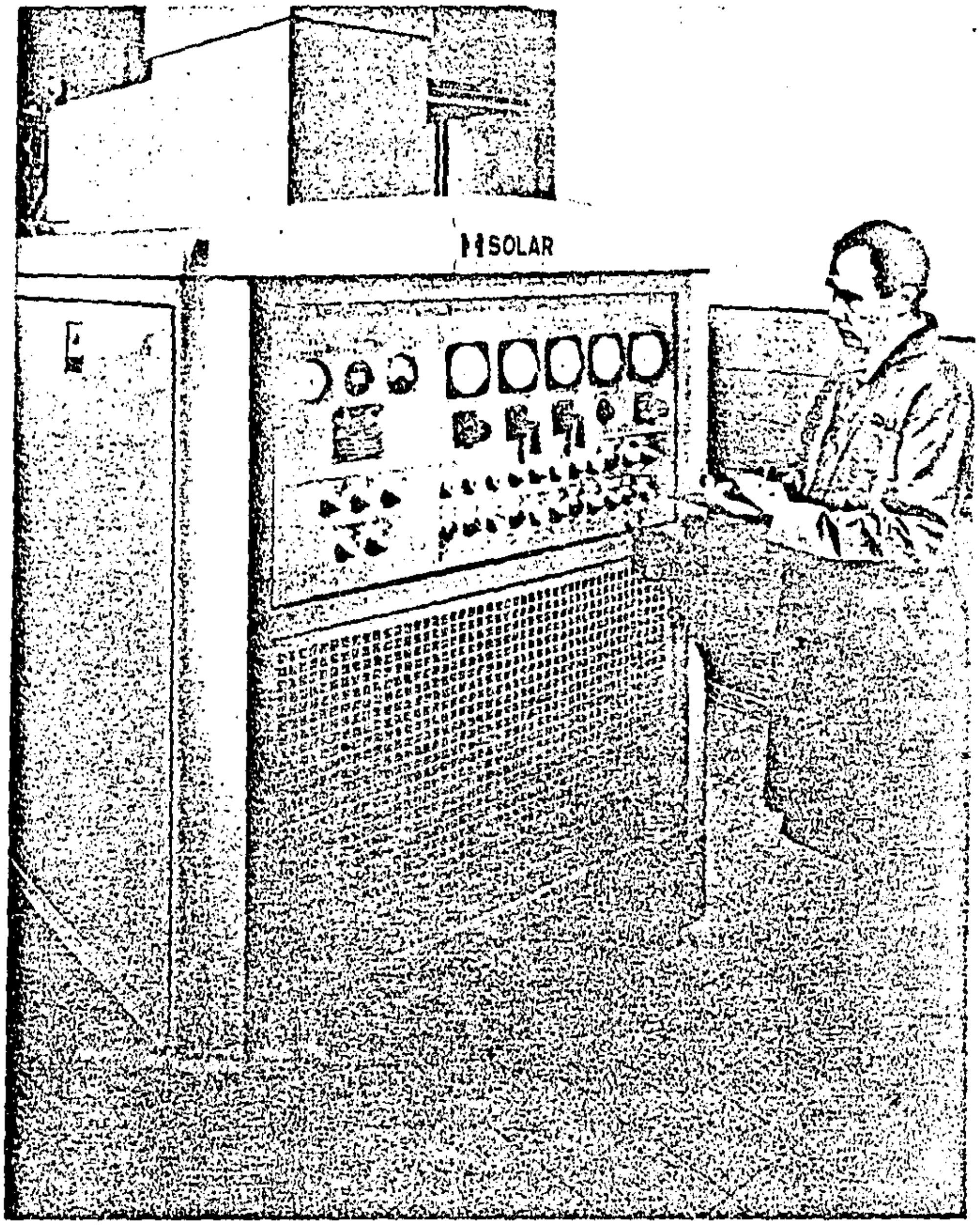
THE NOISE PROBLEM

It is true that reciprocating and turbine engines may be noisy if nothing is done to control their noise. A bare engine running in a non-acoustically treated engine room could be noisy enough to require that ear protectors be worn by operators in the same room; the noise could be disturbing to nearby occupants in the same building, if the engine room does not adequately confine the noise; and escaping engine noise or engine exhaust noise could even be a source of annoyance to nearby residential neighbors during the quiet of the night.

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 quiet suburban homes would not tolerate very much intruding engine noise, while office workers in a busy mid-city office could have greater amounts of noise without even noticing it; and, finally, factory workers in a continuously noise manufacturing space might not even hear a noisy nearby engine installation.

1.1 Noise Criteria

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 discus-



Hundreds of standby generator sets are located inside telephone company buildings. An adequate generator set enclosure, vibration isolation, and suitable mufflers permit installation in virtually any part of building. *(Photograph is courtesy of Solar Division, International Harvester Company.)*

sions of noise criteria can be found in other literature[1], and only a 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.

In Table 1, a number of typical indoor living and working spaces are grouped together into "categories." A low category number indicates areas in which relatively low noise levels are desired; higher category numbers indicate areas in which relatively higher noise levels are permissible.

The various indoor areas listed in Table 1 are intended to be illustrative; areas that are not specifically named here can be assigned, from an acoustic point-of-view, to the various categories on the basis of their similarity with the spaces that are named. From time-to-time in the manual, these areas may be referred to as the "noise-sensitive area," the "acoustically critical area" or some other similar term.

Table 1 also includes a "Noise Criterion Designation" for each of these areas. For example, Category 1 areas include "bedrooms, sleeping quarters, hospitals, residences, apartments, hotels, motels, etc. (for sleeping, resting, relaxing)." This group of activity areas is then given the noise criterion designation of NC-20 to NC-30.

1.2 Sound Pressure Level ("SPL")

The ear is sensitive to tiny oscillations of pressure in the air. Sound waves produce these pressure oscillations. The intensity of sound waves may be described quantitatively by the term "sound pressure level" (abbreviated "SPL" in the manual and popularly called sometimes "noise level"). Sound pressure levels used throughout the manual are given in decibels ("dB") relative to the standard reference pressure of 0.0002 microbar, although the reference pressure is often omitted.

Human hearing covers a full frequency range of from about 20 Hz (cps) to 20,000 Hz (cps)[2]. In acoustical engineering applications it is normal practice to divide this frequency range into eight "octaves" (as in "octaves" of a piano scale) and to work with the "octave frequency bands" of noise. The octave frequency bands that are in common usage now are identified by their "center frequencies" as follows: 63 Hz, 125, 250, 500, 1,000, 2,000, 4,000, and 8,000 Hz. Actually, each of these frequencies is the "geometric mean frequency," and the actual bandwidth extends from 0.7 of the center frequency to 1.4 times the center frequency. An octave band centered at 31 cps is sometimes used, but it is not included here.

In Table 2, actual sound pressure levels (SPL) are listed for the eight octave frequency bands for each of the noise criterion (NC) designations used in Table 1. In later applications within the manual, it will be the goal of a particular "noise control design" to meet or not exceed the noise levels of the appropriate NC designation for a given critical area. In applying the indoor NC designations of Table 1, the lower limit of the range for a given category (for example, NC-20 for Category 1) should be used for a critical situation, while the upper limit (or some intermediate value) may be used for a situation known to be not so critical.

A special note of concern is given for the Category 1 and 2 areas of Table 1. 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

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[1] 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), "Handbook of Noise Control," C.M. Harris, Editor, McGraw-Hill Book Company (1954), the latest issue of the ASHRAE "Guide and Data Book," American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 345 East 47th Street, New York, N.Y. 10017 or selected topics of the Journal of the Acoustical Society of America.

[2] The recently accepted U.S. and international standard unit of frequency is the "Hertz," abbreviated "Hz." Thus, Hertz has the same meaning and value as the traditional and familiar term "cycles per second" or "cps." The new unit Hz is used in this manual.

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.

1.3 Sound Power Level ("PWL")

The sound pressure level in a room is a function of (1) the "acoustics" of the room and (2) the strength of the sound source. The strength of the source may be described quantitatively as the "sound power" radiated by the source, and, when used in the proper units, this becomes "sound power level" (abbreviated "PWL" in this manual), expressed in dB relative to the reference power of 10^{-12} watt[3].

In Sections 3 and 4 of the manual, sound power level data are given for reciprocating and turbine engines, and engine components. As a simple analogy, "sound power" of a source might be likened to the power rating for an electric lamp (e.g., "100-watt" bulb), while "sound pressure" might be likened to the illumination in a room provided by that electric lamp. This analogy also helps explain the role of acoustic absorption in a room.

In a room with white-painted walls and ceiling, a weak light source will give fairly good general illumination, but if the room had only dark, unreflecting surfaces, the general illumination would be very poor. Similarly, a sound *power* source may produce high or low sound *pressure* levels in a room depending on the degree of absorption or reflection of the surfaces inside the room. The acoustics of the room are discussed more quantitatively in Section 5.

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[3] For a more technical description of PWL and SPL refer to an acoustics textbook.

Section 2

THE VIBRATION PROBLEM

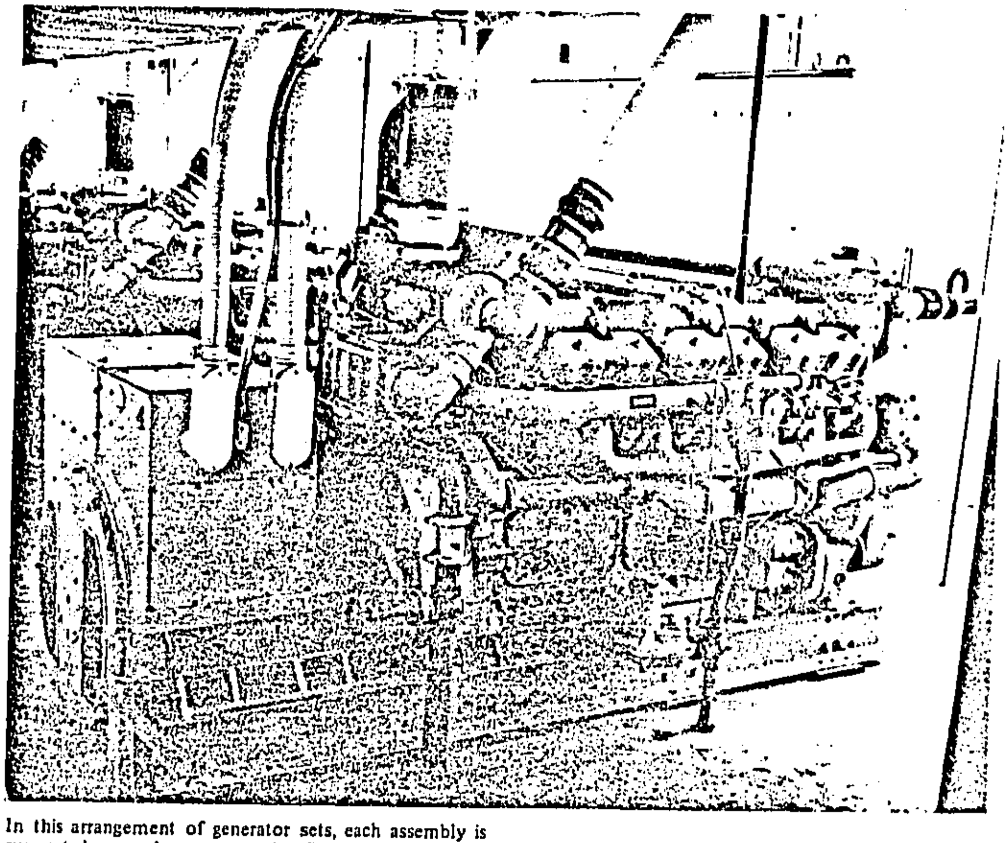
The vibration produced in a building by an operating machine can manifest itself in two ways to a person in that building: (1) by being "feelable" and (2) by being "audible." Building structures and the earth are good transmitters of vibrational energy, and if the transmitted vibration is intense enough, it will be "feelable" as vibration. If the vibration is not intense enough to be "feelable," it may still be "audible."

In a very quiet environment, a large vibrating surface can radiate "audible" sound when its vibration level is only 1/100th to 1/1,000th of that required to be feelable. In fact, a wall that vibrates with an *unfeelable* amplitude of approximately one millionth of an inch can still be *heard* to radiate sound in a very quiet room.

Thus, it is not enough just to prevent an engine from producing feelable vibration in a building; it is usually necessary to reduce building vibration to a level much lower than this in order that building occupants will not hear the noise radiated by the vibrating walls, floors, ceilings, columns, beams, doors, windows, etc. It is this type of vibration that is known as "structure-borne noise."

When designing or specifying a vibration isolation mounting assembly for an engine, the acoustic objective of that design is to reduce the "structure-borne noise" so that it will be considered acceptable in those various categories of spaces given in Table 1. This becomes a particularly difficult requirement when a reciprocating engine is mounted on the upper floor of a building, because the low-speed, unbalanced forces of the reciprocating engine may "drive" the building elements (walls, beams, floors, etc.) at or near their natural resonance frequencies and produce audible or feelable vibration at remote parts of the building away from the engine.

In some special-purpose buildings there may be some vibration-sensitive equipment, instruments or processes that are more sensitive than people to the background vibration in the building. Some medical, magnetic, electric and electronic equipment may be subject to this low-level vibration pickup. It is usually necessary to provide special isolation mountings on this type of equipment so that it will not be sensitive to normal building vibration, whether caused by mechanical equipment or by occupants of the building (walking, closing doors, typing, etc.). In this manual, engine mountings will aim at achieving acceptable vibration and structure-borne noise levels in the building to meet the requirements of human occupants. Special mountings to meet the needs of vibration-sensitive equipment are beyond the scope of the manual.



In this arrangement of generator sets, each assembly is supported on pad mounts at the floor and includes flexible connections to overhead air intake duct, exhaust mufflers and power outlet. *(Photograph is courtesy of Caterpillar Tractor Company.)*

Section 3

RECIPROCATING ENGINE NOISE

For the preparation of this manual, noise data were obtained and studied for over 75 reciprocating natural gas and diesel-fuel engines covering a power range of 10 to 6,000 hp. For the purposes of the manual, interest is limited to reciprocating engines in the power range of 20 to 2,500 hp.

Measured noise data have been studied extensively to determine approximately the influence of certain operational variables on the amount of noise generated by these engines. From this study it now appears possible to predict fairly accurately the amount of noise generated by any reciprocating engine fueled by natural gas or liquid fuel. In this manual, technical details of the study and the results are not given, but useful conclusions are summarized and used[1]. For reciprocating engines, there are three principal noise sources of concern: (1) the engine casing; (2) the engine exhaust; and (3) the air intake to the engine. These noise sources are now considered one at a time.

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[1] Two concurrent but independent projects, carried out by Bolt Beranek and Newman Inc. during 1966-1967, provided the noise data used in this study. The first of these projects was sponsored by the Office of the Chief of Engineers of the Department of the Army. One part of that project was the preparation of the engineering manual "Power Plant Acoustics: Noise Control for Diesel, Gas and Gas Turbine Engine Installations" by Laymon N. Miller, September 1967. That manual summarizes noise data in technical detail. The second project has been sponsored by the American Gas Association and has led to the preparation of this manual. This manual minimizes the technical detail of the noise data but attempts to use the data for practical engineering and architectural purposes.

Data Sheet 1 *(Paragraph 3.2 Example)*

Estimated Sound Power Level (PWL) of Reciprocating Engine Casing Noise

1. Continuous Rating of Engine: 200 hp or ______ kw

2. Engine Speed: 1,200 rpm

3. Fuel: Gas Only ☑ Liquid Only ☐ Gas and Liquid ☐

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

4. Base PWL from Table 3 for Item 1 Rating:

105	109	109	108	108	107	101	94

5. Speed Correction from Table 3 for Item 2 Speed:

-2	-2	-2	-2	-2	-2	-2	-2

6. Fuel Correction from Table 3 for Item 3 Fuel:

-3	-3	-3	-3	-3	-3	-3	-3

7. Estimated PWL (in dB re 10^{-12} watt) of Casing Noise: (Item 7 = Item 4 + Item 5 + Item 6)
Caution: Observe algebraic signs in combining items!

100	104	104	103	103	102	96	89

3.1 Engine Casing Noise

In terms of a rather simplified approach, the noise radiated by the engine casing is related to the following influencing factors: (1) continuous rating of the engine in hp (horsepower)[2], (2) shaft speed of engine, and (3) type of fuel used by the engine. Table 3, at the end of the text, summarizes the procedure for estimating the sound power level (PWL) of an engine casing for a particular rating, speed and fuel combination. This does not include the noise of the exhaust from the engine nor the noise of the exhaust from the engine or of the larger engines that have turbochargers. As seen in Table 3, the "Estimated PWL" is made up of a "Base PWL" plus two correction terms. The Base PWL values differ for the eight octave frequency bands, but the correction terms apply equally to all bands.

3.2 Example, Data Sheet 1

As an example of the use of Table 3, suppose that it is desired to know the approximate PWL of a 200-hp, 1,200-rpm natural gas engine used to drive an electric generator. Sample Data Sheet 1 indicates the steps taken to make the calculation. For this example, each blank of Data Sheet 1 is filled in with the appropriate material from Table 3. (A blank copy of each data sheet used in the manual is given at the end of the text. Reproduced copies of these data sheets may be used for specific design problems.)

3.3 Engine Exhaust Noise

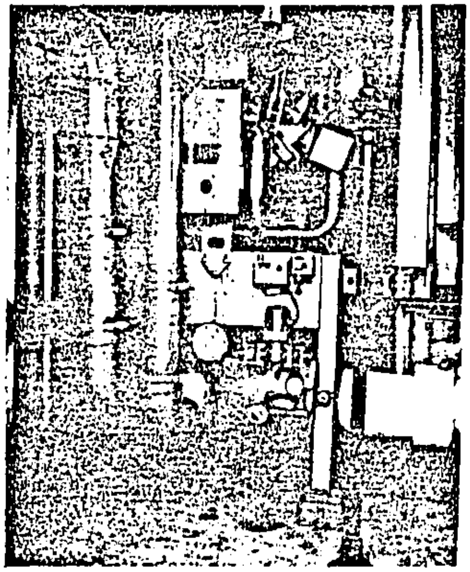
The noise generated by the unmuffled engine exhaust and radiated from the open end of the exhaust pipe is related to the following influencing factors: (1) continuous rating of the engine in hp (horsepower) or kw (kilowatts); (2) the presence or absence of an air intake turbocharger driven by the exhaust gases; and (3) the length of the exhaust pipe leading exhaust gases away from the engine. Table 4 summarizes the data used for estimating the PWL of the exhaust noise. When a turbocharger is driven by the released exhaust gases, some energy is removed from the exhaust, and less noise is produced. A nominal reduction of 6 dB is shown in Table 4 for this effect.

A long exhaust pipe also helps reduce the escaping exhaust noise, approximately as shown by the correction term in Table 4. It is not suggested here, however, that a long exhaust pipe may take the place of a muffler as a noise control device.

It is cautioned that Table 4 data may not accurately predict the low frequency exhaust noise of some of the large low-speed engines. In the cases of some of these engines, the muffler manufacturers design special mufflers to control the exhaust noise.

3.4 Example, Data Sheet 2

Suppose that the 200-hp engine of Example 3.2 is fitted with a turbocharger and a



Ready Power chiller assembly includes flexible connections in piping and vibration isolation mounts at floor. When enclosed springs are used, as here, check carefully that mounting system is properly aligned and that springs are not binding or shorted out. *(Photograph is courtesy of The Ready Power Company.)*

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[2] To convert hp (horsepower) output to kw (kilowatts) produced, or vice versa, use the relationship:

$$kw = hp/1.5$$

Data Sheet 2 *(Paragraph 3.4 Example)*

Estimated Sound Power Level (PWL) of Unmuffled Reciprocating Engine Exhaust Noise

1. Continuous Rating of Engine: 200 hp or ______ kw

2. Air Intake: Turbocharger ☒, No Turbocharger ☐

3. Exhaust Pipe Length: 20 ft.

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

4. Base PWL from Table 4 for Item 1 Rating:

132	138	134	126	122	116	106	98

5. Turbocharger Correction from Table 4 for Item 2 Air Intake:

-6	-6	-6	-6	-6	-6	-6	-6

6. Exhaust Pipe Length Correction from Table 4 for Item 3 Length:

-5	-5	-5	-5	-5	-5	-5	-5

7. Estimated PWL (in dB re 10^{-12} watt) of Unmuffled Exhaust Noise: (Item 7 = Item 4 + Item 5 + Item 6)
Caution: Observe algebraic signs in combining items!

121	127	123	115	111	105	95	87

Estimated Sound Power Level (PWL) of Untreated Turbocharger Noise at Air Inlet Opening of Reciprocating Engine

1. Continuous Rating of Engine: 2,000 hp or ______ kw

2. Inlet Air Duct Length: 30 ft.

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

3. Base PWL from Table 5 for Item 1 Rating:

100	98	98	99	102	103	102	94

4. Inlet Air Duct Length Correction from Table 5 for Item 2 Length:

-5	-5	-5	-5	-5	-5	-5	-5

5. Estimated PWL (in dB re 10^{-12} watt) of Untreated Turbocharger Noise: (Item 5 = Item 3 + Item 4)
Caution: Observe algebraic signs in combining items!

95	93	93	94	97	98	97	89

20-foot-long exhaust pipe leading to the roof of the engine room. Sample Data Sheet 2 summarizes the steps involved in calculating the PWL of the unmuffled exhaust using data from Table 4.

3.5 Engine Air-Intake Noise

Some of the smaller engines covered by this manual do not have turbochargers; instead, the intake air is drawn into the engine by "natural aspiration" (designated by "NA" in some engine catalogs). On the other hand, many of the smaller- and medium-sized engines are provided with turbochargers (a turbine-type device, driven at high speed by the released exhaust gases, that increases the flow and pressure of air into the engine). These smaller turbochargers are usually installed directly in the short air-intake duct to the engine just beyond an air-intake cleaner or filter.

The high frequency whine of the turbocharger is usually masked by the total noise of the engine, although it can be heard up close to the air cleaner opening. The noise of these turbochargers may be assumed to be contained within the total noise of the engine, and no special noise control treatment will be required—other than a caution against listening with the unprotected ear directly at the air cleaner inlet.

For many of the larger and slower engines (say, over 1,000 hp, although there is no clear size distinction), the manufacturer may specify a ducted connection to bring room air or outside air into the engine intake and turbocharger. In this event, the turbocharger noise radiated into the room or out-of-doors may be of concern. A rough estimate of the sound power level of the escaping turbocharger noise is given in Table 5. It is cautioned here also that this may not hold true for all engines and turbochargers. It is important, however, to recognize that turbocharger noise should be considered and that Table 5 data will provide a reasonable approximation of this noise.

3.6 Example, Data Sheet 3

As a sample calculation using Table 5, suppose that a turbocharged 2,000-hp engine has a 30-foot ducted air inlet from outside the engine room. Sample Data Sheet 3 is used to carry out the steps of this calculation. These values are not claimed to be highly accurate because different turbochargers make different amounts of noise, depending on design variables. And the inlet duct may reduce the escaping noise by different amounts than that given in Table 5. Nevertheless, the estimate may serve as a guideline for potential noise.

3.7 Vibration Data

Vibration data have been taken on several of the reciprocating engines studied in this program; however, because of the many variables in engines, mounting arrangements and mounting locations and because vibration data are not readily understood or appreciated by most non-acousticians, summaries of the data are not given here. In Section 8 of the manual, vibration isolation recommendations for various types of reciprocating engine installations are given. The most critical installations, of course, are those located on upper floors of buildings and directly over quiet occupied spaces.

Section 4

TURBINE ENGINE NOISE

For the preparation of this manual, noise data were obtained and studied for over 45 turbine engines covering a power range of 200 to 20,000 kw. For the purposes of the manual, interest is limited to the power range of 200 to 5,000 kw.

300 - 7,500 hp

From the study of turbine engine noise data, it is possible to estimate for engineering purposes the noise from three principal sources: (1) engine casing, (2) engine exhaust, and (3) air intake to the engine. These three noise sources are now considered one at a time.

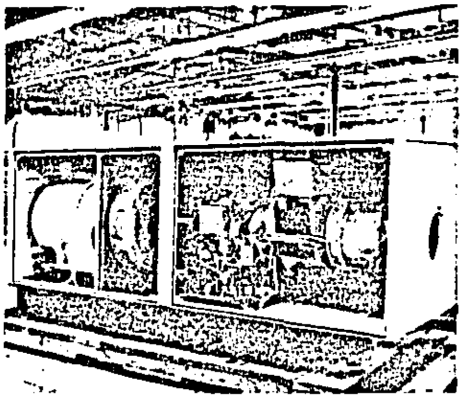
4.1 Engine Casing Noise

The approximate PWL values for the casing noise of unenclosed, untreated gas turbine engines are given in Table 6[1]. These values apply to a bare engine having no thermal insulating cover or form of enclosing cabinet.

If there is some form of cover for the entire engine casing, the radiated noise will probably be reduced, depending on the acoustic quality of the cover. The footnote of Table 6 lists a few typical covers that may be provided with some engines and suggests an approximate noise correction that might be attributed to each general type of cover.

4.2 Engine Exhaust Noise

The approximate PWL values for the unmuffled exhaust noise from gas turbine en-



AiResearch turbine engine-generator set is packaged inside cabinet for shipment and later installation at computer center. Exhaust heat from engines is used to operate absorption refrigeration machine which air conditions computer installation. *(Photograph is courtesy of Garrett-AiResearch Manufacturing Division.)*

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[1] Turbine engine ratings are usually quoted in kilowatt or megawatt values. To convert kilowatts (kw) output and horsepower (hp) produced, use the relationship:

$kw = hp/1.5$

gines are given in Table 7. These values apply to the noise radiated directly from the end of the engine, assuming no exhaust duct. Actually, most turbine engine installations will have connecting ducts from the engine exhaust to some outdoor opening, and these ducts may be counted on for some degree of silencing, depending on the nature of the acoustic or thermal lining of the duct.

The approximate attenuation provided by straight ducts and duct turns for a typical engine exhaust duct may be estimated from data in Tables 9 and 10 or from data in the "ASHRAE Guide"[2]. Attention is directed to the qualifying comments in Tables 9 and 10. It is especially important to realize that, at elevated exhaust temperatures, the wavelength of sound in the exhaust gas is longer than at ordinary temperature, and, as a result, the duct appears shorter, in terms of wavelengths, to the sound wave. Thus, the attenuation will be less. As a first approximation, it is suggested that two-thirds of the normal attenuation be used when hot exhaust ducts are being considered. It is hardly necessary to be more exacting here for estimates of duct losses. It is more important later to take this temperature effect into account when considering mufflers for controlling the noise.

If the turbine engine manufacturer supplies with the engine an exhaust muffler of known insertion loss, the PWL values of Table 7 may be reduced by the muffler insertion loss values for each of the frequency bands (corrected for exhaust temperature). Provision for this is made in the appropriate data sheet.

4.3 Engine Air Intake Noise

The approximate PWL values for the unsilenced air-intake noise of gas turbine engines are given in Table 8. These values apply to the noise radiated directly from the air intake end of the engine. If an inlet duct brings air into the engine, the approximate sound attenuation provided by the duct can be estimated from Tables 9 and 10 or from the "ASHRAE Guide." Please note the qualifications associated with data of Tables 9 and 10.

If the engine manufacturer supplies an air-intake silencer of known insertion loss, the PWL values of Table 8 may be reduced by the silencer insertion loss values for each of the frequency bands.

4.4 Example, Data Sheet 4

Suppose a 2,500-kw turbine-driven generator set is to be installed in a building. As supplied by the manufacturer, the packaged set is housed in a metal cabinet with inside acoustic lining but with many open holes for ventilation. The circular exhaust duct is 30 feet long and extends to the roof of the building. This duct is wrapped on the outside with thermal insulation. The exhaust duct also contains a packaged exhaust muffler which has the following insertion loss, corrected for exhaust temperature, according to the muffler manufacturer:

Hz	63	125	250	500	1,000	2,000	4,000	8,000
PWL, dB	5	10	13	15	19	15	12	10

The air intake to the engine is provided by a 12-foot-long rectangular duct to the outside wall of the building. A 2-inch-thick internal acoustic lining is offered by the manufacturer for this intake duct. It is desired to know the PWL for the casing, exhaust and intake for this unit.

Sample Data Sheet 4 summarizes the steps for arriving at the desired data, making use of the material from Tables 6-10. The desired PWL values for the three principal noise sources are given in Items 9, 13 and 17 of Data Sheet 4 after the appropriate corrections are made for the various noise reduction treatments supplied for this particular installation.

4.5 Vibration Data

Because of the rotary action of the turbine engine, vibration is much lower than for reciprocating engines. However, structure-borne noise may still be of concern in some installations, and suggestions for vibration isolation are given in Section 9 of the manual.

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[2] Refer to the latest available "ASHRAE Guide and Data Book," published by the American Society of Heating, Refrigeration and Air-Conditioning Engineers, 345 East 47th Street, New York, New York 10017.

(Paragraph 4.4 Example)

Data Sheet 4

Sheet 1 of 2

Estimated Sound Power Level (PWL) of Casing, Exhaust and Intake Noise of Gas Turbine Engine

1. Continuous Rating of Engine: ______ hp or 2,500 kw

2. Engine Casing Cover: None ☐ or Type 4 from Table 6

3. Exhaust Duct: Round ☒, Rectangular ☐, Length 30 ft.

Duct Lining: Type 2 from Table 9

Duct Turns: No. 0 Type ______ from Table 10

4. Intake Duct: Round ☐, Rectangular ☒, Length 12 ft.

Duct Lining: Type 4 from Table 9

Duct Turns: No. 0 Type ______ from Table 10

5. Exhaust Muffler Supplied: Yes ☒ No ☐

6. Intake Muffler Supplied: Yes ☐ No ☒

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

7. PWL (in dB re 10^{-12} watt) of Casing Noise from Table 6 for Item 1 Rating:

116	118	119	119	119	119	119	119

8. Noise Reduction Provided by Casing Cover (If Any) of Item 2 as Estimated in Table 6 Footnote:

-4	-4	-5	-6	-7	-8	-8	-8

9. PWL of Casing Noise with Cover (If Any) Item 9 = Item 7 + Item 8
Caution: Observe algebraic signs in combining items!

112	114	114	113	112	111	111	111

(Continued)

(Paragraph 4.4 Example)

Data Sheet 4 (continued)

Sheet 2 of 2

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

10. PWL (in dB re 10^{-12} watt) of Exhaust Noise from Table 7 for Item 1 Rating:

130	132	132	131	129	127	123	117

11. Attenuation Provided by Exhaust Duct and Turns of Item 3, from Tables 9 and 10 or "ASHRAE Guide" (include hot temperature correction):

-3	-2	-1	-0	-0	-0	-0	-0

12. Insertion Loss of Exhaust Muffler or Item 5, If Provided (use muffler manufacturer's data for appropriate bands, corrected for exhaust temperature):

-5	-10	-13	-15	-19	-15	-12	-10

13. PWL of Exhaust Noise Out of Duct and Muffler (as applicable): Item 13 = Item 10 + Item 11 + Item 12
Caution: Observe algebraic signs in combining Items!

122	120	118	116	110	112	111	107

14. PWL (in dB re 10^{-12} watt) of Air-Intake Noise from Table 8 for Item 1 Rating:

117	118	118	121	127	132	132	129

15. Attenuation Provided by Intake Duct and Turns of Item 4, from Tables 9 and 10 or "ASHRAE Guide":

-4	-3	-5	-11	-13	-13	-11	-7

16. Insertion Loss of Intake Muffler of Item 6, If Provided (use muffler manufacturer's data for appropriate bands; see Table 39):

17. PWL of Intake Noise Out of Duct and Muffler (as Applicable)
Item 17 = Item 14 + Item 15 + Item 16
Caution: Observe algebraic signs in combining Items!

113	115	113	110	114	119	121	122

Section 5

CONTROL OF AIRBORNE SOUND INDOORS

Principal concern with the noise of engine-generator sets is the airborne sound radiated directly from the noise-making components. As the term "airborne" suggests, this sound is carried essentially by air paths from the source to the receiver. In some cases the noise may strike a massive wall and even be transmitted "through" the wall to the opposite side where it continues to be radiated and considered as airborne noise.

The control of airborne sound involves: (1) some fundamentals about the distribution of sound inside rooms and outdoors, (2) some quantitative data on the ability of a wall to reduce sound transmission, and (3) some quantitative data on the effectiveness of sound-attenuating mufflers placed in sound passages. Indoor aspects of airborne sound are discussed in this section, while outdoor aspects are considered in Section 6.

5.1 Sound Pressure Level in a Room

In Paragraph 1.3 it was mentioned that the sound pressure level (SPL or noise level) in a room is a function of (1) the acoustics of the room, and (2) the strength of the source. In Sections 3 and 4, data have been given on the "strength of the source," namely, the sound power level (PWL) of the major noise sources of reciprocating and turbine engines. Now, in this part of Section 5, the acoustics of a room are considered.

In a brief and simplified procedure, the acoustics of a room may be described by the volume of the room and the relative amount of "acoustic absorption" or "acoustic treatment" in the room. Then, the SPL (sound pressure level) in the room can be estimated for any PWL (sound power level) in the room by applying a "room correction term" that takes into account the volume and acoustic absorption of the room. Thus,

$$SPL = PWL + \text{Room Correction Term}$$

Table 11 gives average values of the "room correction term" that will yield the "reverberant" noise level in a room, that is, the noise level that may be found to exist fairly uniformly all around the room at a distance of more than about 5 to 10 feet from the source for smaller rooms and more than about 10 to 20 feet from the source for larger rooms. At closer distances to the noise source, the noise will be higher than this "reverberant" noise level by a few decibels, depending on a number of details which will not be discussed here. As a rough approximation of the SPL within a few feet of an engine, the reverberant SPL can be increased by 3 dB. The room correction term actually varies with frequency but that variation is ignored here and the values given in Table 11 may be applied equally to all octave frequency bands. For a more rigorous and detailed analysis of a room acoustics problem, the reader is referred to a textbook on acoustics.

The footnotes of Table 11 describe four different degrees of room treatments that might be used. Various areas and thicknesses of acoustic absorption material are suggested. Where a 3/4 inch to 1 inch thick absorption material is used, it is intended that such a material have a published "NRC" value ("noise reduction coefficient") in the range of 0.65 to 0.75. Where a 1-1/2 inch to 2 inch thick absorption material is used it is intended that such a material have a published NRC value in the range of 0.75 to 0.85. Bulletins on acoustic absorption materials list all of the marketed products that meet these values[1].

5.2 Example, Data Sheet 5

As an example of the use of Table 11, suppose that the 200-hp, 1,200-rpm natural gas engine of Example 3.2 is to drive an electric generator. Suppose the engine room is 12 feet high, 15 feet wide and 20 feet long. On the underside of the roof there is a 2-inch-thick layer of exposed glass fiber insulation that serves both as thermal insulation and as acoustic absorption. It is desired to know the estimated "noise levels" (sound pressure levels) in the room when the engine is running.

To aid in the estimating procedure, Data Sheet 5 may be used. This sheet indicates each of the steps necessary to determine the estimated reverberant SPL in the engine room. For this example, each blank of Data Sheet 5 is filled in with the appropriate material. The Item 8 PWL values are taken from sample Data Sheet 1 used with Example 3.2. Item 9 gives the reverberant SPL in the engine room, and Item 10 gives the approximate SPL near an engine

5.3 Addition of Decibel Levels

When several engines or noise sources are present in a given space, the total sound power level or the total noise level will usually be greater than that for any single engine or noise source in the space. It is necessary to take this into account in any particular noise control design, especially since there frequently are several engines operating in the same room. Since PWL and SPL values are logarithmic-based numbers, they cannot be added together by simple algebraic addition. For example, 90 dB + 90 dB does *not* give 180 dB, instead, as in electrical power addition of decibels, 90 dB + 90 dB = 93 dB, i.e., two equal sources added together produce a total level that is 3 dB greater than the level of either source. Table 12 summarizes four simple rules to be followed for adding SPL or PWL contributions to obtain the total SPL or PWL.

As illustrations of the use of Table 12, according to the first rule,

$$\begin{aligned}95\,dB + 97\,dB &= 99\,dB\\ 90\,dB + 97\,dB &= 98\,dB\\ 85\,dB + 95\,dB &= 95\,dB\end{aligned}$$

According to the second rule,

$$\begin{aligned}80\,dB + 80\,dB &= 83\,dB\\ 95\,dB + 95\,dB + 95\,dB &= 100\,dB.\end{aligned}$$

To illustrate the third rule, the following four sample SPL values are added by using two different orders of addition:

First order,

$$\left.\begin{array}{l}\left.\begin{array}{r}88\,dB\\ 90\,dB\end{array}\right\} = 92\,dB\\ \left.\begin{array}{r}93\,dB\\ 98\,dB\end{array}\right\} = 99\,dB\end{array}\right\} = 100\,dB$$

Second order,

$$\left.\begin{array}{l}\left.\begin{array}{r}88\,dB\\ 98\,dB\end{array}\right\} = 98\,dB\\ \left.\begin{array}{r}90\,dB\\ 93\,dB\end{array}\right\} = 95\,dB\end{array}\right\} = 100\,dB$$

5.4 Transmission Loss of Walls ("TL")

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. In vibrating,

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[1] Refer to manufacturers' published data or see latest applicable annual bulletin of "Performance Data of Architectural Acoustics Materials" or its successor, published by the Acoustical Materials Association, 335 East 45th Street, New York, New York 10017, or its successor.

Data Sheet 5 *(Paragraph 5.2 Example)*

Estimated Sound Pressure Level (SPL) in an Engine Room

1. Volume of Engine Room: 12 ft. X 15 ft. X 20 ft. = 3,600 cu. ft.

2. Total Interior Surface Area of Room (including floor) = 1,440 sq. ft.

3. Area of Acoustic Treatment 300 sq. ft.

4. Per cent Area Covered by Acoustic Treatment (Item 3/Item 2) X 100 = 21 %

5. Thickness of Absorption Material 3/4 in.–1 in. ☐ 1-1/2 in.–2 in. ☒

6. Acoustic Treatment Condition of Room from Items 4 and 5, as Defined in Footnotes of Table 11:
Condition 1 ☐ Condition 2 ☐ Condition 3 ☒ Condition 4 ☐

7. Room Correction Term from Table 11 for Items 1 and 6: -8 dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

8. Total PWL of All Noise Sources in Engine Room (from Item 7 of Data Sheet 1 for Reciprocating Engines or from Item 9 of Data Sheet 4 for Turbine Engines):

100	104	104	103	103	102	96	89

9. Reverberant SPL in Engine Room (Item 9 = Item 8 + Item 7; Item 7 Value Is Same for All Frequency Bands and Is Zero or a Negative Quantity for All Room Conditions)
Caution: Observe algebraic signs in combining terms!

92	96	96	95	95	94	88	81

10. Approximate SPL Near Engine (Item 10 = Item 9 + 3 dB)

95	99	99	98	98	97	91	84

this wall sets into oscillation the air particles along its "back" or opposite surface. These vibrating air particles radiate as sound energy into the space on the back 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 back side. (If the wall is at all porous, some sound–oscillating air particles–can actually pass through the pores of the wall.)

Generally, 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. Of course, there are many qualifications to these generalities. Details of these qualifications are not presented here; but the "transmission loss" data given in the tables reflect these effects.

The approximate "transmission loss" or "TL" values, expressed in dB, of a number of typical wall constructions are given in Tables 13-22 at the end of the text:

Table No.	Construction Material
13	Solid, dense concrete or masonry
14	Hollow-core concrete or masonry
15	Stud-type partitions
16	Metal panel partition and industrial acoustic doors
17	Glass walls or windows
18	Double-glass construction
19	Wood or plywood, including 2-inch thick solid wood door
20	Plaster
21	Aluminum
22	Steel

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 opposite side of 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.

5.5 Noise Reduction of Walls ("NR")

The objective of this discussion on walls is to enable us to determine quantitatively the wall that is required to separate a noisy room from a quiet room. For example, if the noise level in an engine room is 90 dB at a particular octave frequency band and the noise level desired in an adjoining room is 40 dB in that same frequency band, then the wall must provide a "noise reduction" of 50 dB. But, the "noise reduction" provided by that wall involves more than merely the TL of the wall.

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 to an adjoining room, three factors are involved: (1) the TL of the wall (as in Tables 13-22); (2) the area of the wall that is transmitting the noise (that part of the wall area that is common to both the noisy room and the adjoining room); and (3) the acoustic characteristics of the adjoining room that receives the transmitted noise. The term "noise reduction" of a wall (abbreviated "NR") is the term that includes all three of these factors. In practice, the area of the common transmitting wall and the acoustic characteristics of the receiving room can be incorporated into a single "wall correction term"[2] which in turn can be applied to the TL of that wall. Then,

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[2] "Wall correction term" is coined especially for use in this manual. It actually includes several factors combined into one term in the interest of simplifying the details.

the "noise reduction" provided by the wall separating the two rooms is the *the SPL difference between the two rooms.*

However, the NR for any wall is equal to the TL of the wall plus the "wall correction term" (designated here as "C"), or[3]

$$NR = TL + C$$

In these two equations TL and C have known values for specific situations, and the SPL of the source room can be determined from Data Sheet 5. Then, for a given set of conditions, the SPL in the receiver room can be determined, or

$$SPL_{\substack{receiver\\room}} = SPL_{\substack{source\\room}} + TL + C$$

The approximate values of wall TL can be found in Tables 13-22, and the approximate values of "C" can be found in Table 23, where it is necessary to know the total receiving room surface area, the common wall area and the condition of acoustic treatment of the receiving room. In this simplified approach, several approximations are made, and the resulting value of "C" is used equally in all eight frequency bands.

5.6 Example, Data Sheet 6

From all the material thus far presented in the manual, it is now possible to make a very significant calculation. Suppose it is desired to know if an 8-inch hollow-core dense concrete block wall will be adequate to separate an engine room on the top floor of a building from the bedroom of a penthouse apartment immediately beside the engine room.

Suppose the engine and engine room are the same as those used in Example 5.2, where the engine room reverberant and close-in SPLs were found in Items 9 and 10 of the sample Data Sheet 5. Suppose that the bedroom wall common to the engine room is 8 feet high and 12 feet long and suppose that the bedroom has a total inside surface area of 792 square feet and a 12-foot-x-15-foot ceiling of 1-inch-thick acoustic absorption material. Also, the floor is carpeted.

Sample Data Sheet 6 indicates the steps required to estimate the SPL in the bedroom ("receiving room") due to the engine in the engine room ("transmitting room"). In filling in the blanks of Data Sheet 6, note that the footnote of Item 4 permits the addition of extra acoustic absorption to the receiving room due to the presence of carpet, drapes or upholstered furniture.

In a more thorough analysis, the room absorption could be calculated in considerable detail; but here fairly simple estimates are considered adequate. Another judgement decision might be involved in filling in the blanks of Item 12 of Data Sheet 6. If the transmitting wall is quite close to the engine noise source, the noise levels at that wall will be higher than if the wall is more remote from the engine. Hence, for a close wall (say, under 5 feet from the engine), the close-in noise levels of the engine room should be used, i.e., Item 10 of Data Sheet 5; while for a greater distance (say, 5 feet or more from the engine), the reverberant noise levels of the engine room should be used, i.e., Item 9 of Data Sheet 5. In this particular example, assume that the wall is more than 5 feet from the engine and use the reverberant SPL.

Item 13 of sample Data Sheet 6 shows the following SPL values in the bedroom ("receiving room") for the octave frequency bands:

60 62 60 58 53 45 33 21 dB

The question asked by this example is whether the 8-inch-thick hollow-core concrete block wall will yield acceptable noise levels in the bedroom. Recall from the discussion of noise criteria (Paragraph 1.1) and from Table 1 at the end of the text that the noise criterion designation for sleeping (Category 1 in Table 1) is NC-20 to NC-30. From Table 2, the SPL values for NC-20 and NC-30 in the octave frequency bands are as follows:

NC-20: 51 40 33 26 22 19 17 16
NC-30: 57 48 41 35 31 29 28 27

A comparison of the estimated bedroom noise

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[3] Solution to NR = TL + C using the procedure in this manual results in a minus quantity. Caution: Observe algebraic signs in adding "noise" (+) to "reductions of noise" (–) in this manual; e.g., 10 + (–5) + (–2) = 3.

Data Sheet 6 *(Paragraph 5.6 Example)*

Sound Transmission to Adjoining Room Through Common Wall

Sound Transmitting Room: Engine Room

Sound Receiving Room: Bedroom

1. Area of Common Wall That Transmits Noise: 96 sq. ft.

2. Total Interior Surface Area of Receiving Room (including floor): 792 sq. ft.

3. Total Interior Surface Area Divided by Area of Common Wall: 8.2
(Item 3 = Item 2 ÷ Item 1)

4. Area of Acoustic Treatment in Receiving Room*: 270 sq. ft.

5. Per cent Area of Receiving Room Covered by Acoustic Treatment 34 %
(Item 5 = 100 X Item 4 ÷ Item 2)

6. Thickness of Absorption Material: 3/4 in.–1 in. ☒ 1-1/2 in.–2 in. ☐

7. Acoustic Treatment Condition of Receiving Room from Items 5 and 6, as Defined in Footnotes of Table 23:
Condition 1 ☐, Condition 2 ☐, Condition 3 ☒, Condition 4 ☐

8. Wall Correction Term "C" from Table 23 for Item 3 and Item 7 Conditions: +1 dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

9. Expected Wall Construction Material: 8 in. hollow-core dense concrete block

10. Estimated "TL" of Wall from Tables 13-22 or Other Source

31	33	35	36	41	48	54	59

11. Estimated "NR" of Wall in Equation $NR = TL + C$, Using Values from Items 8 and 10
(Item 11 = Item 10 + Item 8):
Caution: Observe algebraic signs in combining items!

32	34	36	37	42	49	55	60

12. Estimated SPL in Sound Transmitting Room from Data Sheet 5 (Use Item 9 Reverberant SPL If Source Is 5 ft. or More from Common Wall or Item 10 Close-in SPL If Source Is Less Than 5 ft. from Common Wall):

92	96	96	95	95	94	88	81

13. Estimated SPL in Receiving Room (Item 13 = Item 12 – Item 11)

60	62	60	58	53	45	33	21

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*Add 50% of floor area to Item 4 if receiving room floor is carpeted or if room has drapes or upholstered furniture. If this is the only acoustic material in the room, treat it as having 3/4 in. thickness in Item 6 for determining Item 7 condition.

levels with the NC-20 to NC-30 noise levels shows that the bedroom noise levels would be about 20 to 30 dB too high. If these noise levels were only a few decibels too high, this might be an almost acceptable condition (a discussion of "tolerances" appears later in Section 7), but a 20 to 30 dB noise excess would be completely unacceptable. Thus, the use of an 8-inch hollow-core concrete block wall would not be adequate to separate the bedroom from the engine room.

Actually, if this example were carried out again using a 12-inch-thick, solid dense concrete block wall, it would still be found inadequate to separate a bedroom from an engine room. This, then, essentially dictates that a sleeping area involving NC-20 to NC-30 noise criteria *not* be located immediately adjoining an engine room on the same floor. (Later comments are devoted to some special design requirements for (1) allowing a Category 3 or 4 area to adjoin immediately an engine room and (2) allowing a Category 1 or 2 area to be reasonably near an engine room.) Recall that the example used here was introduced primarily to illustrate the use of Data Sheet 6 for calculating the SPL in a room adjoining the engine room, but the example was also chosen to illustrate a potentially serious problem.

5.7 Doors and Windows

It is probably fairly obvious that if a 12-inch-thick, solid concrete block wall (at 144 lb/sq ft surface weight) were required to separate a noisy room from a quiet room, an ungasketed lightweight wood door or a single thickness of 1/4-in. glass window (each less than 4 lb/sq ft surface weight) would certainly represent a noticeable noise leakage path through that wall. If the noise reduction value of a wall is to be preserved, the doors and windows must be specially selected.

Where the area of a door or window is only a small part of the area of the wall in which it is installed, it is possible to permit the door or window to have a somewhat lower TL than the wall without significantly harming the total effectiveness of the wall. As examples, if the door or window area is 20% of the wall area, the TL of the door or window can be 3 dB lower than that of the wall; if 10%, 6 dB lower; and if only 5%, the door or window TL can be 10 dB lower than the wall TL. For these conditions the total effectiveness of the wall would be decreased only 1 dB, which can be considered negligible for most practical installations. These and other combinations of window or door area and relative TL and the resulting loss of TL of the composite wall containing them are given in Table 24 (including an example). (This table 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 a 10-inch-thick poured solid concrete wall having a knockout panel of 6-inch-thick concrete block.)

The approximate TL of a 2-inch, solid wood door, gasketed around all edges, is given in Table 19 (*see* Footnote 2 of Table 19), and the approximate TL of a 4-inch-thick and a 6-inch-thick industrial-type "acoustic door" is given in Table 16. The approximate TL of single thicknesses of glass is given in Table 17 and that of a few double-glass combinations is given in Table 18.

In some situations, the structural requirements may exceed the acoustical requirements of a wall, in which case the door or window can have a TL much lower than that of the wall. A few generalizations 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 exactly meet the values suggested as a function of their area relative to the total wall area, are:

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

2. Where the acoustic design requires somewhat more than minimum wall construction (such as staggered stud construction, 4-inch or 6-inch solid core concrete or masonry, or acoustically filled metal panel partitions), use gasketed solid-core wood doors or minimum 1-3/4-inch hol-

low metal doors packed with dense mineral or glass fiber, or special 1-3/4-inch- to 2-inch-thick acoustic doors with gasketing, and use windows of minimum area made up of double panes of at least 1/4 inch-thick glass with at least 2-inch air space, or windows of larger but limited area made up of double panes of at least 1/4-inch-thick glass with 4-inch to 6-inch air space. For area details, check against Table 24 data.

3. Where stringent acoustic requirements must be met, adhere to the door or window TL requirements given in Table 24 as a function of per cent area of the total wall. Use special acoustic doors or provide "sound locks" with gasketed double doors, as in Item (2) above, such that doors are spaced at least 5 to 6 feet 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 "receiving room."

5.8 Transmission Loss of Floor-Ceiling Combinations

The transmission loss of a simple dense concrete floor slab alone is approximately the same as given in Table 13 for dense poured concrete. However, in most building situations, a ceiling of some type is supported below the floor slab.

Five different floor-ceiling combinations are considered here. Under no condition should a reciprocating or turbine engine be mounted on framed wood flooring or on typical lightweight metal deck with 2- to 3-inch-thick concrete surface. These floor constructions are not stiff enough or massive enough to support heavy machinery or to give an adequate base for an engine mounting system.

The five floor-ceiling combinations are discussed in the following paragraphs. (Note: All floor slabs are assumed to be of *dense concrete* [140-150 lb/cu ft density].)

Type 1 floor-ceiling combination is the concrete floor slab that has acoustic tiles cemented directly to the underside. 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, as shown by the calculations in Data Sheets 5 and 6. The estimated TL of a Type 1 floor-ceiling is given in Table 25 for a few typical floor slab thicknesses.

Type 2 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 inch thick, and should have a noise reduction coefficient ("NRC") of at least 0.65 (when mounted as specified by the Acoustical Materials Association; *see* Footnote 1 in Paragraph 5.1). The air space between the suspended ceiling and the concrete slab above should be at least 15 inches, 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 26 for a few typical dimensions of concrete floor slab thickness and air space.

Type 3 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.) One possible 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 27 for a few typical dimensions of concrete floor slab thickness and air space.

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 inch thickness of high density plaster (minimum 12 lb/sq ft surface weight), and the air space should be at least 18 inches thick.

The ceiling should be supported on resilient ceiling hangers that provide at least 1/10-inch 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-inch 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 28 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.

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 the engine assembly or any other large operating equipment. It should extend over all the engine room floor area within 20 feet of the engine assembly, 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 inches 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 inch. The 2-inch space between the floating slab and the structure slab should be covered with a 1-inch 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-inch gaps that are later packed with mastic or fibrous filling and then sealed with a waterproof non-hardening caulking or sealing material. It would be advisable to provide a curb arrangement around the perimeter of the floated slab to help discourage water leakage into the sealed perimeter joints, because, if the floor becomes flooded, water should not fill up the 2-inch space under the floated slab.

As a prevention against this occurrence, 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 29 for a few typical dimensions of floating floor slab in combination with the Type 4 structures of Table 28. The Type 5 combination may be required for an engine room floor in certain critical situations but it probably would not be required in any other applications within the scope of this manual. 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. The floating slab is *not* designed here to support pumps, fans, compressors, engines, motors, refrigeration equipment, and the like.

As a general rule, to be reinforced later in the section under vibration isolation, the engine-room structural-floor slab for an upper floor in a multi-floor building should not be less than 6 inches thick for a gas turbine engine used to drive a completely rotary-action device, or less than 8 inches thick for a gas turbine engine used to drive a reciprocating-action device. For a reciprocating engine drive, the structural floor slab should not be less than 10 inches thick. 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 slightly more beneficial acoustically. Where possible, the engine assembly should be located over principal or secondary beams in the flooring layout.

In the upper frequency bands of Tables 25-29, 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 concentrated effort to stop all escape paths of airborne and structure-borne noise.

5.9 Noise Reduction of Floor-Ceiling Combinations

Paragraph 5.5 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 Paragraph 5.5 and in Table 23. 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 engine room 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." It is also obtained from Table 23, based on (1) room absorption; (2) total interior area; and (3) 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 an engine room.

5.10 Example, Data Sheet 7

Suppose an on-site total energy plant is considered for location on the top floor of an office building. Suppose three 450-hp, 1,800-rpm reciprocating natural gas engines would be located in the engine room directly over a suite of private offices. Suppose a typical office is 10 feet x 15 feet x 8 feet and has a carpet and drapes but no acoustic tile ceiling. Because of the very critical location, suppose a Type 5 floor-ceiling is first calculated to determine the feasibility of this location.

Sample Data Sheet 7 indicates the steps to be taken in the calculation. Before Data Sheet 7 can be completed it is necessary to determine the engine room SPL from Data Sheets 1 and 5. These sample calculations are not given here, but for an assumed engine room of dimensions of 40 feet x 80 feet x 15 feet with a full 50% of all interior surfaces covered with 2-inch acoustic absorption material, the following engine room SPLs can be calculated:

(a) The reverberant SPL due to one engine would be
87 91 91 90 90 89 83 76 dB

(b) The close-in SPL due to one engine would be
90 94 94 93 93 92 86 79 dB

(c) The reverberant SPL due to three engines would be
92 96 96 95 95 94 88 81 dB

Since the floor of concern is immediately under the engines, the close-in SPL of each single engine would normally apply. But in this room, the reverberant SPL due to all three engines is slightly higher than the close-in SPL of each engine, so the reverberant SPL for all three engines is used, i.e.,

92 96 96 95 95 94 88 81 dB

Suppose the Type 5 floor-ceiling consists of a

Data Sheet 7

Sound Transmission to Adjoining Room Through Common Floor-Ceiling

(Paragraph 5.10 Example)

Sound Transmitting Room	Sound Receiving Room
Engine Room	Private Office

1. Area of Common Floor-Ceiling That Transmits Noise: 150 sq. ft.

2. Total Interior Surface Area of Receiving Room (including floor): 700 sq. ft.

3. Total Interior Surface Area Divided by Area of Common Floor-Ceiling: (Item 3 = Item 2 ÷ Item 1) 4.7

4. Area of Acoustic Treatment in Receiving Room*: 75 sq. ft.

5. Per cent Area of Receiving Room Covered by Acoustic Treatment (Item 5 = 100 X Item 4 ÷ Item 2) 11 %

6. Thickness of Absorption Material: 3/4 in.–1 in. ☒ 1-1/2 in.–2 in. ☐

7. Acoustic Treatment Condition of Receiving Room from Items 5 and 6, as Defined in Footnotes of Table 23: Condition 1 ☐, Condition 2 ☒, Condition 3 ☐, Condition 4 ☐

8. Floor Correction Term "C" from Table 23 for Item 3 and Item 7 Conditions: -4 dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

9. Approximate Floor-Ceiling Type (from Paragraph 5.8 of Text): Type 1 ☐ Type 2 ☐ Type 3 ☐ Type 4 ☐ Type 5 ☒

10. Approximate "TL" of Floor-Ceiling Type of Item 9 for Nearest Applicable Dimensions, from Tables 25-29 (Interpolate Between TL Values Shown, If Desired, But Do Not Exceed TL Values Shown in Right-Hand Column of Tables 25-28):

50	54	58	64	72	78	82	88

11. Estimated "NR" of Floor-Ceiling in Equation $NR = TL + C$, Using Values of Items 8 and 10 (Item 11 = Item 10 + Item 8): Caution: Observe algebraic signs in combining items!

46	50	54	60	68	74	78	84

12. Estimated SPL in Sound Transmitting Room from Data Sheet 5 (Use Item 9 Reverberant SPL for Floor-Ceiling Above the Engine or Item 10 Close-in SPL for Floor Under Engine):

92	96	96	95	95	94	88	81

13. Estimated SPL in Receiving Room (Item 13 = Item 12 – Item 11)

46	46	42	35	27	20	10	-3

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*Add 50% of floor area to Item 4, if receiving room floor is carpeted or if room has drapes or upholstered furniture. If this is the only acoustic material in the room, treat it as having 3/4 in. thickness in Item 6 for determining Item 7 condition.

4-inch-thick floating concrete slab supported resiliently 2 inches above a 10-inch-thick dense concrete structure floor slab, with a 30-inch air space to a 1-1/2-inch-thick dense plaster ceiling supported resiliently below the slab. Tables 28 and 29 provide the TL estimate for this combination.

Item 13 of sample Data Sheet 7 yields the following estimated SPL values in the typical office immediately under an engine

46 46 42 35 27 20 10 −3 dB

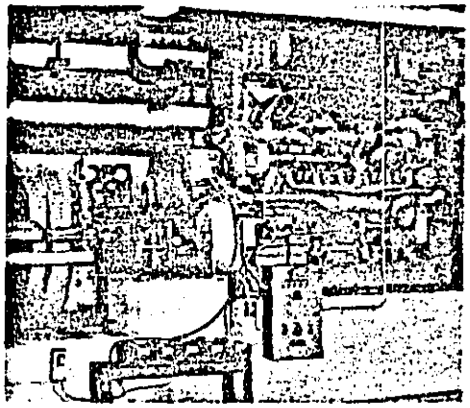
These levels may be compared with the applicable NC-30 to NC-35 criterion levels for a private office (Tables 1 and 2) as follows.

NC-30: 57 48 41 35 31 29 28 27 dB
NC-35: 60 52 45 40 36 34 33 32 dB

It is seen that the NC-30 criterion would be exceeded by 1 dB in only one frequency band, and the NC-35 criterion would be met in all frequency bands. This would probably be an acceptable solution, especially since the office ventilation system air supply would probably also produce some masking noise in the NC-30 to NC-35 range.

5.11 Special Situations

In the example shown in Paragraph 5.10 it is seen that a rather special floor-ceiling combination is required to separate a fairly noisy engine room from a fairly quiet private office. Even so, this particular Type 5 floor-ceiling combination would represent only a marginal solution if the same engine room were *directly over* a bedroom in an apartment building layout (and probably should not even be attempted without extra acoustical assistance). This raises the question: Can an engine room be located, say, on the top floor of an apartment building? The answer to this question is "yes" for a number of situations and "no" for a few situations. These are labeled "special situations" here and are discussed briefly.



Refrigeration system assembly is installed rigidly on massive concrete inertia block. Even when there is no nearby acoustically critical area, this type of mounting will reduce vibration of system components and piping, floor slab and building structure itself. *(Photograph is courtesy of Caterpillar Tractor Company.)*

A. Reciprocating Engines:

1. Primarily for vibration considerations, reciprocating engines above about 1,000 hp each or below about 900 rpm should not be located on an upper floor of a building that will house occupants falling into Categories 1, 2 and 3 of Table 1, without the aid of an acoustical consultant to assure that certain special details are met. Vibration isolation is discussed later in the manual.

2. Reciprocating engines above about 50 to 100 hp probably should not be located *directly above* a Category 1 or 2 area of Table 1, without the aid of an acoustical consultant to check for the required details.

3. Reciprocating engines below about 1,000 hp each and above about 900 rpm can be located on an upper-level engine room floor (with proper vibration isolation, as discussed in Section 8) provided the location is horizontally removed from an NC-20 to NC-25 Category 1 or 2 area by at least 40 feet or two column lines of the building (whichever is the greater distance), or is horizontally removed from an NC-25 to NC-30 Category 1 or 2 area by at least 20 feet or one column line (whichever is the greater). These generalizations apply to a critical area on the floor below or be-

side the engine room, and there must be at least one good full-height wall (having a TL of greater than 25 dB in the 250, 500 and 1,000 Hz frequency bands) between the critical area and the engine room *in addition* to the walls and floors that enclose the engine room. This second wall must be located at least 10 feet beyond the enclosing wall of the engine room, preferably as much as 20 to 30 feet, if the building layout permits. These same suggestions may also be applied to a critical area located on the floor above an engine room floor but the horizontal separation distances can be relaxed slightly (if necessary), if all piping associated with the engine assembly are well isolated in and near the engine room.

4. Reciprocating engines of any size and speed can be located in a basement or on-grade position of any building as long as proper vibration isolation and suitable airborne noise control are provided. If a critical Category 1 or 2 area is located on the floor above the engine room floor, the horizontal separations discussed above are advised.

5. There need be no horizontal separation between the engine room and any critical area that is two floors above or below the engine room floor, so long as proper vibration isolation and adequate airborne noise control are provided for the engine.

B. Turbine Engines:

1. In terms of vibration considerations alone, there appears to be no size limit for a turbine engine located on an upper floor of a multi-floor building that houses occupants in Categories 1, 2 and 3 of Table 1, provided the engine drives entirely rotary-action devices (no reciprocating-action devices) and provided the engine assembly is properly vibration isolated. As a practical matter, the scope of this manual limits the size to about 5,000 kw.

2. In terms of noise considerations alone, a turbine engine rated above about 500 kw probably will either require a Type 4 or Type 5 enclosure (as described briefly in the footnotes of Table 6) or horizontal separation, such as discussed above in Item 3 for reciprocating engines, if an NC-20 to NC-30 critical area is located near the engine room in an upper floor of a building. In addition, of course, proper vibration isolation and airborne sound control for the areas immediately surrounding the engine room should be provided, in accordance with Data Sheets 6 and 7, as applicable.

These special situations will become apparent when calculation of Data Sheet 6 or 7 reveals that a particular high-TL wall or floor-ceiling cannot produce low enough noise levels in the adjoining space to meet the noise criterion desired for that space. In effect, the need for horizontal separation, as discussed in the "special situations" above, is evidence of need for "double-wall construction" in order to attain a high TL value. The recommendation of a second wall at least 10 feet horizontally from the engine room wall would meet the "double-wall" requirement. There are more sophisticated double-wall structures that do

Solar gas turbine generator sets produce power and exhaust heat for chemical plant. Exhaust is ducted out of room. Operators have glass-walled control room to reduce noise. *(Photograph is courtesy of Solar Division, International Harvester Company.)*

not require the 10 foot separation but they become rather complex structures and require special detailing and special attention during construction. If the building layout can accommodate the 10-foot horizontal separation of walls (on the same or immediately adjoining floors), no special wall details would be required (other than that the second wall be of such construction as to provide a TL of at least 25 dB in the three frequency bands listed above–see Tables 13-22 for possibilities).

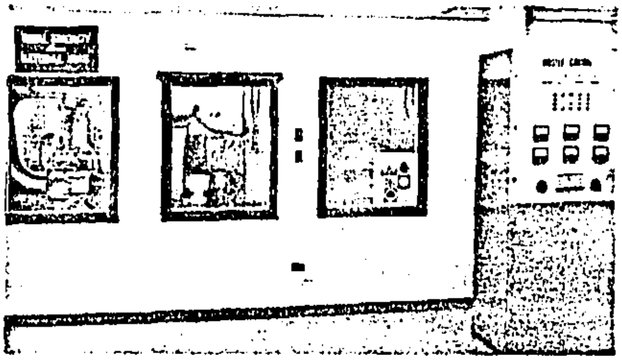
5.12 Engine Room Precautions

Although there may be no obligation to consider the following factors in designing an engine room, there is a growing concern for the well-being of operating personnel and for operational efficiency. Two points are mentioned here regarding engine room occupancy by personnel.

First, conversation in the engine room is usually quite difficult. For this reason, audio communication should not be encouraged: do not install a loud bell to call the operator (use a flashing light instead) and do not install a telephone in the engine room (install the telephone in the control room).

Second, engine room noise levels are sufficiently high that an operator should not be required or expected to remain in the engine room during long periods without ear muffs or ear plugs. To overcome this possibility, a small control room or engineer's office should be provided beside the engine room. Operating personnel should use the control room except when required to be in the engine room.

Detailed analyses of allowable time intervals in engine rooms without the use of ear protectors have been worked out in the manual identified under Footnote 1 at the beginning of Section 3. The details are too lengthy to repeat here, but it would be a fair summary to state that operators should wear ear protectors inside an engine room with reciprocating engines every time they are likely to remain in the engine room for more than 10 minutes at a time. For intervals of less than 10 minutes, separated by at least 30-minute periods in a much *quieter* environment, ear protectors probably need not be worn. Probably, ear protectors should be worn for *all* time intervals in an engine room containing turbine engines. To arrive at more specific recommendations, the reader should know the actual engine room SPLs and check them against the latest data of the "CHABA" committee[4].



Control room beside gas turbine total energy installation. Double-glass viewing windows and interconnecting door through separate alcove reduce noise levels in control room. Installation provides heat, power and refrigeration for large modern office building. *(Photograph is courtesy of Northern Illinois Gas Company.)*

Ear protection is recommended for personnel exposed to high levels in engine room. *(Photograph is courtesy of Northern Illinois Gas Company.)*

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[4] "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.)

Section 6

CONTROL OF AIRBORNE SOUND OUT-OF-DOORS

Section 5 was concerned mostly with the distribution and control of engine casing noise within the building served by the engine. In Section 6, primary concern is for the distribution and control of that portion of the engine noise that is radiated out-of-doors and that might be heard by the neighbors. The term "neighbor" is used here to designate the nearest person or property in a given direction to be protected against excessive noise from the engine. The neighbor may be a nearby resident or a nearby building or, in some cases, part of one's own building or its occupants.

6.1 Sound Pressure Level Out-of-Doors

When a sound source is free to radiate out-of-doors, the sound usually travels out in all directions and becomes less intense at increasing distances from the source. For an "omni-directional" or "non-directional" sound source (i.e., it radiates equally in all directions), the outdoor SPL at any distance, "D", from a source. PWL, is given by the relationship:

$$SPL = PWL - 10 \log (2\pi D^2) + 10 - \alpha D/1000$$

where α is the loss per 1,000 feet due to "molecular absorption" of sound in the air. For the manual, the latter terms of this equation have been combined into one term, called an "outdoor distance term," such that

$$SPL = PWL - \text{Outdoor Distance Term}$$

The "outdoor distance term" has been calculated and summarized in Table 30 for a range of distances from 10 feet to 7,200 feet. This assumes average atmospheric conditions and no sound barrier or obstruction between the source and the receiver. Weather conditions can influence sound transmission, particularly at the large distances, but usually the weather is sufficiently variable that it cannot be relied upon to give a permanent benefit to a noise problem. Hence, in this manual, average sound propagation conditions are assumed.

6.2 Example, Data Sheet 8

As an example of the use of Table 30, suppose it is desired to determine the SPL at a distance of 1,600 feet from the reciprocating engine unmuffled exhaust noise of Example 3.4, as shown by Item 7 of sample Data Sheet 2 for that example, assuming that the noise radiates uniformly in all directions from the exhaust pipe opening.

Sample Data Sheet 8 indicates the steps to be taken for obtaining the outdoor SPL at the 1,600-foot distance.

6.3 Noise Reduction
Provided by a Building

An intruding noise coming from an outdoor noise source 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 neighboring building, can be made to be no greater than the criterion levels of Table 2 for the Table 1 activity areas, 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 (NR) depends on building construction, orientation, wall area, window area, open window area, interior ac-

Data Sheet 8 *(Paragraph 6.2 Example)*

Estimated Outdoor Sound Pressure Level (SPL) Due to an Outdoor Sound Source PWL

1. Distance to Noise Source: 1,600 ft.

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

2. Total PWL of All Outdoor Noise Sources at Source Position:

121	127	123	115	111	105	95	87

3. Outdoor Distance Term from Table 30 for Item 1 Distance:

62	62	62	63	64	67	75	85

4. Approximate Outdoor SPL at Distance of Item 1: (Item 4 = Item 2 $\overset{-}{+}$ Item 3)
Caution: Observe algebraic signs in combining items!

59	65	61	52	47	38	20	2

oustic absorption, etc. For practical purposes, however, the average NR values of Table 31 can be used in the manual.

To estimate the indoor SPL of a noise coming inside the building from outdoors, it is merely necessary to reduce the outdoor SPL by the amount of NR found in Table 31 for the appropriate wall and window condition. If the building has no windows at all, an approximate NR value can be estimated for the wall from the TL values of Tables 13-22. When the indoor SPL is estimated, it can be compared with the noise criterion levels of Tables 1 and 2 considered appropriate for the situation. If the indoor SPL is equal to or less than the criterion levels, there will generally be no noise problem. If the indoor SPL exceeds the criterion levels by more than a few decibels in one or more of the frequency bands, there may be a problem if no further noise reduction is applied.

6.4 Noise Reduction Provided by a Solid Barrier

A solid 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 some noise reduction for a receiver located within the "shadow" produced by the barrier.

Table 32 shows the important dimensions that influence the amount of noise reduction provided by a barrier. For a barrier to be effective, it should extend as far as possible beyond the line-of-sight between any part of the noise source and any part of the receiver in both the *vertical* and *horizontal* directions. This distance beyond the line-of-sight is represented by "H" in the simple sketch included with Table 32.

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 32 must be large compared to the distance R and height (or width) H. The attenuation values given in Table 32 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 32. This is usually not a very important factor in a problem, however, because at a one-mile distance, the noise has become so reduced by the distance effect that the loss of effectiveness of the barrier is hardly noticed.

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.

It should be noted 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 about 2 or 3 dB higher levels in the direction of the reflected sound.

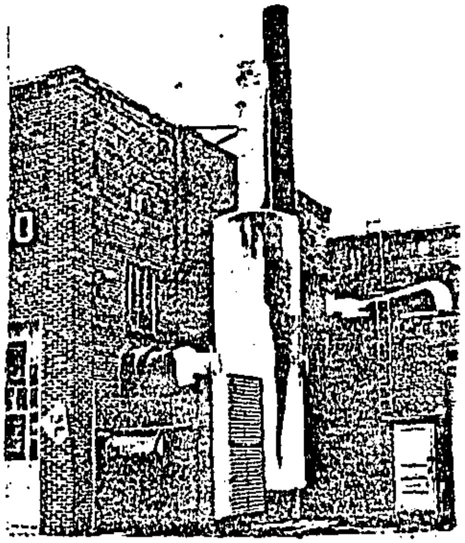
6.5 Noise Reduction Provided by Dense Woods

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 through the lower, less dense parts of the tree growth. 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 32, treating the woods as though it were a solid barrier.

Table 33 gives the approximate noise reduction through dense woods, where dense woods are taken as having an average "visibility penetration" of about 70 to 100 feet. 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 still can be seen. Footnotes in Table 33 pertain to reduced noise reduction rates for less dense and for shorter growths of trees.

6.6 Directivity Effect of a Stack Opening

If noise is emitted to the outdoors from the open end of a large duct, there would be more noise directly in front of the duct opening than there would be at the side of the duct. Table 34 gives some values to this "directivity effect" of a large ducted opening, assuming in this case it applies to the hot, high speed gases and noise emitted by the *vertical exhaust stack* of a large turbine engine. At positions directly in line with the stack opening (0° in Table 34), noise levels are higher than if this were a non-directional unducted source free to radiate equally in all directions. At positions to the side (90° and 135°), the noise levels are lower than if this were a non-directional source. This effect is quite pronounced for large openings and tapers off to a negligible effect for small openings, where the duct size is not very large compared to the wavelength of the sound.



Air inlet filter (with louvered face) and exhaust muffler for large reciprocating engine. *(Photograph is courtesy of Diesel and Gas Turbine Progress.)*

Field tests have shown that the same general effect holds for a large air *intake opening* that emits noise, with the exception that the effect at 90° and 135° is slightly greater, as shown by the footnote in Table 34.

If the inlet or exhaust duct is oriented horizontally and does not form a vertical stack, the general directivity effect of Table 34 can still be assumed, always measuring the angle from the axis of the opening, provided there are no nearby structures than can reflect higher sound levels into the shadow zone of the 90° and 135° regions.

Field data upon which Table 34 is based are quite sketchy and not very definitive. Thus, the actual directivity effects seem to be quite variable depending upon the geometrical variables of the several stacks tested. Some effect is known to exist, however, and reasonable averages are offered in Table 34.

Note that these data are based on vertical stacks that project at least 10 to 20 feet or more above other portions of the building and usually 20 to 40 feet above the ground. If such a stack now is considered as lying horizontally along the ground, the ground and any projections or structures on the ground in front of the stack opening can distort the directivity effect considerably.

If a large duct opening is located in the side wall of a large building, the building itself serves as a barrier that enhances the directivity effect. Less noise will be radiated in a direction of 180° from the axis of the duct, but all the geometric factors make it difficult to give general rules on "how much less." The values will fall somewhere between those of Tables 32 and 34, but this requires some judgement in the use of Table 32 for such a situation.

6.7 Noise Reduction Provided by "Reactive Mufflers"

"Reactive mufflers" are used almost entirely for gas and diesel reciprocating engine exhausts. Reactive mufflers usually consist of two or three large-volume chambers containing an internal labyrinth-like arrangement of baffles, compartments and perforated or slotted tubes. Sometimes also called "snubbers," these mufflers smooth out the flow of impulsive-type exhaust discharge and, by the arrangement of the internal components, attempt to reflect some sound energy back toward the source. These mufflers usually have no acoustic absorption material, although some manufacturers produce some models that contain an inside lining of high-temperature material. Many manufacturers produce a low-pressure-drop and a high-pressure-drop line of mufflers, and each line comes in three different classes of noise reduction. The low-pressure-drop line is advertised largely for turbocharged engines, but the high-pressure-drop line can also be used with turbocharged engines, if the engine manufacturer approves the general exhaust layout. The high-pressure-drop line provides greater noise reduction. Typically, the three different classes of noise reduction are indicated by labels that somewhat relate to the degree of criticalness of the potential noise problem, such as "commercial," "standard," and "residential," and "industrial," and "semi-critical," and "critical," or similar series of names and models.

Table 35 gives approximate noise reduction values for the three classes of mufflers in the low- and high-pressure-drop lines. The three classes are identified here by relative size: small, medium and large. The small size has the lowest noise reduction; the large size has the highest noise reduction.

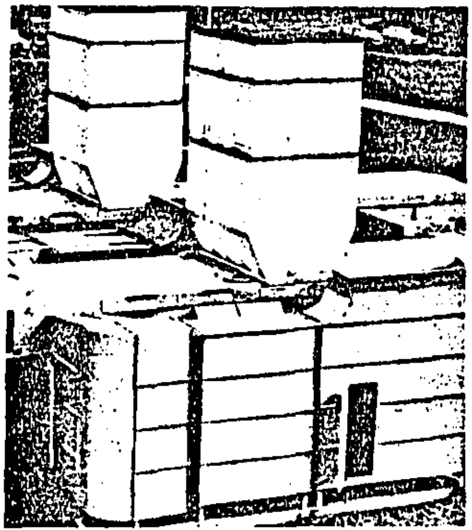
The engine manufacturer may recommend a maximum length and minimum diameter exhaust pipe for his engine, as these influence the back pressure applied to the engine exhaust. Exhaust pipe layout and location of the muffler also influence the noise reduction provided by the muffler.

Muffler manufacturers stress that the muffler should be located as close as possible to the engine in order to avoid exhaust pipe resonances that coincide with engine firing rates. This is not always possible or practical; but it is well to know that if such a resonance should occur and the pipe vibrates strongly or the muffler seems to be ineffective, it may be possible to eliminate or reduce the pipe resonance by changing the pipe length 5 to 8 feet in between the engine and the muffler. For given engine configurations, some critical pipe lengths can be calculated but there is no assurance that resonance will or will not occur for those lengths, so the calculations are not included here.

6.8 Noise Reduction Provided by "Dissipative Mufflers"

"Dissipative mufflers" are made up of various arrangements of acoustically absorbent material that actually absorbs sound energy from the moving air or exhaust stream.

Dissipative mufflers are used in the air intake and gas exhaust of turbine engines. The most popular configuration is an array of "parallel



Twin set of gas turbine engines used to generate electrical power. Air intake to engines enters large muffled ducts at each side of engine housing (nearest air inlet duct shown with open access door in foreground.) Engine exhaust gases and noise pass through vertical mufflers in this outdoor installation. *(Photograph is courtesy of Diesel and Gas Turbine Progress.)*

baffles" placed in the air stream. The baffles may range from 2 inches to 16 inches thick, filled with glass fiber or mineral wool of such type as to stand up under the operating temperature (possibly 900° to 1,200°F in engine exhausts), with adequate internal construction and surface protection to resist the destruction and erosion by high-speed, turbulent flow. These baffles have been proved with many years' use in jet engine test cells. An experienced reputable manufacturer should be selected for these mufflers in order to insure proper quality of materials, design and workmanship and ultimately long-life and durability of the installation.

Where large amounts of attenuation are required in relatively short dimensions, it is necessary to install baffles at relatively close spacing, such that the total open area for air flow may be 30% to 70% of the total cross-section area of the stack or opening. This, in turn, produces pressure drop in the flow, so it is necessary to reach a compromise of cost, area, length and pressure drop in the final choice of muffler arrangement.

In general, thin parallel baffles (2 inches to 4 inches thick) are more effective in the higher frequencies, and thick parallel baffles (8 inches to 16 inches thick) are more effective in the lower frequencies. Unfortunately there are no baffles that are highly effective at very low frequencies, although it is known that large amounts of low frequency attenuation require small percentage open areas of mufflers (such as 30% to 40%).

The field of muffler design and construction is so specialized that an architect or mechanical engineer should not undertake the design of these mufflers. Instead, it is suggested that noise reduction requirements for muffler application be worked out by use of the manual and in conjunction with the turbine engine manufacturer, and then be specified to the engine installer, contractor or muffler manufacturer. Tables 36 and 37 give the approximate noise reduction provided by some typical parallel-baffle mufflers.

For air intakes into gas turbine engines, the noise reduction requirement may be set by any one of the frequency bands; for turbine engine exhausts, the noise reduction requirement will probably be set by one of the low frequency bands.

In some critical installations it has been found that high flow speed through the muffler can produce noticeable "self-noise" that is itself a cause of concern. There is no exact schedule of air speed vs self-noise, but it has been found that a flow speed of about 175 to 200 feet per second is probably acceptable for engines located in fairly noisy commercial or industrial areas and of about 125 to 150 feet per second probably acceptable for engines located in or near quiet residential areas. Remember that, at engine exhaust temperatures, hot exhaust gas is of much lower density and has a higher total volume flow than the cool air intake into the engine.

6.9 Example, Data Sheet 9

The various noise reduction components discussed briefly under Paragraphs 6.3-6.8 can be summarized by use of an example and Data Sheet 9.

Suppose the example of Paragraph 6.2 is continued here. The unmuffled exhaust noise of a 200-hp turbocharged reciprocating engine was found to produce an outdoor SPL at 1,600 feet distance of:

59 65 61 52 47 38 20 2 dB

These values become Item 1 of sample Data Sheet 9. Suppose the neighboring building at 1,600 feet distance is a suburban residence having normally open windows. There is no major obstacle between the engine and the residence that is large enough to serve as a useful sound barrier, but there is a 200-foot-wide densely wooded area separating the engine from the residential area. The exhaust pipe opening is too small to have any significant directivity effect. Items 2-7 of Data Sheet 9 summarize the effects of these noise reduction contributions. The tentative indoor SPL of Item 7 is as follows:

49 53 47 36 28 16 –4 –24 dB

This can be compared with the desired NC-20

Data Sheet 9 *(Paragraph 6.9 Example)*

Estimated Indoor SPL at Neighboring Location, Including All Noise Reduction Contributions

Noise Source Considered in This Data Sheet: `Exhaust of 200-hp Recip. Engine`

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

1. Outdoor SPL at Neighboring Location for No Noise Reduction Contributions (from Item 4 of Data Sheet 8):

59	65	61	52	47	38	20	2

2. Noise Reduction from Outdoors to Indoors Provided by Neighbor Building, from Table 31:

9	10	11	12	13	14	15	16

3. Noise Reduction Provided by Sound Barrier, from Table 32:

0	0	0	0	0	0	0	0

4. Noise Reduction Provided by Woods, from Table 33:

1	2	3	4	6	8	9	10

5. Tentative Indoor SPL to This Point (Item 5 = Item 1 – Item 2 – Item 3 – Item 4):

49	53	47	36	28	16	-4	-24

6. Directivity Effect of Stack Opening, If Applicable, from Table 34:

0	0	0	0	0	0	0	0

7. Tentative Indoor SPL to This Point (Item 7 = Item 5 + Item 6)
Caution: Observe algebraic signs in combining items!

49	53	47	36	28	16	-4	-24

8. Approximate Noise Reduction of Muffler Planned for Use with Noise Source, If Applicable, from Tables 35-37 or from Muffler Data Supplied by Manufacturer:

10	15	13	11	10	9	8	8

9. Estimated Indoor SPL at Neighboring Location, Due to This Noise Source (Item 9 = Item 7 – Item 8)

39	38	34	25	18	7	-12	-32

criterion for the residence, from Tables 1 and 2:

NC-20: 50 40 33 26 22 19 17 16 dB

This comparison shows that the tentative indoor SPL exceeds the NC-20 criterion by the following amounts:

– 13 14 10 6 – – – dB

These excesses could be met by the use of a reactive muffler (Table 35) for the engine exhaust. A small-sized, low pressure-drop muffler, having the following approximate noise reduction

10 15 13 11 10 9 8 8 dB

would be considered adequate to meet the requirement even though it is shy by 1 dB in the 250 Hz band, provided this is the only noise source to be considered. If there were several noise sources present, their total effect on the residence should be considered. And it might be necessary to use a larger muffler with greater noise reduction.

For a gas turbine engine, the amount by which the tentative indoor SPL of Item 7 of Data Sheet 9 exceeds the desired noise criterion for that indoor area would give the noise reduction requirement for a muffler for either the air intake or gas exhaust of the engine. This requirement would then be specified to the engine installer or muffler supplier. A very rough idea of the muffler size might be obtained by checking the requirement with the muffler examples given in Tables 36 and 37.

In using Data Sheet 9, it is cautioned that if the noise reduction provided by a muffler was included in estimating the PWL of the noise source (as in Data Sheet 2, 3 or 4), then the noise reduction of that same muffler cannot be used again in Item 8 of Data Sheet 9.

If a very unusual amount of noise reduction is required or if it appears that conventional muffling devices cannot meet the requirement, re-check all calculations very carefully and, if necessary, obtain the assistance of the muffler company engineer or of an acoustical consultant.

6.10 Noise Control for an Outdoor Engine

If it is expected that an engine is to be located out-of-doors with *no engine room or acoustic enclosure*, the SPL of the engine casing noise at the desired distance should be determined with the use of Data Sheet 8 and appropriate tables of data. Then, with the use of Data Sheet 9, the indoor SPL values at the nearest neighbor of interest can be determined. These SPL values should then be compared with the appropriate indoor noise criterion levels for that neighbor. For a nearby residential neighbor in a quiet community situation, the engine noise might be found to be 10-30 dB too high for neighbor acceptance. Such a situation would clearly require that the engine be housed *inside an engine room.* As a rule-of-thumb, the TL of the walls of such an engine room should be *at least 15 dB greater* in all frequency bands than the noise excess found for the outdoor engine, and the TL of the roof of the engine room should be *at least 6 dB greater* in all frequency bands than the noise excess found for the outdoor engine.

These required TL values would apply if there is no acoustic absorption material inside the engine room. These wall and roof TLs can be reduced 3, 6, or 9 dB for acoustic absorption inside the engine room that meets condition 2, 3 or 4, respectively, in the footnotes of Table 23. Doors and windows (if the latter are necessary) should be compatible with the walls in terms of noise reduction (*see* Paragraph 5.7).

If an engine is expected to be enclosed in some type of lightweight weather enclosure, the same type of calculation as outlined above should first be carried out. If the lightweight enclosure does not meet the TL requirement, the construction should be improved so that it will meet the TL requirement.

6.11 Noise Escape from an Opening

Although one may try to minimize the escape of noise from a room, there are times when the noise will escape, and, at those times, it may be necessary to know the amount of escaping noise. An example of such a situation is the ventilation ducts that serve an engine room. In order to allow air movement through the room, the ducts may be open to the outside, and noise would escape through the open ducts unless suitable muffling is provided.

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

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

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

$$AF = 10 \log A - 10.$$

Then,

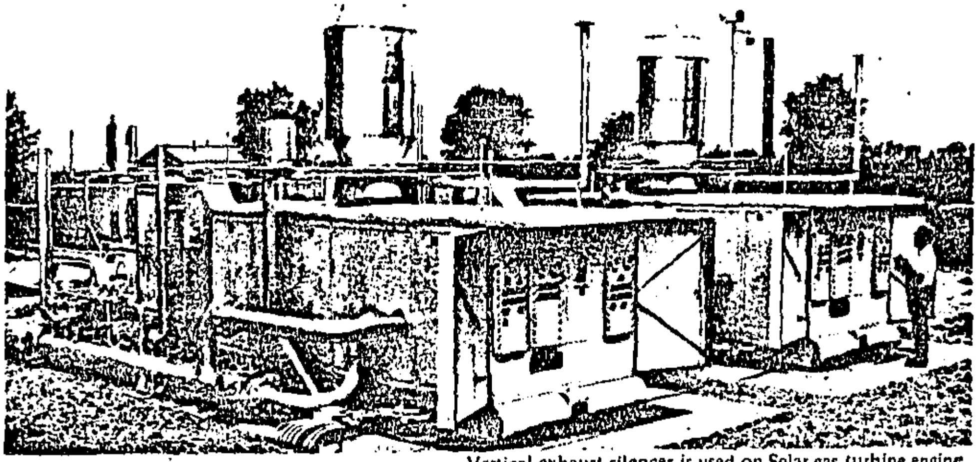
$$PWL = SPL + AF$$

Table 38 gives a range of values of AF for a representative group of areas.

If an engine room has a large opening to the outside to provide ventilation air to the engine, the PWL of the noise escape through that opening should be determined. The SPL is taken as the engine room SPL, and the AF term is found from Table 38 for the area of the opening. The resulting PWL should then be treated as a noise source that radiates to a neighbor, using Data Sheets 8 and 9 as applicable.

There is an important caution to be introduced at this point. In a small room with little or no acoustic absorption, the reverberant SPLs are quite high. For a large area duct or opening to the outside, it is possible to calculate a PWL of escaping noise that is actually larger than the PWL of the noise source inside the room in the first place. Of course, this is a fallacy of the calculations and cannot really exist. Therefore, when calculating the PWL of escaping noise through an opening, check back to the original PWL of the engine or noise source in the room to be certain that the PWL of the escaping noise is never greater than the PWL of the basic noise source inside the room.

Excess noise escaping through holes, ducts, open windows or ventilation ports can best be controlled with the use of a muffler in the opening. Table 39 gives the approximate noise reduction of typical "packaged duct mufflers" supplied by several acoustic products manufacturers. These mufflers may also be used in air-intake openings for turbine engines, if the air speed does not exceed the upper limits suggested by the muffler manufacturer. In fact, so that air flow noise itself is not excessive, the air speed through the muf-



Vertical exhaust silencer is used on Solar gas turbine engine to reduce noise radiated from this natural gas pumping station to the neighbors. *(Photograph is courtesy of Solar Division of International Harvester Company.)*

fler passages probably should not exceed about 4,000 ft/min.

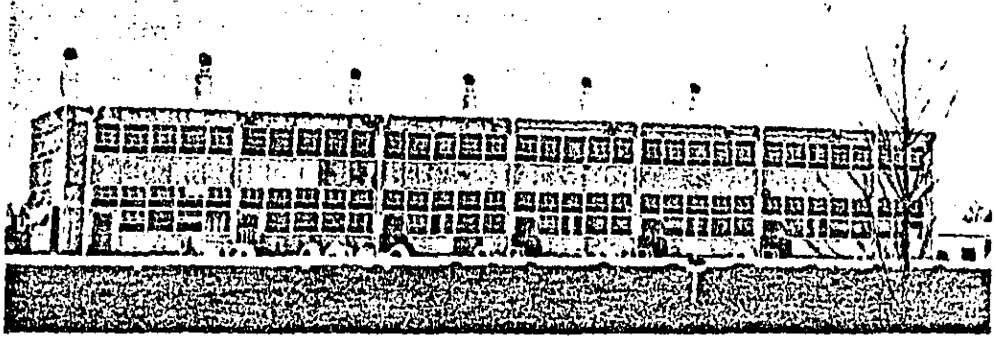
Noise escape through ducted openings can also be controlled by duct lining. This is not discussed in detail here, since duct treatments can be referred to in the "ASHRAE Guide and Data Book" (*see* Footnote 2 under Paragraph 4.2 of the manual).

6.12 Caution on Outdoor Engine Noise

In Paragraph 6.3, it was stated: "If the noise, when heard indoors in the neighbor building, can be made to be no greater than the criterion levels of Table 2 for the Table 1 activity areas, it is quite likely that there will be no complaint against the noise." This is generally true, but it is also important to realize that neighbors who value their outdoor surroundings may not tolerate very much outdoor audible noise even if it is essentially inaudible when heard indoors. For such situations it would be well to compare the outdoor noise made by the engine with the ambient or background noise for the area. If actual measured noise levels cannot be made, an approximate estimate of outdoor nighttime background noise for any area can be determined from a table and set of curves given in the latest "ASHRAE Guide and Data Book" in the chapter on noise control. In general, critical neighbors would not willingly accept outdoor engine noise levels that are more than about 5 to 10 dB above the nighttime ambient noise levels. If the outdoor engine noise is above this amount, additional noise control should be included until the engine noise levels equal or are only slightly above outdoor ambients at the neighboring location.

6.13 Noise Codes and Ordinances

A major objective of this manual is to provide the guidance necessary to achieve a sufficiently quiet engine installation so that neighbors will not complain of the noise. If this objective is met, there may be no reason to invoke a noise ordinance against such an installation. Nevertheless, where local codes or ordinances exist, it is desirable to check the expected noise levels of the engine installation, including all the noise control designs, to determine if they conform to the code requirements. If additional noise reduction is shown to be necessary, determine the additional treatments that must be incorporated into the total design. Data Sheet 8 may be used to determine the outdoor noise levels at any specified distance from the total noise source, and, where applicable, the noise reduction steps of Items 3, 4, 6 and 8 of Data Sheet 9 and other treatments can be used to estimate the ultimate noise levels to be expected at the point specified by the ordinance.



Arrangement of mufflers control engine exhaust noise for bank of engine-driven compressors at this gas transmission line compressor station. (Photograph is courtesy of Universal Silencer Corporation.)

Section 7

TOLERANCES ON AIRBORNE SOUND PRESSURE LEVEL ESTIMATES

When numerical values are assigned to noise criteria, PWLs, SPLs, TLs, NRs, etc., this raises the question of accuracy of data and possible tolerances on the solutions. Each estimate may involve an accuracy of 2 or 3 dB, and the cumulative effect of several estimates could produce an over-design or an under-design of a few decibels. The results, however, have been based on many measurements and many experiences with engine noise sources and noise control problems. It is believed that the methods and procedures given in the manual will produce reliable and workable acoustic solutions to most engine installations.

In general, the noise estimation procedure used in this manual will tend to over-estimate the noise of an "average" engine by about 2 or 3 dB, in order to protect the designs against many of the engines that are just slightly noisier than "average." It is believed that this degree of over-estimation will yield noise control designs that will encompass about 85% to 95% of all diesel and gas reciprocating engines and about 75% to 90% of all gas turbine engines. The remaining reciprocating engines may fall as high as 2 or 3 dB above this "design group" (possibly higher for turbocharger air-intake noise, which is usually not a major problem, however), and a few of the remaining gas turbine engines may fall as high as 5 to 10 dB above this design group.

Acoustic designs are rated here in four classes as to their relative ability to meet calculated needs. These four classes are described as follows:

1. A design may be rated *preferred* if it equals or surpasses the noise requirements of the analysis in all frequency bands.

2. A design may be considered *acceptable* if it produces no more than the following noise excesses above the design goal:

   4 dB in the 63 and 125 Hz band,
   3 dB in the 250 Hz band, or
   2 dB in all higher frequency bands.

3. A design should be considered *marginal* if it produces the following noise excesses above the design goal:

   5-7 dB in the 63 and 125 Hz band,

4-6 dB in the 250 Hz band, or
3-5 dB in all higher frequency bands.

4. A design should be considered *unacceptable* if it produces any higher noise excesses than those listed immediately above in the "marginal" category.

The above ratings and tolerances take into account the general accuracy of data fed into the analysis as well as the "average" reaction of people to noise.

A basic philosophy on two points is offered. First, if a design involves a permanent structure, such as a building wall or floor, that is not easily modified after construction, use a conservative approach and avoid relaxing the design decisions. On the other hand, if a design involves a portable or replaceable item, such as a muffler or a movable partition, that can be modified or corrected later if necessary at relatively small extra cost, then, if desired, design compromises might be made. Second, if the client, customer or building owner is willing to support a step-by-step noise reduction program in an attempt to "get-by" with the least expensive design, it might be justifiable to try some compromise decisions in pursuit of an acceptable solution. (It is cautioned that this may not result in the least expensive total solution, however.) On the other hand, if the client does not wish to be bothered with attempts at intermediate solutions, then the conservative approach should be followed.

If a compromise solution is carried out, there is the possibility that additional noise control may be needed later; however, if no complaint arises from the compromise decision, possibly a less expensive solution will have been reached.

It is pointed out here, in addition to being mentioned in Paragraph 5.3, that when several noise sources are present, their total noise must be considered in any complete evaluation. If just three noise sources, say, the casing, inlet and exhaust of a turbine engine, were each controlled to yield alone an NC-25 at a critical neighboring location, the total noise of those three sources might equal an NC-30 condition. Thus, when multiple sources or paths of noise are present, allow adequate excess noise control to assure that the final total still meets the desired criterion goal.

Section 8

VIBRATION ISOLATION OF RECIPROCATING ENGINES

In this section, vibration isolation of reciprocating engines is discussed for two general locations: (1) on an on-grade slab, such as in a basement or ground floor location, and (2) on an upper-floor of a multi-floor building. All suggestions given here are based on acoustical considerations only; these are not intended to represent structural design requirements. These suggestions apply to both the engine and all attached equipment driven by the engine and apply equally to rotary- or reciprocating-action driven equipment. It is assumed that the mechanical engineer, structural engineer, or equipment manufacturer will specify a stiff, integral base assembly for the mounting of the equipment and that all equipment will be properly aligned. The base assembly should be stiff enough to permit mounting of the entire equipment load on individual point supports, such as "soft" steel springs.

8.1 On-Grade Location

There is no limit on the size of the reciprocating engine that can be installed at an on-grade location.

It is conventional practice, in the installation of large gas or diesel reciprocating engines, to install the engine assembly directly and securely onto a large concrete inertia block having a weight of about 2 to 5 times the total weight of the supported load and having sufficient stiffness to provide the desired degree of alignment. The engine manufacturer will usually specify the minimum dimensions for the concrete block and the maximum tolerances for alignment.

It is recommended here that an engine assembly on-grade be installed rigidly on a large and massive concrete inertia block having a weight not less than twice the weight of the total supported load; even greater weight is desirable. This mass adds stability to the machine and reduces the vibration of the entire assembly. The following schedule is offered as a guide for *vibration isolation* of the total "assembly," which is here taken to consist of the engine, the driven equipment, and the concrete inertia block (here called "engine base") to which these pieces are rigidly mounted. The term "engine assembly" denotes the engine and all the equipment that it drives. Distances named in this schedule may be either horizontal or vertical distances in the earth, in a building structure or in a combination of both.

A. For engines at or under 600 rpm *or* over 1,200 hp:

   1. No vibration isolation of the assembly is required if there is no Category 1 (Table 1) area within 500 feet, or no Category 2 or 3 area within 250 feet, or no Category 4 or 5 area within 150 feet of the engine base. It is good practice, nevertheless, to give the engine base its own footings, separated from the footings of the engine room, with a structural break between the floor slab or floor grille of the engine room and the engine base.

   2. For distances closer than those listed immediately above, for the indicated Categories of Table 1, the engine base should be supported on steel spring vibration isolation mounts that have a static deflection[1] of at least 2 inches for engine speeds of 301 to 600 rpm or at least 3 inches for engine speeds of 200 to 300 rpm. For any isolated base, there must be no rigid structural connections between the engine assembly and the building proper.

   3. In addition, the steel springs listed above should rest on pads of compressed glass fiber, ribbed or waffle-pattern neoprene, if the engine assembly is located within 200 feet of a Category 1 area, within 100 feet of a Category 2 or 3 area or within 50 feet of a Category 4 or 5 area. (It is assumed throughout this schedule that feelable vibration is acceptable in Category 6 areas. If this is not an acceptable assumption, Category 6 should be considered along with Categories 4 and 5.)

B. For engines above 600 rpm and under 1,200 hp:

   1. No vibration isolation of the assembly is required if there is no Category 1 area within 300 feet, no Category 2 or 3 area within 150 feet, or no Category 4 or 5 area within 75 feet of the engine base. It is good practice, nevertheless, to give the engine base its own footings, separated from the footings of the engine room, with a structural break between the floor slab or floor grille of the engine room and the engine base.

   2. For distances closer than those listed immediately above, for the indicated Categories of Table 1, the engine base should be supported on steel spring vibration isolation mounts that have a static deflection of at least 2 inches for engine speeds of 601 to 1,200 rpm or at least 1 inch for engine speeds above 1,200 rpm. For any isolated base, there must be no rigid structural connections between the engine assembly and the building proper.

   3. In addition, the steel springs listed above should rest on pads of compressed glass fiber or ribbed or waffle-pattern neoprene, if the engine assembly is located within 200 feet of a Category 1 area, within 100 feet of a Category 2 or 3 area or within 50 feet of a Category 4 or 5 area. (It is assumed throughout this schedule that feelable vibration is acceptable in Category 6 areas. If this is not an acceptable assumption, Category 6 should be considered along with Categories 4 and 5.)

C. For engines above 1,200 rpm and under 400 hp:

   1. All vibration isolation conditions are

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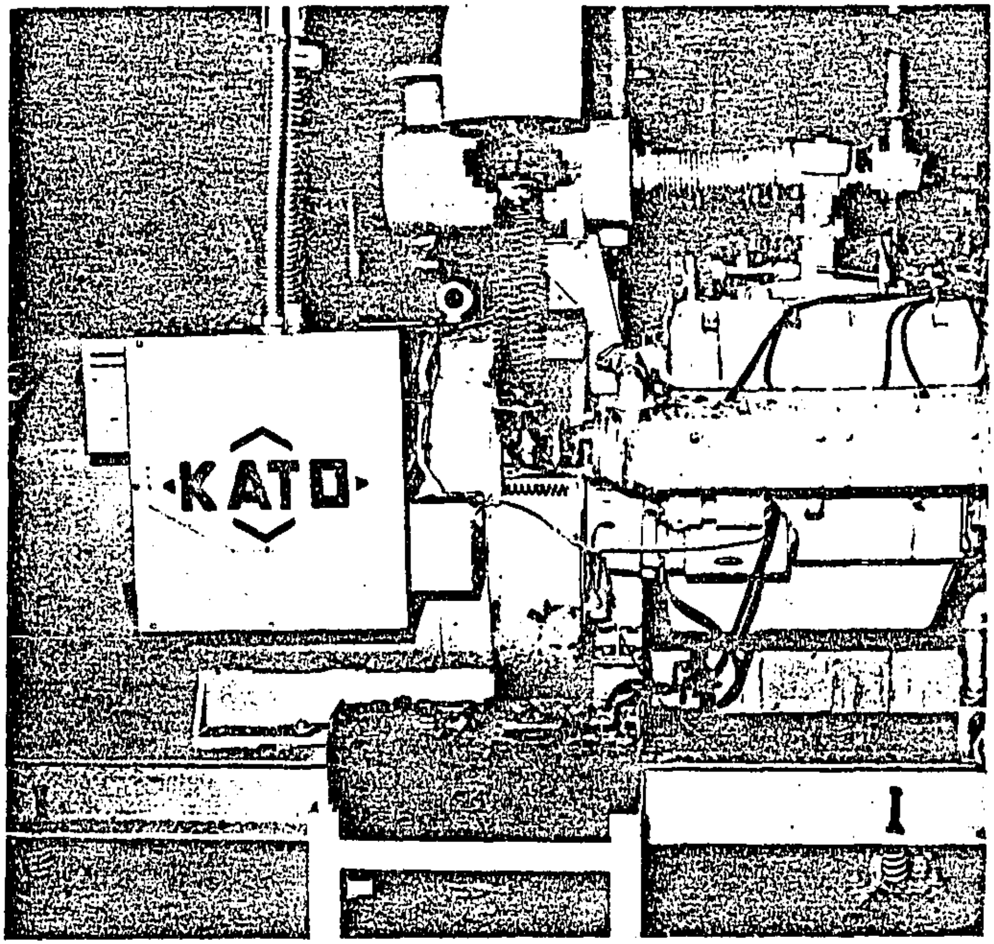
[1] "Static deflection" of an isolation mount is the distance that the mount will compress or deflect when the full weight of the load is applied. The mounts should be selected and located so that all mounts in a given system compress approximately uniformly all around the base and each mount meets the required minimum static deflection. The isolation mount manufacturer will usually recommend specific mounts with his submittal on a job.

the same as in Group B immediately above, except that the concrete inertia block can be eliminated, if desired. If the concrete block is eliminated, a substantial housekeeping pad should be provided under the engine assembly, and the engine assembly should be mounted on a rigid steel frame that is stiff enough to be supported off the floor on individual steel spring isolators without introducing stability or alignment problems. Even here, the concrete inertia block is recommended as a means of providing this stability and stiffness, but it is not absolutely necessary. Items 1, 2 and 3 of Group B above still apply throughout.

D. In this basic guide, wherever vibration isolation of the assembly is required, the following suggestions also apply:

1. Ribbed or waffle-type neoprene pads



Gas reciprocating engine-generator set provides power for research facility. Engine assembly is thoroughly vibration-isolated from on-grade slab, and all piping and conduit have flexible connections. Note control room viewing window in rear wall. *(Photograph is courtesy of Northern Illinois Gas Co.)*

should be made up of three layers of the conventional pad-type material, giving a total thickness of approximately 1 inch of neoprene, with adjacent pads separated by sheet steel plates of approximately 1/16 inch to 1/8 inch thickness. Compressed glass fiber pads should have a thickness of at least 1 inch. The area of the pads should be such as to provide the surface loading recommended by the pad manufacturer. For critical locations, provision should be made to permit replacement of the pads after about 25 years, as the pad material may deteriorate by that time.

2. For an isolated engine assembly there should be no structural, rigid connections between the engine assembly and the building proper. This includes piping, conduit, and ducts to and from the assembly.
    a. A long bellows-type thermal expansion joint in the exhaust piping meets this requirement, as does a flexible connection in the inlet-air ducting to the engine.
    b. Piping to the engine assembly may contain long flexible connections (length at least 6 times the outside diameter of the piping) that are not short-circuited by steel bars that bridge the flanges of the flexible connections; or piping may be used without flexible connections, if the piping is supported on vibration isolation hangers or mounts for a distance along the pipe of at least 100 pipe diameters. The vibration isolation hangers should have a static deflection of at least one-half the static deflection of the mounts that support the engine base. If steel springs are used in the pipe hangers, neoprene or compressed glass fiber pads should be in series with the springs.
    c. Electrical bus bars from the generator should either contain a 6-foot length of braided, flexible conductor across the vibration isolation joint, or the rigid bars should be supported from resilient hangers for a distance of about 25 feet from the isolated assembly.
3. Where steel springs are used, unhoused stable steel springs are preferred. If housed or enclosed springs must be used, special attention must be given to the alignment of the mounts so that they do not tilt or bind in any direction inside their housings. Further, there should be some visual means to check the spring mount in its final position to be certain that binding or tilting does not take place.

8.2 Upper-Floor Location

It is strongly suggested that no reciprocating engine rated over about 1,000 hp or no reciprocating engine operating at a speed lower than about 900 rpm be installed in an upper-floor location. There are no known installations beyond these limits that give rise to these restrictions, but there is concern that the low-speed reciprocating action of very large engines might excite some of the resonances of various building structural elements and produce unwanted noise or vibration, even if the engine is vibration isolated. Then, too, a large concrete inertia block for such a large engine produces structural support problems for an upper-floor location. If engines above 1,000 hp or below 900 rpm must be located in an upper-floor location, it is advisable to engage the services of an acoustical consultant to assist with the design.

The following suggestions apply for any building containing Category 1-5 occupancy areas (*see* Table 1).

1. The entire engine assembly should be mounted rigidly to a concrete inertia block having a weight at least 3 times the total weight of the supported load. The concrete inertia block may be eliminated, if desired, for any engine of less than 100 hp that is located two or more floors away from a Category 1 or 2 area, or that is not located directly over a Category 3

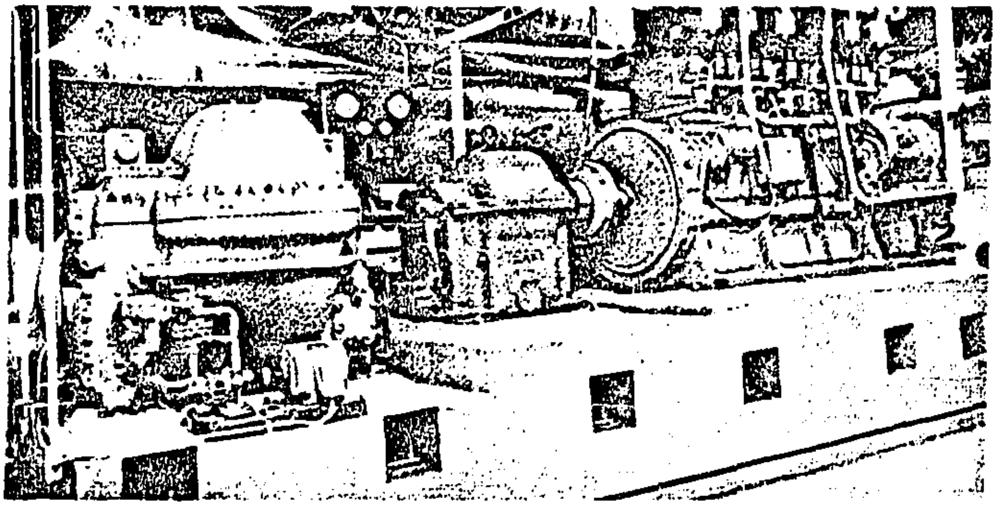
area on the floor beneath the engine room floor. These exceptions assume, nevertheless, that proper airborne noise control is provided in accordance with earlier parts of this manual. If a Type 5 floor-ceiling is required (this involves a "floated floor" over the structure floor—*see* Paragraph 5.8), an inertia block is required under the engine assembly because the inertia block also serves partially as a sound barrier immediately under the engine.

When the concrete inertia block is used, it should be thick enough to assure stiffness and good alignment to the entire assembly. Its area should be at least as large as the overall area of the equipment that it supports. If the engine drives a refrigeration compressor which is connected directly to its evaporator and condenser cylinders, all this equipment should be mounted together onto the same concrete block. The bottom of the inertia block should rest at least 4 inches above the top of the structure slab. If a Type 5 floating-floor slab is involved, this 4-inch air space under the concrete inertia block should be covered with 2-inch-thick low-cost glass fiber or mineral wool as first mentioned under the Type 5 floor-ceiling combination.

If a concrete inertia block is not used, a substantial housekeeping pad should be provided under the engine assembly and the engine assembly should be mounted on a rigid steel frame that is stiff enough to be supported off the floor on individual steel spring isolators without introducing stability or alignment problems.

2. The concrete inertia block or the stiff steel frame of Item 1 above should be supported off the structure floor slab with steel spring vibration isolation mounts. The static deflection of the steel springs should be at least 3 inches for engines in the speed range of 900-1,200 rpm that are mounted on concrete inertia blocks and at least 2 inches for engines above 1,200 rpm mounted on concrete inertia blocks. Any engine mounted on a steel frame (in accordance with Item 1 above) should be supported off the floor on 2-inch static deflection steel springs.

3. Each steel spring should rest on a pad of



Waukesha gas engine drives refrigeration system equipment in this top-floor installation above office suites. Integral concrete inertia block is isolated from floor with steel springs in series with compressed glass fiber pads. *(Photograph is courtesy of Consolidated Kinetics Corporation.)*

compressed glass fiber, or ribbed or waffle-pattern neoprene.

4. To give a stiff support to a reciprocating engine, either with or without a concrete inertia block, the structural floor slab should be at least 10 inches thick and made up of dense concrete (140-150 lb/cu ft). In addition, the engine assembly should be located over primary or secondary beams supporting the floor slab, if at all possible.

5. All suggestions listed under Group D of Paragraph 8.1 apply here also.

If an engine assembly is mounted on the upper floor of a building containing only Category 6 occupancy areas, a concrete inertia block is not required for acoustical purposes but a stiff steel frame would be required for alignment purposes. The engine assembly should be vibration isolated on steel springs having at least 2-inch static deflection for 900-1,200 rpm engines or 1-inch static deflection for engine speeds above 1,200 rpm. Compressed glass fiber or neoprene pads are not required under the springs. The recommendations given under Item 2, Group D of Paragraph 8.1 would also apply here except that isolated pipe hangers can be limited to the confines of the engine room.

8.3 Special Situations

In the preceding paragraphs regarding on-grade locations and upper-floor locations, generalizations have been given that cover a wide range of situations. These generalizations will probably provide adequate protection for most typical equipment installations. Obviously, general rules cannot cover all marginal cases or complex situations, however. Thus, if unusual installations are planned or if some of the general suggestions or limitations given here cannot be met, it would be advisable to seek the assistance of an acoustical consultant who has worked on some of the specific installation problems. Vibration problems are sometimes quite complex and unpredictable.

Section 9

VIBRATION ISOLATION OF TURBINE ENGINES

Because of the high-speed rotary-action of the turbine engine, the engine alone does not induce very much vibration into its supporting floor slab. However, the equipment driven by the turbine engine may produce significant vibration. If a turbine engine is used to drive a piece of reciprocating equipment (such as a reciprocating compressor driven through a speed reduction gear), the entire assembly should be vibration isolated in accordance with the Section 8 suggestions for a reciprocating engine, where the speed and power considerations are based on the speed and power of the driven reciprocating device. For such a condition, the structure floor slab for an upper-floor installation should not be less than 8 inches of dense concrete (140-150 lb/cu ft).

For a turbine engine driving an entirely rotary-action load (centrifugal compressor or generator, with or without a gear), suggestions for vibration isolation are given in the paragraphs that follow, and the engine room floor slab should be not less than 6 inches thickness of dense concrete.

9.1 On-Grade Location

1. No vibration isolation of the assembly is required if there is no Category 1 area (*see* Table 1) within 200 feet, no Category 2 or 3 area within 100 feet, or no Category 4 or 5 area within 50 feet of the engine assembly. These distances should be measured along either horizontal or vertical directions in the earth, in building structures or in any combination of earth and/or structural paths.

2. If the engine assembly is located closer to critical areas than the distances just given, the complete assembly should be supported off the floor slab on pads of 2-inch-thick compressed glass fiber or on pads made up of at least three layers of ribbed or waffle-pattern neoprene. Sheet metal spacers may be used, but are not required between adjacent layers of neoprene. Pipes, ducts and conduit to the assembly either should contain flexible connections or be supported from compressed glass fiber or neoprene-in-shear pipe hangers for a distance of at least 25 feet from the engine. The turbine engine manufacturer must approve the isolation mounting of the assembly because alignment is very critical for a turbine engine.

9.2 Upper-Floor Location

1. An engine assembly located on an upper floor (any floor above grade) of a multifloor building having Category 1 or 2 occupancy areas should be vibration isolated at least in accordance with Item 2 above, under Paragraph 9.1, including pipe, conduit and duct isolation.

2. If a Category 1 or 2 area is within about 30 feet, by structural path distances, of the engine room (either above, below or beside the engine room), stable steel springs having 1-inch static deflection should be inserted in the isolation mounting arrangement between the engine assembly and the glass fiber or neoprene pads on the floor.

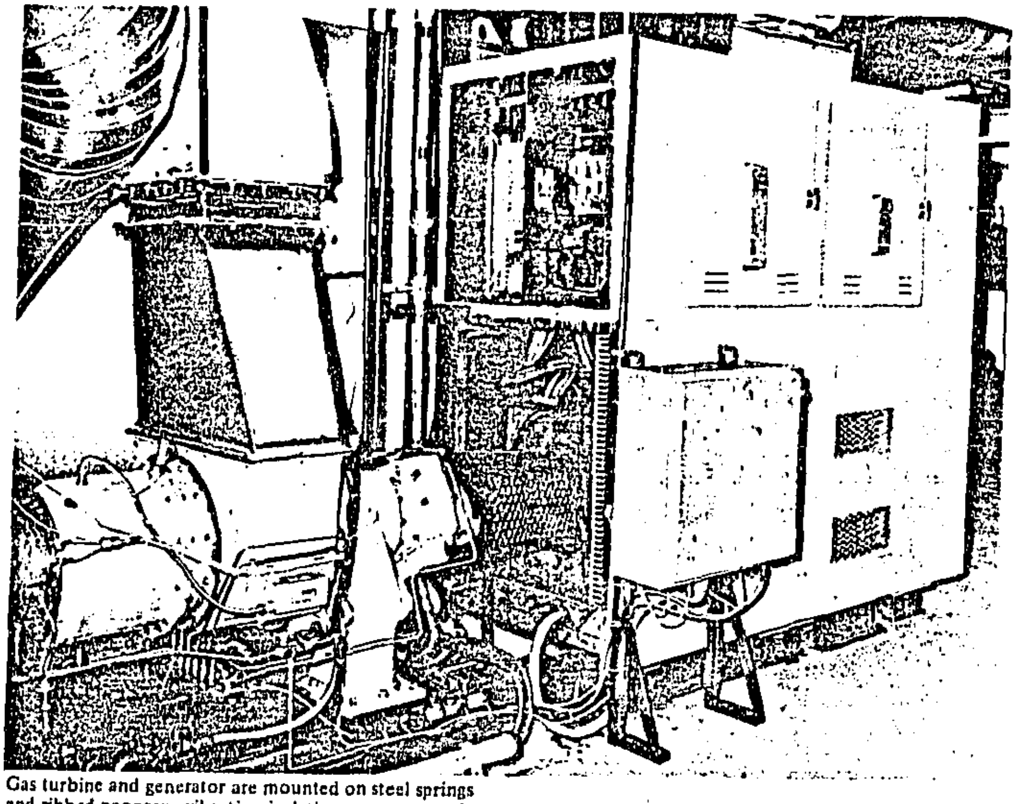
3. If the building contains only Category 3-6 occupancy areas and a Category 3 or 4 area is within about 40 feet or a Category

5 area is within about 20 feet of the engine room, by combined horizontal and vertical structural path distances, the engine assembly should be vibration isolated in accordance with Item 2 of Paragraph 9.1, including pipe, conduit, and duct isolation.

4. If a Type 5 floor-ceiling combination (*see* Paragraph 5.8) is required for air-borne sound control to the floor beneath the engine room, a concrete inertia block is required under the engine assembly. The concrete should be at least 8 inches thick and it should be supported off the structure floor with 1-inch static deflection steel springs in series with 1-inch thickness of ribbed or waffle-pattern neoprene or compressed glass fiber pads. The engine assembly should be mounted rigidly onto this concrete inertia block and there should be at least 3-inch air space between the bottom of the inertia block and the top of the structure slab.

5. No vibration isolation of the turbine engine assembly is required if the building occupancy areas and distances from possible critical areas to the engine room are not covered by Items 1-4 immediately above.

For any of these upper-floor installations, proper airborne sound control should be provided in accordance with earlier parts of the manual, and the turbine engine manufacturer must approve the vibration isolation mounting of the installation.



Gas turbine and generator are mounted on steel springs and ribbed neoprene vibration-isolation mounts to reduce noise transmission to office area and executive conference room immediately below engine room. *(Photograph is courtesy of Northern Illinois Gas Company.)*

Section 10

SUMMARY

This summary is intended primarily to serve as a reminder to not only be aware of, but seek out carefully each noise source and each noise path and provide for each an appropriate noise or vibration control treatment where necessary.

Each engine has three principal noise sources (the casing, the air intake and the gas exhaust). The casing noise, and sometimes the air-intake noise, is radiated directly into the engine room, and the engine room may adjoin a critical area of a building. The procedures of the manual will show the design options available for containing the noise with the use of suitable walls and floor-ceiling combinations, depending on the noise criterion requirements of the adjoining areas.

The engine exhaust noise, and sometimes the air intake noise and casing noise, is radiated out-of-doors and may be heard by the neighbors, depending on the distance away and the nature of the noise criteria applicable to the neighbors. Certain noise reduction effects may be determined, when they are applicable to the situation, and the need or the effectiveness of intake and exhaust mufflers can be calculated from the procedures given in the manual.

In some situations, different neighbors may have different noise criteria at different distances. It may be necessary to work out tentative solutions on paper for each critical neighbor and then to select the design solution that will encompass all neighbor situations. A thorough analysis must not ignore any of the noise sources, paths, or receivers.

Finally, vibration isolation is an important requirement for almost any engine assembly located near a critical occupancy area. Although vibration data are not generally susceptible to calculation and prediction, various vibration isolation suggestions are keyed into engine size, speed and location.

Just as a bucket with many holes will leak water until all the holes are stopped up, so a noisy engine will leak noise until all its paths are identified and treated. This manual has attempted to help identify and quantify the noises so that they can be located and treated even before the building is built or the engine installed. Although some of these engines are admittedly noisy, their noise can be controlled quite easily when the right steps are taken.

Data Sheet 1

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	1,000	2,000	4,000	8,000

4. Base PWL from Table 3 for Item 1 Rating:

5. Speed Correction from Table 3 for Item 2 Speed:

6. Fuel Correction from Table 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: Observe algebraic signs in combining items!

Data Sheet 2

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	1,000	2,000	4,000	8,000

4. Base PWL from Table 4 for Item 1 Rating:

5. Turbocharger Correction from Table 4 for Item 2 Air Intake:

6. Exhaust Pipe Length Correction from Table 4 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: Observe algebraic signs in combining items!

Data Sheet 3

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	1,000	2,000	4,000	8,000

3. Base PWL from Table 5 for Item 1 Rating:

4. Inlet Air Duct Length Correction from Table 5 for Item 2 Length:

5. Estimated PWL (in dB re 10^{-12} watt) of Untreated Turbocharger Noise: (Item 5 = Item 3 + Item 4)
Caution: Observe algebraic signs in combining items!

Data Sheet 4

Sheet 1 of 2

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 6

3. Exhaust Duct: Round ☐, Rectangular ☐, Length______________ft.

   Duct Lining: Type______________from Table 9

   Duct Turns: No.______________Type______________from Table 10

4. Intake Duct: Round ☐, Rectangular ☐, Length______________ft.

   Duct Lining: Type______________from Table 9

   Duct Turns: No.______________Type______________from Table 10

5. Exhaust Muffler Supplied: Yes ☐ No ☐

6. Intake Muffler Supplied: Yes ☐ No ☐

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

7. PWL (in dB re 10^{-12} watt) of Casing Noise from Table 6 for Item 1 Rating:

8. Noise Reduction Provided by Casing Cover (If Any) of Item 2 as Estimated in Table 6 Footnote:

9. PWL of Casing Noise with Cover (If Any): Item 9 = Item 7 + Item 8
   Caution: Observe algebraic signs in combining items!

(Continued)

Data Sheet 4 (continued) *Sheet 2 of 2*

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

10. PWL (indB re 10^{-12} watt) of Exhaust Noise from Table 7 for Item 1 Rating:

11. Attenuation Provided by Exhaust Duct and Turns of Item 3, from Tables 9 and 10 or "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: Observe algebraic signs in combining items!

14. PWL (in dB re 10^{-12} watt) of Air-Intake Noise from Table 8 for Item 1 Rating:

15. Attenuation Provided by Intake Duct and Turns of Item 4, from Tables 9 and 10 or "ASHRAE Guide":

16. Insertion Loss of Intake Muffler of Item 6, If Provided (use muffler manufacturer's data for appropriate bands; see Table 39 for example):

17. PWL of Intake Noise Out of Duct and Muffler (as Applicable)
Item 17 = Item 14 + Item 15 + Item 16
Caution: Observe algebraic signs in combining items!

Data Sheet 5

Estimated Sound Pressure Level (SPL) in an Engine Room

1. Volume of Engine Room:_______ft. X _______ft. X _______ft. = _______cu. ft.

2. Total Interior Surface Area of Room (including floor) = _________ sq. ft.

3. Area of Acoustic Treatment ________ sq. ft.

4. Per cent Area Covered by Acoustic Treatment (Item 3/Item 2) X 100 = __________ %

5. Thickness of Absorption Material: 3/4 in.–1 in. ☐ 1-1/2 in.–2 in. ☐

6. Acoustic Treatment Condition of Room from Items 4 and 5, as Defined in Footnotes of Table 11:
Condition 1 ☐ Condition 2 ☐ Condition 3 ☐ Condition 4 ☐

7. Room Correction Term from Table 11 for Items 1 and 6: ______________ dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

8. Total PWL of All Noise Sources in Engine Room (from Item 7 of Data Sheet 1 for Reciprocating Engines or from Item 9 of Data Sheet 4 for Turbine Engines):

9. Reverberant SPL in Engine Room (Item 9 = Item 8 + Item 7; Item 7 Value Is Same for All Frequency Bands and Is Zero or a Negative Quantity for All Room Conditions)
Caution: Observe algebraic signs in combining terms!

10. Approximate SPL Near Engine (Item 10 = Item 9 + 3 dB)

Data Sheet 6

Sound Transmission to Adjoining Room Through Common Wall

Sound Transmitting Room ________________ Sound Receiving Room ________________

1. Area of Common Wall That Transmits Noise: ____________ sq. ft.

2. Total Interior Surface Area of Receiving Room (including floor): ____________ sq. ft.

3. Total Interior Surface Area Divided by Area of Common Wall: __________
(Item 3 = Item 2 ÷ Item 1)

4. Area of Acoustic Treatment in Receiving Room*: ____________ sq. ft.

5. Per cent Area of Receiving Room Covered by Acoustic Treatment ________ %
(Item 5 = 100 X Item 4 ÷ Item 2)

6. Thickness of Absorption Material: 3/4 in.–1 in. ☐ 1-1/2 in.–2 in. ☐

7. Acoustic Treatment Condition of Receiving Room from Items 5 and 6, as Defined in Footnotes of Table 23:
Condition 1 ☐, Condition 2 ☐, Condition 3 ☐, Condition 4 ☐

8. Wall Correction Term "C" from Table 23 for Item 3 and Item 7 Conditions: ____________ dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

9. Expected Wall Construction Material: ____

10. Estimated "TL" of Wall from Tables 13-22 or Other Source

11. Estimated "NR" of Wall in Equation $NR = TL + C$, Using Values from Items 8 and 10
(Item 11 = Item 10 + Item 8):
Caution: Observe algebraic signs in combining items!

12. Estimated SPL in Sound Transmitting Room from Data Sheet 5 (Use Item 9 Reverberant SPL If Source Is 5 ft. or More from Common Wall or Item 10 Close-in SPL If Source Is Less Than 5 ft. from Common Wall).

13. Estimated SPL in Receiving Room (Item 13 = Item 12 – Item 11)

---

*Add 50% of floor area to Item 4 if receiving room floor is carpeted or if room has drapes or upholstered furniture. If this is the only acoustic material in the room, treat it as having 3/4 in. thickness in Item 6 for determining Item 7 condition.

Data Sheet 7

Sound Transmission to Adjoining Room Through Common Floor-Ceiling

Sound Transmitting Room ____________________ Sound Receiving Room ____________________

1. Area of Common Floor-Ceiling That Transmits Noise:______________sq. ft.

2. Total Interior Surface Area of Receiving Room (including floor):______________sq. ft.

3. Total Interior Surface Area Divided by Area of Common Floor-Ceiling:
(Item 3 = Item 2 ÷ Item 1) ___________

4. Area of Acoustic Treatment in Receiving Room*:______________ sq. ft.

5. Per cent Area of Receiving Room Covered by Acoustic Treatment
(Item 5 = 100 X Item 4 ÷ Item 2)______________ %

6. Thickness of Absorption Material: 3/4 in.–1 in. ☐ 1-1/2 in.–2 in. ☐

7. Acoustic Treatment Condition of Receiving Room from Items 5 and 6, as Defined in Footnotes of Table 23:
Condition 1 ☐, Condition 2 ☐, Condition 3 ☐, Condition 4 ☐

8. Floor Correction Term "C" from Table 23 for Item 3 and Item 7 Conditions:______________dB

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

9. Approximate Floor-Ceiling Type (from Paragraph 5.8 of Text):
Type 1 ☐ Type 2 ☐ Type 3 ☐ Type 4 ☐ Type 5 ☐

10. Approximate "TL" of Floor-Ceiling Type of Item 9 for Nearest Applicable Dimensions, from Tables 25-29 (Interpolate Between TL Values Shown, If Desired, But Do Not Exceed TL Values Shown in Right-Hand Column of Tables 25-28):

11. Estimated "NR" of Floor-Ceiling in Equation $NR = TL + C$, Using Values of Items 8 and 10
(Item 11 = Item 10 + Item 8):
Caution: Observe algebraic signs in combining items!

12. Estimated SPL in Sound Transmitting Room from Data Sheet 5 (Use Item 9 Reverberant SPL for Floor-Ceiling Above the Engine or Item 10 Close-in SPL for Floor Under Engine):

13. Estimated SPL in Receiving Room (Item 13 = Item 12 – Item 11)

---

*Add 50% of floor area to Item 4 if receiving room floor is carpeted or if room has drapes or upholstered furniture. If this is the only acoustic material in the room, treat it as having 3/4 in. thickness in Item 6 for determining Item 7 condition.

Data Sheet 8

Estimated Outdoor Sound Pressure Level (SPL)
Due to an Outdoor Sound Source PWL

1. Distance to Noise Source:_______________ ft.

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

2. Total PWL of All Outdoor Noise Sources at Source Position:

3. Outdoor Distance Term from Table 30 for Item 1 Distance:

4. Approximate Outdoor SPL at Distance of Item 1: (Item 4 = Item 2 – Item 3)
Caution: Observe algebraic signs in combining items!

Data Sheet 9

Estimated Indoor SPL at Neighboring Location, Including All Noise Reduction Contributions

Noise Source Considered in This Data Sheet: ____________________

Frequency Band in Hz							
63	125	250	500	1,000	2,000	4,000	8,000

1. Outdoor SPL at Neighboring Location for No Noise Reduction Contributions (from Item 4 of Data Sheet 8):

2. Noise Reduction from Outdoors to Indoors Provided by Neighbor Building, from Table 31:

3. Noise Reduction Provided by Sound Barrier, from Table 32:

4. Noise Reduction Provided by Woods, from Table 33:

5. Tentative Indoor SPL to This Point (Item 5 = Item 1 – Item 2 – Item 3 – Item 4):
Caution: Observe algebraic signs in combining items!

6. Directivity Effect of Stack Opening, If Applicable, from Table 34.

7. Tentative Indoor SPL to This Point (Item 7 = Item 5 + Item 6)
Caution: Observe algebraic signs in combining items!

8. Approximate Noise Reduction of Muffler Planned for Use with Noise Source, If Applicable, from Tables 35-37 or from Muffler Data Supplied by Manufacturer.

9. Estimated Indoor SPL at Neighboring Location, Due to This Noise Source (Item 9 = Item 7 – Item 8)

TABLE 1

Category Classification for Power Plant Noise as Heard in Various Indoor Functional Activity Areas

Category	Area (and Acoustic Requirements)	Noise Criterion Designation
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 2

Octave Band Sound Pressure Level (SPL) Values Associated with the Noise Criterion Designations of Table 1

Noise Criterion Designation	SPL (in dB re 0.0002 microbar) in Octave Frequency Band 63 Hz	125 Hz	250 Hz	500 Hz	1,000 Hz	2,000 Hz	4,000 Hz	8,000 Hz
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 3

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**	**1,000 Hz**	**2,000 Hz**	**4,000 Hz**	**8,000 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
950–1,499	113	117	117	116	116	115	109	102
1,500–2,399	115	119	119	118	118	117	111	104
2,400–3,800	117	121	121	120	120	119	113	106

For Engine Speed:	**Speed Correction (in all bands)**
Under 600 rpm	–5 dB
600–1,500 rpm	–2 dB
Above 1,500 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

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*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 4

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	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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
950–1,499	140	146	142	134	130	124	114	106
1,500–2,399	142	148	144	136	132	126	116	108
2,400–3,800	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 5

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	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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
950–1,499	99	97	97	98	101	102	101	93
1,500–2,399	100	98	98	99	102	103	102	94
2,400–3,800	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 6

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	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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
850–1,299	114	116	117	117	117	117	117	117
1,300–1,999	115	117	118	118	118	118	118	118
2,000–3,299	116	118	119	119	119	119	119	119
3,300–5,000	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 lightweight 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 7

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	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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
850–1,299	126	128	128	127	125	123	119	113
1,300–1,999	128	130	130	129	127	125	121	115
2,000–3,299	130	132	132	131	129	127	123	117
3,300–5,000	132	134	134	133	131	129	125	119

TABLE 8

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	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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
850–1,299	111	112	112	115	121	126	126	123
1,300–1,999	114	115	115	118	124	129	129	126
2,000–3,299	117	118	118	121	127	132	132	129
3,300–5,000	120	121	121	124	130	135	135	132

TABLE 9

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")

Values given are in dB attenuation per foot of duct length, for the eight octave bands from 63 Hz to 8,000 Hz. These values assume duct cross dimensions in the region of 3 to 5 feet. 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.

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.

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

Round Duct:	0.10	0.07	0.03	0.02	0.02	0.02	0.02	0.02
Rectangular:	0.20	0.14	0.07	0.05	0.04	0.04	0.04	0.04

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

Round Duct:	0.15	0.09	0.04	0.02	0.02	0.02	0.02	0.02
Rectangular:	0.30	0.20	0.09	0.05	0.04	0.04	0.04	0.04

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

Round Duct:	0.15	0.16	0.22	0.6	0.8	0.8	0.6	0.4
Rectangular:	0.30	0.25	0.25	0.7	0.8	0.8	0.6	0.4

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

Round Duct:	0.20	0.20	0.3	0.9	1.1	1.1	0.9	0.6
Rectangular:	0.30	0.28	0.4	0.9	1.1	1.1	0.9	0.6

TABLE 10

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")

Values given are in dB attenuation (or insertion loss) for the eight octave bands from 63 Hz to 8,000 Hz. These values assume duct cross dimension in the region of 3 to 5 feet. 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.

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.

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 feet 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 inches (dB per turn):

1 4 5 4 3 3 3 3

Type 3. For ducting with continuous 1-inch or 2-inch-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-inch or 2-inch-thick internal acoustic absorption duct lining, 90° square turn of rectangular duct, with turning vanes of chord length less than 6 inches (dB per turn):

1-inch lining:	1	5	6	6	7	8	9	10
2-inch lining:	1	6	7	8	9	10	11	12

TABLE 11

Room Correction Term for Converting PWL of a Source into the Reverberant SPL in a Room

SPL = PWL + Room Correction Term

Volume of Room (cu ft)	Room Correction Term Acoustic Treatment of Room Condition 1	Condition 2	Condition 3	Condition 4
Under 1,000	0 dB	0 dB	− 3 dB	− 6 dB
1,000– 2,100	0	− 3	− 6	− 9
2,200– 4,600	− 2	− 5	− 8	−11
4,700– 9,900	− 4	− 7	−10	−13
10,000– 21,000	− 6	− 9	−12	−15
22,000– 46,000	− 8	−11	−14	−17
47,000–100,000	−10	−13	−16	−19
Over 100,000	−12	−15	−18	−21

Condition 1: No significant amount of sound absorption material (less than that of Condition 2).

Condition 2: 10–25% of total room area covered with 3/4 in.–1 in. thick sound absorption material.

Condition 3: 26–50% of total room area covered with 3/4 in.–1 in. thick sound absorption material; or 10–30% of total room area covered with 1-1/2 in.–2 in. thick sound absorption material.

Condition 4: Over 50% of total room area covered with 3/4 in.–1 in. thick sound absorption material; or over 30% of total room area covered with 1-1/2 in.–2 in. thick sound absorption material.

TABLE 12

Rules for Adding SPL or PWL Contributions by "dB Addition"

1. For adding 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, they may be added as follows:

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

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

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

TABLE 13

Approximate Transmission Loss (in dB) of Dense* Poured Concrete or Solid-Core Concrete Block or Masonry

(140-150 lb/cu ft density)

Octave Frequency Band (Hz)	Thickness of Concrete or Masonry (inches)					
	4	6	8	10	12	16
	Approximate Surface Weight (lb/sq ft)					
	48	72	96	120	144	192
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
1,000	42	46	50	52	54	56
2,000	49	53	56	58	59	61
4,000	55	58	61	63	64	66
8,000	60	63	66	68	69	70

---

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

TABLE 14

Approximate Transmission Loss (in dB) of Hollow-Core Dense* Concrete Block or Masonry

Octave Frequency Band (Hz)	Thickness of Concrete Block (inches)					
	4	6	8	10	12	16
	Approximate Surface Weight (lb/sq ft)					
	28	36	44	52	60	76
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
1,000	37	39	41	43	45	48
2,000	42	46	48	50	52	55
4,000	49	52	54	56	58	60
8,000	55	57	59	61	63	65

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

TABLE 15

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]
63	15	17	1
125	20	22	2
250	26	30	3
500	34	38	4
1,000	40	44	4
2,000	45	47	5
4,000	43	45	5
8,000	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] 2 x 4 wood studs on 16 in. centers, nailed to 2 x 4 wood plates; 5/8 in. thick gypsum board nailed on both sides of studs; fill and tape joints and edges.

[3] 2 x 4 wood studs staggered on 2 x 6 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.

[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 16

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 Inch Thick	6 Inch Thick
63	22	29	35
125	26	33	37
250	31	36	39
500	36	42	46
1,000	43	47	50
2,000	48	53	56
4,000	50	56	61
8,000	52	59	65

---

[1] Constructed of two 16 gage 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 17

Approximate Transmission Loss (in dB) of Glass* Walls or Windows (13 lb/sq ft/in. Surface Density)

Octave Frequency Band (Hz)	Thickness of Glass (inches)			
	1/8	1/4	1/2	3/4
	Approximate Surface Weight (lb/sq ft)			
	1-1/2	3	6-1/2	10
63	5	11	17	20
125	11	17	23	24
250	17	23	25	25
500	23	25	26	27
1,000	25	26	27	28
2,000	26	27	28	29
4,000	27	28	30	33
8,000	28	30	36	39

---

*Special laminated safety glass ("Acousta-Pane," Amerada Glass Corporation) 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 18

Approximate Transmission Loss (in dB) of a Few Typical Double-Glass Windows*

Octave Frequency Band (Hz)	Glass-Air Space-Glass Thicknesses (inches) 1/4–1/4–1/4	1/4–1-1/2–1/4	1/4–6–1/4
63	18	19	20
125	23	23	24
250	24	25	28
500	24	27	31
1,000	26	31	37
2,000	28	34	40
4,000	30	37	43
8,000	36	42	46

---

*Thermal insulation double-glass windows typically have 1/4 in. to 1 in. sealed air space between 1/4 in. to 3/8 in. 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 19

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 (inches)				
	1/4	1/2	1	2[2]	4
	Approximate Surface Weight (lb/sq ft)				
	1	2	4	8	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
1,000	18	19	20	26	32
2,000	19	20	26	32	37
4,000	20	26	32	37	41
8,000	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 20

Approximate Transmission Loss (in dB) of Dense* Plaster (9 lb/sq ft/in. Surface Density)

Octave Frequency Band (Hz)	Thickness of Plaster (inches)				
	1/2	3/4	1	1-1/2	2
	Approximate Surface Weight (lb/sq ft)				
	4-1/2	7	9	13	18
63	15	18	21	24	26
125	21	24	26	27	27
250	26	27	27	28	28
500	27	28	28	29	29
1,000	28	29	29	30	33
2,000	29	30	33	37	40
4,000	33	37	40	44	47
8,000	40	44	47	50	53

---

*If lightweight, non-porous plaster is used, these TL values may be used for equal values of surface weight. These data *must not be used* 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 15 for the applicable stud construction.

TABLE 21

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
63	1	7	13
125	7	13	19
250	13	19	23
500	19	23	25
1,000	23	25	26
2,000	25	26	27
4,000	26	27	28
8,000	27	28	32

TABLE 22

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
63	9	15	21
125	15	21	27
250	21	27	33
500	27	33	38
1,000	33	38	39
2,000	38	39	39
4,000	39	39	37
8,000	39	37	40

TABLE 23

Approximate Wall or Floor Correction Term "C" For Use in the Relationship $NR = TL + "C"$

Total Surface Area Inside Receiving Room Divided by Area of Common Wall or Floor	Acoustic Treatment of Receiving Room Condition 1	Condition 2	Condition 3	Condition 4
1.4– 2.7	10 dB	–7 dB	–4 dB	–2 dB
2.8– 5.5	- 7	–4	–2	+1
5.6–10	- 4	–2	+1	+3
11 –21	- 2	+1	+3	+4
22 –43	+ 1	+3	+4	+5
44 –80	+ 3	+4	+5	+6

Condition 1: No significant amount of sound absorption material (less than that of Condition 2).

Condition 2: 10–25% of total room area covered with 3/4 in.–1 in. thick sound absorption material.

Condition 3: 26–50% of total room area covered with 3/4 in.–1 in. thick sound absorption material; or 10–30% of total room area covered with 1-1/2 in.–2 in. thick sound absorption material.

Condition 4: Over 50% of total room area covered with 3/4 in.–1 in. thick sound absorption material; or over 30% of total room area covered with 1-1/2 in.–2 in. thick sound absorption material.

TABLE 24

Approximate Transmission Loss of a Wall Containing Doors or Windows

I Door or Window Area as Percent of Total Wall Area	II If TL of Door or Window Is Less Than TL of Wall by	III 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

Example: The TL of a filled metal partition at 125 Hz is 26 dB (Table 16). An 1/8-inch-thick window, accounting for 40% of the over-all partition area, has a TL of +11 dB (Table 17). What is the TL of the composite partition?

Solution: Table 24 indicates that for a 40% window area (Column I) and a $TL_{wall} - TL_{door}$ of 15 dB, [26 – 11 = 15](fourth line of Column II), the TL of the composite wall = 26 – 11 = 15 dB (fourth line of Column III).

TABLE 25

Approximate Transmission Loss (in dB) of Type 1 Floor-Ceiling Combination

(See Paragraph 5.8 of 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
63	33	34	35	36
125	35	36	37	38
250	36	38	40	41
500	40	43	45	47
1,000	46	50	52	54
2,000	53	56	58	59
4,000	58	61	63	64
8,000	63	66	68	69

TABLE 26

Approximate Transmission Loss (in dB) of Some Type 2 Floor-Ceiling Combinations

(See Paragraph 5.8 of 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.)			
	16	18	24	24
63	35	37	39	40
125	38	40	42	43
250	40	43	46	47
500	45	49	52	54
1,000	52	57	60	62
2,000	59	63	66	67
4,000	64	68	71	72
8,000	69	73	76	77

TABLE 27

Approximate Transmission Loss (in dB) of Some Type 3 Floor-Ceiling Combinations

(See Paragraph 5.8 of 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
63	38	40	42	43
125	41	43	45	46
250	43	46	49	50
500	48	52	55	57
1,000	55	60	63	65
2,000	62	66	69	70
4,000	67	71	74	75
8,000	72	76	79	80

TABLE 28

Approximate Transmission Loss (in dB) of Some Type 4 Floor-Ceiling Combinations

(See Paragraph 5.8 of 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
63	41	43	46	48
125	45	47	50	53
250	47	50	54	57
500	52	56	60	64
1,000	59	64	68	72
2,000	66	70	74	77
4,000	71	75	78	82
8,000	76	80	84	87

TABLE 29

Approximate Transmission Loss (in dB) of Some Type 5 Floor-Ceiling Combinations

(See Paragraph 5.8 of test for description of Type 5)

For Floating Floor Slab of 3 Inches Thickness Supported Resiliently 2 Inches Above Structure Slab: Add 3 dB to Table 28 Values

For Floating Floor Slab of 4 Inches Thickness Supported Resiliently 2 Inches Above Structure Slab: Add 4 dB to Table 28 Values

For Floating Floor Slab of 5 Inches Thickness Supported Resiliently 2 Inches Above Structure Slab: Add 5 dB to Table 28 Values

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TABLE 30

Outdoor Distance Term for Determining SPL at Any Distance from a Non-Directional Unobstructed Outdoor Sound Source PWL

SPL = PWL – Outdoor Distance Term

Distance (ft)	Outdoor Distance Term					
	63, 125, 250 Hz	500 Hz	1,000 Hz	2,000 Hz	4,000 Hz	8,000 Hz
10–11	18	18	18	18	18	18
12–14	20	20	20	20	20	20
15–18	22	22	22	22	22	22
19–23	24	24	24	24	24	24
24–29	26	26	26	26	26	26
30–37	28	28	28	28	28	28
38–47	30	30	30	30	30	30
48–59	32	32	32	32	32	32
60–75	34	34	34	34	34	34
76–94	36	36	36	36	36	36
95–111	38	38	38	38	38	38
112–140	40	40	40	40	41	42
141–177	42	42	42	42	43	44
178–223	44	44	44	45	46	47
224–281	46	46	46	47	48	50
282–355	48	48	48	49	50	53
356–447	50	50	51	51	53	56
448–563	52	52	53	54	56	59
564–711	54	54	55	56	59	63
712–899	56	57	57	58	62	67
900–1,110	58	59	59	61	66	72
1,120–1,400	60	61	62	64	70	78
1,410–1,770	62	63	64	67	75	85
1,780–2,230	64	65	67	70	79	93
2,240–2,810	66	68	70	74	85	102
2,820–3,550	68	70	72	77	92	114
3,560–4,470	70	73	76	82	101	128
4,480–5,630	72	76	79	87	111	145
5,640–7,200	74	78	83	93	123	165

TABLE 31

Approximate Noise Reduction of Outside Noise Provided by Typical Exterior Wall Construction with Open and Closed Windows

Octave Frequency Band (Hz)	Walls with Windows Open[1] (dB)	Walls with Air Vents Open[2] (dB)	Walls with Closed Windows[3] (dB)
63	9	13	19
125	10	14	20
250	11	15	22
500	12	16	24
1,000	13	17	26
2,000	14	18	28
4,000	15	19	30
8,000	16	20	30

---

[1] Any typical wall construction, with open windows covering about 5% of exterior wall area.

[2] Any typical wall construction, with small open air vents of about 1% of exterior wall area.

[3] Any typical wall construction, with closed windows covering about 10-20% of exterior wall area.

TABLE 33

Approximate Noise Reduction (in dB) Provided by Dense Woods

(Mixed deciduous and evergreen trees; 20–40 foot height, visibility penetration of 70 to 100 feet)

Octave Frequency Band (Hz)	Excess Attenuation (in dB per 100 feet of woods)
63	1/2
125	1
250	1-1/2
500	2
1,000	3
2,000	4
4,000	4-1/2
8,000	5

Notes:

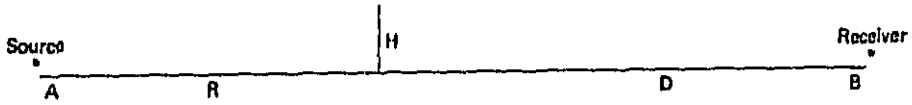
1 For average 10–20 foot height, use one-half the rate given in the table.

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

TABLE 32

Approximate Noise Reduction (in dB) Provided by a Solid Barrier

(Do not go outside of the tabulated range for attenuation in any bands. See text for discussion. Use one-half of attenuation for D greater than 1 mile.)



Ratio H²/R (ft)	Noise Reduction in Frequency Band 63 Hz	125 Hz	250 Hz	500 Hz	1,000 Hz	2,000 Hz	4,000 Hz	8,000 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 34

Approximate Directivity Effect (in dB) of a Large Vertical Exhaust Stack Compared to a Non-Directional Source of the Same Power

(See paragraph 6.6 of text for discussion)

Approximate Dimensions of Stack Opening (ft) and Octave Band (Hz)	Relative SPL for Indicated Angle from Vertical Axis				
	0° (vert.)	45°	60°	90°* (horiz.)	135°*
Over 10x10 ft					
63-250 Hz	8	5	2	− 3	− 4
500-1,000 Hz	9	6	1	− 9	−11
2,000-8,000 Hz	10	7	−1	−14	−18
3x3 ft to 10x10 ft					
63-250 Hz	4	2	0	− 2	− 3
500-1,000 Hz	5	3	−1	− 4	− 7
2,000-8,000 Hz	6	4	−2	− 8	−12
Under 3x3 ft					
63-250 Hz	0	0	0	0	0
500-1,000 Hz	0	0	0	− 1	− 2
2,000-8,000 Hz	0	0	0	− 2	− 4

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*For air intake openings, values in the 90° and 135° columns can be increased by 50%.

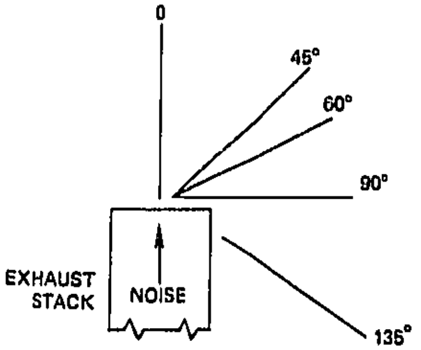

TABLE 35

Approximate Noise Reduction (in dB) of Typical Reactive Mufflers Used with Reciprocating Engines

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
1,000	17	20	25
2,000	15	19	24
4,000	14	18	23
8,000	14	17	23
	"Low Pressure-Drop Line"		
63	10	15	20
125	15	20	25
250	13	18	23
500	11	16	21
1,000	10	15	20
2,000	9	14	19
4,000	8	13	18
8,000	8	13	18

Refer to manufacturers' literature for more specific data.

TABLE 36

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 Inch	8 Inch	12 Inch	16 Inch	24 Inch
	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
1,000	56	38	19	16	12
2,000	48	35	13	11	8
4,000	40	26	10	8	6
8,000	20	18	7	6	4

Refer to manufacturers' literature for more specific data.

TABLE 37

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
1,000	27	23	24	−25
2,000	22	19	19	−22
4,000	18	14	14	−17
8,000	15	10	10	−14

Refer to manufacturers' literature for more specific data.

TABLE 38

"Area Factor" ("AF") for Use in Determining the PWL of an Area "A" That Transmits Sound of Level SPL

(See paragraph 6.11 of text for discussion)

$$\begin{aligned} PWL &= SPL + (10 \log A - 10) \\ &= SPL + \text{“}AF\text{”} \; (\text{in dB re } 10^{-12}\ w) \end{aligned}$$

Area "A" (sq. ft.)	"AF" (dB)	Area "A" (sq. ft.)	"AF" (dB)
1.0	−10	10	0
1.25	−9	12.5	1
1.6	−8	16	2
2.0	−7	20	3
2.5	−6	25	4
3.2	−5	32	5
4.0	−4	40	6
5.0	−3	50	7
6.3	−2	63	8
8.0	−1	80	9

TABLE 39

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
1,000	15	19	22
2,000	16	20	24
4,000	14	18	22
8,000	9	14	18
	"High Pressure-Drop Class"		
63	8	11	13
125	10	14	18
250	15	23	30
500	23	32	40
1,000	30	38	44
2,000	35	42	48
4,000	28	36	42
8,000	23	30	36

Refer to manufacturers' literature for more specific data.