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DEVELOPMENT OF A MECHANICAL EQUIPMENT NOISE-CONTROL PERMIT SCHEME FOR MODEL BUILDING CODE

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7. AUTHOR(S)
Warren E. Blazier, Jr.   Nicholas P. Miller
David A. Towers

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Bolt Beranek and Newman Inc.
50 Moulton Street
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16. ABSTRACT

This report deals with the development of noise-control approaches applicable to a mechanical equipment permit scheme for commercial, business, institutional, and residential high-rise buildings.

The report identifies and categorizes mechanical equipment as noise sources. The equipment is classified and rank-ordered on the basis of potential noise impact. A series of worksheets and guidelines are presented for use in performing the calculations necessary to evaluate a given mechanical design for exterior noise impact.

A mechanical equipment permit scheme is explained for use by a community. In developing this scheme the authors first examined the strengths and weaknesses of enforcement strategies currently being used by various jurisdictions throughout the country. These existing strategies are evaluated in terms of effectiveness, feasibility, enforcement costs and legal provisions. In making an evaluation emphasis is placed upon these enforcement practices which increase the probability that mechanical-equipment noise will be controlled.

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1. INTRODUCTION

This report is concerned with the development of noise-control approaches applicable to a mechanical equipment permit scheme for commercial, business, institutional, and residential high-rise buildings. The application to low- and medium-rise structures has also been considered. This project has been confined to those aspects of noise control associated with building mechanical equipment and systems that can influence the quality of the exterior environment in the vicinity of the structure.

It is anticipated that the mechanical equipment permit scheme will be an integral part of the U. S. Environmental Protection Agency (EPA) model building code for noise control. The model building code and mechanical equipment permit scheme will be valuable tools which communities can use to construct their own building codes and permit approaches suited to local needs and conditions.

Sections 2 and 3 of the report deal with the identification and categorizing of equipment as noise sources; how these can be classified and rank-ordered on the basis of potential noise impact.

Section 4 discusses the sources for the data base from which the generalized noise characteristics of each equipment category were established in terms of magnitude and frequency structure, as a function of size or other operational parameters.

In Sections 5, 6, and 7, the technical procedures are developed for calculating the magnitude of noise produced by specific items of equipment, within each category, as a function of the details concerning mechanical performance which ordinarily are available from building plans and specifications.

Sections 8 and 9 contain procedures for calculating the noise level at an assigned point of reference outdoors due to mechanical equipment and systems serving the building that is being screened for permit purposes.

Section 10 contains a series of worksheets and guidelines for their use in performing the calculations necessary to evaluate a given mechanical design for exterior noise impact. A discussion of the potential for errors and the uncertainties that affect the reliability of calculated results is also included.

Having developed the procedure and worksheets for calculating mechanical-equipment noise levels at specific points of reference Section 11 discusses how the procedure might be incorporated into a mechanical equipment permit scheme. A recommended scheme is developed by first examining the strengths and weaknesses of enforcement strategies currently being used by various jurisdictions across the country. These existing strategies are evaluated in terms of effectiveness, feasibility, and enforcement costs. This evaluation identifies the importance of legal provisions; but more importantly shows which enforcement practices increase and which decrease the probability that mechanical-equipment noise will be controlled.

2. IDENTIFICATION OF MECHANICAL EQUIPMENT NOISE SOURCES IN BUILDINGS THAT CAN SIGNIFICANTLY INFLUENCE THE EXTERIOR ENVIRONMENT

The identification of building mechanical equipment that can potentially impact the exterior noise environment depends on four factors:

1. The noise power produced by the equipment.
2. The location of the equipment (i.e. whether it is outdoors or indoors).
3. The noise transmission path from the equipment source to receiver.
4. The ambient noise level at the receiver location due to other sources.

2.1 Outdoor Equipment

Equipment normally located outdoors (such as a cooling tower, for example) usually will have a greater potential impact on the exterior environment, for a given noise power, than equipment that is normally located indoors; the sound attenuation between source and receiver at comparable distances is typically less. This generalization leads to the grouping of building mechanical equipment on the basis of normal installation location, when classifying specific items with regard to potential noise impact.

If the analysis is restricted to machinery associated with building heating, ventilating, or air-conditioning systems (HVAC), the equipment most likely to be located outdoors may be identified as follows:

1. Cooling Towers and Evaporative Condensers
2. Air-Cooled Condensers (with or without integral compressors)
3. Packaged Roof-Top HVAC Units
4. Exhaust Fans
5. Room Air Conditioners (condenser side)

In addition to the above, outdoor equipment may include transformers, emergency power generator sets, circulating pumps, etc., although these items are more generally located indoors in built-up urban areas.

2.2 Indoor Equipment

With regard to equipment normally located indoors but coupled to the outdoors through ventilation openings or ducts, the following items will most frequently be of concern:

1. Air Return/Exhaust Fans (discharge side)
2. Air Supply Fans (intake side)
3. Centrifugal Chillers
4. Reciprocating Chillers
5. Refrigeration Compressors
6. Air Compressors
7. Transformer Substation Equipment
8. Emergency Power Generators
9. Radiator Fans
10. Circulating Pumps

3. CATEGORIZATION OF EQUIPMENT NOISE SOURCES

By far the most frequently encountered environmental noise problems associated with building mechanical systems are with fans. Most building equipment, with the possible exception of pumps and compressors, contain fans of one type or another; the noise of a cooling tower or condensing unit, for example, is principally that due to the fan components. For this reason, considerable emphasis has been placed on the prediction scheme for fan-noise evaluation in developing the technical backup for noise-estimation procedures presented in Section 10 of this report.

Compressor equipment establishes another important category of building noise sources; these are frequently located in mechanical spaces or penthouses that contain large areas of openings to the exterior for ventilation. Of particular significance are reciprocating and centrifugal compressors used in both packaged and built-up water/brine chillers for air-conditioning or refrigeration-system applications. Air compressors are of less significance in most commercial or residential high-rise buildings; they are generally small in capacity because their use is primarily for powering pneumatic control devices in HVAC systems. However, air compressors used in laboratory processing and pneumatic conveyor applications can be very significant in their potential for noise impact on the surrounding environment.

Electrical substation transformers rated at or above 500 KVA must also be considered. There are electrical substations serving most large buildings that contain transformers and cooling fans which are typically either located outdoors or in vaults. Those in vaults are vented to the outside through openings in the building either at grade level or from a sub-grade level through sidewalk grilles.

A less significant category of building noise sources includes pumps, electric motors, emergency power systems, and miscellaneous building support equipment. The noise problems traceable to motors are usually caused by the motor-cooling fan. Pump noise is hard to separate from that of the drive motor except in cases where a strong tone is generated at the impeller blade-passage frequency. Emergency power generators can be very noisy. However, in normal times, the units are typically operated only for test during a 30-minute interval on a once-a-week basis; test operations are generally scheduled to minimize noise impact. However, we are aware of several instances (a hospital in San Diego, for example) where emergency power systems are used during peak-load periods, on a daily basis, to supplement the power available from the local utility; in these cases, "emergency" generator systems can have significant noise impact.

Thus, there are four broad categories of building-equipment noise sources, summarized below in decreasing order of significance:

1. Fan systems and fans used as components in packaged equipment (including boilers).
2. Compressor equipment.
3. Transformer substations.
4. Pumps, motors, and emergency power units (when operated only for test).

4. CHARACTERIZATION OF BUILDING MECHANICAL EQUIPMENT NOISE

4.1 Sources for the Data Base

A complete description of the noise produced by a particular piece of mechanical equipment requires the detailed measurement and analysis of the following characteristics:

1. Noise magnitude (sound power level or sound pressure level at a reference distance).
2. Frequency Spectrum (whether predominantly broadband, predominantly pure-tone, or, a mixture of the two).
3. Temporal variations in magnitude and frequency distribution that occur with changes in operating conditions, operating points, or sudden changes due to cycling of components.

At the present time such detailed data on the noise of specific products rarely, if ever, appear in a manufacturer's catalog (although they may exist as part of a product improvement or development program). Part of the reason is the manufacturer's concern that the building mechanical designer would have great difficulty in evaluating such detailed information in terms of his own requirements; he might easily become confused, possibly reaching the wrong conclusion, when making comparisons with competitive products whose catalog information on noise is "played down" as a potential source of problems.

In equipment areas where no industry standards have yet been adopted, the completeness of an individual manufacturer's published data on product noise tends to match that of the competition. Furthermore, unless some form of certification requirement exists, the published data can become so "massaged" that any significant differences between competing products tend to vanish. Thus, some uncertainty exists about the reliability of published catalog data in product areas where industry standards for noise rating either do not exist or are not subject to certification or other forms of "policing."

Where industry standards pertaining to equipment noise do exist, it is important to distinguish between standards for noise measurement and standards for noise rating. A further distinction is required between certified noise ratings and ratings "based" on industry standards.

For example, the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), in the role of a technical society, over the years has developed or adopted standards for the *measurement* of equipment noise in several HVAC product areas.[1] The American National Standards Institute (ANSI) is also developing standards for noise measurement that cover a broader scope of equipment categories.[2] However, these standards deal primarily with the problems of noise measurement and do not address the question of noise rating (how to process and interpret measured data).

In practice, industry trade associations such as the Air-Conditioning and Refrigeration Institute (ARI) and the Air Moving and Conditioning Association (AMCA) assume the responsibility for developing rating standards. (In some cases it may also include a measurement procedure.) These standards are available for use by member companies and others, as a basis for the product noise-rating information that is supplied to the field.[3,4,5]

In a few instances, these rating standards are used as the basis for certification programs within the trade associations, whereby member companies can measure and publish product noise ratings that are backed up by independent laboratory testing of random samples drawn from the participating manufacturers. At the present time, however, few of these certification programs apply to products in the size ranges of concern in commercial or residential high-rise buildings.[6]

More generally, the test and rating standards that presently exist are used by manufacturers on an optional basis to describe product noise, *without* the requirement for certification at the trade-association level. Presumably a member company may be asked by the assocation to verify the capability of his *laboratory* to meet the requirements embodied in the test standard, should the procedure be specifically referenced in the published data. However, in the case of non-member companies and others who state that their tests are "based" on a particular standard, the quality of the published data is subject to wide variation in credibility.

Consequently, many of the data obtained from manufacturer's catalogs and other published sources during this Project have required careful screening and interpretation for applicability in characterizing noise sources of equipment used in buildings. These data have been supplemented by measurements made by Bolt Beranek and Newman (BBN) on other related projects[7] and by new data obtained in field surveys conducted specifically for this Project.

In addition, ARI and several of their member companies undertook a cooperative program to furnish the Project with current data in several equipment categories for which little information has yet appeared in the literature.

4.2 Selection of Scheme for Classifying Equipment Noise

From the outset, we have taken the position that the technical review process, which is required in support of a mechanical equipment permit scheme, must be of minimum complexity if it is to find acceptance at the Building-Official level or that of the Builder/Mechanical Designer. True! Therefore, the task has been one of developing a procedure that is simple enough to follow with a minimum of experience in acoustical engineering, but yet is capable of providing answers that are meaningful and dependable.

A second factor of importance is that the current trend and format of environmental noise-control criteria should strongly influence the metric chosen for the technical review process incorporated in the mechanical equipment permit scheme.

Taking these two factors into account, a strong case can be made for using A-weighted sound levels, rather than octave-band or one-third-octave-band levels, for the numerical output of the technical review process. Once this conclusion is reached, two other decisions are required:

1. The use of an A-weighted sound level to describe mechanical noise does not differentiate between a piece of equipment whose spectrum contains significant narrowband or pure-tone components and one whose spectrum is principally broadband in character. For example, a power transformer vs. a fan.

   There is also a concern that the A-weighted level does not properly classify machinery such as large air compressors and vibration-induced noise radiation from rotating equipment because the principal energy is typically at low frequencies.

These concerns are serious if the question of subjective response is to be adequately addressed. However, considering that the present trend in setting limits on environmental noise is to use a metric that is essentially only "level-sensitive," there is no justification for making the ratings of machinery noise more complex than the criteria that are likely to be used to evaluate it.

2. If the *output* of the technical procedure that is embodied in the permit scheme is to be an A-weighted sound level, the input information must in some way refer to the shape of the frequency spectrum.

   This information is required because many noise-control elements that can appear in the path between source and receiver are frequency-sensitive in their performance. For example, the noise of a vane-axial fan peaks about one-octave higher in the frequency spectrum than a centrifugal fan. Because the attenuation provided by most commercial sound traps increases significantly with frequency over the range of interest, the resulting A-weighted noise reduction is generally greater when used with vane-axial as opposed to centrifugal fans.

Taking all of these considerations into account, the simplest base for classifying mechanical-equipment noise that retains sufficient information about the source for use in the permit scheme requires two elements.

1. A-Weighted Sound Power Level (the use of A-weighted sound pressure level at a reference distance is also acceptable in some situations).

2. Identification of the spectrum shape, for example, by comparison with several generalized reference spectra.

4.3 Classifications of Mechanical Equipment Noise

4.3.1 Sound Power Level vs. Sound Pressure Level

The argument generally advanced for using Power Level rather than Pressure Level in quantifying equipment noise is that the power radiated is not dependent on the acoustical properties of the surroundings, whereas the pressure depends on distance, room absorption, or the geometry of the installation.

For simple point sources (and ones whose power output is not appreciably affected by the radiation impedance presented by the test environment) there is a classical relationship between power and pressure that permits using *either* descriptor, with only small uncertainties in the translation from one to the other. In the classical relationship, the sound pressure level is assumed to drop off at the rate of 6 dB per doubling of distance, initially, until the reflected energy begins to dominate; at some point the level becomes constant and independent of greater distances.

However, for larger sources outdoors, or for a distribution of several sources over an extended surface, the sound pressure in the vicinity of the source is not related to the radiated power by the simple inverse-square distance law; and in the indoor situation, there is growing evidence that the relationship between the radiated sound power and the resulting sound pressure at a point in the room does *not* follow the direct/reverberant field relationship generally assumed in current practice. For the actual situation, in typical mechanical rooms, the drop in sound pressure level is observed to be closer to 3 dB per doubling of distance and the "constant" reverberant field level cannot be found.

This is particularly true for an extended source such as a large piece of equipment and in "cluttered" rooms where the absorptive treatment is concentrated on a single surface, such as the ceiling. In these situations it is more reliable to estimate the sound pressure level at some desired distance, based on the sound level measured close to the machine (such as one meter), rather than to calculate it using a known sound power level and the classical reverberant room equation.

Thus, there is no simple answer to the question of whether sound power level or sound pressure level should be used for the input data in the technical review procedure of the Mechanical Equipment Permit Scheme. In some situations greater reliability will result by using sound pressure level at a reference distance; in others, the use of sound power level will be preferable.

4.3.1.1 *Outdoor Equipment*

In the case of outdoor equipment, the use of sound power level for source characterization is the better choice in most instances. If sound pressure level is used, then the reference distance should be large with respect to the source dimensions (at least one, but preferably two to three times the major equipment dimension).

4.3.1.2 *Indoor Equipment*

The use of sound power level for source characterization is recommended for all fan equipment and for certain small packaged refrigeration or air-compressor equipment; this is consistent with the present trend in industry standards for noise rating.

We recommend the use of sound pressure level at a distance such as 3 feet (or 1 meter) for large machinery such as centrifugal chillers, transformer substation equipment, etc; this is also consistent with the present trend in industry rating standards.*

4.3.2 Classification by Spectrum Shapes

The use of an A-weighted metric for machinery noise must be supplemented by some identification of the typical frequency spectrum associated with the equipment, in order to compute the attenuation provided by frequency-sensitive noise-control devices.

We have reviewed many noise data representing a number of equipment categories to determine how best to classify differences in the frequency spectra encountered. Since pure-tone components and other narrowband characteristics cannot be dealt with (for the reasons discussed above) the review was confined to an analysis of spectral distributions in octave bands.

We found, if the octave-band noise spectra are first adjusted by A-weighting and then normalized at 500 Hz or 1000 Hz, that a plateau region exists, typically three-octaves wide, where the levels are nominally constant; and also that these three bands can be identified with specific types of equipment. From this evolved a classification scheme that requires only four characteristic spectra for identifying frequency content. A possible exception is with transformers where significant differences exist between fan-cooled and radiant-cooled equipment.

The four classifications are as follows: (The octave-band levels, adjusted by A-weighting, are typically constant within ±1 dB in the indicated bands.)

Class I - 250, 500, 1000 Hz
Class II - 500, 1000, 2000 Hz
Class III - 1000, 2000, 4000 Hz
Class IVA - 125, 250, 500 Hz (radiant-cooled transformers)
IVB - 125, 250, 500, 1000 Hz (fan-cooled transformers)

Pure Tones?

It was found that the (logarithmic) sum of A-weighted octave-band levels at the center frequencies indicated was generally within 1 dBA of the value for the entire spectrum. Thus, a feasible scheme for rating equipment noise can be established by using the A-weighted sound power (or pressure) level with an added subscript to denote the spectrum classification. For example, $L_W = 105\ dBA_I$ would refer to an A-weighted sound power level of 105 dBA associated with a Class I type spectrum.

4.4 Examples of Equipment with Similar Characteristic Spectrum Shapes

The following equipment lists, grouped by spectrum classification, have been established from an analysis of the available data. The data base came from manufacturers' catalogs, the technical literature, BBN files on previous projects, new field data obtained by BBN, and information recently furnished us by ARI member companies.

1. Class I (250, 500, 1000 Hz A-weighted octave-band levels similar)
    a. Centrifugal Fans (airfoil, backward inclined, forward curved, and modified radial)
    b. Cooling Towers with Propeller Fans
    c. Rotary Screw Compressors
    d. Large Air Compressors

2. Class II (500, 1000, 2000 Hz A-weighted octave-band levels similar)
    a. Vane-Axial Fans
    b. Air-Cooled Condensing Units
    c. Packaged Roof-top HVAC Units
    d. Cooling Towers with Centrifugal Fans
    e. Chillers with Reciprocating Compressors
    f. Chillers with Direct-Drive Hermetic Centrifugal Compressors
    g. Pumps
    h. Electric Motors
    i. Diesel-Engine Generators

3. Class III (1000, 2000, 4000 Hz A-weighted octave-band levels similar)
    a. Chillers with Internally-Geared, Hermetic Centrifugal Compressors
    b. Large Chillers (> 1,000 TR) Both Direct and Gear-Driven

4. Class IVA (125, 250, 500 Hz A-weighted octave-band levels similar)
    a. Transformers (radiant cooled)

5. Class IVB (125, 250, 500, 1000 Hz A-weighted octave-band levels similar)
    a. Transformers (fan-cooled)

The majority of the equipment of concern to the Permit Scheme falls into either the Class I or II characteristic spectrum categories. The noise-reduction elements most frequently used in HVAC systems, such as packaged sound attenuators and duct lining materials, also will most generally be applied with equipment in these spectrum classifications. This suggests that only two A-weighted noise-reduction ratings may be required for such elements.

5. RECOMMENDED PROCEDURE FOR ESTIMATING THE A-WEIGHTED SOUND POWER LEVEL OF FAN EQUIPMENT IN GENERAL

With few exceptions, the A-weighted sound power or sound pressure levels of equipment used in buildings are predictable in terms of a constant (which varies with the specific type of equipment) plus corrections for size and other operational parameters.

In the case of fans, the correction factors are based on air volume, total static pressure, and percent of peak static efficiency at the point of operation. Several types of components, such as condensing units and packaged HVAC units, correlate best with a correction factor based on the cooling capacity. Still others correlate better by a simple relationship to the total horsepower of the drive motors.

5.1 Influence of Static Efficiency on Fan Noise Levels

A number of studies of fan-noise characteristics are found in the literature. Most papers on this subject within the last five to ten years agree that fan noise is predictable on the basis of fan type, air volume, and static pressure; but only a few references can be cited to show the effect of fan efficiency on the noise level actually produced in a field application.[8-12]

As stated previously, air-moving devices can be considered one of the most significant sources of noise in the exterior environment resulting from operation of building mechanical equipment and systems. Thus, we have investigated the potential error in fan-noise estimates likely to be introduced by neglecting the static efficiency at the point of operation on the fan curve. We conclude that this factor *must not* be ignored. The fan may actually be as much as 12-15 dBA noisier than estimated by neglecting static efficiency, if the operating point, for example, should lie within the range of 50 percent - 60 percent of *peak* static efficiency.

Many fan applications in the field operate at points substantially to the right of the peak in the static-efficiency curve. This is the reason that an installation frequently turns out to be noisier than anticipated in design. Although the ASHRAE Handbook, which is the basis for many current design estimates of fan noise, cautions against making a fan selection too far off the point of peak efficiency, no relationship is given to demonstrate the importance of this factor.

To gain an insight to this problem, we queried several major manufacturers for data on noise at operating points other than peak static efficiency (most tests for noise rating are customarily run in the range of peak static efficiency). Although the response was mixed, we were able to obtain enough data to observe the trend of changes in noise level with point of operation on the fan curve. These data were normalized to specific sound power level at *peak static efficiency* and compared with computations based on the method published in the current ASHRAE Handbook;[13] good agreement was found.

A sample of the static-efficiency correlation obtained for airfoil-type fans, using data furnished by two manufacturers, is shown in Figure 1. Note that a prediction based on the current (1976) ASHRAE Handbook would be in agreement at about 95 percent

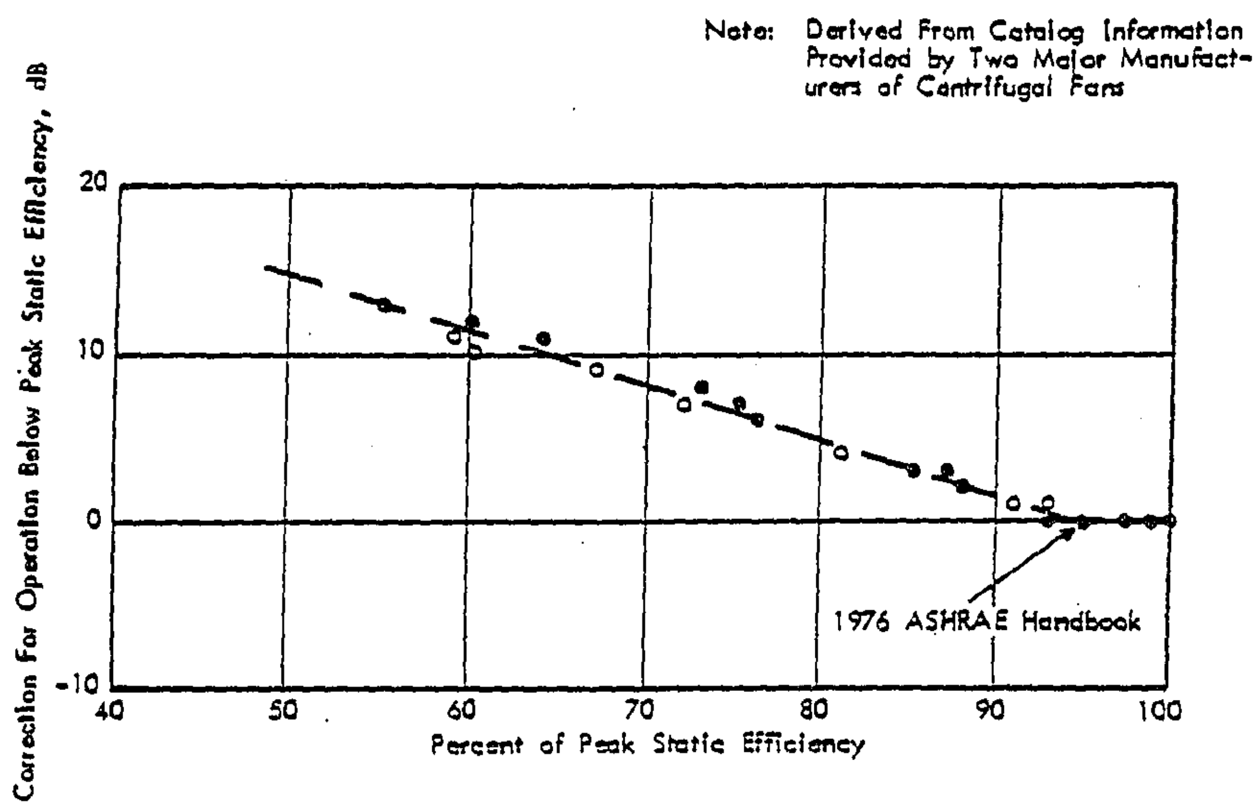

FIGURE 1. - CHANGE IN A - WEIGHTED SOUND POWER LEVEL WITH POINT OF OPERATION ON FAN PERFORMANCE CURVE

peak static efficiency, and significantly underestimates the noise level if the fan actually runs at lower efficiency. The slope of the curve implies that fan-noise level increases 1 dB per 3 points reduction in percent of peak static efficiency. A detailed analysis of the data indicated that these changes with static efficiency are relatively constant over a wide frequency range and thus can be applied to the A-weighted level.*

Thus, it is clear that the estimation of fan-noise power must include a correction for operating point on the fan curve.**

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*One manufacturer who has tested several other types of centrifugal as well as vane-axial fans recommends that a similar correction of this magnitude be applied in all cases. Although the peak static efficiency varies with the type of fan, the correction factor is expressed as a percent of peak efficiency and is assumed to be independent of the absolute peak value. The differences in the absolute noise levels for fans of other types whose characteristic peak efficiency is lower is presumably accounted for in the value identified with the specific sound power level given in the ASHRAE Handbook.

**The implications of this conclusion are an important consideration in energy conservation. A fan operating at low static efficiency requires a significantly greater power input for the same cfm and static pressure than one providing the same duty but a higher efficiency. For a given duty, a lower efficiency fan selection will represent a *lower first cost* (some space saving also) but a *higher operating cost*. However, considering that a low-efficiency fan selection may require also a larger first-cost investment for *noise control*, one should question the decision to use the less efficient fan at all!

5.2 Procedure for Estimating the A-Weighted Sound Power Level

For estimating the A-weighted sound power level of fan equipment, the recommended equation assumes the following form:

$$L_W(A) = K_A + A + B + C \quad \text{dBA}$$

where

$L_W(A)$ = A-weighted Sound Power Level re 10^{-12} Watt
K_A = A-weighted Specific Sound Power Level re 10^{-12} Watt*
A = Correction for Air Volume, dB (10 log cfm)
B = Correction for total static pressure, dB (20 log P_s)
C = Correction for percent of peak static efficiency, dB (108 - 54 log % peak S.E.).

The values of K_A and the spectrum classification for fans of various types are provided in Table 1 below. These values are based on the octave band spectra corresponding to Specific Sound Power Level (as a function of fan type) published in the 1976 ASHRAE Handbook. Adjustments have been made to account for the blade-passage frequency. (Typographical errors present in the Handbook tables have also been corrected.)

The values of the correction factors, A, B, and C, are tabulated in Tables 2, 3, and 4, respectively, for the range of typical operating conditions encountered in practice.

*Specific Sound Power Level is that level corresponding to a fan supplying 1 cfm at 1" w.g. static pressure.

TABLE 1
Fan Noise Estimation Parameters

Fan Type	K_A (dBA)	Spectrum Classification	Per Ton
A. Centrifugal Design			
1. Airfoil Backward Curved Backward Inclined			
a. Wheel diameter, 36" & over*	35	I	
b. Wheel diameter, less than 36"	40	I	
2. Forward Curved (independent of wheel dia. ?)	39	I	
3. Modified Radial			
a. Wheel diameter, 40" & over*	45	I	
b. Wheel diameter, less than 40"	50	I	
B. Vane-Axial Design			
50% Hub/Diameter Ratio			
a. Wheel diameter, 40" & over*	46	II	
b. Wheel diameter, less than 40"	52	II	
C. Propeller (Exhaust or Ventilation Applications)	52	I	*

*Recent data from two fan equipment manufacturers indicate that the specific sound power level is also dependent upon fan diameter; the peak fan efficiency is reduced at wheel diameters below the breakpoint indicated. This fact is not recognized in the 1976 ASHRAE Handbook.

TABLE 2

Correction Factor "A," For Fan Air Volume

Air Volume 1000 CFM	Correction Factor dB
1.0	30
1.3	31
1.6	32
2.0	33
2.5	34
3.2	35
4.0	36
5.0	37
6.0	38
8.0	39
10	40
13	41
16	42
20	43
25	44
32	45
40	46
50	47
60	48
80	49
100	50
130	51
160	52
200	53

TABLE 3

Correction Factor "B," For Fan Static Pressure

Fan Static Pressure Inches, Water Gauge	Correction Factor dB
1.0	0
1.25	2
1.5	4
1.75	5
2.0	6
2.25	7
2.5	8
2.75	9
3.0	10
3.5	11
4.0	12
4.5	13
5.0	14
5.5	15
6	16
7	17
8	18
9	19
10	20
11	21
12	22
14	23
16	24

TABLE 4

Correction Factor "C," For Percent of Peak Static Efficiency

Static Efficiency % of Peak	Correction Factor dB
93-100	0
90-92	1
87-89	2
84-86	3
81-83	4
78-80	5
75-77	6
72-74	7
69-71	8
66-68	9
63-65	10
60-62	11
57-59	12
54-56	13
51-53	14
48-50	15

5.3 Example Illustrating Use of the Procedure to Estimate Sound Power Level of Fans

An example to illustrate the procedure for estimating fan noise power level using Tables 1, 2, 3, and 4 is given below.

5.3.1 Equipment Selection Based on First-Cost Considerations

A building exhaust fan is to handle 60,000 cfm at 2.5" w.g. static pressure. Based on first-cost considerations, the mechanical engineer has selected a 40"-diameter, double-width, double-inlet airfoil fan (40 AF DWDI).

From the catalog performance data it is determined that this particular fan is capable of operating at a peak static efficiency of 78 percent; however, for the selected duty, the point of operation on the fan curve corresponds to a static efficiency of only 44 percent. The percentage of peak static efficiency, therefore, is 56 percent (44 ÷ 78 x 100).

K_A = 35 dBA (Table 1, Airfoil fan > 36")
A = 48 dB (Table 2, correction for 60,000 cfm)
B = 8 dB (Table 3, correction for 2.5" static pressure)
C = 13 dB (Table 4, correction for 56% of peak static efficiency

$$L_W(A) = K_A + A + B + C$$

$$= 35 + 48 + 8 + 13 = 104 \text{ dBA re } 10^{-12} \text{ Watt.}$$

5.3.2 Equipment Selection Based on Optimizing Operating Costs

Consider now a case for similar fan duty (60,000 cfm @ 2.5" P_s), except that the mechanical engineer makes a fan selection based on optimizing mechanical efficiency, thereby reducing operating costs.

The fan selected might be a 60 AF DWDI rather than the 40 AF DWDI chosen in the previous example. The peak static efficiency for this fan would still be 78 percent; however, the point of operation on the fan curve would be at about 70 percent static efficiency. Thus, the percent of peak static efficiency would be 90 percent; and the C-correction term would be only 1 dB. This change to a more efficient fan reduces the sound power level by 12 dBA! This reduction in noise, achieved by choosing an efficient fan, is comparable to the attenuation provided by a 3-foot commercial packaged sound trap. Thus, to offset the increase in noise resulting from the selection of a less efficient fan would require the installation of a sound trap that otherwise might not be necessary.

5.3.3 First Cost vs. Operating Cost

In practice, the choice of a less efficient fan is generally based on lower first cost and smaller space requirements. For the fan and drive-motor selections used in the above examples, the cost difference is about $3900. However, the cost of adding a sound trap to obtain the equivalent noise level would be about $1800. Therefore, the net savings in first cost would be in the range of $2100.

Consider now the impact on long-term operating costs: The less efficient fan will require about 54 brake-horsepower to meet the design operating conditions; the more efficient fan to do the same job requires only 34 brake-horsepower. This is a saving of 20 hp, or 15 kW. At 4 cents/kWh, operating 12 hours/day, 5 1/2 days/week, there is a saving in power costs in excess of

$2000/year. Thus the savings in first cost with the less efficient fan would be offset by operating cost after the first year's operation. In addition, for the local utility providing the power, the 20 hp reduction in motor size for the efficient fan would correspond to a saving of about 106 barrels of oil per year! Impress,

6. RECOMMENDED PROCEDURES FOR ESTIMATING THE A-WEIGHTED SOUND POWER LEVEL OF OUTDOOR EQUIPMENT

We recommend the use of Sound Power Level for expressing the noise magnitude of outdoor equipment. This is consistent with the present trend of industry measurement and rating standards for HVAC equipment. A possible exception is with cooling towers and transformer substations, which, due to their large physical size, are more frequently rated in terms of sound pressure level at some reference distance.

6.1 Packaged HVAC Roof-Top Units

These units typically are self-contained, packaged, HVAC systems composed of three component parts:

1. A cooling section containing one or more reciprocating compressors, plus an air-cooled condenser generally using propeller-type fans.

2. A heating section containing either a gas/oil-fired furnace, or resistance heater elements in conjunction with a reverse-cycle heat pump; the heat pump would utilize the same components described in Item 1 above.

3. A fan section containing a supply fan and a return/exhaust fan.

The exterior noise level generated by these units is typically a mixture of the four principal sources listed below in order of significance.

1. Air-cooled condenser fan
2. Discharge side of return/exhaust fan
3. Intake side of supply fan
4. Compressor and unit casing.

Because very little information on the exterior noise levels of these units could be found in the manufacturers' current catalogs, we sought the assistance of Air-Conditioning and Refrigeration Institute (ARI). A request was made that ARI member companies contribute whatever data were available to an information pool, which would then be used as the basis for bracketing the range of noise levels likely to be encountered in field applications. It was hoped that a data correlation might be found that would permit the estimation of noise levels based on a simple relationship to unit size or capacity.

The Industry responded by making available data on 21 different units within the capacity range of 10-82 Tons Refrigeration (TR).* These data, together with those measured independently by BBN on 11 additional units, were found to correlate with refrigeration capacity in a predictable manner.

The correlation of the data provided by ARI is shown in Figure 2; the best fit is $L_W(A) = 77 + 12 \log (TR)$, dBA, with a standard error of estimate, s, equal to 3 dB. We recommend that noise estimation for use in the Permit Scheme be based on the best-fit curve plus one standard error as follows:

$$L_W(A) = 80 + 12 \log (TR), \text{ dBA};$$

the characteristic frequency spectrum was found to be Class II.

*One ton is equivalent to a capacity of 12,000 Btu/hour.

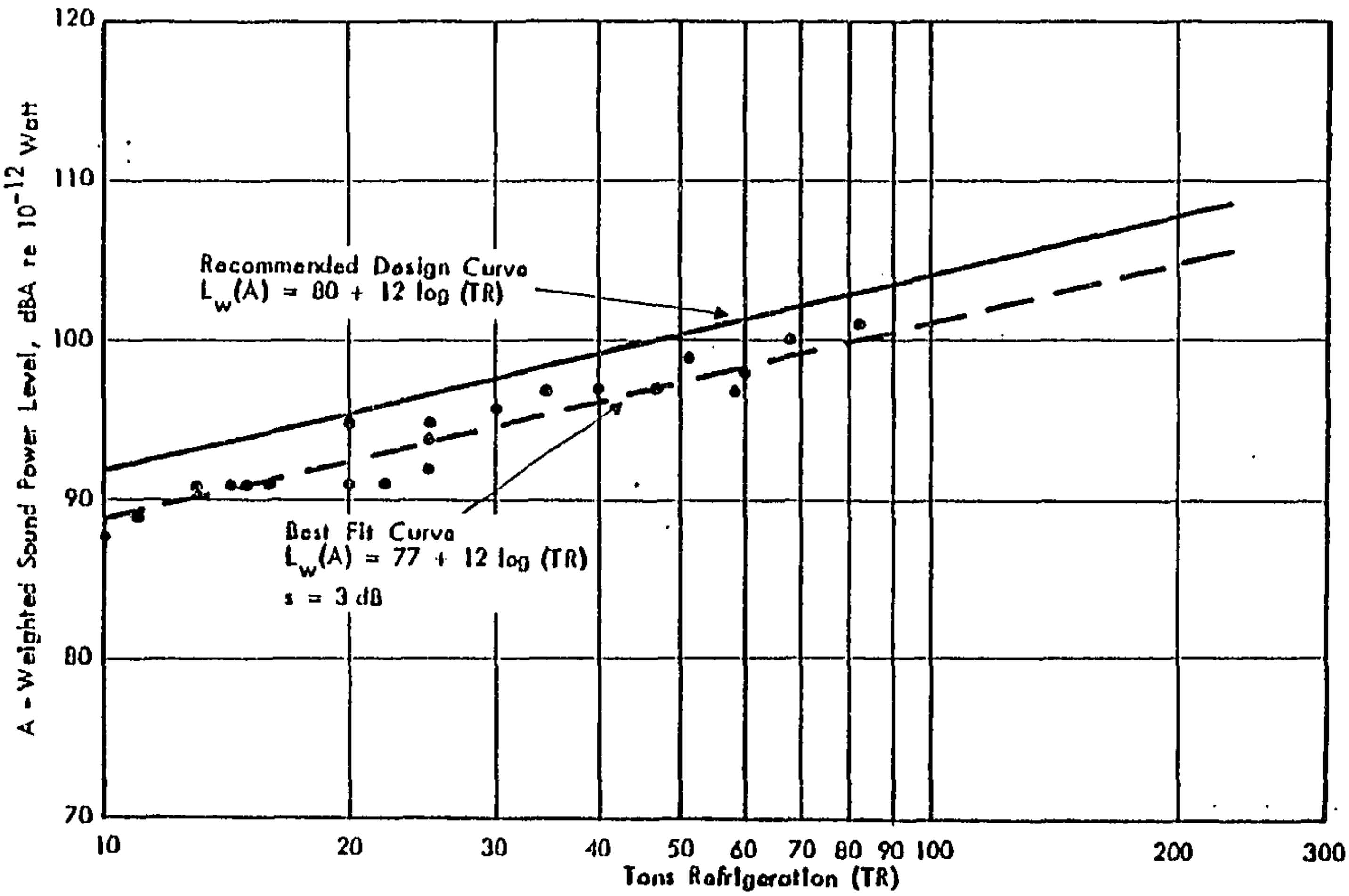


FIGURE 2. NOISE ESTIMATION CURVES FOR ROOF - TOP PACKAGED AIR CONDITIONERS

Table 5A lists the estimated A-Weighted Sound Power Levels in increments of 1 dBA for the capacity range of 10-200 TR.

6.2 Air-Cooled Condensing Units/Chillers

These units are components of many HVAC systems used in buildings. They are frequently remote from the rest of the system and are located outdoors. The units may or may not contain integral compressor equipment; however, they will always contain fans--generally of the propeller type.

Very little information on the operating noise levels of these units was found in manufacturers' current catalogs, and the assistance of ARI was requested again. The Industry responded with data on 40 units of various designs within a capacity range of 10-100 TR. Included in the sample were data for air-cooled condensers, with and without integral compressors, and for air-cooled chillers.

The best fit to the data, as shown in Figure 3, is

$$L_W(A) = 75 + 12 \log (TR), \text{ dBA},$$

with a standard error of estimate equal to 3 dB. Thus, for use with the Permit Scheme we recommend estimating the A-weighted sound power level on the basis of:

$$L_W(A) = 78 + 12 \log (TR), \text{ dBA};$$

the characteristic frequency spectrum was found to be Class II.

Table 5B lists the estimated A-Weighted Sound Power Levels in increments of 1 dBA for the capacity range of 10-200 TR.

TABLE 5

A-Weighted Sound Power Levels of Outdoor HVAC Equipment

Equipment Type	Rated Capacity Tons	Sound Power Level (A-Weighted) dBA re 10^{-12} Watt
A	10 - 11	92
	12 - 13	93
	14 - 16	94
Packaged "Rooftop" HVAC Equipment	17 - 20	95
	21 - 24	96
	25 - 29	97
Range 10-200 TR	30 - 35	98
	36 - 42	99
Frequency Spectrum Class II	43 - 51	100
	52 - 62	101
	63 - 75	102
	76 - 91	103
	92 - 110	104
	111 - 133	105
	134 - 162	106
	163 - 200	107
B	10 - 11	90
	12 - 13	91
	14 - 16	92
Air-Cooled Condensing Units/ Chillers	17 - 20	93
	21 - 24	94
	25 - 29	95
	30 - 35	96
Range 10-200 TR	36 - 42	97
	43 - 51	98
Frequency Spectrum Class II	52 - 62	99
	63 - 75	100
	76 - 91	101
	92 - 110	102
	111 - 133	103
	134 - 162	104
	163 - 200	105

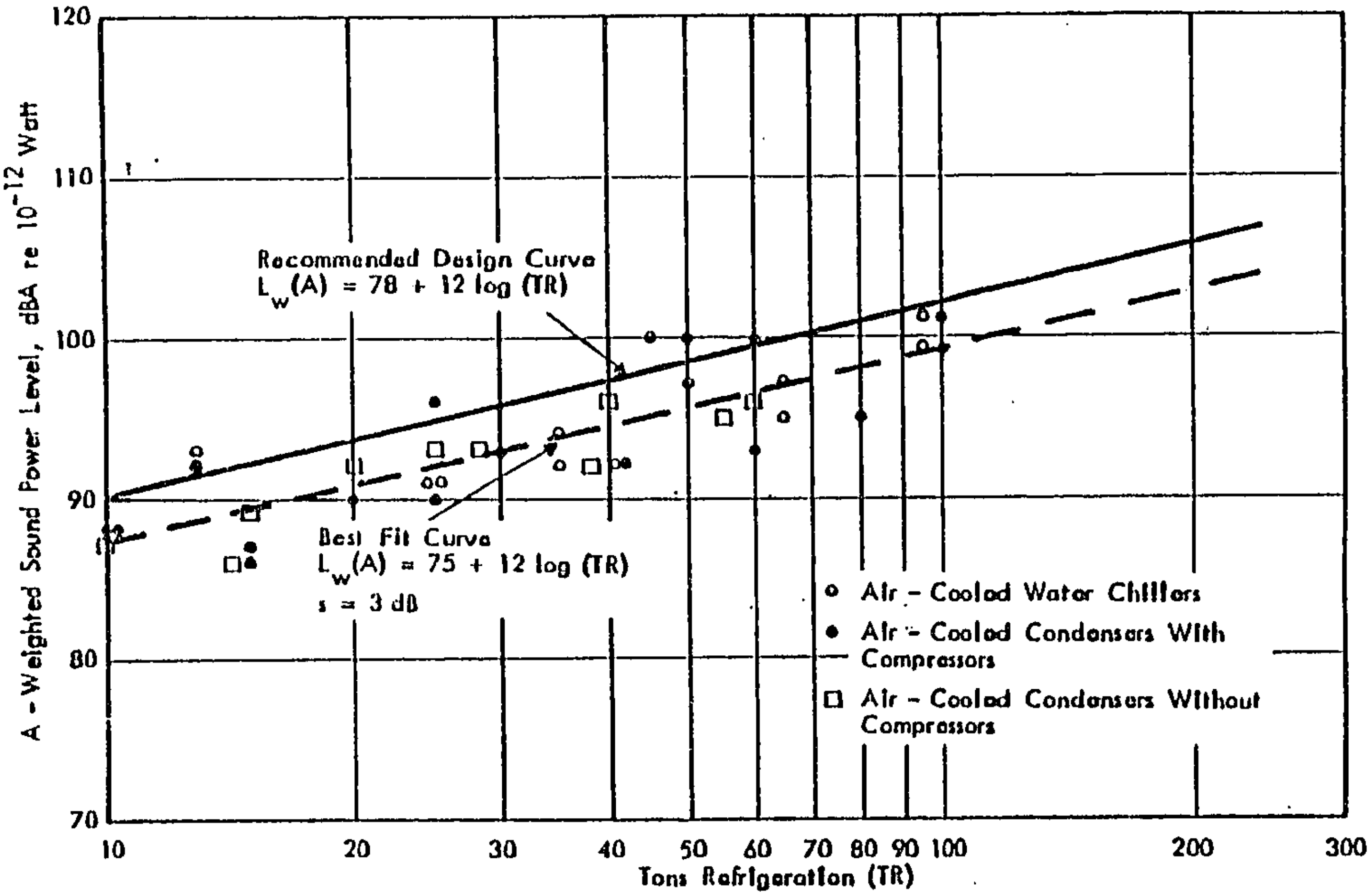


FIGURE 3. NOISE ESTIMATION CURVES FOR AIR - COOLED CONDENSERS AND CHILLERS

6.3 Cooling Towers And Evaporative Condensers

These units are components of many HVAC systems used in buildings. They are located outdoors and provide essentially the same function as air-cooled condensing units in the heat-transfer process associated with the compressor equipment.

At the present time, there appears to be no industry standard for measurement and rating of cooling tower noise. The Cooling Tower Institute (CTI) has set up a working group to develop a procedure for sound measurement (although we were unable to determine the present status of this program).

Jack we should contact this group

Two major manufacturers of cooling tower equipment were contacted for noise level data representative of their product lines. The information received from these sources was combined with additional data present in BBN files and analyzed to develop a noise level prediction scheme.

The best correlation of equipment size with noise level appears to be on the basis of total fan-motor horsepower. However, a different equation is required for each of the three types of towers in common usage today (Centrifugal, Propeller, Vane-Axial).

6.3.1 Centrifugal Fan Cooling Towers

Figure 4 shows the A-Weighted Sound Power Levels obtained from one equipment manufacturer for 166 towers with centrifugal fans, plotted as a function of fan-motor horsepower. The best curve fit to the data is

$$L_W(A) = 78 + 9.5 \log(hp), \text{ dBA},$$

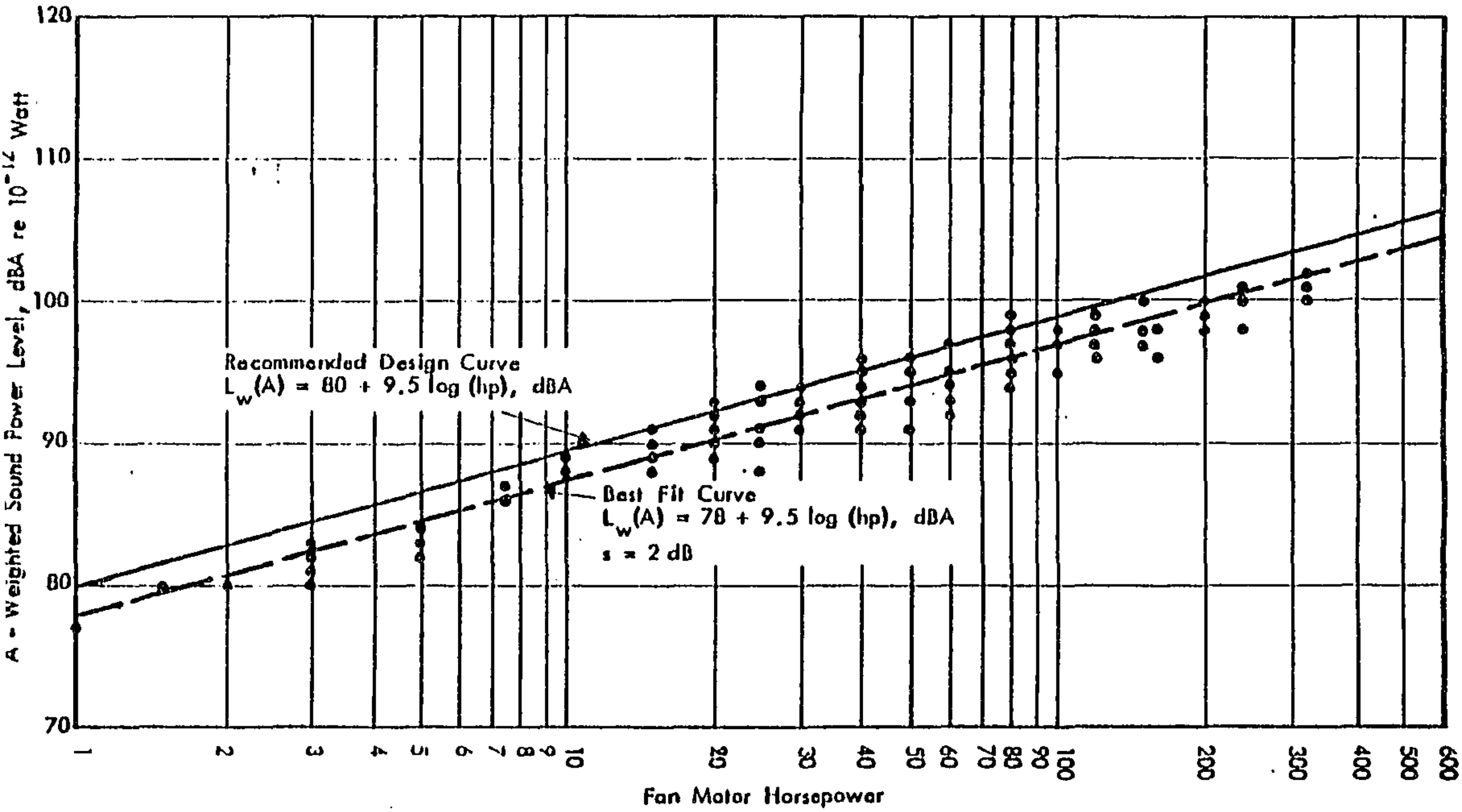


FIGURE 4. CENTRIFUGAL FAN COOLING TOWER NOISE ESTIMATION CURVES

with a standard error of estimate equal to 2 dB. Thus, for use in the Permit Scheme we recommend estimating the A-Weighted Sound Power Level as follows:

$$L_W(A) = 80 + 9.5 \log (hp), \text{ dBA};$$

the characteristic frequency spectrum was found to be Class II.

Table 6A lists the estimated A-Weighted Sound Power Levels in 1 dBA increments for fan-motor drives in the 10-350 hp range.

6.3.2 Propeller Fan Cooling Towers

Figure 5 shows the noise correlation obtained for 55 towers with propeller fans. The best fit to the data is

$$L_W(A) = 88 + 7.5 \log (hp), \text{ dBA},$$

with a standard error of estimate equal to 3 dB. Thus, for use in the Permit Scheme we recommend estimating the A-Weighted Sound Power Level as follows:

$$L_W(A) = 91 + 7.5 \log (hp), \text{ dBA};$$

the characteristic frequency spectrum was found to be Class I.

Table 6B lists the estimated A-Weighted Sound Power Levels in 1 dBA increments for fan-motor drives in the 5-100 hp range.

6.3.3 Vane-Axial Fan Evaporative Condensers

Figure 6 shows the noise correlation obtained for 11 evaporative condensers with vane-axial fans. The data base is from

TABLE 6

A-Weighted Sound Power Levels
of Cooling Towers and Evaporative Condensers

Equipment Type	Total Fan-Motor hp hp	A-Weighted Sound Power Level dBA re 10^{-12} Watt
A	10 - 13	90
	14 - 16	91
Centrifugal	17 - 21	92
Range 10-350 hp	22 - 26	93
	27 - 34	94
Frequency Spectrum Class II	35 - 43	95
	44 - 55	96
	56 - 70	97
	71 - 89	98
	90 - 113	99
	114 - 144	100
	145 - 183	101
	184 - 234	102
	235 - 298	103
	299 - 350	104
B	5 - 7	97
	8 - 10	98
Propeller	11 - 14	99
	15 - 19	100
Range 5-100 hp	20 - 25	101
	26 - 34	102
Frequency Spectrum Class I	35 - 46	103
	47 - 63	104
	64 - 85	105
	86 - 100	106
C	5.0 - 6.2	100
	6.3 - 7.5	101
Vane-Axial	7.6 - 9.1	102
	9.2 - 11.0	103
Range 5-75 hp	11.1 - 13.3	104
	13.4 - 16.2	105
Frequency Spectrum Class II	16.3 - 19.6	106
	19.7 - 23.7	107
	23.8 - 28.7	108
	28.8 - 34.8	109
	34.9 - 42.2	110
	42.3 - 51.1	111
	51.2 - 61.9	112
	62.0 - 75.0	113

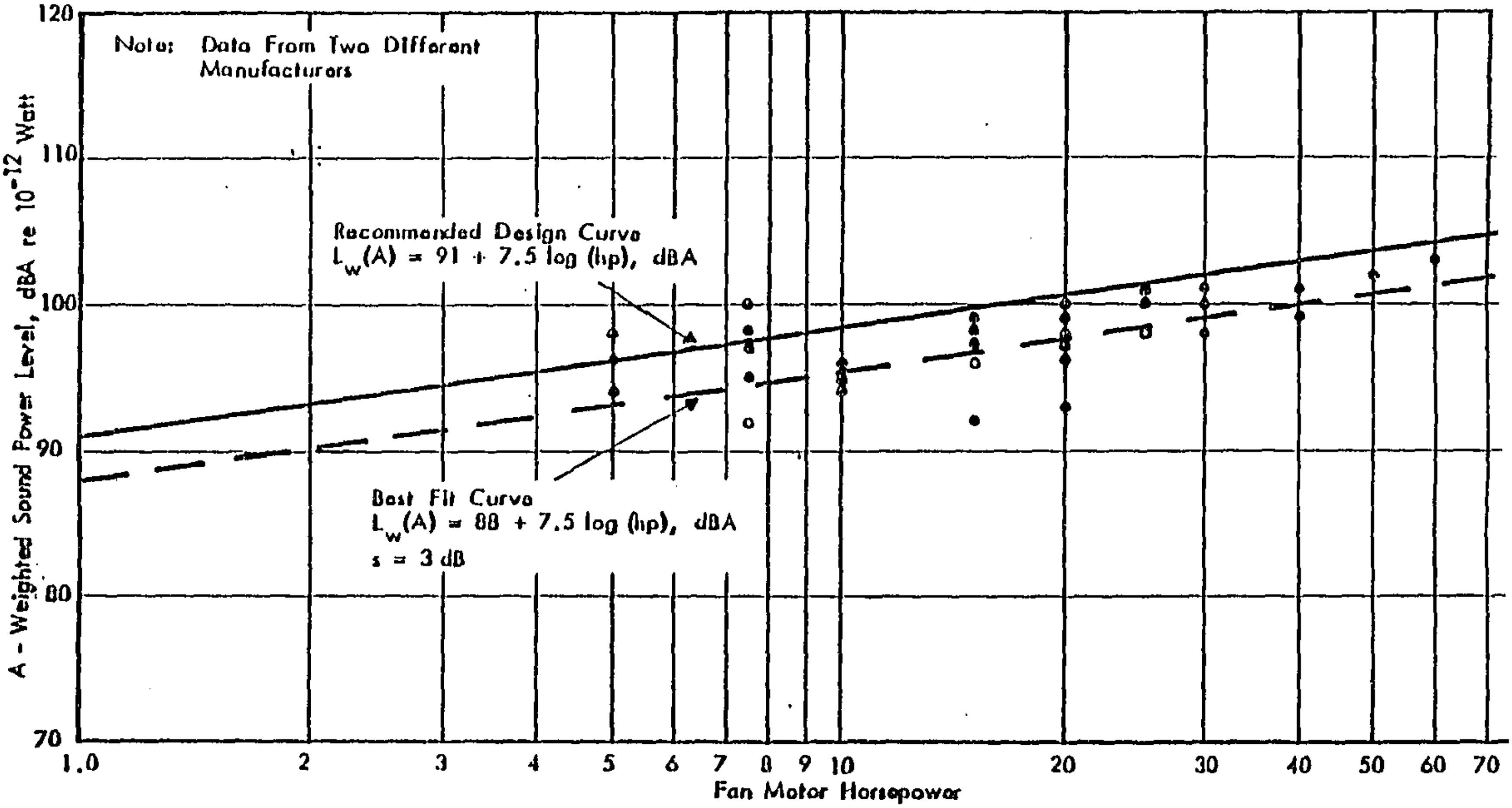


FIGURE 5. PROPELLER FAN COOLING TOWER NOISE ESTIMATION CURVES

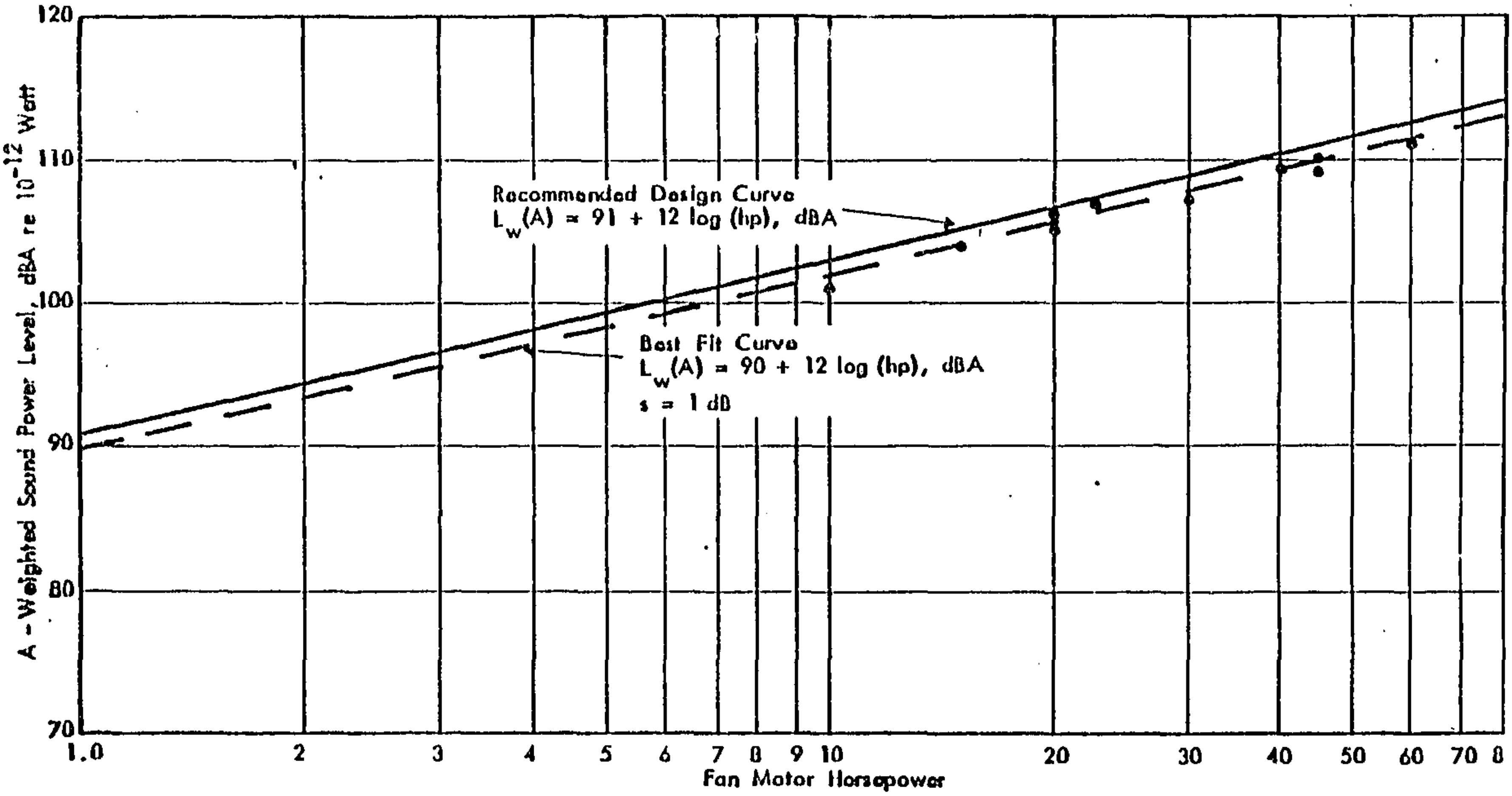


FIGURE 6. VANE - AXIAL FAN COOLING TOWER NOISE ESTIMATION CURVES

only one equipment manufacturer. The best fit to the data is:

$$L_W(A) = 90 + 12 \log (hp), \text{ dBA},$$

with a standard error of estimate equal to 1 dB. Thus, for use in the Permit Scheme we recommend estimating the A-Weighted Sound Power Level as follows:

$$L_W(A) = 91 + 12 \log (hp), \text{ dBA};$$

the characteristic frequency spectrum was found to be Class II.

Table 6C lists the estimated A-Weighted Sound Power Levels in 1 dBA increments for fan-motor drives in the 5-75 hp range.

6.4 Room Air Conditioners (Condenser Side)

This type of unit has two principal applications. The first is as a window installation in older buildings that were not planned for central air conditioning; the second is in newer construction where air conditioning is provided on a room-by-room basis with a "through-the-wall," non-ducted unit.

With respect to this project, the multiple application in remodeled or new building construction is of chief concern; an array of these units installed in a building facade can represent a significant source of exterior noise.

Products of this type fall within the jurisdiction of the American Home Appliance Manufacturers Association (AHAM). Although industry standards are available for measurement and rating of room air-conditioner noise levels, AHAM does not at the present time have a certification program to police the claims made by the manufacturers.

We found it very difficult to obtain sound data on these units from the individual manufacturers--not because data are unavailable, but because these companies are reluctant to disclose such information until an industry certification program is adopted. However, one of the companies was willing to discuss the subject with us on an informal, non-official basis and provided an indication of the probable range of noise levels likely to be encountered with present industry products. This information, coupled with data in BBN files from a study made in 1967[14] has been used to develop a prediction scheme for the noise level on the condenser (outdoor) side of room air conditioners.

The recommended equation for estimating the A-Weighted Sound Power Level of room air conditioners is:

$$L_W(A) = 37 + 10 \log (\text{Btu/h}), \text{ dBA;}$$

the characteristic frequency spectrum shape is Class II.

The basis for the prediction scheme is shown in Figure 7. It will be noted that the correlation between the curve based on an "unofficial" industry source and that derived by extrapolating the ARI data on air-cooled condensers is quite good. Furthermore, the range of data obtained in the 1967 BBN study is well bracketed by either curve. We have chosen to use the curve based on the industry source because it is slightly more conservative; it is estimated that 10-15 percent of the typical units encountered in practice may be 2-3 dBA higher than predicted

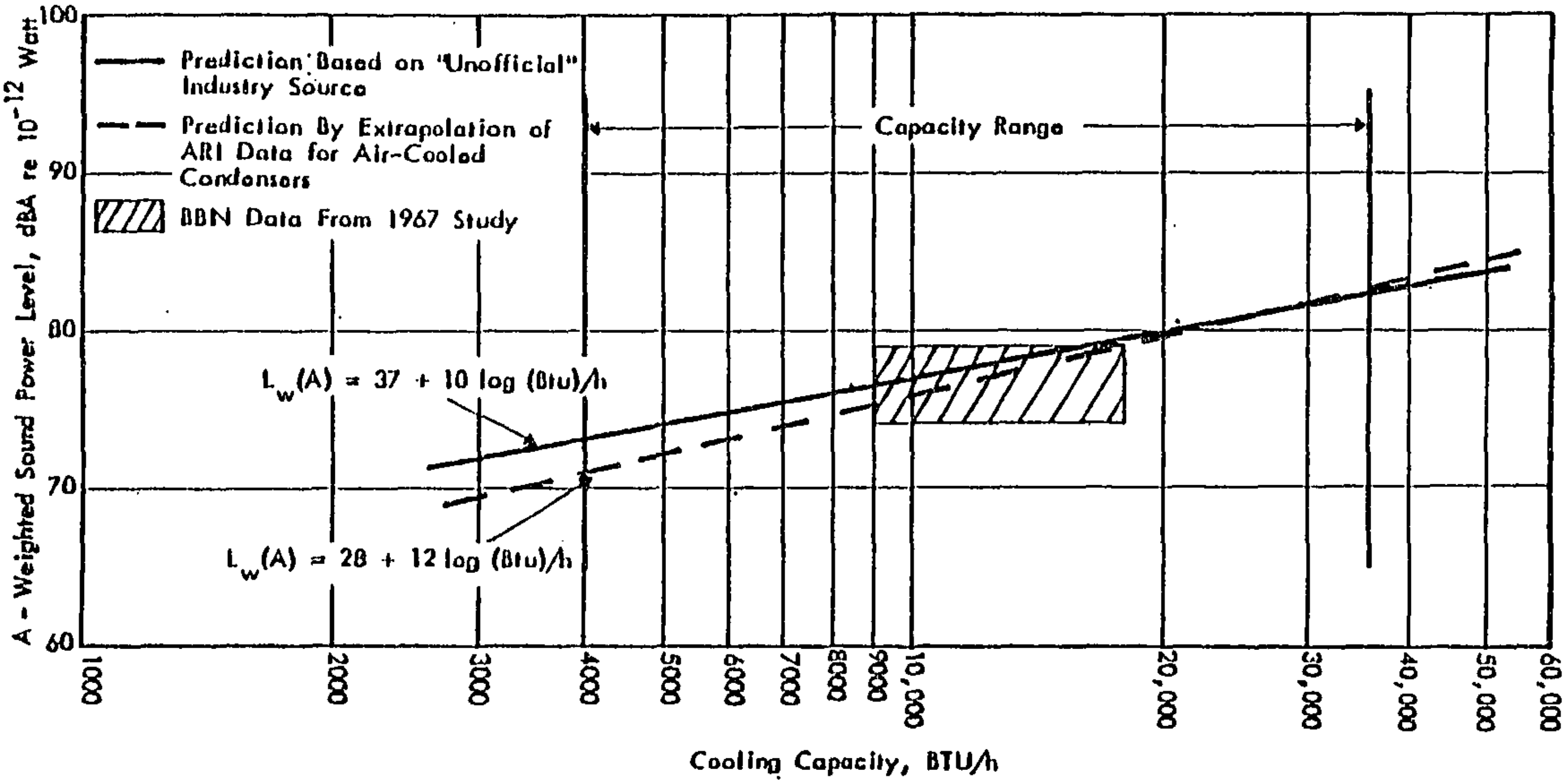


FIGURE 7. NOISE ESTIMATION CURVE FOR ROOM AIR - CONDITIONERS

by this curve. In Table 7 below, the A-Weighted Sound Power Levels are listed in 1 dBA increments for room air conditioners (condenser side) over the capacity range of 3500-36,000 Btu/h.

TABLE 7

A-Weighted Sound Power Levels of Room Air Conditioners (Condenser Side)

Capacity Btu/h	A-Weighted Sound Power Level dBA re 10^{-12} watt
3,500- 4,500	73
4,600- 5,600	74
5,700- 7,100	75
7,200- 8,900	76
9,000-11,200	77
11,300-14,100	78
14,200-17,800	79
17,900-22,400	80
22,500-28,200	81
28,300-36,000	82

6.5 Transformer Equipment

Sub-station transformers associated with buildings typically have power ratings in the range between 0.5 and 20 MVA, and may be either radiant-cooled or fan-cooled units. These units may be located outdoors or in vaults below street level, and are often owned by the utility. Smaller transformers (< 0.5 MVA), purchased by the building owners, are typically located in mechanical rooms in several areas of the building.

It is proposed that the A-Weighted Sound Power Levels of outdoor transformers be predicted based on MVA rating, readily obtainable manufacturer's data. If, however, transformer dimensional data (obtained by the NEMA Standard [15] method) are available, it is recommended that noise prediction be done by direct calculation. This procedure is outlined in Appendix 1 of this report.

A relationship between transformer MVA rating and A-weighted sound level was obtained in a recent study carred out on behalf of the Bonneville Power Administration.[16] Measurements for 59 transformers rated between 6 and 1200 MVA, as illustrated in Figure 8, resulted in the following correlation:

$$L_{(500)} = 29.7 + 8.2 \log (\mathrm{MVA})$$

where

$L_{(500)}$ = sound level at 500 ft., dBA
MVA = transformer power rating

The standard deviation on the above regression line at the mean rating is approximately 3 dB. The fact that the sound level varies as 8.2 log (MVA) rather than the expected 10 log (MVA) is probably explained by the increased cooling capacity of the larger power transformers.[16] Furthermore, the study indicates that due to the close-in measurement procedure, the above correlation is over-predicting the far-field A-weighted sound levels by approximately 4 dB. Thus, if the above equation is adjusted to a 10 log slope, reduced by 4 dB and converted to sound power level (assuming free-field hemispherical sound radiation) the following relation results:

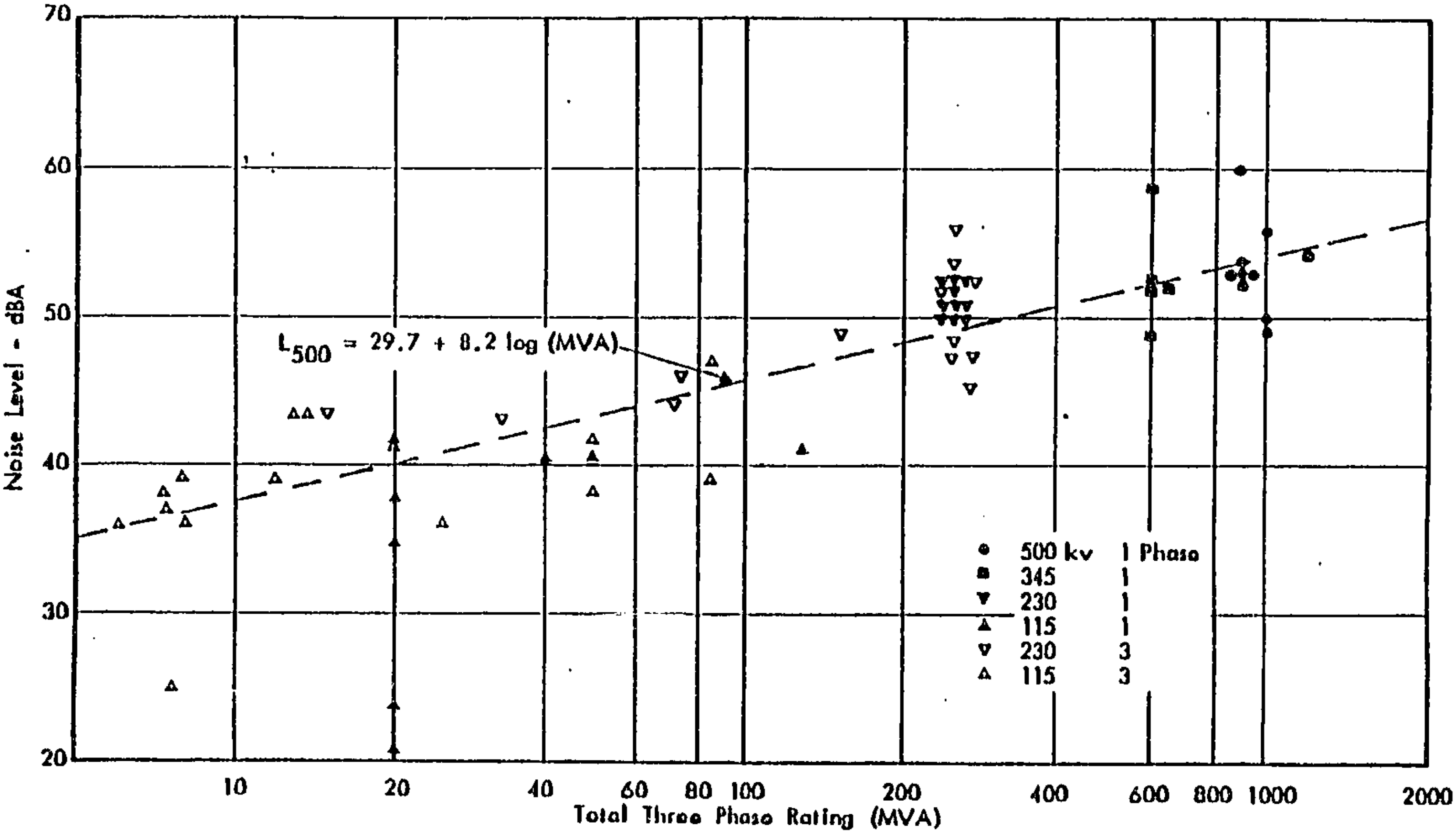


FIGURE 8. NOISE LEVELS AT 500 FT DISTANCE BASED ON NORMALIZATION OF TRANSFORMER CLOSE - IN DATA (REF. 16)

$$L_w(A) = K_A + 10 \log (MVA) \ , \quad dBA$$

where

$L_w(A)$ = A-weighted Sound Power Level re 10^{-12} Watt

K_A = A-weighted specific sound power level re 10^{-12} Watt

= 75 for radiant-cooled transformers

= 77 for fan-cooled transformers

MVA = transformer power rating

Thus, it is proposed that the A-Weighted Sound Power Level of outdoor transformers be predicted based on MVA rating using the above relation. Table 8 tabulates the function 10 log (MVA).

Since the BPA correlation was based on relatively large transformers, other correlation methods were developed based on data from References 17, 18 and 19. The resulting prediction curves agreed reasonably well with the proposed curve and thus it is felt that the proposed method is justifiable for design purposes within the transformer size range of interest.

In summary, it is recommened that outdoor transformer noise prediction be based on MVA rating as described above. If, however, measured NEMA data are available, the transformer noise prediction may be based on measured NEMA sound data and transformer size, as described in Appendix 1. The characteristic frequency spectra are Class IVA for radiant-cooled transformers and Class IVB for fan-cooled transformers.

TABLE 8

Correction Factor For MVA Rating

MVA Rating	10 log (MVA)
0.447-0.562	-3
0.563-0.708	-2
0.709-0.891	-1
0.892- 1.12	0
1.13- 1.41	1
1.41- 1.78	2
1.79- 2.24	3
2.25- 2.82	4
2.83- 3.55	5
3.56- 4.47	6
4.48- 5.62	7
5.63- 7.08	8
7.09- 8.91	9
8.92-11.2	10
11.3 -14.1	11
14.2 -17.8	12
17.9 -22.4	13

7. RECOMMENDED PROCEDURE FOR ESTIMATING THE A-WEIGHTED SOUND PRESSURE LEVELS OF INDOOR EQUIPMENT

In Section 4.3 reference was made to growing evidence that the relationship between radiated sound power and the resulting sound pressure at a point in most typical rooms does not generally follow the direct/reverberant field relationship usually assumed in current practice. For example, in typical mechanical rooms, the drop in sound pressure level is observed to be closer to 3 dB per doubling of distance and the "constant" reverberant field level cannot be found.

To demonstrate this departure from the classical direct/reverberant field relationship measurements were made in five typical building mechanical rooms that ranged in size from 29' x 15' x 12' to 44' x 25' x 18'. These rooms contained varying amounts of machinery, piping, ductwork, plenums, and surface acoustical treatment. The measurement periods were chosen at times when enough equipment could be shut down to permit using an Ilg reference sound source as a noise generator without interference from other machinery. In each room the A-Weighted Sound Pressure Levels were measured at several distances from the Ilg source. The relationship, in dB, between the known sound power being rediated by the reference source and the resulting sound pressure at each measurement distance was then determined. The results of this experiment are plotted in Figure 9.

Note that the scatter of the data at comparable distances from the reference source is generally on the order of only 1 dB around the best curve fit, which has a slope of about -3 dB per doubling of distance. It is also significant that for distances comparable to the maximum room dimensions there is no "leveling off" of the sound pressure level that would indicate the existence of a reverberant field.

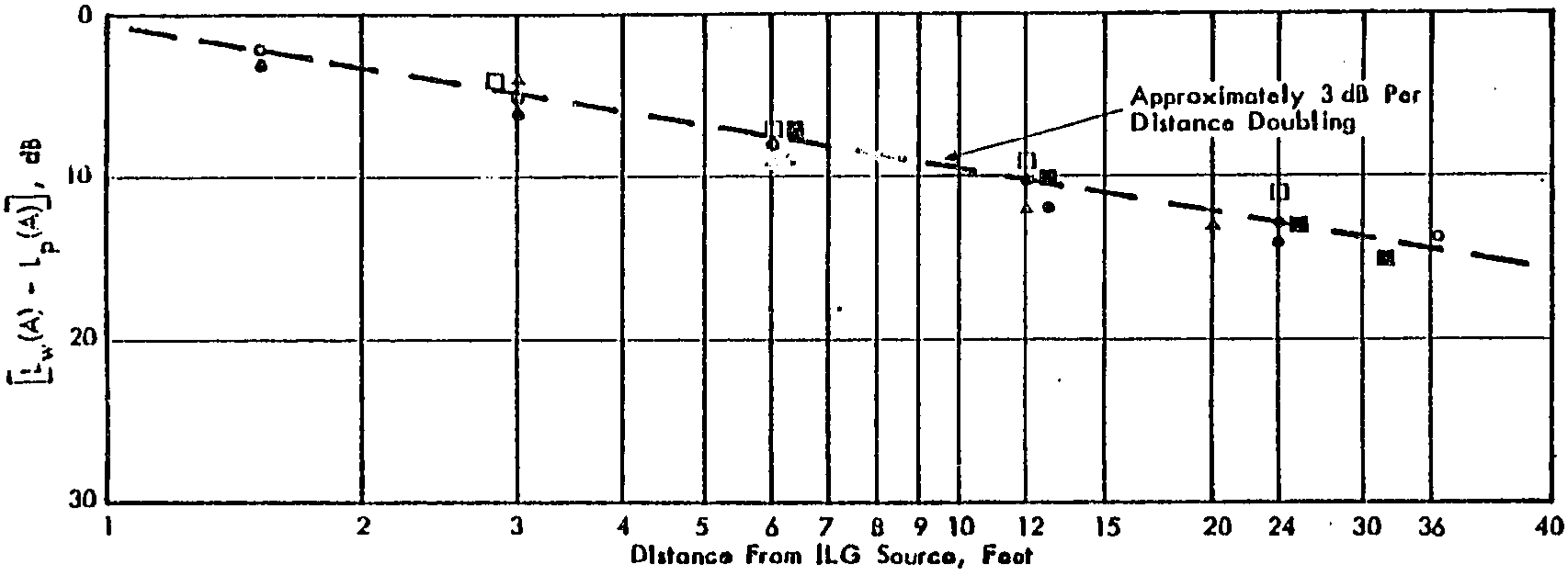


FIGURE 9. RELATIONSHIP BETWEEN A - WEIGHTED SOUND POWER AND SOUND PRESSURE LEVEL FOR AN ILG REFERENCE SOURCE LOCATED IN 5 TYPICAL MECHANICAL ROOMS

With regard to the Permit Scheme, the prediction of exterior noise levels, due to a source located indoors, requires the estimation of equipment sound levels at an opening in the building face. Based on the above discussion, we believe better reliability will be obtained using equipment sound pressure levels at a reference distance from the source, such as one meter, rather than to use sound power level. Were the latter chosen instead, the conversion to sound pressure at a point in the room would require the use of a relationship that doesn't seem reliable in typical mechanical spaces--even for a small source such as the Ilg fan. Furthermore, the conversion would require the determination of room absorption which could be subject to gross error if estimated from the information likely to appear on a set of building plans.

7.1 Centrifugal Chiller Equipment

Centrifugal compressors are very common in chiller equipment used for HVAC applications in most medium and large sized buildings. Unless the cooling capacity of individual machines is required to be much in excess of 1,000 tons, the compressors are generally hermetics (motor-drive and compressor-integrated in a sealed housing). In the size range above 1000 tons, the motor-drive system is frequently separate from the compressor.

There are two basic types of centrifugal machines; the direct-drive design, where the compressor speed is the same as the motor, and the gear-drive, where the compressor speed may be several times greater than that of the motor. The radiated noise spectra of these two machine designs are distinctly different. The direct-driven equipment noise spectrum typically peaks in the region of 1000 Hz, whereas the geared-machine spectrum peaks one to two octaves higher in the frequency range.

Because little catalog data on noise has been published for centrifugal machines, ARI was asked to request whatever data were available from member companies for use on this project. The response was very good; data were supplied on some 35 different machines of both designs, and covering a range of sizes from 100-7,000 tons. For most of these machines, data were furnished for operation at light, medium, and full-load conditions.

The first attempt to establish a noise-level correlation with equipment size was to plot the data independently of differences in machine design, refrigerants, or operating load. The result is shown in Figure 10. Note that the standard error of estimate is approximately 4 dB relative to the best-curve fit.

After a series of different correlation schemes were tried, in an attempt to reduce the scatter in results, it was determined that the data should be separated on the basis of drive design, operating load, and equipment capacity range.

7.1.1 Internally-Geared Machines

In Figure 11, the correlation between sound level and equipment capacity is shown for machines in the size range of 100-1000 tons. The data are separated on the basis of operation at light loads and medium to full load. Note that the standard error of estimate has been reduced to about 2 dB and also that operation at light loads is about 3 dB noiser than at medium to full load.

For use in the Permit Scheme, we recommend that the A-Weighted Sound Pressure Level at one meter be estimated using the following equation:

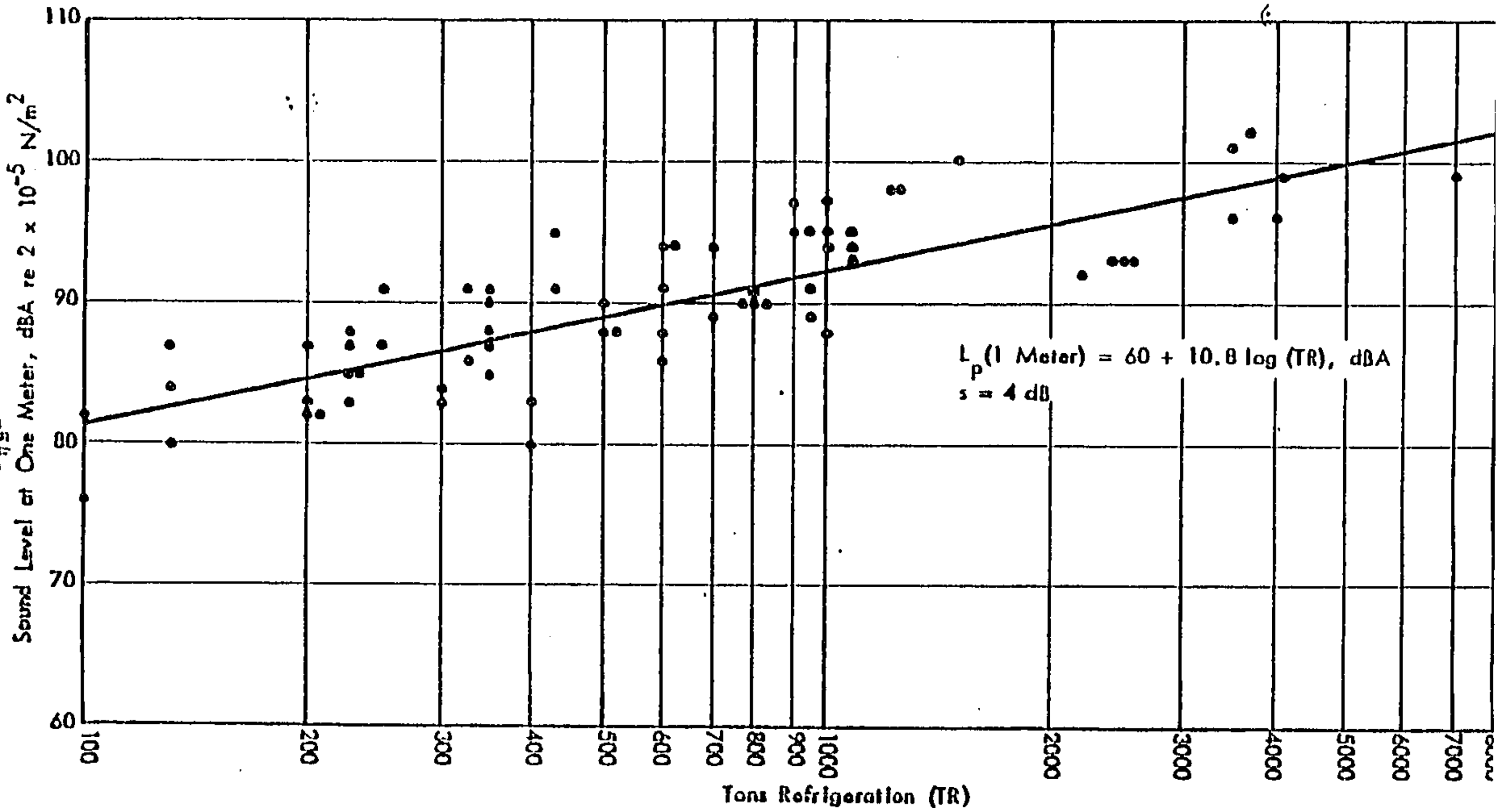


FIGURE 10. CENTRIFUGAL CHILLER EQUIPMENT NOISE: ALL TYPES, SIZES AND LOADS

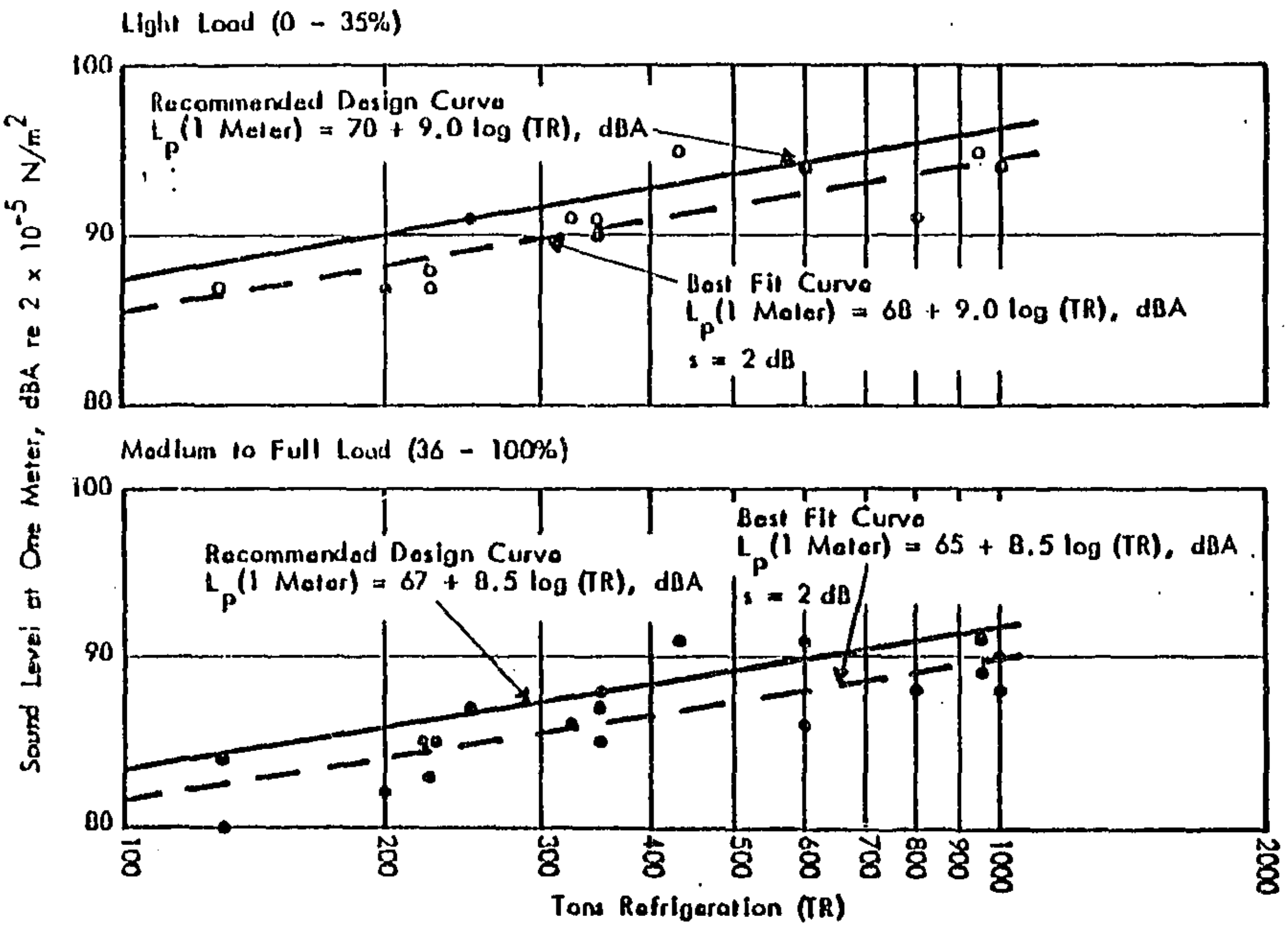


FIGURE 11. INTERNALLY - GEARED, HERMETIC CENTRIFUGAL COMPRESSORS (100 - 1,000 TONS)

$$L_p(A) = 70 + 9 \log (TR), \text{ dBA*};$$

the characteristic frequency spectrum is Class III.

The recommended curve corresponds to the best-fit of the data plus one standard error for light load conditions. This should yield a conservative estimate for machines at any operating load.

7.1.2 Direct Drive Machines

Figure 12 shows the correlation of noise with equipment size determined for direct-drive machines in the capacity range of 100-1000 tons when operating in the medium to full-load range. No data were made available for operation at light-loads. The standard error of estimate is 4 dB and thus the correlation is not as good as for the geared machines.

We recommend that the curve for estimation of maximum noise levels be rasied 4 dB above that corresponding to the medium to full-load range. On this basis the relationship recommended for estimating the A-weighted sound pressue level at 1 meter becomes:

$$L_p(A) = 46 + 17 \log (TR), \text{ dBA};$$

the characteristic shape of the frequency spectrum is Class II.

*"TR" is an abbreviation for Tons-Refrigeration. 1 Ton equals 12,000 BTU per hour.

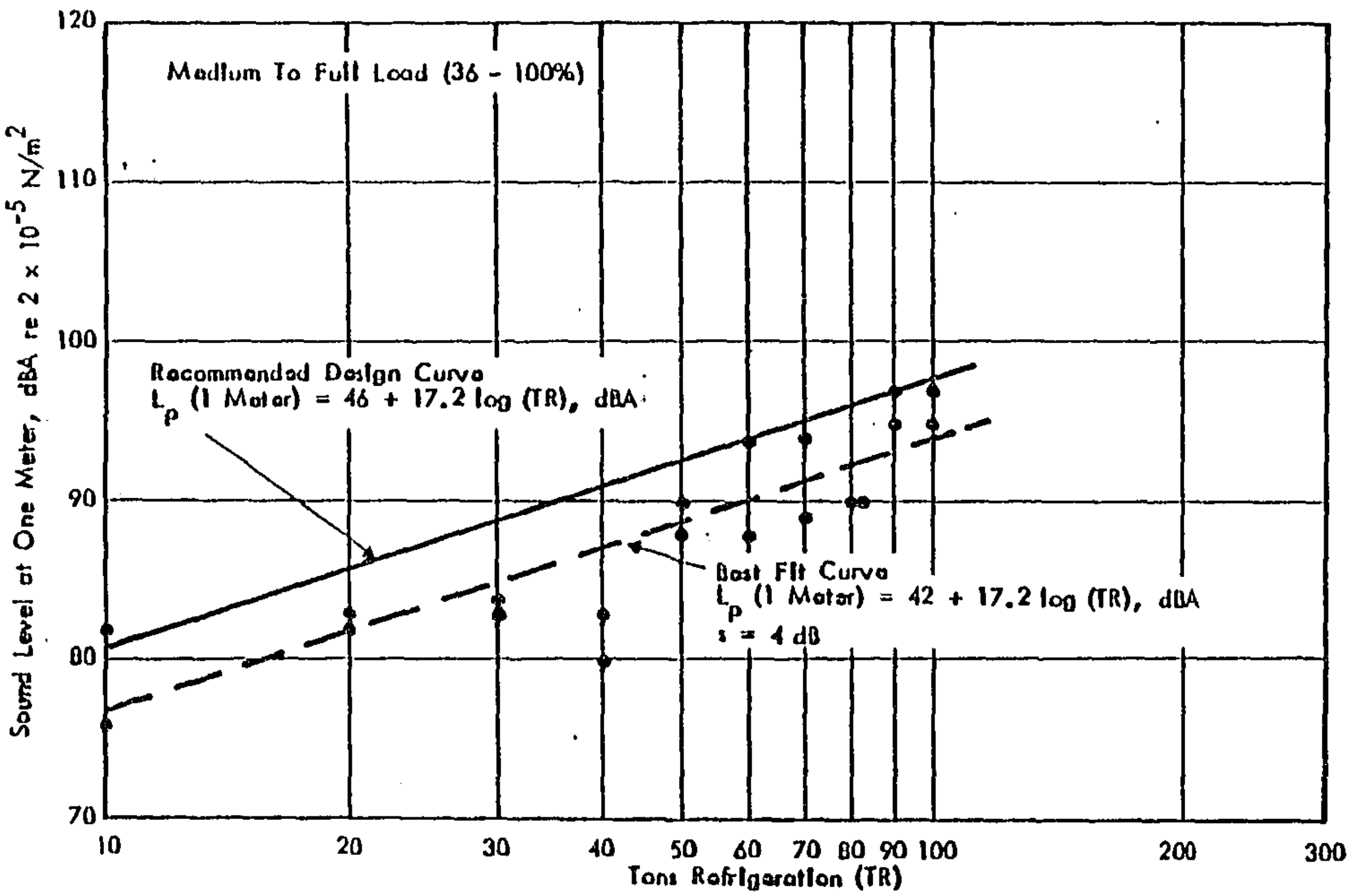


FIGURE 12. DIRECT - DRIVE, HERMETIC CENTRIFUGAL COMPRESSORS (100 - 1000 TONS)

7.1.3 Large Centrigugal Equipment (> 1000 tons)

There were less noise data provided on large centrifugal machines in the size range greater than 1000 tons. However, the trend clearly indicated the need for a different equation for estimating noise levels than either of those derived for the hermetic machines.

In Figure 13 it will be seen that the best curve-fit has a standard error of estimate equal to 3 dB. Machines using different refrigerants and types of drives (motor, turbine and motor-gear) are included in the data base. To be on the conservative side we recommend using a noise estimating curve 3 dB higher than the best curve-fit. The equation for estimating the A-weighted sound pressure level at 1 meter thus becomes:

$$L_p(A) = 81 + 6.8 \log (TR), \text{ dBA};$$

the shape of the characteristic frequency spectrum is Class III.

7.1.4 Noise Level Comparisons

In Figure 14, the three curves recommended above for estimating the sound level of centrigugal equipment at 1 meter are drawn for comparison. It can be seen that direct-driven machines are characterized by somewhat lower sound levels than geared machines in the capacity range below 1000 tons. The offset for large centrifugal equipment is about +1 dB near 1000 tons, but the rate of increase in sound level with capacity is less than that of either type of hermetic machine.*

*The study made of similar machines for ARI in 1971-1972 and reported in Reference 20 found noise levels for hermetic machines some 4-8 dBA higher than those contained in current survey. However, the two surveys are in fairly good agreement for large centrifugal machines.

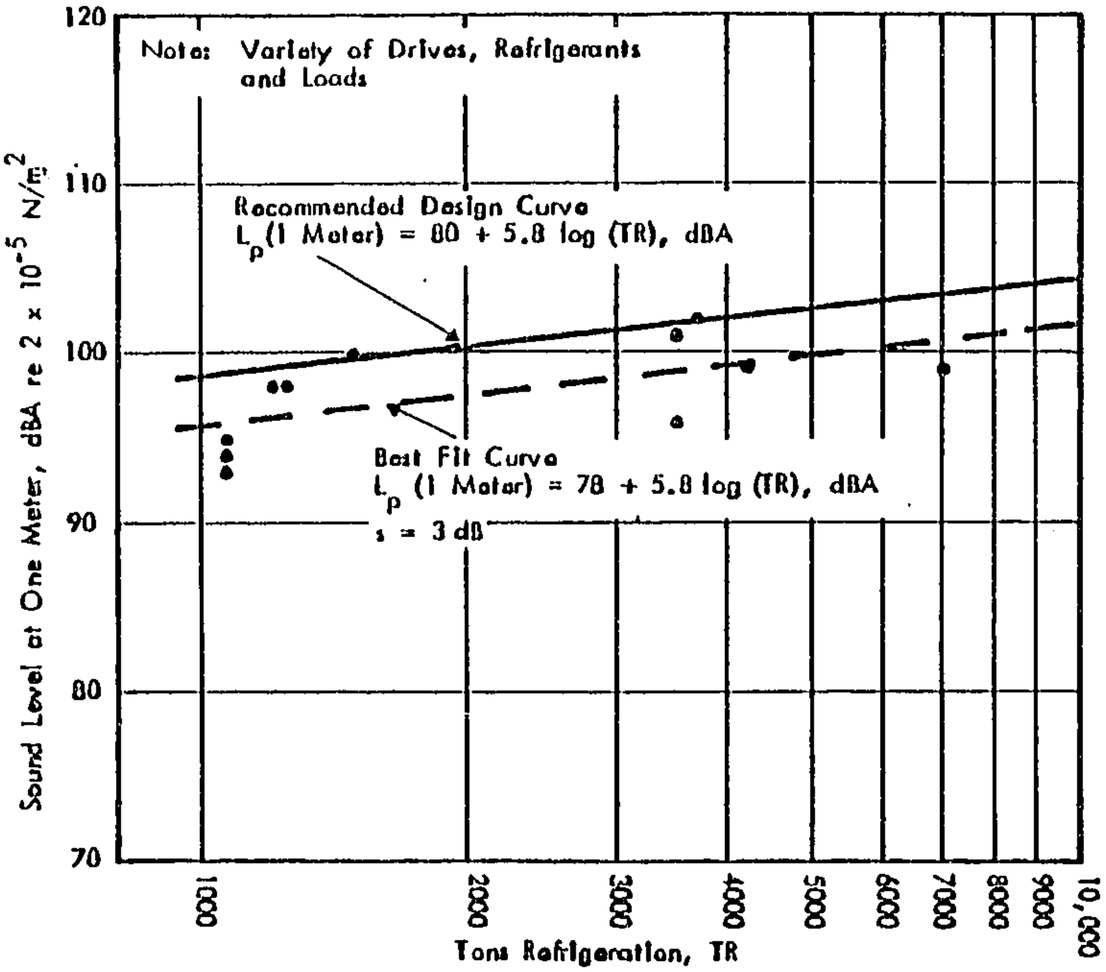


FIGURE 13. LARGE CENTRIFUGAL COMPRESSORS (> 1000 TONS)

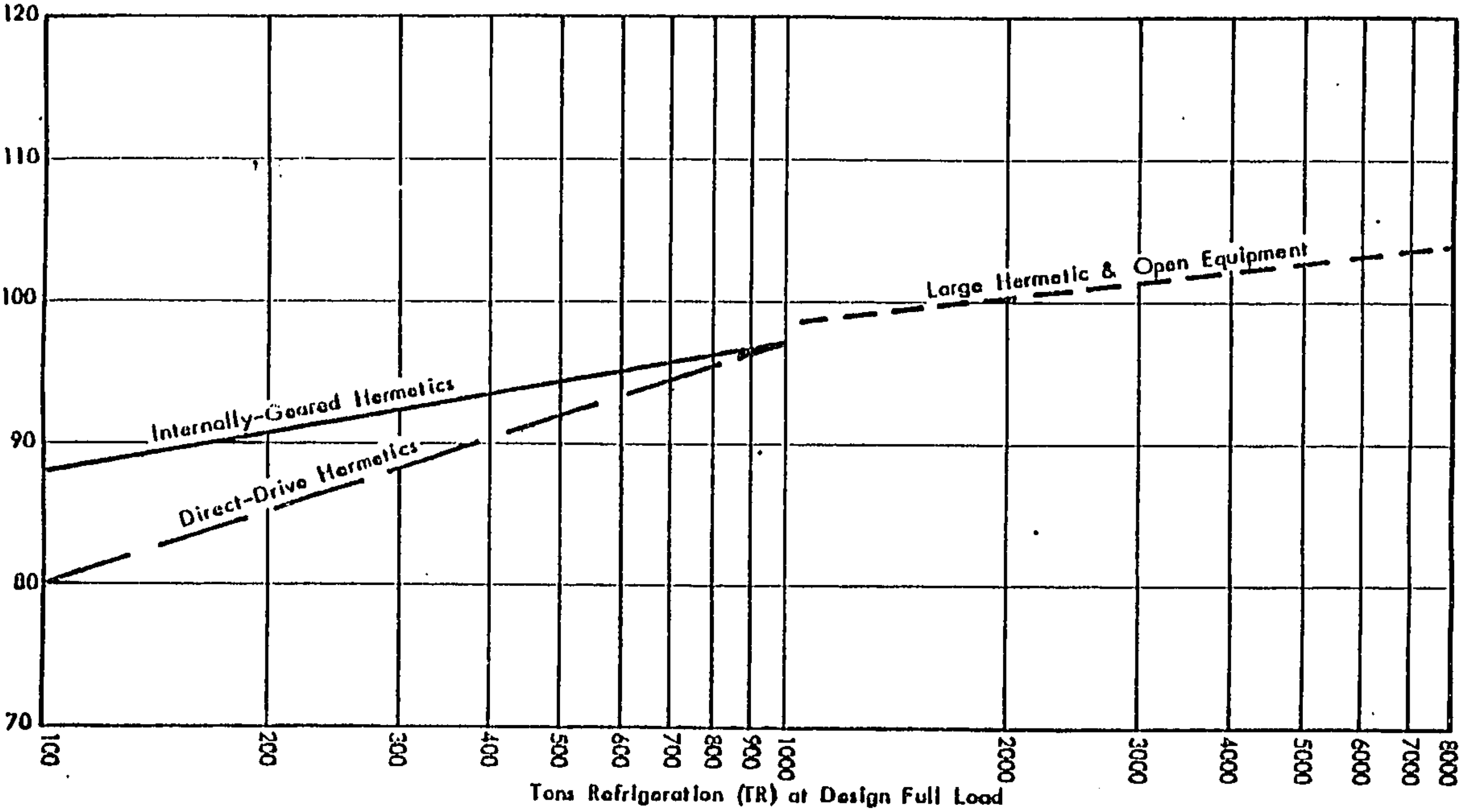


FIGURE 14. NOISE ESTIMATION CURVES FOR CENTRIFUGAL CHILLER EQUIPMENT

Tables 9A, 9B and 9C list the sound levels at 1 meter in 1 dBA increments as a function of equipment type and design full-load capacity.

7.2 Reciprocating Chiller Equipment

Reciprocating compressors are most generally found in chiller equipment used in HVAC and refrigeration applications where the capacity requirements for individual machines are ordinarily 100 tons or less.

The design of these "packaged" chillers is subject to wide variation; for the same capacity, the number and types of compressors used (open vs. hermetic) are the principal variables. The refrigerant used (R-12 vs. R-22, etc.) may also vary since this permits the same compressor to provide several ranges of capacity.

Each of the above design factors affects the noise level of the machine for a given duty; our attempts to correlate the measured data on this equipment with capacity resulted in a scatter greater than ±5 dBA.

In Figure 15, the data furnished by ARI on 11 different machines, operating at several loads, have been combined with those used in the 1971-1972 study reported in Reference 20 and plotted versus capacity in tons. It will be seen that the scatter about the best curve-fit to the data is +10, -8 dBA; the standard error of estimate is 5 dB.

A better correlation could probably be found for units separated on the basis of compressor design, number of compressors/package and refrigerant used. However, the sample size of the data made available by industry was too small to permit this type of an analysis. Therefore, a conservative approach must be taken in the estimation of noise levels for this equipment.

TABLE 9

Chiller Equipment

A-Weighted Sound Pressure Level at 1 Meter

Equipment Type	Rated Capacity Tons	Sound Level at 1 Meter dBA re 2×10^{-5} N/m^2
A	100-114	88
Internally-Geared	115-147	89
Hermetic Centrifugal	148-190	90
Range 100-1000 TR	191-245	91
	246-316	92
Frequency Spectrum	317-408	93
Class III	409-528	94
	529-681	95
	682-880	96
	881-1000	97
B		
Direct-Drive	100-116	81
Hermetic Centrifugal	117-132	82
Range 100-1000 TR	133-151	83
	152-173	84
Frequency Spectrum	174-198	85
Class II	199-226	86
	227-259	87
	260-296	88
	297-338	89
	339-387	90
	388-442	91
	443-505	92
	506-578	93
	579-660	94
	661-755	95
	756-864	96
	865-1000	97

TABLE 9 (continued)

Chiller Equipment
A-Weighted Sound Pressure Level at 1 Meter

Equipment Type	Rated Capacity Tons	Sound Level at 1 Meter dBA re 2×10^{-5} N/m^2
C		
Large Centrifugal	1050-1550	99
Machines	1551-2300	100
> 1000 TR	2301-3400	101
All Drive Types	3401-5100	102
	5101-7600	103
Frequency Spectrum	7601-10,000	104
Class III		
D		
Hermetic Reciprocating	20-26	88
Range 20-200 TR	27-34	89
	35-44	90
Frequency Spectrum	45-58	91
Class II	59-75	92
	76-97	93
	98-125	94
	126-164	95
	165-200	96

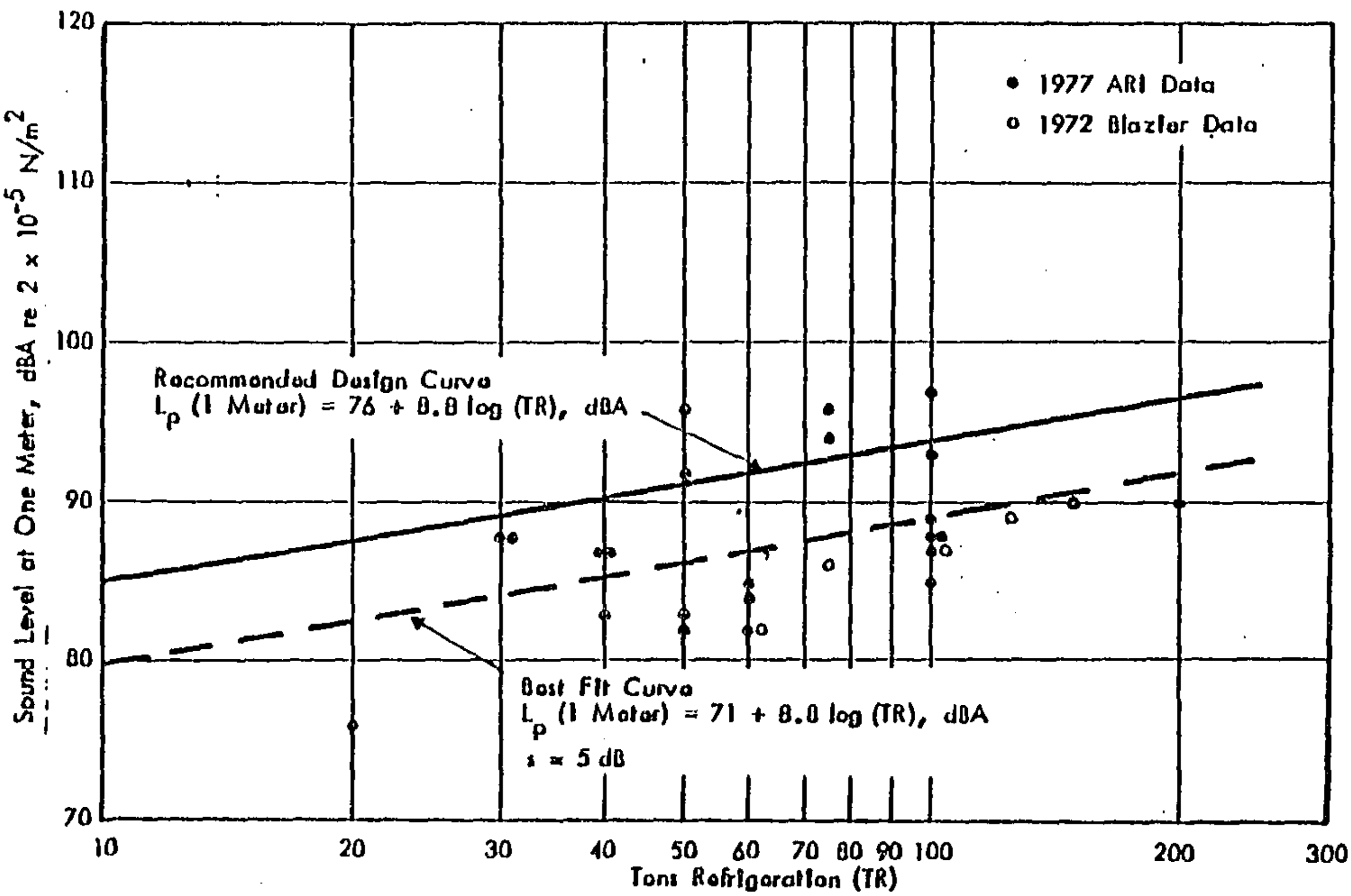


FIGURE 15. HERMETIC RECIPROCATING CHILLER EQUIPMENT (WATER COOLED)

For application to the Permit Scheme, the following relationship is recommended:

$$L_p(A) \text{ @ 1 meter} = 76 + 8.8 \log (TR), \text{ dBA};$$

the characteristic frequency spectrum in Class II.

Although the above relationship may potentially underestimate about 18% of the equipment, pegging the estimation curve at a higher level does not seem justified considering the small size of the data sample. Table 9D lists the sound level of this equipment in 1 dBA increments vs. capacity, based upon the relationship recommended above.

7.3 Miscellaneous Chiller Equipment

Two other types of chillers are occasionally found in HVAC systems in buildings. These are: (1) absorption machines, and (2) chillers using rotary screw-compressors.

Very little published information is available on these two types of chillers and the data in BBN files on measurements obtained during past projects have been used as the basis for the noise level estimates presented below.

7.3.1 Absorption Machines

The noise level associated with these machines is primarily that resulting from the solution pump and the auxiliary equipment required to operate the system.

For application to the Permit Scheme we recommend using 85 dBA for the sound level at 1 meter from these machines, independent of equipment size. The characteristic frequency spectrum is Class II.

7.3.2 Rotary Screw Compressors

Based on data for only 5 units, we recommend using a value of 90 dBA for the sound level at 1 meter from this equipment. The characteristic frequency spectrum is Class I. The above value is representative of machines operating in the 100 - 300 ton capacity range, at or near 3600 rpm.

7.4 Circulating Pumps

The noise environment near pumps used in HVAC applications is typically that due to the pump itself plus significant contributions from the motor-drive.

Several references have been reviewed to establish a basis for estimating the sound level of pump equipment in HVAC applications.[21,22,23] The best correlation seems to be horsepower of the motor-drive. On the basis of this review we recommend using the following equation for estimating the sound level at 1 meter from pump equipment:

$$L_p(A) = 77 + 10 \log (hp), \text{ dBA};$$

the characteristic spectrum shape is Class II.

Table 10, following, lists pump sound levels in 1 dBA increments as a function of motor-drive horsepower.

TABLE 10

Pump Equipment

Drive-Motor Size HP	Sound Level at 1 Meter dBA re $2x10^{-5}$ N/m^2
3.0 - 3.5	82
3.6 - 4.5	83
4.6 - 5.5	84
5.6 - 7.0	85
7.1 - 9.0	86
9.1 - 11.0	87
11.1 - 14.0	88
14.1 - 18.0	89
18.1 - 23.0	90
23.1 - 28	91
29 - 35	92
36 - 45	93
46 - 56	94
57 - 71	95
72 - 89	96
90 - 112	97
113 - 141	98
142 - 178	99
179 - 225	100

7.5 Boilers

On previous BBN projects[21] noise data have been measured or collected for at least 36 boilers, ranging in size from 50 - 2,000 boiler-horsepower (one "BHP" equals 33,500 BTU per hour). It has not been possible to correlate noise with heating capacity alone or any other known design parameter. Noise levels, normalized to 1 meter distance, may be as high for the smallest as for the largest units. Considering the wide variety of blower assemblies, burners and combustion chambers found on various boilers, it is not surprising that the noise output cannot be simply associated with heating capacity.

For boilers of the forced-draft type (these are the ones most likely to produce noise problems in buildings), we recommend that a sound level of 1 meter of 88 dBA be used for Permit Schéme estimation purposes, independent of boiler size. This level is representative of those measured at the front (combustion air-intake) of units in the 50 - 2000 BHP range. The characteristic frequency spectrum is Class I.

7.6 Air-Compressors

Two types of air-compressors are frequently found in buildings: one is a relatively small compressor (usually under 5 hp) used to supply high-pressure air for operating controls of the HVAC systems; the other is a medium size compressor (possibly up to 100 hp) used to supply air to maintenance and machine shops or to laboratory spaces.

The operation of this equipment tends to be cyclic, on demand, and thus is widely variable from building to building. Furthermore, the principal noise problems that arise from this equipment are most frequently because of the absence of an effective air-intake muffler in the installation. For example, noise levels at 1 meter from typical building air-compressors, without efficient intake mufflers installed, may be as high as 95 - 100 dBA. However, a noise reduction of 20 dBA can be anticipated by using an efficient muffler on the air-intake. Noise reductions greater than this are generally not obtained in practice because of flanking by casing radiated noise.

For application to the Permit Scheme, we recommend using a value of 95 dBA for the sound level at 1 meter, independent of equipment size, for unmuffled machines.* The characteristic frequency spectrum is Class III.

When an efficient air-intake muffler is incorporated in the installation we believe the noise of this equipment can be neglected in most instances.

7.7 Emergency/Auxiliary Electrical Power Generators

Emergency electrical power systems are found in nearly all new buildings. These systems are of two types (diesel engine or turbine) and vary widely in physical capacity, depending on the requirements for emergency power.

*On many occasions, the filter assembly on the air-intake to the compressor is called a "muffler." Our experience is that the noise reduction provided by such filters is negligible.

Auxiliary electrical power systems are beginning to be used in many buildings to reduce the demand on local utilities during peak periods or to permit more than emergency operation in the case of a major power failure.

The noise impact of these systems depends on the frequency and duration of their use. Emergency systems are typically operated for test once a week for a 30-minute period and the noise-control precautions taken are generally minimal for this reason. Auxiliary power systems used, for example, to balance out peak-demand loads are a different matter; the periods of operation may be several hours in duration, on a daily basis. There is growing evidence that systems initially installed for use as emergency power sources are now also being used periodically for auxiliary power.

With regard to the Permit Scheme, the need for a detailed analysis of these systems depends on their utilization factor. For those situations where analysis is required, the following discussion is pertinent. The data base used to develop the procedures for noise estimation appearing below were drawn from BBN files on other projects and, in particular, References 24 & 25.

7.7.1 Diesel-Engine Equipment

The noise of diesel-driven equipment is a function of three sources:

- combustion air-intake;
- machine casing;
- combustion exhaust.

When these systems are installed in buildings, there are typically two paths through which the noise is coupled to the outdoors: the first path is through openings to the exterior in the walls of the mechanical room housing the machine, required for combustion air and for ventilation. It is through these openings that the noise associated with the air-intake and casing radiated components is of concern. The second path is the combustion exhaust system which is piped to the outdoors and discharged at or near the roof.

7.7.1.1 Estimation of Interior Sound Levels

Within the mechanical room, the sound level at 1 meter from a diesel-driven generator can be estimated by use of the following expression:

$$L_p\ (A) = 87 + 10 \log (KW)^*,\ dBA;$$

the characteristic frequency spectrum is Class II.

The sound levels at 1 meter are listed in Table 11 for the range of equipment sizes normally found in buildings.

7.7.1.2 Estimation of Exhaust Sound Power Levels

Because the exhaust system is piped to the outdoors, sound power rather than sound pressure is the preferred basis for estimating the level of this component.

The A-Weighted Sound Power Level of the exhaust at the point it leaves the engine can be estimated from the following expression:

*The correlation has been expressed in kilowatts since the systems are typically rated on this basis. The brake-horsepower of the engine is about 1.5 times the KW capacity assuming an electrical efficiency of 90%.

TABLE 11

Diesel-Driven Electrical Power Generators

Equipment Capacity, KW	Room Sound Level @ 1 Meter dBA re $2x10^{-5}$ N/m^2	Exhaust Sound Power Level dBA re 10^{-12} Watt
40 - 56	104	123
57 - 71	105	124
72 - 89	106	125
90 - 112	107	126
113 - 141	108	127
142 - 178	109	128
179 - 224	110	129
225 - 282	111	130
283 - 355	112	131
356 - 447	113	132
448 - 562	114	133
563 - 708	115	134
709 - 891	116	135
892 -1122	117	136

$$L_w\ (A) = 106 + \log\ (KW),\ \text{dBA re } 10^{-12}\ \text{Watt;}$$

the characteristic frequency spectrum is Class I.

The sound power levels of the "unmuffled" exhaust are listed in Table 11 for the range of equipment sizes normally found in the buildings.

7.7.2 Gas Turbine Equipment

The noise control problems with turbine-driven equipment are similar to those for the diesels discussed above. However, there are generally three separate paths for the noise to reach the outdoors:

- Casing radiated noise; path is generally through ventilation openings in the mechanical room;
- Combustion air-intake noise; path is typically ducted to the exterior at or near the level of the mechanical room;
- Exhaust noise; path is ducted to the exterior through a roof-stack.

7.7.2.1 *Casing Radiated Noise*

The estimation of casing radiated noise impact on the exterior environment involves a room-acoustics analysis; therefore sound level at 1 meter is the preferred metric. This can be estimated using relationship:

$$L_p\ (A) = 101 + 5 \log\ (KW),\ \text{dBA;}$$

the characteristic spectrum shape is Class III.

The sound levels at 1 meter are listed in Table 12 for equipment in the capacity range found in buildings.

TABLE 12

Gas Turbine - Driven Electrical Power Generators

Equipment Capacity KW	Casing Radiated Noise Sound Level at 1 Meter dBA re $2x10^{-5}$ N/m^2	Intake Noise A-Weighted Sound Power Level dBA re 10^{-12} Watt	Exhaust Noise A-Weighted Sound Power Level dBA re 10^{-12} Watt
200 - 329	113	122	124
330 - 529	114	125	126
530 - 849	115	128	128
850 - 1299	116	131	130
1300 - 1999	117	134	132
2000 - 3299	118	137	134
3300 - 5000	119	140	136

7.7.2.2 *Combustion Air-Intake Noise*

Since the combustion air is normally ducted from outdoors, noise estimation on a power level basis is preferred. The A-weighted sound power level may be estimated from the following equation:

$$L_w\ (A) = 86 + 15 \log (KW),\ \text{dBA re } 10^{-12} \text{ Watt;}$$

characteristic frequency spectrum is Class III.

Table 12 lists the sound power levels of this noise component over the range of equipment sizes found in buildings.

7.7.2.3 *Exhaust Noise*

The exhaust noise is always ducted to the outside; therefore, sound power level is the preferred basis for noise estimation. The recommended equation is:

$$L_w\ (A) = 100 + 10 \log (KW),\ \text{dBA re } 10^{-12} \text{ Watt;}$$

the characteristic frequency spectrum is Class II.

The sound power levels of this noise component will be found listed in Table 12 for the range of equipment sizes found in buildings.

7.8 Transformer Equipment

It is proposed that noise estimation for indoor transformers be based on the NEMA Standard [15] sound level.* Due to the

*Since NEMA ratings are *maximum* allowable levels, it is preferable to use a *measured* NEMA level, when available, for estimation purposes.

characteristic dimensions of these units, we recommend using the NEMA value for estimating the sound level at 1 meter from the transformer tank or cabinet. The characteristic frequency spectra are Class IVA for radiant-cooled transformers and Class IVB for fan-cooled transformers.

8. DEVELOPMENT OF PROCEDURES FOR ESTIMATING THE SOUND LEVEL AT A POINT OUTDOORS DUE TO EXTERIOR BUILDING MECHANICAL EQUIPMENT

For building mechanical equipment installed outdoors, there are generally only two noise reduction elements in the source to receiver path that require consideration in the Permit Scheme. The first is the natural attenuation, due to spreading, over the distance between source and receiver. The second is the attenuation due to shielding by barriers in some form that break the line-of-sight between source and receiver. The directivity of the noise source due to local reflections must, of course, be considered in the calculations, but this factor affects only the initial level of the source and not the attenuation over the path to the receiver.

8.1 Attenuation Due to Spreading

In using the worksheets developed in Section 10 to compute the sound level at a point outdoors resulting from an equipment source at distance, d, the assumption is made that the source-directivity factor, Q, will always be at least 2.*

In the Permit Scheme, the noise of exterior equipment is to be expressed as an A-Weighted Sound Power Level the sound level at the receiver point will thus be a function of the distance, d, and the corresponding L_W (A).

*This is true for all sources considered in the Permit Scheme except point sources emanating from roof-stacks; this is accounted for on the specific worksheets for diesel and turbine-driven generators.

The attenuation due to spreading is tabulated in Table 13 in 1 dB increments for source to receiver distances from 10 - 300 feet. The table is valid for source to receiver distances greater than about one-third of a major equipment-dimension. Where this restriction is violated at a source distance of concern, a method is described in Section 9.3 for dealing with the situation.

8.2 Attenuation Due to Barriers

In using the worksheets developed in Section 10 to compute the sound level outdoors, a credit of 5 dB is applied for the insertion loss of a barrier that just breaks the line-of-sight between the source and receiver. Greater attenuations than this can, of course, be realized for a barrier that extends some distance beyond that necessary to just provide a break in the sight-line. A procedure for calculating the actual barrier insertion loss is provided in Appendix 2 and its use in the worksheets is made optional.

8.3 Corr. due to reflections

TABLE 13

Outdoor Equipment: Attenuation Due to Spreading

Source to Receiver Distance, D. Feet	Attentuation $[L_w(A) - L_p(A)]$* dB
10	18
11	19
12	20
13 - 14	21
15 - 16	22
17 - 18	23
19 - 21	24
22 - 24	25
25 - 27	26
28 - 30	27
31 - 34	28
35 - 38	29
39 - 42	30
43 - 47	31
48 - 53	32
54 - 60	33
61 - 67	34
68 - 75	35
76 - 84	36
85 - 94	37
95 - 106	38
107 - 119	39
120 - 133	40
134 - 150	41
151 - 168	42
169 - 189	43
190 - 212	44
213 - 238	45
239 - 267	46
268 - 300	47

*The reference level for L_w is 10^{-12} Watt; for L_p, the reference level is 2×10^{-5} N/m^2

9. DEVELOPMENT OF PROCEDURES FOR ESTIMATING THE SOUND LEVEL AT A POINT OUTDOORS DUE TO BUILDING MECHANICAL EQUIPMENT LOCATED INDOORS

For equipment installed indoors, the calculation of the sound level at a point outdoors is significantly more complex than that for an outdoor source. The procedure must be carried out in two steps: the first step determines the sound level present at an opening in the building face due to the interior noise source; the second step determines the sound level at the receiver point by taking into account the attenuations due to spreading, directivity and shielding.

9.1 Procedure for Use with Ducted Fan Equipment

This procedure, in general, applies to the discharge side of exhaust and return-air fan systems, and to the intake side of supply-air fans. The basis for the procedure is that the sound power level of the equipment, on the side of interest, is ducted to the outside.

In the procedure, the attenuation provided by such elements as absorptive duct-lining and commercial sound traps is determined by taking into account the characteristic spectrum shape of the noise source and the frequency-dependent properties of these noise-reduction elements.

To use the procedure the first step is to obtain the following information about the fan equipment:

1. Fan-type (air-foil, forward curved, etc.)
2. Wheel diameter
3. Design cfm

4. Design total static pressure
5. Static efficiency at design operating point
6. Peak static efficiency for the fan design
7. Noise spectrum classification (I or II).

Using the above information, the second step is to determine the A-weighted sound power level of the fan at the specified operating conditions. This may be done by following the procedure outlined in Section 5.2 using Tables 1, 2, 3 and 4.

The third step is to determine the noise-reduction of the elements in the duct between the source and the building opening, as described below.

9.1.1 Attenuation Due to Ductlining

In Table 14, the attenuation due to the presence of absorptive lining in straight ducts is tabulated for the following combination of variables: Rectangular vs. round duct; one-inch vs. two-inch ductlining; Class I vs. Class II input spectrum shape.* For convenience the table is established in increments of 1 dB. To use this table, first select the column corresponding to the appropriate input spectrum, duct shape and lining thickness; then, determine the attenuation in dB provided by the length of lined duct shown on the drawing.

In the worksheets developed in Section 10, a credit of 5 dB is also given for elbows that are followed by a minimum of 10 feet of lining downstream of the turn. However, this lining beyond the elbow is not counted in the length used when entering Table 14.

*The values shown in this table are tentative. They are based on empirical evidence developed from many years of practical experience. However, recent work in this field sponsored by ASHRAE may lead to a revision of this table upon publication of the results.

TABLE 14

Attenuation of Ductlining*

Input Spectrum	Class I				Class II			
Duct Shape	Rectangle		Round		Rectangle		Round	
Lining Thickness	1"	2"	1"	2"	1"	2"	1"	2"
Required Length → For This Attenuation ↓	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.	Ft.
1	1	1	1	1	1	1	1	1
2	3	2	3	2	2	2	2	1
3	5	3	5	3	3	3	3	2
4	7	4	7	4	4	4	4	2
5	9	6	9	6	6	5	5	3
6	11	8	12	8	8	6	7	4
7	13	10	15	10	10	7	9	5
8	16	12	18	13	12	8	11	6
9	19	14	21	16	14	9	13	7
10	22	16	24	19	16*	10	15	9
11	25	18	27	22	18	11	17	11
12	28	20	30	25	21	12	19	13
13	31	22	34	28	24	14	21	15
14	34	24	38	31	27	16	23	17
15	37	27	42	34	30	18	25	19
16	40	30	46	38	33	20	27	21
17	43	33	50	42	36	22	29	23
18	46	36	54	46	39	24	31	25
19	50	39	58	50	42	26	33	27
20	54	42	62	54	45	28	35	29
21	58	45	66	58	48	30	37	31
22	62	48	70	62	52	32	40	33
23	66	51	74	66	56	34	43	36
24	70	54	79	71	60	36	46	39
25	74	57	84	76	64	38	49	42
26	78	60	89	81	68	40	52	45
27	82	63	94	86	72	42	55	48
28	86	66	99	91	76	44	58	51
29	90	69	104	96	80	46	61	54
30	94	72	109	101	84	48	64	57

*For example: To obtain 10 dB attenuation of a Class II input spectrum using rectangular ductwork with 1"-thick lining requires a length of 16 feet.

effect of 90° bends in ducting?

9.1.2 Attenuation of Packaged Sound Attenuators

To establish a data base for classifying commercial sound attenuators according to their insertion-loss as a function of input spectrum shape, the catalog data published by eight manufacturers were subjected to a detailed review.

The best basis found for normalizing the performance of these devices was in terms of static pressure-drop, unit-length and shape (rectangular vs. round).

The best classification in terms of static pressure-drop, at 1,000 fpm entering face-velocity, was as follows:

Shape	Pressure Drop	Classification
Rectangular	< 0.10 in. w.g.	"Low"
Rectangular	0.10 - 0.30 in. w.g.	"Medium"
Rectangular	> 0.30 in. w.g.	"High"
Cylindrical	< 0.03 in. w.g.	"Low"
Cylindrical	> 0.03 in. w.g.	"High"

There was some standardization of length found with the rectangular sound attenuators (3', 5', 7' long). The length of the cylindrical attenuators generally was on the order of 2 - 3 diameters, but this was not standardized among the various manufacturers.

The average insertion loss for typical packaged sound attenuators is tabulated in Table 15 in line with the classification scheme discussed above. Insertion loss values are given for the two types of input spectra (Class I and II) that bracket most of the fan designs likely to be found in HVAC systems.

TABLE 15

Insertion Loss of Packaged Sound Attenuators

Shape	Length	Pressure Drop	Average Insertion Loss, dB Class I	Class II
	3'		11	16
Rectangular	5'	Low	16	21
	7'		18	25
	3'		14	20
Rectangular	5'	Medium	18	25
	7'		22	29
	3'		18	26
Rectangular	5'	High	22	33
	7'		24	35
Cylindrical	2-3	Low	15	20
> Duct Diam.	Diameters	High	19	26
Cylindrical	2-3	Low	11	11
= Duct Diam.	Diameters	High	14	17

Need corr. due to duct air vel.

9.1.3 Correction for Area of Duct Opening in Building Face

A conversion is required between the sound power in the duct and the resulting sound pressure at the opening in the building face. If the duct terminates in a louvered plenum, but there is an on-axis opening to the outside, no credit is taken for the area of the louvered face. A correction of -3 dB is allowed if the duct discharges into a louvered plenum where the opening to the outside is at least 45° off-axis of the duct centerline. The corrections for duct cross-sectional area are given in Table 16.

At this point in the procedure the sound level at the building face can be determined by subtracting the corrections for attenuation in the ductwork and for duct cross-sectional area.

For example, consider the installation of an exhaust fan of the Airfoil type that is selected for a duty of 60,000 cfm @ 2.5" total static pressure. The fan is 40" in diameter; for this service it will operate at 56% of peak static-efficiency (this is the fan used in the example given in Section 5.3.1.

The fan is connected to an exhaust duct of dimensions, 54" x 80" (30 sq. ft.), in series with a 5 ft. "low" pressure-drop sound attenuator. The duct terminates at a louvered opening in the building face, on-axis with the duct centerline: Determine the sound level at the building face.

Step 1. Calculate the A-weighted sound power level of the fan (Refer to Section 5.2)

$$L_W(A) = K_A + A + B + C$$
$$= 35 + 48 + 8 + 13$$
$$= 104 \text{ dBA re } 10^{-12} \text{ Watt; the spectrum is Class I.}$$

TABLE 16

Correction for Duct Cross-Sectional Area

Duct area, ft.^{2}	Correction Factor, dB
9-11	0
11.1 - 14	1
14.1 - 18	2
18.1 - 22	3
22.1 - 28	4
28.1 - 35	5
36 - 44	6
45 - 56	7
57 - 70	8
71 - 89	9
90 - 112	10
113 - 141	11
142 - 180	12
181 - 225	13

• corr. needed for branches or take-offs in duct system.

• corr. needed for changes in duct size downstream.

Step 2. Determine the correction for the sound attenuator. From Table 15, a 5 ft. "low" pressure-drop unit has an attenuation of 16 dB for a Class I input spectrum.

Step 3: Determine the area correction. From Table 16, the area correction for a 30 ft.^{2} duct is 5 dB.

The sound level at the building face is determined by subtracting the results of Steps 2 and 3 from Step 1.

$$L_p\ (A) = 104 - 16 - 5 = 83 \text{ dBA.}$$

9.2 Procedure for Use with Non-Ducted Mechanical Equipment

This procedure deals with the noise radiated by mechanical equipment into an enclosing space that is coupled to the outdoors by ventilation openings in the building. The sound transmission through exterior walls of the building is not considered an important factor in this analysis because in nearly all practical situations the walls are "flanked" by the ventilation openings.

The first step is to determine the sound levels at 1 meter for each machine of interest using the appropriate equations developed in Section 7. List these in descending order of magnitude and concentrate first on only those noise sources that are the highest.

The second step is to determine the distance in feet from the machine of interest to the closest opening to the outside; use Table 17, following, to obtain the correction factor to be subtracted from the sound level at 1 meter to obtain the sound level on the room side of the building opening. An additional correction of 3 dB is permitted if the machine is shielded from the opening by plenums or other large barriers.

TABLE 17*

Correction for Distance Between Machine and Building Opening

Distance from Openings ft.	Correction Factor dB
5	2
6-7	3
8-9	4
10-11	5
12-14	6
15-18	7
19-22	8
23-29	9
30-36	10
> 36	10

*Table 17 is based on the discussion in Section 7 concerning the departure of typical mechanical rooms from classical direct/reverberant field theory. The correction is limited to 10 dB, maximum, because of the small size of the data base.

The next step is to compute the sound level on the room side of the building opening by subtracting the correction factor determined in step two from the machine sound level at 1 meter determined in step one.

At this point it is best to determine whether there are other noise sources in the mechanical room that will significantly influence the sound level at the building opening that was computed for the noisiest machine. In general, noise sources 6 dB lower in level will have negligible effect unless they are closer to the opening, or several in number.

We recommend that the mechanical installations in a given room be screened first on the basis of the noisiest sources; if this analysis indicates that no environmental problems exist at the receiver point (outdoors), then other lower level sources probably can be neglected. However, for those situations where the sound levels of several sources are to be combined, a procedure is provided in Appendix 3 for the addition of decibels; decibels are logarithmic values and cannot be summed by normal algebraic additions.

The final step in the analysis is to adjust the sound level, determined for the room-side of the building opening, for any losses through the opening; the result will be the sound level at the exterior face of the building. The losses to be accounted for, if any, generally will be those due to the installation of packaged sound attenuators or acoustical louvers at the opening. The average insertion loss of sound attenuators may be found in Table 15; the insertion loss of typical acoustical louvers is given in Table 18, following.

TABLE 18

Insertion Loss of Acoustical Louvers

Pressure Drop	Average Insertion Loss, dB		
	Class I	Class II	Class III
Low: < 1.0" w.g. @ 1000 fpm	8	10	10
High: ≥ 1.0" w.g. @ 1000 fpm	10	13	12

As an example in using this procedure, consider the installation of a centrifugal chiller, of the internally-geared type, having a capacity of 600 tons. The chiller is located in a mechanical room at a distance of 12 feet directly opposite a ventilation opening in the exterior wall. A low-pressure drop acoustical louver is installed in the opening.

Step 1: Determine the sound level of the chiller using Table 9A. The 600 ton unit is found to produce a sound level of 95 dBA at a distance of 1 meter. For the internally-geared machine, the characteristic spectrum is Class III.

Step 2: Find the magnitude of the distance correction factor using Table 17. For 12 feet, the factor is 6 dB.

Step 3: Determine the insertion loss of the acoustic louver, using Table 18, for a Class III input spectrum. This factor is 10 dB.

The sound level at the exterior face of the building due to this compressor installation will be found by subtracting the results of steps 2 and 3 from step 1.

$$L_p(A) = 95 - 6 - 10 = 79 \text{ dBA}.$$

Had the acoustical louver not been installed, the sound level would be 10 dB higher, or 89 dBA.

9.3 Procedure for Determining the Sound Level at a Point Outdoors Due to a Source at the Building Opening

In subsections 9.1 and 9.2 above, procedures were given for determining the sound level at the exterior face of a building opening resulting from a machine located inside. This subsection presents a procedure for determining the corresponding sound level at some distance from the building opposite the opening.

The attenuation of sound with distance from a finite rectangular plane source (or opening) of dimensions a and b ($a < b$) has the following characteristics along an axis perpendicular to the opening[26]:

1. In the region between the opening and a distance of about a/3, there is no change in sound level.

2. In the region between about a/3 and b/3, the sound level drops at the rate of 3 dB per doubling of distance.

3. In the region beyond about b/3, the sound level drops at the rate of 6 dB per doubling of distance.

Thus, to determine the sound level at some distance, d, from a building opening, the "source dimensions" must be taken into consideration; for a given distance, the attenuation reduces with an increase in the size of the opening.

For distances equal to or greater than about b/3, the attenuation can be determined from the following equation:

$$L_{p_o} - L_{p_d} = 20 \log d - 10 \log (ab) + 10, \text{ dB}$$

where L_{p_o} is the sound level at the opening,

L_{p_d} is the sound level at distance d, and

a & b are the dimensions of the opening; thus (ab) is the area of the opening.

In most practical field situations, the distance to some point of reference outdoors more than likely will be equal to or greater than b/3. For example, consider a reference distance of 20 feet from the building opening. In this case, the "effective" building opening (that portion coupled to the noise source) would need to exceed 60 feet in width before use of the above equation would be invalid. For ducted fan equipment, the effective building opening will, in general, be the same as the duct cross-sectional area. For ventilated mechanical rooms, the opening is not likely to be much greater than one bay in width. Therefore, the 60 feet allowable width for an opening to receiver distance of 20 feet would accommodate most practical situations.* The use of this

*One frequently observes buildings with a louvered facade extending throughout an entire floor. However, behind the louvers there will generally be a series of plenums or otherwise compartmentalized spaces associated with individual noise sources. The "effective" opening relative to a specific source in most cases will be less than observed from outside.

equation might be marginal at a reference distance less than 10 feet because the corresponding maximum effective opening of 30 feet is close to a typical bay spacing.

In addition to sound levels along the axis varying with distance from the opening, there will be a variation with angle *away* from the axis because the radiation of sound from a rectangular opening is directional[27]. The sound level is about 3 dB less for ref angles in the range of 30° - 60° than on-axis; for angles in the range of 60° to 90°, the radiation is down about 6 dB with respect to the on-axis level.

To calculate the outdoor sound level at some point of reference, then it is necessary to determine four quantities: the sound level at the opening, the area of the opening, the on-axis distance to the point of interest, and the vertical angle between the receiver and the perpendicular axis through the center of the opening. This can be expressed as:

$$L_{p_d} = L_{p_o} - (\text{Directivity Factor}) - (\text{Distance Factor}) + (\text{Area Factor})$$

where Directivity Factor = 0 dB for angles between 0°-30°;
3 dB for angles between 30°-60°;
6 dB for angles between 60°-90°;
Distance Factor = $20 \log d + 10$ dB;
Area Factor = $10 \log (ab)$.

Values of Distance and Area Factors appropriate for use in the worksheets in Section 10 are listed in Table 19.

TABLE 19

DISTANCE AND AREA FACTORS FOR USE IN
DETERMINING SOUND ATTENUATION FROM A RECTANGULAR OPENING

Distance Factor

Distance (feet)	Factor (dB)
14 - 15	33
16 - 17	34
18 - 19	35
20 - 21	36
22 24	37
25 - 27	38
28 - 30	39
31 - 34	40
35 - 38	41
39 - 42	42
43 - 47	43
48 - 53	44
54 - 60	45
61 - 67	46
68 - 75	47
76 - 84	48
85 - 94	49
95 - 106	50
107 - 119	51
120 - 133	52
134 - 150	53
151 - 168	54
169 - 188	55
189 - 211	56
212 - 237	57
238 - 266	58
267 - 300	59

Area Factor

Area of Opening (sq ft)	Factor (dB)
9 - 11	10
12 - 14	11
15 - 18	12
19 - 22	13
23 - 28	14
29 - 35	15
36 - 45	16
46 56	17
57 - 70	18
71 - 89	19
90 - 112	20
113 - 141	21
142 - 178	22
179 - 224	23
225 - 282	24
283 - 355	25
356 - 447	26
448 - 562	27
563 - 708	28
709 - 891	29
892 -1120	30

10. DEVELOPMENT OF MECHANICAL EQUIPMENT NOISE GUIDELINES AND WORKSHEETS FOR MODEL PERMIT SCHEME

The guidelines and worksheets developed in this section are intended for use in screening building mechanical system designs and the corresponding installation details for potential noise impact on the exterior environment.

A number of steps are involved in this screening process that require the availability of detailed mechanical drawings and specifications for the building, in order to obtain the necessary data inputs for the worksheet analysis. In addition, certain architectural drawings are required, such as site plans and building elevations in order to establish the locations of exterior equipment and openings to interior mechanical rooms with respect to the outdoor reference points designated for analysis.

In general, the worksheets proceed in a step-wise manner, beginning with an identification of the piece of equipment of concern, together with technical data concerning its operation sufficient to establish the level of sound energy produced by the machine. The location of the equipment, relative to the outdoor reference point, is then established and those factors are listed that will affect the sound attenuation in the path between source and receiver. The magnitudes of these sound attenuating components are then calculated and combined to determine the total losses in the transmission path. The final step is to calculate the sound level at the chosen outdoor point due to the building equipment source under consideration.

Two basic types of worksheet procedures are needed: one to deal with outdoor sources; the other to deal with sources located indoors but coupled to the outdoors through ducts or ventilation openings. Sample worksheets are provided in Appendix 4.

10.1 <u>Worksheet for Calculation of the Sound Level at an Outdoor Reference Point Due to Exterior Building Mechanical Equipment</u>.

Worksheet A illustrates the procedure to be followed in calculating the outdoor sound level at some point of concern due to a piece of mechanical equipment installed outside the building. The worksheet requires the use of the tables provided in previous sections of the report for data entries with regard to source noise levels and sound attenuating elements in the path between source and receiver. The other information required for the worksheet must be obtained from the drawings and specifications prepared by the Architect/Mechanical Designer.

Part 1 of the worksheet lists the data required concerning the item of equipment to be analyzed and the necessary details about the installation configuration. Part 2 contains the procedure for calculating the sound level at the reference point based on the information developed in Part 1.

To illustrate the use of Worksheet A, the following examples are given:

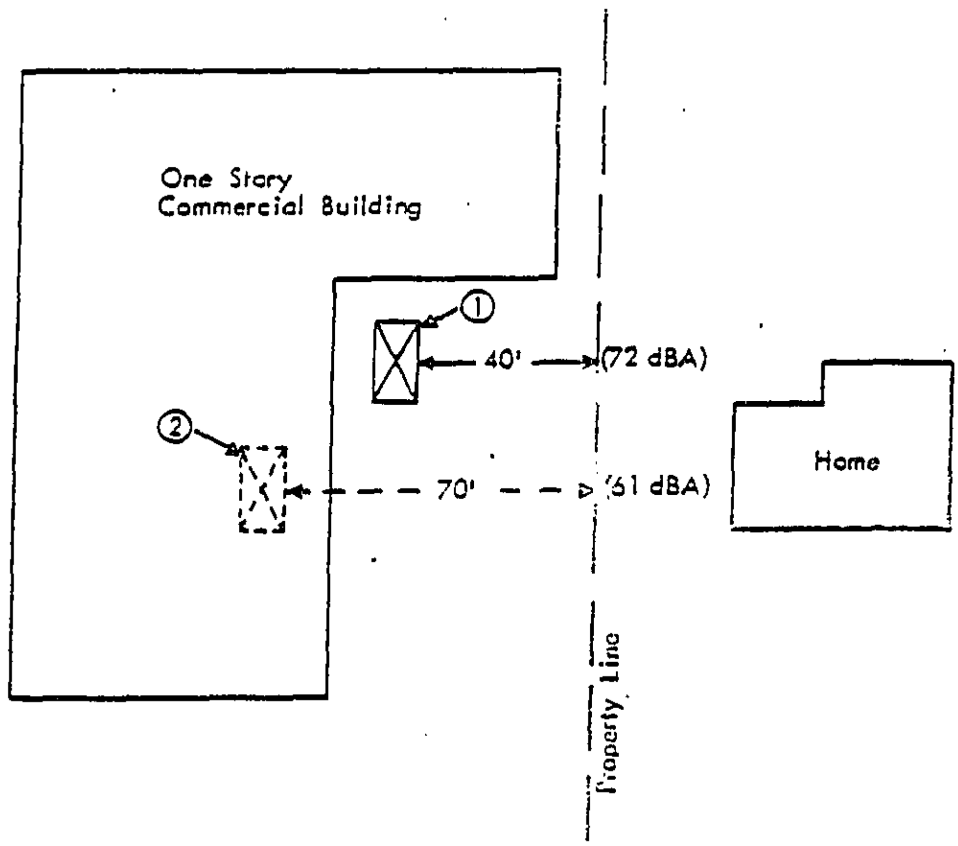
Example 1: Cooling Tower Installation

A 50-hp centrifugal-type cooling tower is to serve a one-story commercial building that is located adjacent to a residential neighborhood. The reference point for evaluation is the nearest point on the intervening property line as illustrated in Figure 16.

Two installation options exist: ?

1. Location of the cooling tower on-grade, near an inside corner.
2. Location on the roof, with a set-back sufficient to prevent its being seen from the close-in approach to the building. However, *it will be visible from the residences.*

50 Horsepower Centrifugal Cooling Tower



Option 1: Location on Grade
Option 2: Location on Roof

FIGURE 16. OUTDOOR EQUIPMENT INSTALLATION - EXAMPLE 1

On the sample worksheet shown in Table 20, the solution is first obtained for option 1; the solution for option 2 is shown in parentheses.

Part 1

Steps 1-4: Identify the equipment type and size.

Step 5: List the sound power level of the equipment based on Table 6 (96 dBA re 10^{-12} Watt).

Step 6: Identify the installation locations for the two options.

Step 7: Indicate that two reflecting surfaces exist for option 1 and none exist for option 2. (The tables used in conjunction with the worksheet assume all sources have a reflecting plane at the base.)

Step 8: Indicate that the line of sight to the reference point (property line) is unobstructed in both cases.

Step 9: Indicate the equipment distances to the reference points.

Part 2

Step 10: Enter the sound power level determined in step 5.

Step 11: Correction for directivity; 6 dB for option 1, 0 dB for option 2.

Step 12: Correct the radiated sound power level for directivity; 96 ⟶ 102 for option 1, no change for option 2.

Step 13: Correction for shielding; none.

Step 14: Correct radiated sound power level for shielding; no change.

Step 15: Correct for distance using Table 13; 30 dB for option 1, 35 dB for option 2.

Step 16: Subtract the distance correction from the adjusted sound power level to find the sound level at reference point: 72 dBA for option 1, 61 dBA for option 2.

TABLE 20

Outdoor Equipment Installation - Example 1 Worksheet

WORKSHEET A

Outdoor Equipment

Procedure for Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description Centrifugal Fan Cooling Tower
2. Identification Symbol on Drawings CT-1
3. Manufacturer and Model Number XYZ 1040 A
4. Operating Conditions 50 hp
5. A-Weighted Sound Power Level 96 dBA re 10^{-12} Watt
   Spectrum Class II
   ✓ Calculated from tables (attach worksheet)
   ___ Certified test data (attach substantiation)

6. Installation Location:

   ✓ On-grade
   (✓) Roof-top

7. Presence of Nearby Reflecting Surfaces:

   (✓) a. None ___ b. One ✓ c. Two

8. Line of Sight between Equipment and Reference Point:

   ✓, (✓) a. Unobstructed
   ___ b. Broken by solid barrier, roof setback, etc.

9. Distance, Equipment to Reference Point 40, (70) feet

   ✓, (✓) Perpendicular distance
   ___ Slant distance

TABLE 20 (continued)

Outdoor Equipment Installation - Example 1 Worksheet

Worksheet A (continued)

Part 2: Sound Level Estimation

10. Sound Power Level (from line 5) 96 dBA re 10^{-12} Watt
11. Correction for Directivity:
    a. If 7a checked, enter 0 (0) dB
    b. If 7b checked, enter 3 ____dB
    c. If 7c checked, enter 6 6 dB

12. Add lines 10 and 11 (a, b, or c) 102, (96) dBA re 10^{-12} Watt
13. Correction for Shielding:
    a. If 8a checked, enter 0 0, (0) dB
    b. If 8b checked, enter:
        (1) 5 (allowance w/o calc.) or ____dB
        (2) Result of computation using Appendix 2 (attach calc's.) ____dB

14. Subtract line 13 from line 12 102, (96) dBA re 10^{-12} Watt
15. Distance Correction (from Table 13 using distance shown on line 9) 30, (35) dB
16. Subtract line 15 from line 14 to get Sound Level at Reference Point 72, (61) dBA re $2 \times 10^{-5} N/m^2$

This example demonstrates the influence of directivity and distance on the sound level at the boundary of the receiver. The 11 dB change from option 1 to option 2 in this case could well represent the difference between complaint and acceptance by the adjoining neighborhood.

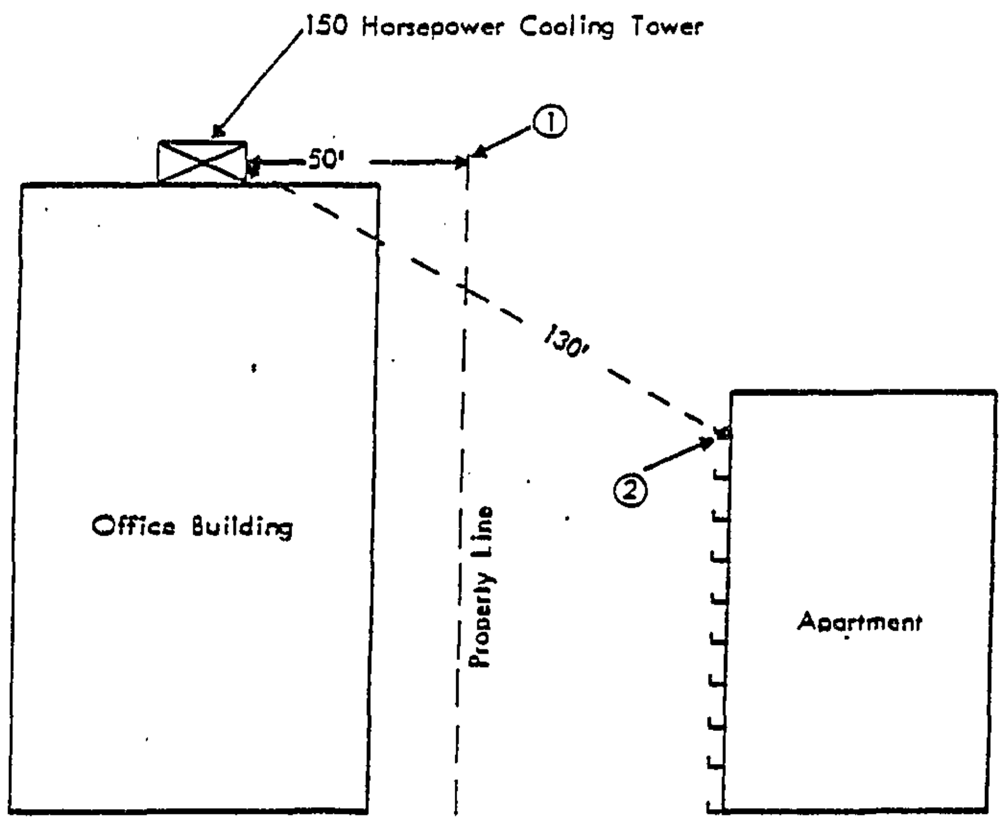
Example 2: Cooling Tower Installation (High-Rise)

A 150 hp centrifugal-type cooling tower is to be installed on the roof of a high-rise office building. The closest property line boundary perpendicular to the tower is 50 feet away. There is a lower apartment structure with exterior balconies across the street. The slant distance from tower to the closest balcony is 130 feet, but the line of sight is broken. (See illustration in Figure 17.) Determine the sound level due to the tower at two points: 1) the property line and 2) the balcony face.

The worksheet for Example 2 is illustrated in Table 21, first with the calculations for case 1 (the property line) and then with the calculations for case 2 (the balcony face) shown in parentheses. It will be noted that the solution relative to the property line predicts a sound level of 69 dBA, whereas the level at the balcony face is 13 dBA lower (56 dBA). This demonstrates the difference between choosing a reference point at the property line as opposed to a location at the closest receiver in an existing land-use. The shielding correction accounts for 5 dB of this difference; the remaining 8 dB is the effect of the greater distance.

10.2 Worksheet for Calculation of the Sound Level at an Outdoor Reference Point Due to Interior Building Mechanical Equipment

Two types of worksheets are required for analysis of building mechanical equipment located indoors. Worksheet B-1 is to be



Case 1: Evaluation in Plane of Property Line
Case 2: Evaluation at Closest Receiver

FIGURE 17. OUTDOOR EQUIPMENT INSTALLATION - EXAMPLE 2

TABLE 21

Outdoor Equipment Installation - Example 2 Worksheet

WORKSHEET A

Outdoor Equipment

Procedure for Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description Centrifugal Fan Cooling Tower
2. Identification Symbol on Drawings CT-2
3. Manufacturer and Model Number XYZ 2050 B
4. Operating Conditions 150 hp
5. A-Weighted Sound Power Level 101 dBA re 10^{-12} Watt
   Spectrum Class II
   ✓ Calculated from tables (attach worksheet)
   ___ Certified test data (attach substantiation)

6. Installation Location:

   ___ On-grade
   ✓, (✓) Roof-top

7. Presence of Nearby Reflecting Surfaces:

   ✓, (✓) a. None ___ b. One ___ c. Two

8. Line of Sight between Equipment and Reference Point:

   ✓ a. Unobstructed
   (✓) b. Broken by solid barrier, roof setback, etc.

9. Distance, Equipment to Reference Point 50, (130) feet
   ✓ Perpendicular distance
   (✓) Slant distance

TABLE 21 (continued)

Outdoor Equipment Installation - Example 2 Worksheet

Worksheet A (continued)

Part 2: Sound Level Estimation

10. Sound Power Level (from line 5) 101 dBA re 10^{-12} Watt

11. Correction for Directivity:
    a. If 7a checked, enter 0 0, (0) dB
    b. If 7b checked, enter 3 ____ dB
    c. If 7c checked, enter 6 ____ dB

12. Add lines 10 and 11 (a, b, or c) 101, (101) dBA re 10^{-12} Watt

13. Correction for Shielding:
    a. If 8a checked, enter 0 0 dB
    b. If 8b checked, enter:
       (1) 5 (allowance w/o calc.) or (5) dB
       (2) Result of computation using Appendix 2 (attach calc's.) ____ dB

14. Subtract line 13 from line 12 101, (96) dBA re 10^{-12} Watt

15. Distance Correction (from Table 13 using distance shown on line 9) 32, (40) dB

16. Subtract line 15 from line 14 to get Sound Level at Reference Point 69, (56) dBA re 2×10^{-5} N/m^2

used for ducted air-handling equipment. Worksheet B-2 is to be used for equipment located within a mechanical room which is vented to the outdoors through openings in the side or roof of a building. The format of these procedures is similar to that used for Worksheet A discussed above: Part 1 lists the reference data required in the analysis; Part 2 contains the calculation procedure for determining the sound level at some exterior reference point.

10.2.1 Worksheet for Ducted Air-Handling Equipment

To illustrate the use of Worksheet B-1, the following example is given:

Example 3: Building Exhaust Fan

A building exhaust fan, of the vane-axial type, is ducted to an opening in the side of a high-rise office building. There is a lower apartment building across the street with balconies. The geometry is similar to that illustrated in Figure 17 which was used in conjunction with Example 2, above.

The vane-axial fan is 60" in diameter and handles 37,800 cfm against a total static pressure of 1.5 inches, w.g. The static efficiency at the chosen operating point is 64 percent; the peak static efficiency of the fan design is 80 percent.

The discharge duct connected to the fan is 54" x 64" (24 square feet) and is acoustically treated for a length of 16 feet with one-inch-thick ductliner. The duct opening at the building face is connected to a louvered plenum, on-axis with the duct centerline.

The solution shown on the sample worksheet (Table 22) is for two situations:

1. Evaluation at the property line directly opposite the opening at 20 feet.

2. Evaluation at the balcony face of the closest apartment across the street, which is a distance of 95 feet away, at an angle of 40° below the normal to the opening.

The solution for case 2 is shown in parentheses.

The reference data based on the above description is entered in Part 1 in Steps 1-10. The calculation procedure (Part 2) begins with Step 11, the calculation of fan-sound power level, using the information located in Section 5 of this report. Steps 12-17 are used to calculate the sound level at the exterior face of the building, using the information provided in Section 9.1. Steps 18-20 determine the sound level at the exterior reference point based on the procedure given in Section 9.3.

It will be noted that a 17 dB difference exists between the two reference points chosen for analysis; 3 dB is the result of a directivity correction for the 40° off-axis line of sight; 14 dB results from the greater distance to the boundary of the closest receiver.

10.2.2 Worksheet for Indoor Building Mechanical Equipment

To illustrate the use of Worksheet B-2, the following example is given:

TABLE 22
Ducted Building Exhaust Fan Installation - Example 3 Worksheet

WORKSHEET B-1

Indoor Air-Handling Equipment

Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description Vane-Axial Exhaust Fan
2. Designation on Drawings or Schedule RE-2
3. Manufacturer and Model Number XYZ 60-26½-860
4. Service Application:
   ___Supply Air
   ___Return Air
   ✓ Exhaust Air
5. Fan Type:
   ___a. Airfoil
   ___b. Backward Curved/Inclined
   ___c. Forward Curved
   ___d. Radial
   ✓ e. Vane-Axial
   ___f. Propeller
6. Fan Diameter: 60 inches
7. Fan Operating Point:
   a. Volume 37,800 cfm
   b. Total Static Pressure 1.5 inches, w.g.
   c. Brake Horsepower 14 hp
   d. Static Efficiency at Operating Point 64 %
   e. Peak Static Efficiency on Fan Curve 80 %
   f. Percent of Peak Static Efficiency at Operating Point 80 %
8. Configuration:
   ✓ a. Ducted
      (1) Duct Width 64 inches
      (2) Duct Height 54 inches
      (3) Duct Length 16 feet
      ✓ (4) Ductlining:
         (a) Lining thickness 1 inches
         (b) Length of straight lined duct 16 feet
         (c) Lined elbow with minimum 10 ft of lining beyond elbow in direction of sound propagation:
            ___Yes
            ✓ No
      ___ (5) Packaged Sound Attenuator:
         (a) Manufacturer & Model Number ___
         (b) Static Pressure Drop at 1000 fpm ___ inches, w.g.
      (6) Area of Duct Opening at Building Face 24 ft²
         ✓ (a) Plenum Opening on Axis of Duct
         ___ (b) Plenum Opening > 45° off Axis of Duct
   ___ b. Non-Ducted (Applicable to Flush-Mounted Ventilating Fans)

TABLE 22 (continued)

Worksheet B-1 (Continued)

9. Distance, Building Opening to Reference Point 20,(95) feet

 ✓ Perpendicular Distance
 (✓) Slant Distance

10. Line of Sight between Equipment and Reference Point

 ✓,(✓) a. Unobstructed
 ___ b. Broken by solid barrier, roof setback etc.

Part 2: Sound Level Estimation at Reference Point

11. Calculation of Fan Sound Power Level (Based on Lines 5,6,7):

 a. Specific Sound Power Level, K_A (Table 1) 46 dBA re 10^{-12} Watt
 b. Volume Correction, A (Table 2A) 46 dB
 c. Static Pressure Correction, B (Table 3) 4 dB
 d. Static Efficiency Correction, C (Table 4) 5 dB
 e. Sound Power Level (11a + 11b + 11c + 11d) 101 dBA re 10^{-12} Watt
 f. Spectrum Class II

*12. Correction for Lined Ductwork (only if line 8a,(4) checked):

 a. Attenuation of Straight Duct (Table 14) 10 dB
 b. Elbow Attenuation:
 (1) If "Yes" checked, enter 5 ___ dB
 (2) If "No" checked, enter 0 0 dB

*13. Correction for Packaged Sound Attenuator (only if 8a,(5) checked):

 a. Attenuation (Table 15) ___ dB
 b. Certified Test Data (attach substantiation) ___ dB

14. Adjusted Sound Power Level:

 a. Line 11e minus (Lines 12a + 12b)* 91 dBA re 10^{-12} Watt
 b. Line 11e minus Line 13a, b* ___ dBA re 10^{-12} Watt

15. Calculation of Sound Level at Building Opening:

 a. Correction for Duct Cross-Sectional Area (from Table 16 using area of 8a, (6), 4 dB
 b. Correction for Plenum Loss
 (1) If 8a,(6),(a) checked, enter 0: 0 dB
 (2) If 8a,(6),(b) checked, enter 3: ___ dB
 c. Add Lines 15a and 15b: 4 dB
 d. Sound Level at Opening (Line 14(a or b) minus Line 15c): 87 dBA re 2×10^{-5} N/m²

*A correction for either duct lining or a packaged sound attenuator is allowed but not both.

TABLE 22 (continued)

Worksheet B-1 (Continued)

*16. Correction for Directivity (use only if 10a is checked)
Vertical angle between reference point and opening:

0° - 30°, enter 0 — 0 dB
30° - 60°, enter 3: — (3) dB
60° - 90°, enter 6: — ___ dB

*17. Correction for Shielding (use only if 10b is checked):

a. Allowance w/o Calculations, enter 5 ___ dB
b. Computation from Appendix 2 (attach calculations): ___ dB

18. Adjusted Sound Level at Building Opening (Line 15d minus 16, or, Line 17): 87, (84) dBA re $2 \times 10^{-5} N/m^2$

19. Correction for Distance to Reference Point:

a. Distance Factor (From Table 19 and Line 9): 36, (50) dB
b. Area Factor (From Table 19 and Line 8a, (6)): 14 dB
c. Line 19a minus Line 19b: 22, (36) dB

20. Sound Level at Reference Point: (Line 18 minus Line 19c) 65, (48) dBA re $2 \times 10^{-5} N/m^2$

*A correction for either directivity or shielding is allowed, but not both.

Example 4: Chiller Room

A building mechanical room is located on the top floor of the office building described in the previous example. A louvered opening 11' x 25' (275 square feet) used for ventilation is located on the side of the building opposite the apartment structure. Inside this mechanical room is a 950-ton hermetic, internally-geared centrifugal chiller that is a distance of 8 feet from the exterior opening in which low-pressure drop acoustical louvers are installed.

As in the previous example, two solutions are given (see Table 23): one, for the property-line case; the other, based on the location of the closest receiver. The necessary reference data are listed in Steps 1-10 of Part 1; the sound level of the chiller entered in Step 5 was obtained from Table 9 in Section 7.2 of this report.

The calculation procedure in Steps 11-14 determines the sound level at the building exterior, using the information provided in Section 9.2. Steps 15-19 result in the sound levels at the outdoor reference points based on the procedure given in Section 9.3.

In comparison with Example 3, the same 17 dB difference in level results between the two reference points chosen because the exterior geometry is identical. The absolute values of the sound levels at the corresponding reference points, however, differ by 6 dB for the two individual sources (fan and chiller).

10.3 Combining Sound Levels at an Outdoor Reference Point

Examples 2, 3, and 4 were based on the same exterior geometry, with one building considered as the noise source, and an

TABLE 23

Chiller Room Installation - Example 4 Worksheet

WORKSHEET B-2

Building Mechanical Equipment Indoors

Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description Hermetic Internally-Geared Centr. Chiller
2. Identification Symbol on Drawings CH-1
3. Manufacturer and Model Number XYZ IG-95
4. Operating Conditions 950 Tons (Air-Conditioning Service)
5. A-Weighted Sound Level at 1 meter 97 dBA re 2×10^{-5} N/m^2
   Spectrum Class III
6. Distance between Equipment and Closest Opening in Exterior Wall 8 feet
   ✓ a. Opening unshielded from equipment
   ___ b. Opening shielded from equipment
7. Dimensions of Opening:
   a. Height 11 ft
   b. Width 25 ft
   c. Area 275 ft^2
8. Acoustical Treatment of Opening:
   ___ a. None
   ___ b. Packaged Sound Attenuator
   ✓ c. Acoustical Louvers
9. Distance, Building Opening to Reference Point 20,(95) ft
   ✓ Perpendicular Distance
   (✓) Slant Distance
10. Line of Sight between Equipment and Reference Point
   ✓,(✓) a. Unobstructed
   ___ b. Broken by solid barrier, roof setback, etc.

TABLE 23 (continued)

Worksheet 3-2 (Continued)

Part 2. Sound Level Estimation at Reference Point

11. A-Weighted Sound Level at 1 meter (from Line 5) 97 dBA re $2x10^{-5}N/m^2$

12. Correction for Distance to Closest Opening:
    a. Distance (using Table 17 and Line 6) 4 dB
    b. Shielding:
       Unshielded, enter 0 0 dB
       Shielded, enter 3 ___dB
    c. Total Correction (12a + 12b) 4 dB

13. Attenuation across Opening:
    a. If 8a is checked, enter 0 ___dB
    b. If 8b is checked, use
       ___Table 15 or
       ___Certified Ratings(attach substantiation) ___dB
    c. If 8c is checked, use
       ✓ Table 16 or
       ___Certified Ratings(attach substantiation) 10 dB

14. A-Weighted Sound Level at Exterior Side of Opening:
    Line 11 minus Line 12c minus Line 13a,b,c: 83 dBA re $2x10^{-5}N/m^2$

*15. Correction for Directivity (use only if Line 10a is checked:
    Vertical angle between reference point and opening:
    0° - 30°, enter 0: 0 dB
    30° - 60°, enter 3: (3) dB
    60° - 90°, enter 6: ___dB

*16. Correction for Shielding (use only if line 10b is checked):
    a. Allowance w/o Calculations, enter 5: ___dB
    b. Computation from Appendix 2 (attach calc's): ___dB

17. Adjusted Sound Level at Building Opening
    (Line 14 minus Line 15, or, Line 16): 83, (80) dBA re $2x10^{-5}N/m^2$

18. Correction for Distance to Reference Point:
    a. Distance Factor (from Table 19 and Line 9) 36(50) dB
    b. Area Factor (from Table 19 and Line 7c): 24 dB
    c. Line 18a minus Line 18b: 12, (26) dB

19. Sound Level at Reference Point
    (Line 17 minus Line 18c) 71, (54) dBA re $2x10^{-5}N/m^2$

*A correction for either directivity or shielding is allowed, but not both.

adjacent building considered as the receiver. Consider now the combined effect at the balcony face of the closest apartment:

Sound Level due to Cooling Tower (Example 2)	56 dBA
Sound Level due to Exhaust Fan (Example 3)	48 dBA
Sound Level due to Chiller (Example 4)	54 dBA
Combined Sound Level	59 dBA*

From the above analysis the noise of the cooling tower is seen to have the greatest influence; the combination of the other two sources raises the level by 3 dB.

However, in setting up Examples 3 and 4 for the interior equipment, it was assumed that some attempt had been made by the Architectural/Mechanical designers to incorporate at least minimal noise-control measures. In the case of the exhaust fan, 16 feet of lined duct, corresponding to a noise reduction of 10 dB, was used. In the Chiller installation, acoustical louvers were installed in the ventilation opening resulting in a noise reduction of 10 dB. Had these noise-control measures not been taken, the sound levels at the balcony face would have combined as follows:

Sound Level due to Cooling Tower	56 dBA
Sound Level due to Exhaust Fan	58 dBA
Sound Level due to Chiller	64 dBA
Combined Sound Level	65 dBA*

Note that the difference in these comparisons is an increase in total noise level by 6 dB. This is a significant change on most

*For a discussion on the addition of decibels see Appendix 3.

scales of subjective response to noise; typically, such scales are graduated in 5 dB increments in the range between acceptable, marginal, and objectionable.

10.4 Reliability of Calculated Results

The reliability of these procedures in predicting the sound level at a point outdoors depends on several factors. For both indoor and outdoor sources, the primary uncertainty concerns the validity of the input data for the equipment noise characteristics. As discussed in Section 5, 6, and 7, much of the available data on the noise of mechanical equipment have been obtained during a period when standardized test procedures are in various stages of development within a broad segment of the industries producing building mechanical equipment. For this reason, many of the data in specific product categories reflect the use of different procedural techniques, measurement parameters, and test environments. In reducing these to some common denominator for comparison, inherent errors are introduced that depend on how much is known about the way the tests were conducted and how the data were processed.

The data for certain HVAC components that were furnished by ARI, with the cooperation of member companies, probably have the highest reliability because standardized test procedures for each equipment category were used. The data from other segments of the industry are less reliable because the sources are principally from manufacturers' catalogs where competitive pressures to manipulate the results undoubtedly have an influence--particularly in the absence of industry certification programs. In many of these "gray areas," however, we have biased this information on the basis of our own test data obtained under field conditions in deciding what to include and what to reject in developing the formulas for equipment noise estimation recommended earlier in this report.

There was, of course, scatter in the data that were used in the correlation of equipment noise levels with physical size or other operating parameters. In general, we chose an estimation curve representing one standard error of estimate above the linear regression line corresponding to the best fit to the data. Statistically, this means that about 16 percent of all products within a given category could have noise levels above the estimation curve. However, in many cases falling within that 16 percent, the amount by which the curve might be exceeded is only 2-3 dBA; in almost all cases it is less than 5 dBA.

Several factors were considered in deciding where to locate the estimation curve relative to the data scatter; the two obvious choices would be to use the *average* of all the acceptable data, or to use the upper boundary of the data scatter, as a very conservative approach. The disadvantage in using an average value is twofold: First, the noisier equipment might well be more than 5 dBA higher than estimated, and we know that increments of change in subjective response are graduated on this order. Second, we believe that there should be some inducement to the manufacturer to provide certified test ratings on his equipment, for use as an alternative input for the worksheet analysis; if the estimation curve produces only "average" values, no incentive exists.

The disadvantage in using an estimation curve that represents the upper boundary of the data scatter is that such a conservative approach economically penalizes and unnecessarily complicates many installations by leading to overdesign in the required noise-control measures. One argument in favor of this approach would be the strong inducement to industry to provide certified test ratings as an alternative to estimating the sound output of the equipment. However, we believe that the industry as a whole is taking seriously the noise-control problems with their equipment; to set the estimation curve at the upper limit of the data scatter would not reflect any recognition of this effort. Using one standard error of estimate above the best-fit linear regression line seems to be the best compromise among the possible alternatives.

The second area in which potential errors may be present in the prediction of the noise level at an outdoor reference point is in the determination of sound attenuation along the source-to-receiver path. For example, the tables provided for estimating the attenuation of frequency-sensitive elements, such as packaged sound traps and duct linings, have been based on several generalized spectrum shapes that approximate the characteristics of most typically encountered sources. The attenuation values themselves were derived from an analysis of several manufacturers' catalog data, which is not yet subject to industry standardization with regard to test method. In developing these attenuation tables, therefore, a conservative approach was taken. The "downstream" sound level calculated with these tables would tend to be on the high side of the average value by 2-3 dBA. These corrections for attenuation are subject to some uncertainty, in part because of deviations in input spectrum shape from those assumed and, in part, because of the variability in performance of so-called "equivalent" sound traps among the various manufacturers. However, the worksheet provides for the use of alternative attenuation factors when certified ratings are available. This should be an inducement to the manufacturers to develop an industry standards and certification program.

A third possibility for error lies in the provision made for the effects of shielding due to barriers and other geometric factors associated with the installation of outdoor equipment; with indoor sources, the principal uncertainty is in assigning an attenuation for the distance between the equipment and a building opening. In both situations, the worksheets tend to be conservative by using attenuation factors on the low side of what might actually occur in practice. For example, a credit of 5 dB is allowed for barriers that just break the line of sight between source and receiver. Higher barriers would ordinarily introduce an attenuation greater than 5 dB. For th se cases an alternate procedure is provided in Appendix 2 and i made optional on the worksheets.

Our chief concern is with difficulties that may be experienced in trying to verify the predicted sound level on the basis of field measurements after completion of the building. The problems arise not so much from potential errors in the worksheet procedures as from facturs such as installation and operational variables that cannot be realistically assessed until after the installation is completed. One example would be fan systems wherein the differences in point of operation on the fan curve may correspond to changes in noise level as high as 15 dB (see Section 5.1).. The mechanical designer may have made an optimum fan section based on his best knowledge of the operation requirements and his estimate of system resistance. However, if experience in the application should result in a different set of operational requirements, or if for some reason changes in duct configuration and resistive system elements have been made between the permit approval and the actual installation of the system, the fan noise level could be significantly higher (or lower) than predicted. In order to assess the validity of the worksheet analysis by means of measurements at the completed installation, it will be essential to determine first that the design under review is consistent with the actual installation.

11. ENFORCEMENT STRATEGIES FOR THE MECHANICAL EQUIPMENT PERMIT SCHEME

11.1 Introduction

The mechanical equipment permit scheme may be thought of as containing two distinct parts: legal provisions, and an enforcement strategy. The legal provisions are those sections of an ordinance, building or zoning code that specify the objective provisions (e.g., sound level limits) and/or subjective provisions (e.g., nuisance-type provisions) by which the noise from mechanical equipment must be judged. Legal provisions also provide a given individual or office, such as the building department, with the requisite authority to enforce the objective/subjective provisions.

The enforcement strategy, on the other hand, consists of those methods that the responsible office uses to encourage compliance, and to detect non-compliance when it occurs. A typical enforcement strategy for building code provisions consists of first requiring that building plans, associated specifications, and certain engineering calculations are submitted for examination before construction activity on any proposed building may commence. The building department examines the plans, specifications, calculations, etc. to verify correctness and completeness before issuing a building permit. During construction, building inspectors visit the construction site to assure that the structure is being built in accordance with the plans. Finally, the completed building is inspected for conformance, and if all is in order, an occupancy permit is issued.

This section of the report first develops and then evaluates several different mechanical equipment permit schemes for specification and control of mechanical equipment noise. Development is based on an examination of current practice in different jurisdictions across the U.S. The evaluation is based on judgements of the effectiveness, feasibility, and enforcement costs of these different schemes.

Ideally, all existing permit schemes that are directed at controlling mechanical equipment noise would be identified and evaluated. In practice, however, no jurisdictions were found, with the possible exception of New York City, that have a complete mechanical equipment permit scheme containing not only the necessary legal provisions but also a complete enforcement strategy. Rather, several jurisdictions were found that enforce identifiable bits and pieces of a complete scheme. Consequently, each scheme had to be broken down not only into its important legal provisions and enforcement strategy, but the enforcement strategies had to be broken into separate *enforcement practices*. Each enforcement practice could then be evaluated in terms of its contribution to the scheme's effectiveness, feasibility and enforcement costs.

The remainder of this section is divided into three parts. First, existing schemes in various jurisdictions are described and examined critically for strengths and weaknesses. Next, legal provisions and enforcement practices are evaluated. The importance of the various legal provisions is evaluated by comparing these provisions in the law with the provisions as they are actually enforced. Specific enforcement practices that increase permit scheme effectiveness are identified and these practices are evaluated in terms of their contribution to overall permit scheme effectiveness. The feasibility and enforcement costs of the different enforcement practices are also examined. Finally, a recommended mechanical equipment permit scheme is presented.

11.2 Existing Mechanical Equipment Permit Schemes

11.2.1 Introduction

With the possible exception of New York City, no fully developed permit schemes to control mechanical equipment noise could be found. However, a number of jurisdictions have building related noise control procedures and provisions that provide insight into the importance of specific legal provisions and enforcement practices. These jurisdictions can, for the most part be divided into two categories:

- Jurisdictions that have mechanical equipment noise related provisions in their local code or ordinance;
- Jurisdictions that have no specific mechanical equipment noise law provisions, but do have noise control considerations forming a part of the building plans review process.

The following sub-section discusses these jurisdictions, and Table 24 summarizes their noise control provisions.

11.2.2 Jurisdictions with Laws Containing Mechanical Equipment Noise Provisions

11.2.2.1 *Beverly Hills, California*

Provisions in Law

Chapter 8., Noise Regulations, §4-8.206, makes it unlawful to operate any machinery, equipment, pump, fan, or air-conditioning

TABLE 24

Provisions in Law and *De Facto* Provisions

Provisions \ Jurisdiction*	Warwick, RI		San Diego		Massachusetts		Chicago		San Francisco		New York City		New Haven		Miami		Hemet, CA		Hartford		Beverly Hills	
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L
Type of Restrictions Placed on Outdoor Noise																						
On installation/construction of mechanical equipment:																						
• Performance standards	✓				✓		✓		✓		✓	✓									✓	
• Design specifications																			✓	✓		
• Subjective (nuisance provisions)																				✓		
On use of mechanical equipment																						
• Objective (sound level limits)			✓		✓		✓		✓	✓	✓	✓		✓			✓	✓			✓	
Jurisdiction law places no specific restriction on mechanical equipment noise, but does contain:																						
• Zoning-type (sound level limits at property line)		✓		✓				✓								✓						
• Subjective (nuisance provisions)						✓										✓						
Noise Restrictions Apply to:																						
• All occupancy groups				✓		✓	✓	✓	✓	✓	✓	✓		✓					✓	✓		✓
• Commercial/industrial buildings	✓	✓			✓											✓		✓			✓	
• Multi-family buildings	✓				✓																✓	
Mechanical equipment noise dealt with only in response to complaints			■										■		■		■					

*L = provisions in law.
D = *de facto* provisions.

apparatus so as to create any noise which would cause the noise level at the property line of any property to exceed the ambient noise level by more than five decibels. "Ambient noise shall mean the all-encompassing noise associated with a given environment, usually being a composite of sounds with many sources, near and far." Finally, the building official has broad powers to require "plans and specifications to be prepared and designed by an engineer or architect," and to require associated computations and other data sufficient to show the correctness of the plans.

Enforcement Practices

In Beverly Hills, the building department interprets the legal requirements to mean that the ambient noise levels during the quietest period of the day (usually 2 am to 5 am) should not be exceeded by ~~5dB~~ decibels. The ~~5dB~~ decibel rule is applicable to each octave band as well as to the overall sound pressure level, and a report from an acoustical engineer must be submitted demonstrating that the proposed building will comply with the requirements. Such reports must be prepared for most condominiums and apartments and for commercial buildings that will be adjacent or close to residential areas. The building department can evidently exercise some judgement in determining whether or not a report is required. Such judgement is based on the likelihood that mechanical equipment noise will cause problems.

5dB [illegible] requirement is [illegible] than overall 5dB rule.

Though the building department inspects all buildings during construction, it is not always possible for an inspector to be at the site often enough to verify conformance with all plans and specifications. Consequently, for projects in particularly noise sensitive areas, or when a project is likely to result in noise problems, the building department has required builders to have their work inspected periodically and reported on by an acoustical engineer.

As a building is completed, the mechanical equipment is generally started up, and the building inspector can make a judgement as to whether the equipment seems excessively noisy. The department has withheld occupancy permits until noisy mechanical equipment was adequately quieted.

If noise problems arise shortly after a building has been completed, the building department will attempt to contact the contractor, builder, owner, acoustical engineer, and others who were involved in the project, and to encourage remedial actions. If, however, noise problems arise long after a building is finished, the original "cast of characters" has usually disappeared, and the current owner must be persuaded to correct the problem. In these latter cases, the building department has found that corrective action is often difficult to obtain, and that pursuing legal action (prosecuting through the city attorney) is generally too time consuming and is judged not worth the effort.

Conclusions

A. Detailed administrative requirements do not need to be specified in the law for implementation of an active mechanical equipment noise control program.

B. Withholding the occupancy permit is an effective method for gaining compliance.

C. Resolving noise problems after issuance of the occupancy permit can be extremely difficult.

D. Not formulating specific guidelines for requiring an acoustical engineer's report could, under some circumstances, result in arbitrary judgements of when a report must be prepared.

Specifically contradicts part A

11.2.2.2 Hartford, Connecticut

Provisions in Law

Chapter 41-97, Noise Control, under the Health Department nuisance control ordinance, gives mounting and ducting requirements for mechanical equipment so that "excessive" vibration is prevented, "noise nuisance" is eliminated, and "noise transmission" is reduced. These provisions are enforced by the Department of Licenses and Inspections.

Enforcement Practices

When building plans are reviewed, the Department of Licenses and Inspections verifies that vibration isolation is specified where it is required. Primarily, however, for adequate noise control the department relies on the knowledge of the licensed engineer/architect who places a seal on the plans thus certifying that the plans comply with all code/ordinance requirements.

Conclusions

A. Subjective noise control requirements in the law, and specification of design details for installation of mechanical equipment do not of themselves guarantee that mechanical equipment use and design will be closely scrutinized for potential noise problems at plans review.

B. Specification of design details can tie the building department to outdated technology. The building official cannot be expected to keep abreast of all technological advances in mechanical equipment design. Consequently, if the law says that all air-conditioning equipment must be mounted on vibration isolating materials, he may feel compelled to require vibration isolation, even though improved designs may have reduced air-conditioner vibration to insignificant levels.

11.2.2.3 *Hemet, California*

Provisions in Law

Chapter 12, Code of the City of Hemet, §12.2, subsection 11 specifies, by frequency, maximum permissible sound pressure levels for electrical and mechanical noise produced in commercial/industrial zones. Sound pressure levels are to be measured "at the property line of the residential property concerned closest to property upon which the sound or noise is produced."

Enforcement Practices

Mechanical equipment noise requirements are dealt with only on a complaint basis by the police. Plans reviewed by the building inspection department do~~es~~ not include an examination for mechanical equipment noise control provisions.

Projects are, however, reviewed for conformance with other noise control requirements, and this review is divided into three separate areas of responsibility. The City Planner reviews plans for single family home developments to determine whether the proposed location is exposed to noise levels in excess of specified standards. The building inspection department reviews proposed multi-family dwellings for adequate wall and floor-ceiling noise control design, and inspects all buildings during construction for conformance with the plans and specifications. Finally, the City sends the plans for all commercial buildings to either ICBO (International Conference of Building Officials) or, recently, to a private engineering firm. For a fee, these and similar organizations review plans for conformance with state and local codes. For Hemet, these organizations review the commercial building plans for adequate noise control in wall and floor-ceiling assemblies.

Conclusions

A. Property line limits on mechanical equipment noise do not insure that mechanical equipment noise will be considered at the design stage.

B. Any enforcement strategy for a mechanical equipment permit scheme should recognize the possibility that several different offices, in addition to the building department, may have to be coordinated to achieve maximum effectiveness:

1. A planning department may wish to utilize mechanical equipment noise predictions/measurements that could result from the permit scheme. Mechanical equipment noise contributes to "ambient" community noise levels, and these community noise levels may form a part of the information used for planning decisions.

2. Outside consultants/engineers who may review plans for the jurisdiction must be aware of mechanical equipment requirements and procedures. Some

jurisdictions, especially those with small building department staffs, send out plans to such consulting firms or organizations for review.

3. Police departments, or other departments, that respond to noise complaints should know of the mechanical equipment requirements so that complaints about equipment noise from new buildings are coordinated with the building department.

11.2.2.4 *Miami, Florida*

Provisions in Law

(Although Miami has neither specific provisions controlling mechanical equipment noise nor noise related review of projects, it does have noise control provisions in its building code.) The Miami Building Code, Article XXII, Performance standards, specifies that all uses in general commercial, commercial, light industrial, general industrial districts shall conform to sound level limits specified by frequency as measured at the property line. Additionally, these uses shall be so constructed, maintained and operated so as not to be injurious or offensive to the occupants of adjacent premises by reason of the emission or creation of noise.

Enforcement Practices

Noise is not considered at the plans review stage, and the noise control provisions are dealt with on a complaint basis by the zoning department.

Conclusions

A. The City of Miami has unused, though somewhat ill-defined, authority to enforce noise restrictions on mechanical equipment constructed and used in the identified districts. Other jurisdictions having similar restrictions on the *use* of mechanical equipment (see Beverly Hills, Chicago and Table 24) or even less well defined restrictions on noise from buildings (see Massachusetts and Table 24) have developed and implemented plans review programs for control of mechanical equipment noise.

B. Obviously, any model permit scheme must include an associated set of enforcement practices (an enforcement strategy); putting the appropriate language into law may, for many jurisdictions, not be enough to insure development of a working, effective program to control mechanical equipment noise.

11.2.2.5 New Haven, Connecticut

Provisions in Law

Section 18-19 of the Code of Ordinance of the City of New Haven provides that operation of any air-conditioning or mechanical equipment so as to cause excessive noise is unlawful. In an area zoned residential, noise is excessive only if the sound level exceeds 55 dBA as measured on the property line. In an area zoned industrial, noise is excessive if the sound level exceeds 80 dBA on the property line. If air-conditioning or mechanical equipment is found to violate the ordinance, the equipment shall not be operated unless the proper corrections have been made and approved by the Building Department. The Building Department (or other municipal departments) "shall be empowered to enforce the violations of such ordinance."

Enforcement Practices

There is no enforcement of the ordinance at plans review, during construction, or prior to issuance of occupancy permits. The Building Department staff has attempted to enforce the ordinance in response to complaints, including taking offenders to court. But due to lack of training in the use of noise measurement equipment and in noise law enforcement, prosecution of violations was terminated.

Conclusions

If a new law requires enforcement techniques or measurement procedures that differ substantially from existing techniques and procedures, the people who will be responsible for enforcement of the law must receive adequate training in those techniques and procedures.

II.2.2.6 *New York City*

Provisions in Law

Section C26-1208.3 of the Building Code of the City of New York specifies that mechanical equipment spaces shall not be ventilated through openings into yards or courts where living quarter windows open unless such openings have sound attenuating devices to limit the noise transmitted to NC-35 (noise criterion) inside an open window of the exposed dwelling units. For mechanical equipment located outside a building, maximum permissible sound *power* levels are specified by frequency. These maximum levels apply when one or more windows of a residential dwelling is located within 100 feet of the equipment. The maximum permissible sound power levels depend upon the distance from the window to the equipment. The sound *power* levels need not be used if the octave-band sound *pressure* levels measured within the dwelling do not exceed specified maximums. Though not clearly stated in the code, it is apparent that when sound *power* data are not available, the maximum sound *pressure* levels apply.

Additionally, Local Law No. 57, the Noise Control Code, Section 1403.3-5.13, Circulation Devices, states that no person shall operate a circulation device not subject to Title C, Chapter 26 (see above), so as to create a sound level in excess of 45 dB(A) as measured inside the dwelling unit affected. Measurements are made in a line with, and three feet from an open portion of the window nearest the exterior face of the circulation device. Circulation devices circulate a gas or fluid and include, but are not limited to any air-conditioner, pump, cooling tower, fan or blower.

Enforcement Practices

A. Initial

At the time the City adopted the code, in 1970, it also contracted an acoustical consulting firm to provide enforcement training for plan examiners and building inspectors. General training in acoustics was given to 30 City officials, and 11 of these 30 received specialized training in field measurement procedures. Thus, building department personnel knew what to look for when examining plans prior to issuing building permits and when inspecting new buildings both during construction and prior to issuing occupancy permits. If " ... conditions indicate that the installed construction or equipment does not meet the noise control prescribed in [the code] ... "building personnel were trained and equipped to make the necessary field tests.

> "In the event a construction or noise source fails to meet the building code noise control requirement in a field test, the inspector will place a violation against the building. The architect or owner may then engage the services of an acoustical consultant to conduct tests to determine the responsibility for the failure or to challenge the violation. If a violation exists, corrective work must follow before the Certificate of Occupancy is awarded."[28]

It would seem, then, that in 1970, New York City had all the ingredients necessary to make the noise control provisions of its building code effective. The code contained specific objective standards, based on current engineering practice. Measurement procedures had been developed for verification of compliance, and building department personnel were trained in these procedures. Finally, plan examiners and building inspectors had been trained in the code requirements, and presumably knew what their responsibilities were with respect to these requirements.

B. *Current*

Since New York City had a seemingly strong foundation at the program's commencement, the current status of the program may provide insight into what ingredients are important if a mechanical equipment permit scheme is to be effective.

Based on conversations with several officials in the City's building department, one must conclude that the noise control provisions of the code are now essentially non-functional. Because the building department was so overloaded with responsibilities, architects and engineers found that obtaining the necessary permits was excessively time-consuming. In an effort to speed the process, local architects and engineers applied for, and received, permission to assume responsibility at the design stage for conformance with all code requirements except those that apply to safety, egress and zoning. Thus, up to the time of final plans examination, the licensed architect or engineer has sole responsibility for designing to comply with the noise control provisions. At final plans examination, the complete final drawings are reviewed by the building department, but this examination deals primarily with the major items of safety, egress and zoning. Thus, noise control design is often never reviewed by City officials.

But aside from these problems of an overloaded building department, the enforcement procedures had several other shortcomings with respect to the mechanical equipment requirements. The sound *power* level requirements proved unworkable. Most manufacturers do not supply sound power level data for the larger equipment typically used in multiple family dwellings; this equipment is too large to be measured using approved procedures. Consequently, the building department had to rely for proof of conformance on the alternative maximum permissible sound *pressure* levels within nearby dwellings. But determining these levels was time-consuming, and required costly equipment and trained personnel. Over the years, the measurement equipment has not been maintained, and the people who were trained in 1970 have disappeared through normal personnel turnover. The department now assumes that new mechanical equipment conforms to the code noise provisions unless complaints arise after installation and use. When complaints occur, the building department refers them to the building owner. If complaint activity is organized, the building department will "order" the building owner to have noise measurements made. Complaints have been sporadic and are handled separately by the five different borroughs; consequently, there is no collected information on how complaints are finally resolved.

Conclusions

A. An overloaded building department may delegate and/or ignore building code requirements thought to be of minor importance.

B. Knowledge of complex procedures associated with low priority code requirements is not easily passed on from one generation of building inspectors to the next.

C. Without some change in the sound power level determination procedures (ASHRAE No. 36-62) that were available when the New York Code was written, use of maximum permissible sound power levels may prove unworkable. An alternative approach would be for a jurisdiction to establish its own procedures for determination of sound power levels, and require manufacturers of mechanical equipment to provide sound power level data as a condition for use/sale in that jurisdiction.

D. One person alone should be responsible for administering the noise control requirements, and administering these requirements should be one of his/her primary responsibilities. Though this conclusion may not follow directly from the specific problems discussed above, it arises if all problems are considered together. When the noise control program is a primary responsibility, it is less likely to be overlooked in favor of other, more pressing, responsibilities. Uniformity of plans examination and building inspection is more likely, and complaints will be dealt with in a consistent manner. This one person will know the procedures, and could see that new personnel receive adequate instruction.

11.2.2.7 <u>*San Francisco, California*</u>

Provisions in Law

Chapter VII of the Municipal Code (Police Code), Part II, Article 29, Regulation of Noise, Section 2909 specifies maximum permitted A-weighted sound levels for fixed sources as measured at the property line of the receiving land use. Fixed sources include, but are not limited to, industrial and commercial process machinery and equipment, pumps, fans, air-conditioning apparatus, or refrigeration machines.

Enforcement Practices

When applying for a building permit, the applicant recieves a copy of the noise ordinance, and must certify that he understands this and all other legal requirements imposed by the City. The applicant is thus held responsible, both before and after receipt of the occupancy permit, for the conformance of the building with all requirements. After the occupancy permit has been issued, however, if any items of noncompliance are found, requiring the applicant to correct these items proves, in practice, to be extremely difficult. The City does not review the plans to determine potential mechanical equipment noise problems. awkward!!

Mechanical equipment noise is also dealt with by responding to complaints. Since 1972, some 3000 noise-related complaints have been received, and an estimated 50% of these concern air-conditioner noise. Most air-conditioner related complaints involve old/poorly maintained equipment.

Conclusions

A simple procedure (certifying knowledge of noise control requirements) may theoretically help fix responsibility for compliance with mechanical equipment noise requirements. In practice; however, once the occupancy permit has been issued, enforcing this responsibility for a building that is found not in conformance may be difficult. Not only can the defendant use numerous delaying tactics, but it may be difficult, if not impossible, to identify and locate the original owners/permit applicant.

11.2.3 Jurisdictions with no Specific Provisions for Mechanical Equipment Noise, but with Noise Considerations as Part of the Plans Review Process

11.2.3.1 Chicago, Illinois

Provisions in Law

Chapter 17, the Chicago Environmental Control Ordinance, specifies maximum sound pressure levels by octave band, at zoning district and/or lot boundaries. This chapter also gives the Commissioner of Environmental Control authority "to examine and approve the plans of fuel-burning, combustion or process equipment or devices, furnaces ... and noise control devices installed, constructed, reconstructed, repaired, or added to any building ... to assure that they are in accordance with the requirements of this chapter ...)".

Enforcement Practices

The Department of Environmental Control (DEC) presently has two employees working full time in the building department office. These two people review all plans to determine compliance with the environmental control ordinance. If either of these two DEC personnel decides that the plans indicate that a building may cause environmental noise problems, the plans are sent to the DEC offices for closer scrutiny. Generally, two basic criteria are used to identify potential noise problems: building plans need more detailed study if (a) the building volume is greater than 500,000 cubic feet; or if (b) fans capable of delivering more than 2000 cfm will be used. Additionally, the two DEC employees are familiar enough with the noise control parameters to deviate somewhat from these criteria, as when a smaller building will be located in an especially noise-sensitive area, or when a building that exceeds the criteria will be built in an area that is solely industrial.

Once the plans of potential problem buildings have been sent to DEC, more information is required from the architect/engineer. DEC must know what types of equipment will be used (by make and model), what the sound power levels of this equipment are, what noise control devices or designs will be used, and what the sound pressure levels will be at the property line. Usually, the mechanical engineer working on the project prepares this information. DEC checks the calculations, and spot checks the sound power levels with manufacturers' data. If all is reasonable, the project is given "environmental approval." DEC has been reviewing about 50 to 60 plans per year.

As with the New York City department of buildings, Chicago DEC has found that no sound power level data are available for large equipment. In these cases, DEC requires that the owner submit a letter stating that any equipment installed will comply with the maximum permissible sound pressure levels, and if it does not comply, noise control methods will be used. DEC policy is to assume that the owner will be responsible for corrective measures.

Once plans are approved, the building department inspectors insure that the specified equipment is, in fact, installed. (DEC indicated, however, that coordination with the building department was a major problem at program inception.) After construction, DEC has yet to be
has spot checked with noise measurements and found no instances of non-compliance. DEC also handles noise complaints, and though many of the complaints concern air-conditioner or mechanical equipment noise, they have not received any complaints about large equipment since the program began.

Conclusions

A. Responsibility for building noise control requirements can be located outside the building department, but coordination may be facilitated if the program is kept entirely within the building department.

B. A screening procedure can be used to reduce work, both for designer and for inspector, by identifying projects that require detailed noise analysis.

C. The problem of equipment sound power levels must be resolved if all size installations are to be dealt with quantitatively at the design stage and uniformly at the construction stage. Without some quantitative data on large equipment, compliance cannot be determined prior to installation and use of the equipment.

11.2.3.2 *Massachusetts*

Provisions in Law

Regulation 10, Noise, of Regulations for the Control of Air Pollution prohibits unnecessary emissions from a source of sound that may cause noise. Noise means sound of sufficient intensity and/or duration as to cause or contribute to a condition of air pollution. Air pollution means the presence of one or more air contaminants in such concentrations and of such duration as to cause a nuisance, be injurious, or unreasonably interfere with the enjoyment of life and property. Air contaminant means, among other things, sound. Emission means any discharge or release of an air contaminant.

Enforcement Practices

Noise sources that are proposed to be built or modified must submit noise data to the Department of Environmental Quality Engineering. These noise data are to show, at the property line and at the nearest inhabited residences: a) the lowest "ambient" sound levels during the operating hours of the noise source; b) the measured sound levels of the noise source prior to modification (if appropriate); c) the calculated sound levels that will occur during the operating hours of the noise source due to the modification of the noise source. Approval for the modification or construction of the noise source is granted if the modified or new noise source does not "increase the broad band noise level in excess of 10 dB(A) above ambient or produce a pure tone condition."

The Department cannot be certain that it receives noise data for all proposed modifications to noise sources or for all proposed new noise sources. It is possible for new buildings to be constructed without receiving approval from the Department. The Department also enforces air quality requirements for proposed projects, and can thus require noise data for these projects. But not all projects need file for air quality approval.

The Department also responds to complaints, and has received complaints about mechanical equipment noise from newly constructed buildings.

Conclusions

A. Lack of objective restrictions (i.e., sound level limits) in the law did not preclude the use of such restrictions.

B. Though the law does not specifically identify mechanical equipment, enforcement practices result in restrictions on mechanical equipment noise.

C. A mechanical equipment permit scheme should be enforced in such a way that the probability of identifying all potential offenders is high; new buildings containing mechanical equipment should not "slip by" into the construction phase without review.

II.2.3.3 San Diego, California

Provisions in Law

Primarily, San Diego enforces California Administrative Code (CAC), Title 25, "Noise Insulation Standards." However, this enforcement has been made possible in part by San Diego Municipal

Code, Article 9.5, Noise Abatement and Control. This code establishes the office of the Noise Abatement and Control Administrator within the Building Inspection Department, gives the Administrator broad powers to control noise, requires development of noise level contours for the city, and sets maximum permissible sound levels by land use zones.

Enforcement Practices

There are no noise related restrictions placed on mechanical equipment at plans review, and noisy mechanical equipment is dealt with in response to complaints.

San Diego does have, however, a well-developed procedure for enforcement of CAC Title 25. Primary emphasis is on insuring adequate outdoor-to-indoor noise reduction in new multi-family dwellings, with secondary efforts directed at insuring the use of approved party wall, and floor-ceiling constructions in these buildings.

All multi-family project plans are reviewed to determine the location of the project relative to the City's noise level contours, and to verify the use of the approved partition constructions. If the project is located in specified high noise areas, the building permit applicant is notified that he must have an "acoustical analysis report" prepared by a "registered acoustical consultant." (The Building Inspection Department furnishes a list of acoustical consultants that have prepared adequate reports in the past.) The acoustical analysis report must be submitted, be adequately documented, and show that interior noise levels will not exceed the standards. The Building Inspection Department spot checks the calculations, and from time to time conducts simultaneous outdoor-indoor noise measurements on buildings in high noise areas to verify adequate reduction of outdoor noise.

Conclusions

A. A single person in the Building Inspection Department, the Noise Abatement and Control Administrator, is responsible for all the City's noise control efforts. Locating this person in the building department facilitates the coordination of building inspection and plans review; the building inspectors are kept aware of the important, noise related details that must be inspected during construction.

B. Since all noise complaints are directed to the Noise Abatement and Control Administrator, he will be kept aware of not only the major noise problems in the city, but he should be able to judge the effectiveness of the plans review/building inspection noise control program.

C. Acoustical expertise (the "registered acoustical consultant") is incorporated into the building design process according to specific criteria (when a proposed multi-family dwelling is located in an identified high noise area).

D. *All* multi-family dwelling plans are reviewed by the Noise Abatement and Control Administrator. Thus, no new multi-family building will be built without a review of its specific noise control design.

11.2.3.4 Warwick, Rhode Island

Provisions in Law

Section 8.4.9 of the city zoning ordinance specifies maximum permissible sound pressure levels by frequency for Light Industrial and for Heavy Industrial districts. For Light Industrial Districts, noise measurements are made on any property line of the tract on which the industrial operation is located. For Heavy Industrial Districts, noise measurements are made at the nearest Heavy Industrial District boundary line.

Enforcement Practices

For apartments and for industrial buildings, the designer/engineer develops a plan that shows property line noise levels, and this plan is submitted at the time of application for a building permit. The building department compares the submitted property line noise levels with the limits specified in the ordinance. No noise related complaints have been received on buildings built to comply with the limits specified in the ordinance.

Conclusions

An ordinance with only property line sound pressure level limits has given rise to a plans review procedure that includes noise control considerations for mechanical equipment.

11.3 Evaluation of Alternative Mechanical Equipment Permit Schemes

11.3.1 Introduction

This sub-section evaluates the permit scheme components, that is, the legal provisions and the enforcement strategies, that were presented in the preceeding sub-sections. (Recall that an enforcement strategy is composed of many separate enforcement practices.) First, the legal provisions are evaluated by comparing the restrictions that the laws place on noise with the restrictions as they are actually enforced. Such a comparison of provisions in law vs. *de facto* provisions indicates how important the specific provisions in the law are for development of a permit scheme. Second, the enforcement practices that seem to contribute most to the effectiveness of an enforcement strategy are identified and rated, not only in terms of their contribution to the strategy's effectiveness, but also in terms of their feasibility and enforcement costs. Such an evaluation helps more clearly identify the importance of individual enforcement practices in an overall enforcement strategy.

11.3.2 Provisions in Law Compared with *De Facto* Provisions

Table 24 summarizes, by jurisdiction, the provisions in the law (columns labeled "L") and the *de facto* provisions (columns labeled "D"). The table also summarizes the types of buildings to which the provisions apply, and notes if mechanical equipment noise is dealt with *only* in response to complaints.

Generally, the provisions may be thought of as restricting outdoor noise levels in some way. The restrictions may apply specifically to the noise produced by mechanical equipment or more generally to any outdoor noise. If the restrictions apply to mechanical equipment, they may apply to the installation/construction of the equipment, or to the use of such equipment. Restrictions on installation/construction include performance standards such as maximum sound power/pressure level limits, design specifications such as required use of vibration isolation or minimum duct size, or subjective restrictions such as a prohibition against unnecessary noise. The only type of mechanical equipment use restrictions found are objective ones (sound level limits). If the restrictions do not apply specifically to mechanical equipment, they are either zoning-type (sound level limits specified at property lines by land-use) or subjective.

In practice, the most effective type of restriction for mechanical equipment noise is a performance standard enforced on installation of equipment (i.e., at plans review and during building construction). Such a standard specifies, for example, sound level limits at a particular receiver location. The ultimate objective of the permit scheme is to control the sound level that people will be exposed to, and the performance standard (if enforced) can do just that.

Which jurisdictions have enforced a *de facto* performance standard, and what are the provisions that these jurisdictions have in their laws? Six jurisdictions have *de facto* performance standards enforced on equipment installation. Of these six, one (Massachusetts) has in its law only a subjective restriction that is not even specific to mechanical equipment, two (Warwick, Chicago) have zoning-type restrictions not specific to mechanical equipment, two (San Francisco, Beverly Hills) have objective restrictions on mechanical equipment use, and one (New York City) has performance standards that apply to installation of mechanical equipment.

From this comparison, it is undeniable that enforcement of performance standards at plans review can occur for widely differing provisions in the law; it is not necessary, empirically, to have specific performance standards, applicable to equipment installation, written into the law. Therefore, it must be concluded that the provisions in the law as identified here do not play a particularly important role in determining the enforcement practices in a given jurisdiction.

legal provisions are the basis for definitive enforcement practices at specific times in process by specific city offices.

However, it should not be concluded that any and all provisions are unimportant. All six jurisdictions mentioned do have a noise-related provision in the law and, except for Massachusetts, do have *objective* sound level limits in the law. Additionally, except again for Massachusetts, it is believed that the laws of these jurisdictions provide broad statutory authority to the responsible office (building department, health department, department of environmental conservation, etc.) to require from permit applicants any reports, plans, specifications, computations deemed necessary to insure compliance with the laws of that jurisdiction.

Thus, two general provisions can be tentatively identified as playing an important role in development of an effective permit scheme to control mechanical equipment noise: an objective restriction on outdoor noise, and broad authority to request information from permit applicants. The exact form of the objective restriction is unimportant; it may apply to either the use or the installation/construction of mechanical equipment, or it can be a general zoning-type set of sound level limits without specific reference to mechanical equipment. This objective restriction can be used to set the performance standards for mechanical equipment noise.

As the next sub-section will demonstrate, effective enforcement practices depend largely upon a building department's ability (or the ability of the responsible office) to request certain information from permit applicants. Thus, if the permit scheme is to effectively control mechanical equipment noise, the other important provision, the authority to request information, must be present in the law. It is believed that this authority already exists in the building departments of most jurisdictions in the U.S.

Thus, these two provisions form the necessary legal framework for development of the effective mechanical equipment permit scheme. The remaining question is: What enforcement strategy should be implemented to form the complete, and most effective permit scheme? The next sub-section presents an answer to this question.

11.3.3 Enforcement Strategies

The enforcement strategy evaluation presented here is based on judgments of enforcement practice effectiveness. A given enforcement practice increases or contributes to an enforcement strategy's effectiveness if it is judged to increase the probability that mechanical equipment noise will be controlled to some given performance standard. By examining both the enforcement practices of the jurisdictions (identified in Section 11.2) and the associated conclusions, it is possible to identify at least nine practices that increase the effectiveness of any enforcement strategy designed to control mechanical equipment noise. This subsection first lists these practices, then rates each one on the basis of effectiveness, feasibility, and enforcement costs, then, using these ratings, evaluates the enforcement strategies of the identified jurisdiction.

11.3.3.2 *Identified Enforcement Practices*

The following are the enforcement practices that, when present in an enforcement strategy, should increase the effectiveness of the strategy, and that, when absent will result in a less effective strategy.

1. All applicable building plans are reviewed for mechanical equipment noise.

2. The permit applicant must hire an "expert" to prepare an acoustical analysis.

2a. Technical information needed for preparation of the acoustical analysis is readily available.

2b. Acoustical analysis methods are standardized.

3. An "expert" is required to certify that the building plans comply with the mechanical equipment noise control requirements.

4. The "expert" becomes involved in the project only under prescribed circumstances.

5. Responsible official occasionally performs a final, post construction check of mechanical equipment noise.

5a. The final check requires use of noise measuring equipment.

6. The office that enforces the mechanical equipment noise restrictions at the plans review stage also responds to complaints about mechanical equipment noise.

11.3.3.2 *Rating of Enforcement Practices*

Numerical ratings presented in Table 25 are based on the relative effect each practice will have on enforcement strategy effectiveness, feasibility and enforcement costs. Ratings are presented for both the presence and the absence of each practice in an enforcement strategy. These ratings are based on the following guidelines.

Effectiveness

All identified practices increase effectiveness if present in a permit scheme, and either decrease or do not change effectiveness if absent. The more a practice increases effectiveness, the higher the numerical rating on a scale of 1 to 4. If, when a practice is absent, it does not decrease effectiveness, it is rated as zero (0). When absent, the more the enforcement strategy's effectiveness is reduced, the more negative the numerical rating (-1 or -2).

Feasibility

The feasibility of each enforcement practice is rated by judging how much that practice would affect the operations of a building department in a "typical" jurisdiction if the building department adopted an enforcement strategy containing that practice. The "typical" jurisdiction is assumed to currently make no attempt whatsoever to control mechanical equipment noise at the plans review stage. This "typical" jurisdiction is also assumed to have the necessary provisions in its law (statutory authority to require submission of information with building plans and an objective restriction on outdoor noise, see 11.3.2), and to have some established system of building code enforcement that operates routinely for all new/altered/repaired, etc. buildings.

TABLE 25
Enforcement Practice Ratings

Enforcement Practice	Rating*					
	Present			Absent		
	E	F	C	E	F	C
1. All plans reviewed	2	0	2	-1	0	1
2. Analysis required	4	-3	4	-2	0	0
2a. Information available	3	1	0	-2	-1	2
2b. Analysis standardized	1	1	0	-1	-1	1
3. Expert certifies	2	0	0	0	0	0
4. Expert involvement prescribed	1	0	1	-1	0	2
5. Occasional final check	2	-2	1	0	0	0
5a. Final check requires equipment	2	-3	3	0	0	0
6. Office also takes complaints	1	-2	2	0	0	0

-2

How did you arrive at these figures

*E = Effectiveness rating
F = Feasibility rating
C = cost rating

A feasibility rating of zero (0) is given if the *presence* of an enforcement practice is judged to require little or no change in the existing building code enforcement system. Naturally, for all but two of the identified enforcement practices, their *absence* receives a zero feasibility rating since absence implies no change is needed in the current enforcement system. Only 2a and 2b (information needed for, and standardized methods for the acoustical analysis) are given negative feasibility ratings when absent since, if an acoustical analysis is required, their absence will create additional work for those officials who must review the acoustical analyses. (Conversely, the presence of 2a and 2b receive positive feasibility ratings.) Negative feasibility ratings are also given when the presence of a given enforcement practice in the adopted enforcement strategy means that someone in the building department must learn procedures that differ substantially from current procedures.

Costs

Costs are rated by estimating how much the presence or absence of the identified practices in an enforcement strategy will increase the enforcement costs of the responsible department. These ratings are based on subjective judgments, and should be regarded only as indicators of relative costs. For example, the highest cost rating is 4, is given to the enforcement practice of requiring an acoustical analysis, and indicates only that the inclusion of this practice in an enforcement strategy could increase the costs of that enforcement strategy more than would the inclusion of any other single practice. But it would be incorrect to assume that a cost rating of 4 means that the rated practice is twice as costly as one with a rating of 2. Rather, the 4 rating indicates costs substantially higher than the costs associated with 2.

Using these general guidelines, the enforcement practice ratings are presented in Table 25, and resultant evaluations of enforcement strategies are presented in Table 26. Examination of Table 26 shows that some jurisdictions have fairly effective enforcement strategies, and pay a fairly high price; others pay a low price and have relatively ineffective programs, and one (Massachusetts) has a fairly costly program with almost no effectiveness.

Obviously, these enforcement practices could be combined in numerous enforcement strategies, and the reader may wish to try a few combinations. However, a single recommended enforcement strategy is shown, and this strategy is discussed in the next sub-section.

11.4 Recommended Permit Scheme for Control of Mechanical Equipment Noise

11.4.1 Provisions in Law

A. Statutory authority for the appropriate office to require submission of pertinent data, calculations, reports, etc. that it deems necessary for demonstrating compliance of proposed buildings with all laws of the jurisdiction.

B. Objective restrictions on outdoor noise, such as sound level limits specified by land use categories as measured at property lines.

11.4.2 Enforcement Strategies

The recommended enforcement strategy is evaluated in Table 26. This strategy includes most of the identified enforcement practices. Specifically, the enforcement strategy must be implemented in such a way that all relevant plans are reviewed for compliance

TABLE 26

Evaluation of Enforcement Strategies

Enforcement Strategy* / Enforcement Practice	Recommended			Warwick, RI			Massachusetts			Chicago			San Francisco			New York City (current practice)			Hartford			Beverly Hills		
	E	F	C	E	F	C	E	F	C	E	F	C	E	F	C	E	F	C	E	F	C	E	F	C
1. All plans reviewed	2	0	2	2	0	2	-1	0	1	2	0	2	2	0	2	2	0	2	2	0	2	2	0	2
2. Analysis required	4	-3	4	4	-3	4	4	-3	4	4	-3	4	-2	0	0	-2	0	0	-2	0	0	4	-3	4
2a. Information available	3	1	0	-2	-1	2	-2	-1	2	-2	-1	2	-	-	-	-	-	-	-	-	-	-2	-1	2
2b. Analysis standardized	1	1	0	-1	-1	1	-1	-1	1	1	1	0	-	-	-	-	-	-	-	-	-	-1	-1	1
3. Expert certifies	0	0	0	0	0	0	0	0	0	2	0	0	2	0	0	2	0	0	2	0	0	0	0	0
4. Expert involvement prescribed	1	0	1	-1	0	2	-1	0	2	1	0	1	-1	0	2	-1	0	2	-1	0	2	-1	0	2
5. Occasional final check	2	-2	1	0	0	0	0	0	0	2	-2	1	0	0	0	0	0	0	0	0	0	2	-2	1
5a. Final check requires equipment	2	-3	3	0	0	0	0	0	0	2	-3	3	0	0	0	0	0	0	0	0	0	0	0	0
6. Office also takes complaints	1	-2	2	0	0	0	1	-2	2	1	-2	2	0	0	0	1	-2	2	1	-2	2	1	-2	2
Totals for strategy	16	-8	13	2	-5	11	0	-7	12	13	-10	15	1	0	4	2	-2	6	2	-2	6	5	-9	14

*E = Effectiveness rating
F = Feasibility rating
C = Cost rating

with the objective restrictions on outdoor noise (enforcement practice 1). Such a thorough review probably means that either the building department is given responsibility for this plans review, or that the responsible office closely coordinates its efforts with the building department.

Second, an acoustical analysis must accompany building permit applications for any project that will use any of the equipment covered by the worksheets of section 10. This acoustical analysis must follow the procedures identified by the worksheets. These requirements insure that enforcement practices 2, 2a, 2b, and 4 are part of the enforcement strategy.

Third, post-construction measurements of actual mechanical equipment noise levels must be made on a spot check basis prior to issuing the occupancy permit. This requirement (practices 5 and 5a), though improving the strategy's effectiveness, may present serious problems of feasibility. Building inspectors are not accustomed to making final tests of performance and are likely to balk at the prospect of denying an occupancy permit because of a failure, by "a few decibels," to meet appropriate performance standards.

Finally, the office that is responsible for reviewing the acoustical analyses, and performing the final noise measurements, should also be responsible for responding to complaints about mechanical equipment noise (practice 6). This would ensure not only that the officials who respond to the complaints are familiar with mechanical equipment noise, but that a single office is responsible for dealing with mechanical equipment noise from design through construction to occupancy and thereafter. The office would thus be kept aware of the effectiveness of its permit scheme to control mechanical noise.

11.4.3 Problems Not Solved by the Recommended Enforcement Strategy

There are three major problems, all alluded to in earlier sections of this report, that must be left unresolved by the recommended enforcement strategy. First, the statistical accuracy of the worksheet procedures in predicting the noise levels of operating equipment is not known (see Section 10). Second, a Reference Point at which mechanical equipment sound levels are to be predicted must be determined for each installation. Third, the feasibility as well as the effectiveness of the final post-construction test is not really known. why 2/3 ?

All these problems, however, could be better understood, if not resolved, by "trial" implementation of the strategy. If one or more jurisdictions were to adopt the recommended strategy and implement all the enforcement practices, data could be gathered demonstrating the accuracy and utility of all procedures. Such a trial period would provide experience in how building departments and other offices actually react to the practices, and how easy or difficult architects/engineers find the worksheet procedures.

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APPENDIX 1
TRANSFORMER SOUND POWER LEVEL CALCULATION

This appendix provides a procedure for calculating the A-Weighted Sound Power Level for a transformer based on measured NEMA sound levels and transformer dimensions.

A recent study performed for ESEERCO[17] has demonstrated the following relationship between the noise level measured in the far-field and that measured at NEMA positions:

$$L_d = L_N - 20 \log (d) + 10 \log (S) - 8$$

where

L_d = space average sound level measured at distance, d, from the tank wall, in dBA

L_N = sound level measured at NEMA position, circumferential averages, in dBA

S = surface area of the four sides of the transformer tank in square feet

d = distance from the transformer tank, in feet.

The above relation is based on measured data, as shown in Figure A-1.1, with a standard deviation, σ, of 3 dB (due mostly to directivity effects). A limited number of measurements also indicated that the above equation is valid for units with cooling fans operating, provided the measurement positions are not shielded from the fans by the tank wall.

Assuming free-field, hemispherical sound radiation from a point source, the A-weighted Sound Power Level, L_W (A) is related to the

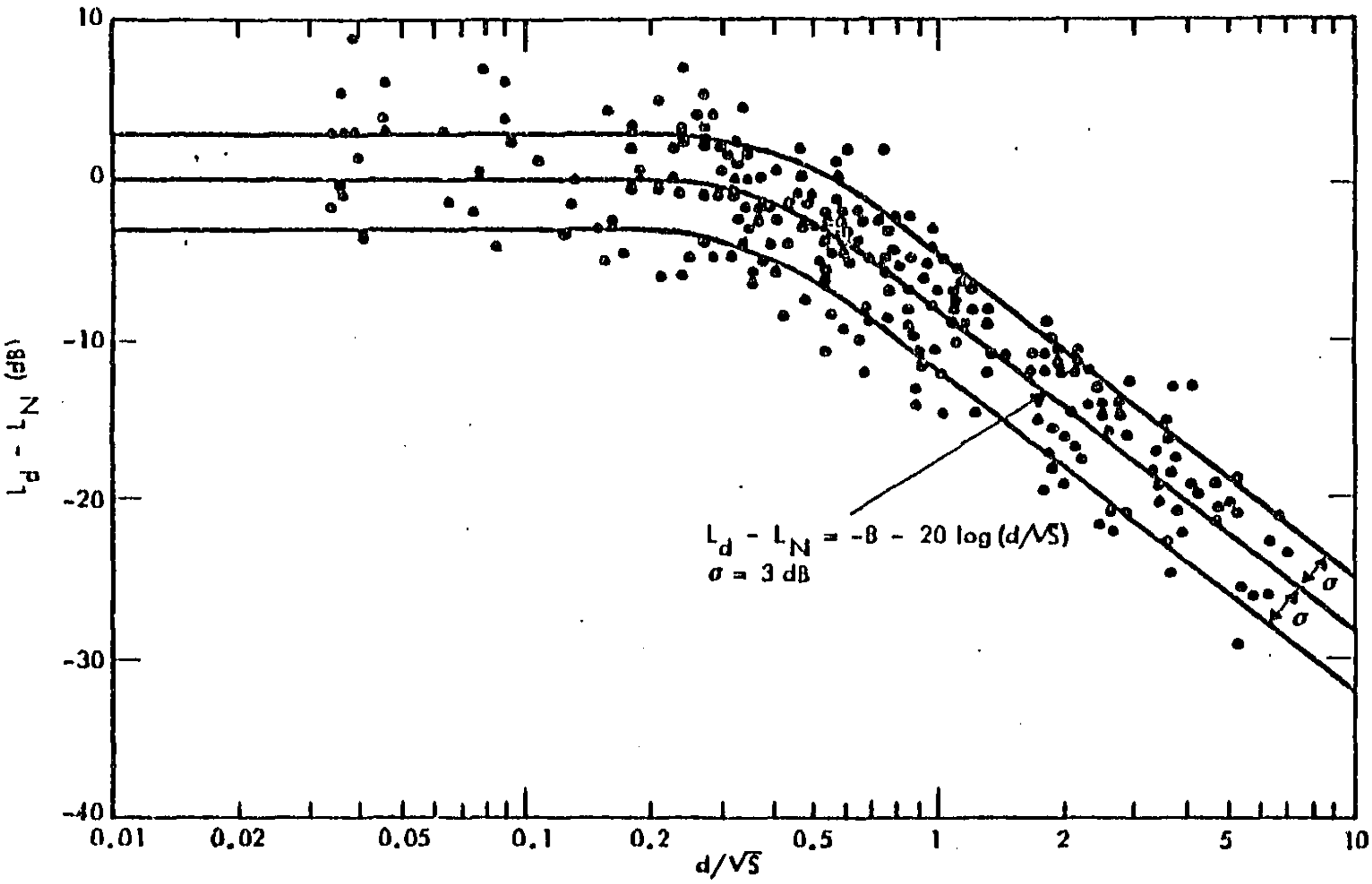


FIGURE A-1.1. NORMALIZED CORE NOISE VS NORMALIZED DISTANCE (REF. 17)
(See Text For Definition of Symbols)

far-field sound level as follows:

$$L_w(A) = L_N + K_A$$

where

$L_w(A)$ = A-Weighted Sound Power Level re 10^{-12} Watt

L_N = average sound level measured at NEMA positions, in dBA

K_A = area correction factor (= 10 log (S) - 10.5), in dB.

Thus, transformer Sound Power Level may be calculated using the above relation. Table A-1.1 provides the area correction factor, K_A, as a function of tank face area.

TABLE A-1.1

TRANSFORMER TANK AREA CORRECTION FACTOR

Tank Face Area, S, ft²	K_A, dB
100 - 125	10
126 - 155	11
156 - 200	12
201 - 250	13
251 - 315	14
316 - 400	15
401 - 500	16
501 - 630	17
631 - 795	18
796 - 1000	19
1001 - 1250	20
1251 - 1585	21
1586 - 1995	22
1996 - 2500	23

APPENDIX 2
SOUND ATTENUATION BY BARRIERS

A barrier is a solid wall or obstruction which breaks the direct line of sight between a sound source and a receiver. An effective barrier has no holes or air gaps and has a surface weight of at least 4 lb/ft^2. Figure A-2.1 shows a typical geometrical configuration of source, receiver, and intervening infinite barrier. The sound attenuation provided by this simple barrier is a function of the source-to-barrier distance (R), the receiver-to-barrier distance (D), and the line-of-sight break distance (h).

It should be noted that for barriers of finite width, sound may propagate around the sides of the barrier as well as over the top and, therefore, such a barrier provides less attenuation than an infinite barrier. The situation becomes even more complex for 2, 3, or 4-sided barriers, where sound reflection between non-absorptive barrier walls may also compromise the attenuation. Finally, barrier attenuation may be compromised by the use of walls with low surface weight. Due to these considerations, the practical upper limit of barrier performance lies in the range between 15 and 20 dB.

A procedure for estimating the sound attenuation provided by finite, single-wall barriers is outlined below. The method, adapted from Reference 29, calculates the reduction in A-weighted sound level provided by a barrier, based on a point source model evaluated at a frequency of 565 Hz. The procedure is considered applicable to all building mechanical equipment spectrum classes, with an expected accuracy of ±3 dB. In practice, where non-ideal noise sources are the rule rather than the exception, greater

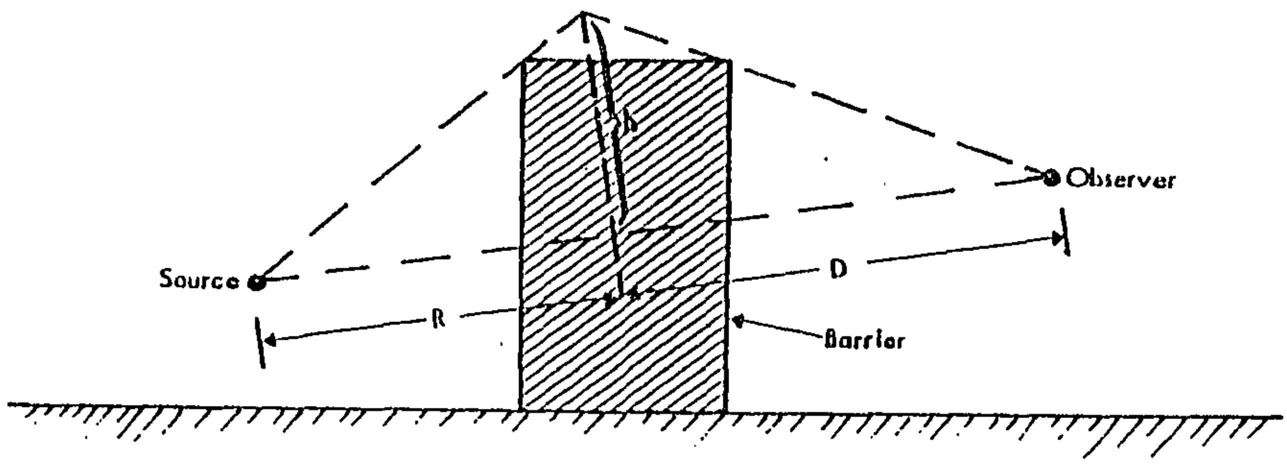


FIGURE A-2.1. SIMPLE BARRIER CONFIGURATION discrepancies are possible. Nevertheless, the calculation procedure presented below yields an estimate of barrier noise reduction which is useful for design purposes.

Barrier Evaluation Procedure

1. Determine the observer location of interest.

2. Locate the source at a point 1/3 down from the top and 1/2 in from the front and sides of the equipment noise source.

3. Locate the barrier profile and obtain accurate values for the following quantities (see Figure A-2.1): h, the shortest distance from the barrier top to the line of sight from source to observer (feet); R and D, the slant distances, along the line of sight, from the barrier to the source and observer, respectively. (Specifically, R and D are the two segments into which h breaks the line of sight.) Note that h is *not* the height of the barrier above ground, but the distance from the barrier top to the line of sight.

4. Enter at the top of Figure A-2.2 with the value of h on the left-hand scale; move right to intersect the curve corresponding to R (or D, whichever is *smaller*).

5. Move down to intersect the curve corresponding to the value of D/R (or R/D, whichever is greater than unity).

6. Move right to intersect the vertical scale in order to find the potential barrier shielding, A_1, in decibels, corresponding to an ideal barrier of infinite length.

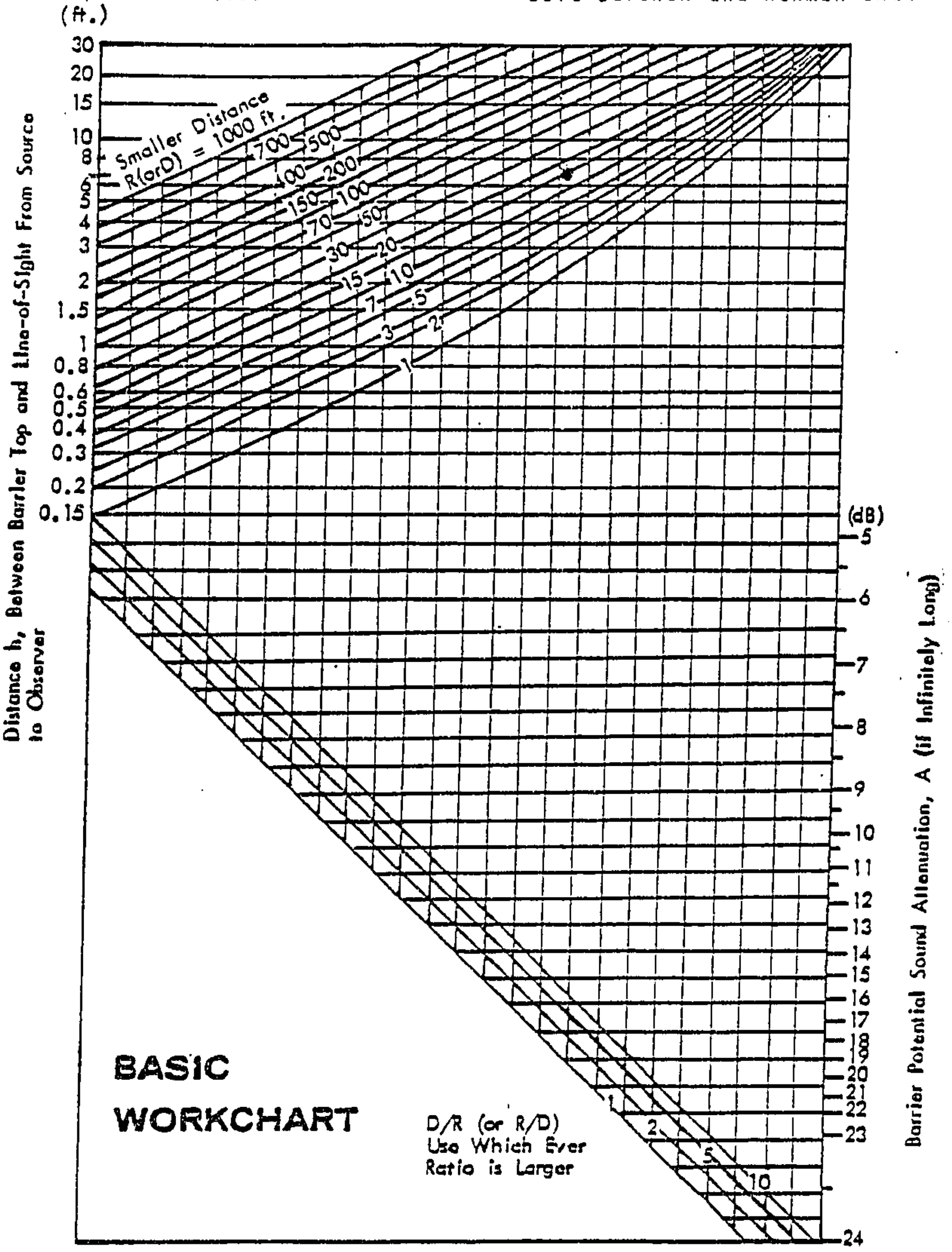


FIGURE A-2.2. BASIC WORKCHART FOR BARRIER SOUND ATTENUATION

Example

Estimate the sound attenuation of the barrier configuration illustrated in Figure A-2.3. The barrier is constructed of 1/2 inch plywood with a surface density of 1.5 lb/ft^2 and a sound transmission loss of 18 dB in the 500 Hz octave band (TL_{500} = 18).

1. The observer location, O, is as indicated on Figure A-2.3.

2. The source location, S_1, for the "over-the-top" sound path is chosen at the center of the machine (plan view) and 1/3 down from the top as shown on Figure A-2.3.

3. The quantities h_1, R_1, and D_1 are obtained from a scale drawing, as indicated in Figure A-2.3.

4. Entering the workchart with h_1 = 6.3 ft on the left-hand scale, a line is drawn to the right to intersect the curve corresponding to R = D = 16 ft (see Figure A-2.4).

5. Moving down, a line is drawn to intersect the curve corresponding to R/D = 1 (see Figure A-2.4).

6. Moving right, a line is drawn to intersect the vertical scale in order to find A_1 = 17 dB (see Figure A-2.4).

Since there are sound paths around the sides of the barrier and the surface weight of the barrier is less than 4 lb/ft^2, the calculation proceeds to Step 9.

7. If there are no sound paths around the sides of the barrier, and the barrier has a surface weight greater or equal to 4 lb/ft², then the estimated barrier attenuation may be taken to be A_1.

8. If there are no sound paths around the sides of the barrier, and the barrier has a surface weight less than 4 lb/ft², then the barrier attenuation may be estimated by combining A_1 with TL_{500} (the transmission loss of the barrier wall evaluated for the 500 Hz octave frequency band). A simplified method for combining decibel attenuations is provided in Table A-2.1.

9. If there are sound paths around one or two sides of the barrier, calculate the barrier attenuations A_2 and A_3 for these paths in the horizontal plane, as described in steps 3 through 6 above. The source locations for these calculations, however, should be at the *side* of the equipment closest to the barrier edge being evaluated.

10. Estimate the overall barrier attenuation by combining A_1, A_2, A_3, and TL_{500} (whichever apply) in a step-wise fashion, using the method in Table A-2.1

TABLE A-2.1

SIMPLIFIED COMBINATION OF DECIBEL ATTENUATIONS

When Two Decibel Attenuation Values Differ by:	Subtract the Following Amount from the Lower Value:
0 or 1 dB	3 dB
2 or 3 dB	2 dB
4 to 9 dB	1 dB
10 dB or more	0 dB

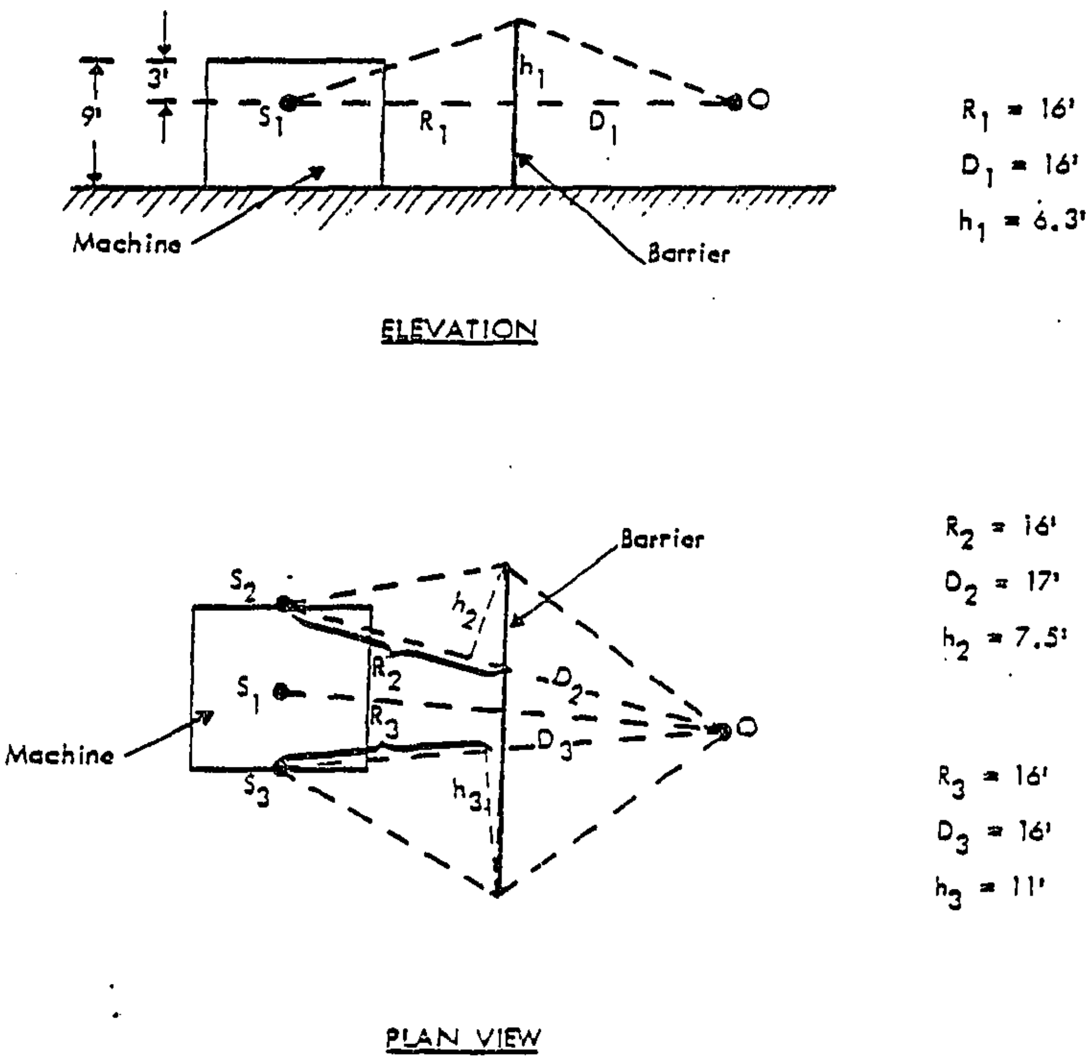


FIGURE A-2.3. BARRIER EVALUATION EXAMPLE

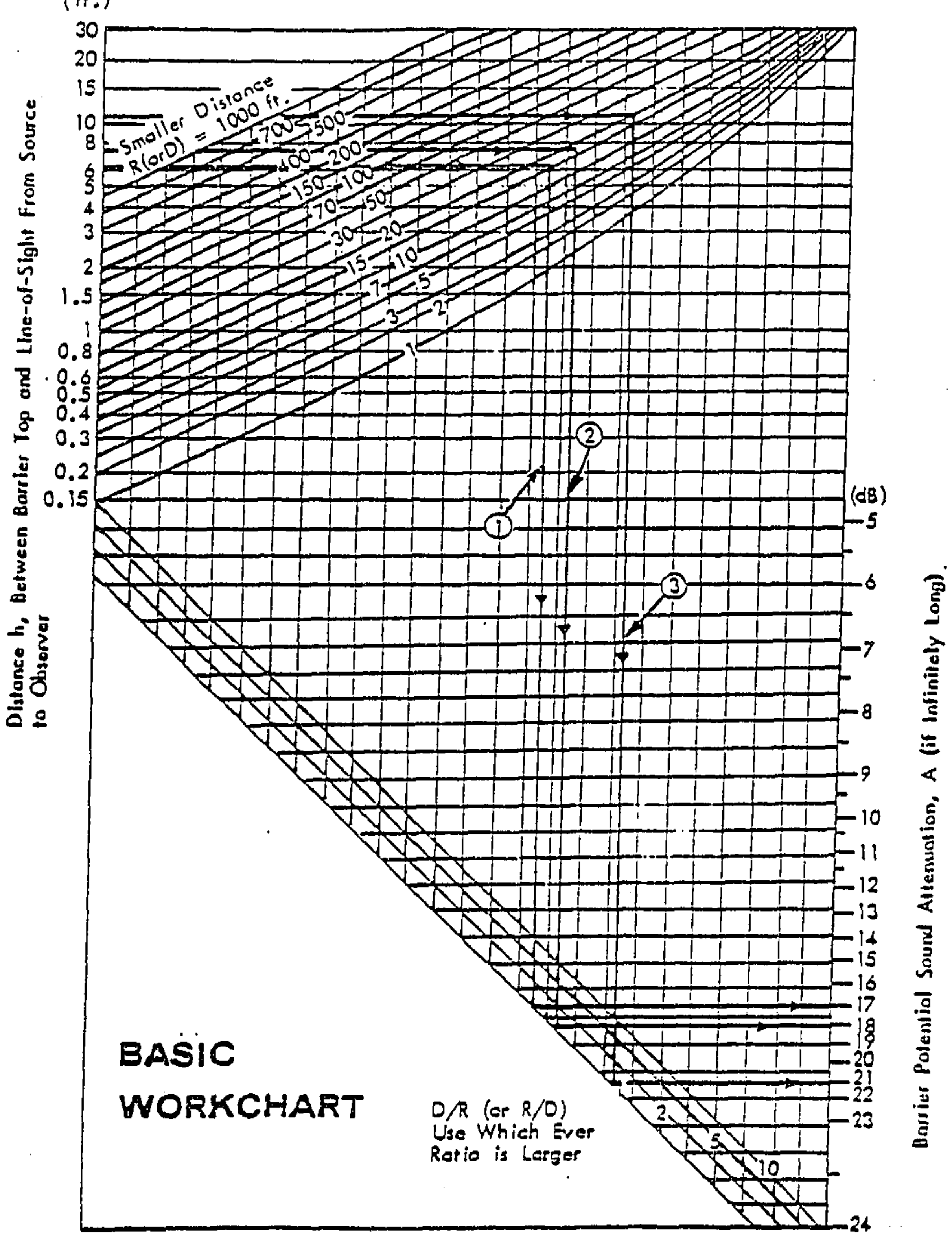
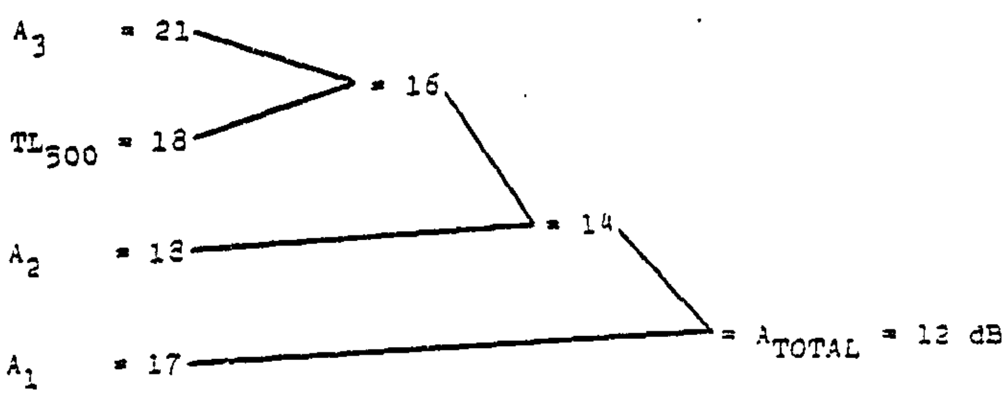


FIGURE A-2.4. WORKCHART SOLUTION TO BARRIER

9. The source locations, S_2 and S_3, for the sound paths around the two sides of the barrier are chosen at the ends of the machine as illustrated in Figure A-2.3. The attenuation values A_2 and A_3 for these paths are calculated in the same manner as A_1. The workchart computations, shown on Figure A-2.4, result in values of A_2 = 18 dB and A_3 = 21 dB.

10. The overall barrier attenuation is estimated by combining A_1, A_2, A_3, and TL_{500} using the method of Table A-2.1. In order to do this, the component attenuations are arranged in descending order and combined as shown below.



Thus, the procedure estimates an overall barrier attenuation of 12 dB, which is well below the attenuation of any one component path.

APPENDIX 3
ADDITION OF DECIBELS

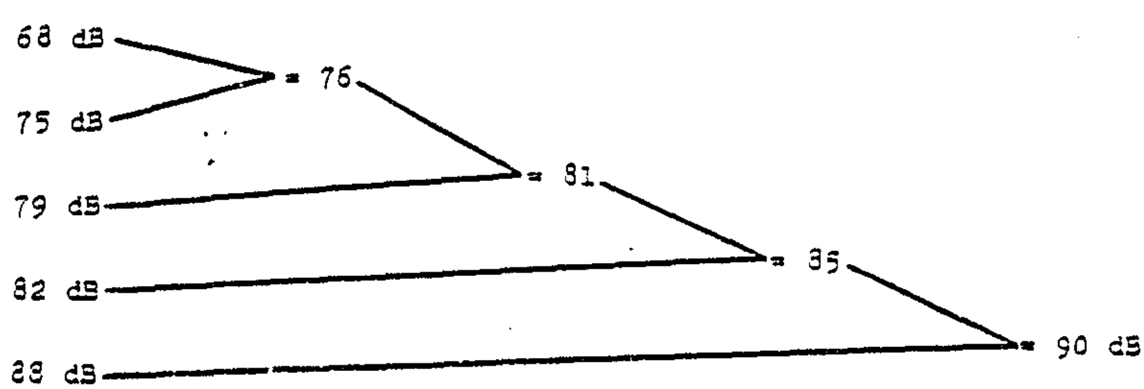
Since decibels are logarithmic values it is not proper to add them by normal algebraic addition. For example, 63 dB plus 63 dB *does not* equal 126 dB but only 66 dB.

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

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

When there are several decibel levels to be added, they should be added two at a time, starting with the lower valued levels and continuing the addition procedure of two at a time until only one value remains.

To illustrate, suppose it is desired to add the following five sound levels, using the above summation procedure:



The simplified addition rules above involve rounding off of some sums to the nearest whole number, resulting in the possibility of a small error. In general, the above procedure will yield sums accurate to the nearest 1 dB.

APPENDIX 4

SAMPLE PERMIT SCHEME WORKSHEETS

WORKSHEET A

Outdoor Equipment

Procedure for Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description ____
   ____
2. Identification Symbol on Drawings ____________________
3. Manufacturer and Model Number ________________________
4. Operating Conditions ______________________________
5. A-Weighted Sound Power Level _____ dBA re 10^{-12} Watt
   Spectrum Class _____
   ___ Calculated from tables (attach worksheet)
   ___ Certified test data (attach substantiation)

6. Installation Location:

   ___ On-grade
   ___ Roof-top

7. Presence of Nearby Reflecting Surfaces:

   ___ a. None ___ b. One ___ c. Two

8. Line of Sight between Equipment and Reference Point:

   ___ a. Unobstructed
   ___ b. Broken by solid barrier, roof setback, etc.

9. Distance, Equipment to Reference Point _________ feet

   ___ Perpendicular distance
   ___ Slant distance

Worksheet A (continued)

Part 2: Sound Level Estimation

10. Sound Power Level (from line 5) _____dBA re 10^{-12} Watt
11. Correction for Directivity:
    a. If 7a checked, enter 0 ____dB
    b. If 7b checked, enter 3 ____dB
    c. If 7c checked, enter 6 ____dB

12. Add lines 10 and 11 (a, b, or c) ____dBA re 10^{-12} Watt
13. Correction for Shielding:
    a. If 8a checked, enter 0 ____dB
    b. If 8b checked, enter:
       (1) 5 (allowance w/o calc.) or ____dB
       (2) Result of computation using Appendix 2 (attach calc's.) ____dB

14. Subtract line 13 from line 12 ____dBA re 10^{-12} Watt
15. Distance Correction (from Table 13 using distance shown on line 9) ____dB
16. Subtract line 15 from line 14 to get Sound Level at Reference Point ____dBA re $2 \times 10^{-5} N/m^2$

WORKSHEET B-1

Indoor Air-Handling Equipment

Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description ______________________________
2. Designation on Drawings or Schedule ____________________
3. Manufacturer and Model Number _________________________
4. Service Application:
   ___Supply Air
   ___Return Air
   ___Exhaust Air
5. Fan Type:
   ___a. Airfoil
   ___b. Backward Curved/Inclined
   ___c. Forward Curved
   ___d. Radial
   ___e. Vane-Axial
   ___f. Propeller
6. Fan Diameter: _____inches
7. Fan Operating Point:
   a. Volume _______cfm
   b. Total Static Pressure ______inches, w.g.
   c. Brake Horsepower _____ hp
   d. Static Efficiency at Operating Point _____%
   e. Peak Static Efficiency on Fan Curve _____%
   f. Percent of Peak Static Efficiency at Operating Point _____%
8. Configuration:

___a. Ducted
   (1) Duct Width _____ inches
   (2) Duct Height _____ inches
   (3) Duct Length _____ feet
   ___ (4) Ductlining:
      (a) Lining thickness _____inches
      (b) Length of straight lined duct_____feet
      (c) Lined elbow with minimum 10 ft of lining beyond elbow in direction of sound propagation:
         _____Yes
         _____No
   ___ (5) Packaged Sound Attenuator:
      (a) Manufacturer & Model Number _______________
      (b) Static Pressure Drop at 1000 fpm ____inches, w.g.
   (6) Area of Duct Opening at Building Face ____ft^2
      ___(a) Plenum Opening on Axis of Duct
      ___(b) Plenum Opening > 45° off Axis of Duct

___b. Non-Ducted (Applicable to Flush-Mounted Ventilating Fans)

Worksheet B-1 (Continued)

9. Distance, Building Opening to Reference Point _____ feet

___ Perpendicular Distance
___ Slant Distance

10. Line of Sight between Equipment and Reference Point

___ a. Unobstructed
___ b. Broken by solid barrier, roof setback etc.

Part 2: Sound Level Estimation at Reference Point

11. Calculation of Fan Sound Power Level (Based on Lines 5,6,7):

a. Specific Sound Power Level, K_A (Table 1) ___ dBA re 10^{-12} Watt
b. Volume Correction, A (Table 2*) ___ dB
c. Static Pressure Correction, B (Table 3) ___ dB
d. Static Efficiency Correction, C (Table 4) ___ dB
e. Sound Power Level (11a + 11b + 11c + 11d) ___ dBA re 10^{-12} Watt
f. Spectrum Class _____

*12. Correction for Lined Ductwork (only if line 8a,(4) checked):

a. Attenuation of Straight Duct (Table 14) ___ dB
b. Elbow Attenuation:
(1) If "Yes" checked, enter 5 ___ dB
(2) If "No" checked, enter 0 ___ dB

*13. Correction for Packaged Sound Attenuator (only if 8a,(5) checked):

a. Attenuation (Table 15) ___ dB
b. Certified Test Data (attach substantiation) ___ dB

14. Adjusted Sound Power Level:

a. Line 11e minus (Lines 12a + 12b)* ___ dBA re 10^{-12} Watt
b. Line 11e minus Line 13a, b* ___ dBA re 10^{-12} Watt

15. Calculation of Sound Level at Building Opening:

a. Correction for Duct Cross-Sectional Area (from Table 16 using area of 8a, (6), ___ dB
b. Correction for Plenum Loss
(1) If 8a,(6),(a) checked, enter 0: ___ dB
(2) If 8a,(6),(b) checked, enter 3: ___ dB
c. Add Lines 15a and 15b: ___ dB
d. Sound Level at Opening (Line 14(a or b) minus Line 15c): ___ dBA re $2x10^{-5}$ N/m^2

*A correction for either duct lining or a packaged sound attenuator is allowed but not both.

Worksheet B-1 (Continued)

*16. Correction for Directivity (use only if 10a is checked)

Vertical angle between reference point and opening:

0° - 30°, enter 0	___dB
30° - 60°, enter 3:	___dB
60° - 90°, enter 6:	___dB

*17. Correction for Shielding (use only if 10b is checked):

a. Allowance w/o Calculations, enter 5 ___dB

b. Computation from Appendix 2 (attach calculations): ___dB

18. Adjusted Sound Level at Building Opening (Line 15d minus 16, or, Line 17): ___dBA re $2x10^{-5} N/m^2$

19. Correction for Distance to Reference Point:

a. Distance Factor (From Table 19 and Line 9): ___dB

b. Area Factor (From Table 19 and Line 8a, (6)): ___dB

c. Line 19a minus Line 19b: ___dB

20. Sound Level at Reference Point: (Line 18 minus Line 19c) ___dBA re $2x10^{-5} N/m^2$

---

*A correction for either directivity or shielding is allowed, but not both.

WORKSHEET B-2

Building Mechanical Equipment Indoors

Calculation of Sound Level at a Reference Point Outdoors

Part 1: Reference Data

1. Equipment Description______________________________
2. Identification Symbol on Drawings______________________
3. Manufacturer and Model Number________________________
4. Operating Conditions______________________________
5. A-Weighted Sound Level at 1 meter ____dBA re 2×10^{-5} N/m^2
   Spectrum Class ________
6. Distance between Equipment and Closest Opening in Exterior Wall _______feet
   ___a. Opening unshielded from equipment
   ___b. Opening shielded from equipment
7. Dimensions of Opening:
   a. Height _____ft
   b. Width _____ft
   c. Area _____ft^2
8. Acoustical Treatment of Opening:
   ___a. None
   ___b. Packaged Sound Attenuator
   ___c. Acoustical Louvers
9. Distance, Building Opening to Reference Point _____ft
   ___Perpendicular Distance
   ___Slant Distance
10. Line of Sight between Equipment and Reference Point
    ___a. Unobstructed
    ___b. Broken by solid barrier, roof setback, etc.

Worksheet 3-2 (Continued)

Part 2. Sound Level Estimation at Reference Point

11. A-Weighted Sound Level at 1 meter (from Line 5) ___dBA re $2x10^{-5}N/m^2$
12. Correction for Distance to Closest Opening:
    a. Distance (using Table 17 and Line 6) ___dB
    b. Shielding:
        Unshielded, enter 0 ___dB
        Shielded, enter 3 ___dB
    c. Total Correction (12a + 12b) ___dB
13. Attenuation across Opening:
    a. If 8a is checked, enter 0 ___dB
    b. If 8b is checked, use
        ___Table 15 or
        ___Certified Ratings (attach substantiation) ___dB
    c. If 8c is checked, use
        ___Table 16 or
        ___Certified Ratings (attach substantiation) ___dB
14. A-Weighted Sound Level at Exterior Side of Opening:
    Line 11 minus Line 12c minus Line 13a,b,c: ___dBA re $2x10^{-5}N/m^2$

*15. Correction for Directivity (use only if Line 10a is checked:
    Vertical angle between reference point and opening:
    0° - 30°, enter 0: ___dB
    30° - 60°, enter 3: ___dB
    60° - 90°, enter 6: ___dB

*16. Correction for Shielding (use only if line 10b is checked):
    a. Allowance w/o Calculations, enter 5: ___dB
    b. Computation from Appendix 2 (attach calc's): ___dB

17. Adjusted Sound Level at Building Opening
    (Line 14 minus Line 15, or, Line 16): ___dBA re $2x10^{-5}N/m^2$
18. Correction for Distance to Reference Point:
    a. Distance Factor (from Table 19 and Line 9): ___dB
    b. Area Factor (from Table 19 and Line 7c): ___dB
    c. Line 18a minus Line 18b: ___dB
19. Sound Level at Reference Point
    (Line 17 minus Line 18c) ___dBA re $2x10^{-5}N/m^2$

*A correction for either directivity or shielding is allowed, but not both.