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ACQUISITION AND STUDY OF THE NOISE DATA OF CERTAIN ELECTRICAL AND MECHANICAL EQUIPMENT USED IN BUILDINGS

Part I of Final Report
by Laymon N. Miller

January 1970

BBN CONFIDENTIAL

Prepared for:

Department of the Army
Office of the Chief of Engineers
Washington, D.C. 20315

CAMBRIDGE NEW YORK CHICAGO LOS ANGELES

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Bolt Beranek and Newman Inc.
50 Moulton Street
Cambridge, Massachusetts 02138

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PREFACE

This report summarizes the accumulation and study of noise data of the electrical and mechanical equipment agreed upon within the terms of Contract No. DACA 73-68-C-0017 between the Office of the Chief of Engineers and Bolt Beranek and Newman Inc. The equipment included in this study represents most major equipment involved in the operation of almost any large occupied building, with the exception of (1) the entire air handling and air distribution system and (2) personnel and material conveyance systems (elevators, escalators and conveyors). The purpose of this report is to arrive at noise estimation methods for the various types of included equipment, so that a reasonably reliable noise estimate can be made of the equipment, on the basis of type, size, speed or other operational characteristic, without having to first define the specific manufacturer and model of the equipment. With such noise estimates, the architect and engineer can proceed with building designs of mechanical spaces that would adequately contain or control the noise of this equipment, before final equipment bids are submitted and selected.

The noise estimation methods derived in this report are used in the final report of this project which is an engineering manual entitled "Mechanical Equipment Noise Control". This manual is a follow-up of the earlier BBN-OCE manual on Power Plant Acoustics (TM 5-805-9) and will be directed toward the design of noise and vibration control for the electrical and mechanical equipment as installed in buildings.

WARNING

Some of the noise data obtained and presented in this report were obtained with the permission of the equipment manufacturers with the understanding that the data would only be used to develop general noise relationships, that the data would be used discreetly and that the manufacturers' specific data would not be made public. This report is prepared for the sole use of the Corps of Engineers of the U.S. Army as back-up information to give technical support and verification of the general noise relationships that have been derived here and that will be used in the Manual. Only a few reports have been printed and distributed to the Corps of Engineers. Further copying of this report is not authorized without permission of the Corps of Engineers or the written permission of all manufacturers represented.

ACQUISITION AND STUDY OF THE NOISE
DATA OF CERTAIN ELECTRICAL AND
MECHANICAL EQUIPMENT USED IN BUILDINGS

SECTION 1. INTRODUCTION

In the pursuit of this project, noise data on equipment have been collected from four general sources: (1) from published literature, (2) from manufacturers, (3) from BBN files on earlier jobs, and (4) from BBN measurements specifically for this job. At this point in the report, the breakdown of all data into these four sources is not attempted; rather, some reference to the sources of data is made within each discussion of the specific types of equipment. Early in the project a form letter was sent to over 130 manufacturers of equipment included in this report seeking noise data or suggestions on control of noise and/or vibration of their equipment. The response was better in some ways than expected, but was quite poor on an absolute scale, indicating the paucity of data (and possibly interest) in the noise of their products. Some companies made generous and important contributions of data and we are particularly grateful to them for their interest and contributions. These contributions are acknowledged separately under the particular equipment discussions. We wish, at this point, to give special credit to Mr. Peter Baade, Senior Acoustical Engineer of the Research Division of the Carrier Corporation, for his assistance to our work. Mr. Baade is in charge of editing the chapter on Noise Control of the ASHRAE Guide and Data Book, so he especially appreciates the magnitude of the task and its value to architects and engineers. Among other positive contributions to this project, he furnished us with a recent paper by Irving Heitner.* Mr. Heitner's paper is a significant contribution in the area of plant noise and much of the equipment given in his paper is within the list of equipment covered by this report. In the separate sections on specific equipment, the Heitner estimates, where applicable, are included along with other data. A copy of Mr. Heitner's paper is included at the end of this report. Although we believe that the total collection of data on the specific equipment listed here is more extensive and up-to-date than some of Heitner's data, we nevertheless admire the thorough work that he obviously has put into his paper.

Three brief explanations are offered here for the data studies that follow:

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

one pertains to SPL vs PWL, one pertains to the development of the noise estimation curves, and one pertains to vibration data.

1-01. SPL DATA. In the collection of noise data for the Power Plant Acoustics manual it was almost essential that the data ultimately be presented in terms of sound power level. This was brought on by the fact that the engines varied greatly in size and their rooms varied greatly in volume and layout. It was also made possible by the fact that most engines were measured by BBN personnel, measurements were made from several positions in the room and the acoustic absorption conditions were noted so that the effect of the room acoustics could be considered in calculating the sound power level. Also, some engines were measured out-of-doors and that effect was also taken into account.

In the present collection of data, most noise levels are measured at close-in positions because equipment space is usually limited; most mechanical equipment spaces are somewhat similar in size (especially the height of the room); and for a very major part of the data, not enough information is given about the room conditions to warrant an evaluation of sound power level. For these reasons, most of the noise data are presented in terms of sound pressure levels rather than sound power levels.

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

1-02. NOISE ESTIMATION CURVES. In the Power Plant Acoustics manual, referring to reciprocating engine casing noise as an example, a best-fit curve was drawn in an attempt to represent the data of a large number of engines having a number of noise-influencing factors. A "noise design curve" was then drawn 2 dB above the best-fit curve. The values of this design curve then were equal to or greater than the PWL values of approximately 80% to 90% of all the engines studied. This approach seemed to give the necessary design protection, recognizing that a few engines would be slightly noisier than the design curve would predict. The detailed data study on engine casing noise produced what appeared to be important parameters to noise generation, so the finally selected design curve, with its appropriate corrections for specific engine characteristics, led to a relatively low standard deviation between measured noise and estimated noise. The agreement between measured and estimated noise for other portions of reciprocating and turbine engine noise was not as good as for reciprocating engine casing noise due to the inability to specify so well the factors that influence noise generation and radiation.

In the present project, several factors make it impossible to be as specific about the noise of this equipment as we were able to be in the engine noise study: (1) a greater diversity and a larger number of types of equipment are included in the present study, (2) less measured field data have been taken on each type of equipment,(3) there may be considerable design variability by different manufacturers for some of the equipment included here, (4) the amount of time and money available for this study does not permit an in-depth evaluation (or attempted evaluation) of some of the more subtle characteristics that might influence noise, and (5) the noise of most of this equipment is not so high but that it can be controlled by normal methods. This latter point is an important one. If a diesel engine were placed in an upper floor of a hospital, apartment building or office building, certain special and somewhat unconventional steps would have to be taken to control the transmission of noise and vibration to nearby critical areas of the building. However, the noise and vibration produced by most of the equipment included in the present study can be controlled by fairly straight-forward methods that are already in general use by many architects and engineers. Thus, it is believed unnecessary to know to a high degree of accuracy the noise of all the equipment, as long as the expected upper limits of the noise can be protected by reasonably conventional methods. In the data summaries that follow, the "Design Curves" tend to represent the 80 to 95 percentile curves of noise where adequate data seem to justify this selection, or possibly the upper limits (or beyond) of the measured noise when very little data were available. To some extent, the resulting designs may be slightly over-designed if the actual equipment is quieter than the design curve would indicate. On the other hand, where limited amounts of data have gone into this study, there is always the possibility that the actual equipment used may be slightly noisier than the design curve would indicate. For upper floor locations, floor requirements are more likely to be imposed by the vibration isolation needs of the installation than by the airborne sound aspects of the problem, so an over-design for airborne sound may not be an over-design in terms of vibration control.

To summarize, depending on the quantity, quality and uniformity of the noise data, Design Curves are developed to provide a high degree of protection in the estimating of equipment noise.

1-03. VIBRATION DATA. Vibration data have been collected and studied for several pieces of equipment in this project. The vibration levels are quite variable, however, due to (1) actual position of the vibration pick-up (accelerometer) on the equipment, (2) nature of the isolation mounting of the equipment, and (3) ambient vibration levels in the building due to other sources that could not be turned off. The data are sufficiently variable and sketchy that it is impossible to draw conclusions on the general vibration characteristics of equipment and of equipment mounting arrangements that would be meaningful to this study. Consequently, vibration data are not presented here or in the Manual, but vibration control recommendations for all included equipment in on-grade and upper-floor locations are given in the Manual.

1-04. HZ VS CPS. In the relatively short time since the completion of the Power Plant Acoustics Manual, the term "Hertz" has moved rapidly into fairly common use as the unit of frequency, replacing the older and more familiar unit "cycles per second". Even though this represents a change from the Power Plant Acoustics Manual, it is proposed that the new term "Hertz" (abbreviated "Hz") be used throughout the new Noise Control Manual. Data given in this report refer to Hz for expressing frequencies.

1-05. EQUIPMENT NOISE SUMMARIES. Individual discussions of the noise of each type of equipment are given in the sections that follow. The types of equipment included in the noise summaries are outlined below:

Refrigeration System Equipment

- Packaged Chillers with reciprocating compressors
- Packaged Chillers with rotary-screw compressors
- Packaged Chillers with centrifugal compressors
- Built-up refrigeration machines
- Absorption machines

Heating System Equipment

Boilers
Steam Valves

Liquid Circulation System Equipment

Cooling Towers
Pumps

Air Compressors

Prime-Mover Equipment

Electric Motors
Steam Turbines
Gears

Electric Equipment

Transformers

SECTION 2.

PACKAGED CHILLERS WITH RECIPROCATING COMPRESSORS

Noise data for 24 reciprocating compressors or packaged chillers with reciprocating compressors have been collected and studied. These units range in size from 15 tons to 150 tons cooling capacity. Two manufacturers, Carrier Corporation and Dunham-Bush, have submitted data on a total of six of their compressors; an earlier paper* by Robert M. Hoover of BBN provided data on eight compressors, and the data for ten compressors have been collected or measured specifically for this job. The noise levels have been reduced to a common 3-ft distance from the front of the compressor. All the known data of the units are summarized in Table RC-1. The rated cooling capacity is

* "Noise Control for Reciprocating Compressors", Robert M. Hoover and Lloyd J. Williams; Heating, Piping and Air Conditioning, November 1962.

given in tons for most of the machines, but when this value was not known, the rated HP of the driving motor or engine has been used. These values are close enough that they are frequently used interchangeably. Measurement positions are selected to emphasize compressor noise, but it is possible that some noise levels are influenced by other noise sources in the area.

In terms of noise production, it appears that the measured compressors can be divided into two groups: 15-50 tons and 51-150 tons. The two ends of the total range have been extended slightly to cover compressors from 10 to 175 tons. The suggested Design Curves based on study of all the noise data are given in Figure RC-1. There is not a large enough range in speed for the machines measured to justify a noise adjustment for speed.

In Figure RC-2 the octave band sound pressure levels of the seven compressors in the 15-50 ton group are compared with Design Curve A. A similar comparison is shown in Figure RC-3 for Design Curve B and the seventeen units in the 51-150 ton group. Only four data points exceed the Design Curves: one exceedance is of 2 dB and three exceedances are of 1 dB only. Thus, these Design Curves represent approximately the 85 to 100 percentile noise curves in the four frequency bands of most concern in considering noise control in buildings, namely the 125, 250, 500 and 1000 Hz bands. In these same bands, the 75 percentile curve would fall about 3 dB below the Design Curve and the 50 percentile curve would fall about 7 dB below the Design Curve. All of this means that noise control based on the Design Curves would give adequate protection for approximately 85% to 100% of these machines (based on the limited sampling of this study). If the Design Curves were 3 dB lower, they would give protection to about 75% of the equipment; or if the Design Curves were about 7 dB lower, they would give protection to about 50% of the equipment.

When cooling requirements exceed about 100 to 150 tons, centrifugal compressors become more economical so there are few reciprocating units rated above about 150 tons. Even in this collection of data, several of the larger units are actually made up of assemblies of two to four smaller compressors.

The noise levels of the two Design Curves of Figure RC-1 are proposed for use in the Manual and the values are tabulated in the accompanying Table RC-2. Although major interest has been concentrated here on the compressor component of a refrigeration machine, an electric motor is usually the drive unit for the compressor. The noise levels attributed here to the compressor will encompass the drive motor most of the time, so these values are taken to be applicable to either a reciprocating compressor alone or to a motor-driven packaged chiller containing a reciprocating compressor. For a more

TABLE RC-1

MEASURED NOISE LEVELS OF RECIPROCATING COMPRESSORS
USED WITH PACKAGED CHILLERS (NORMALIZED TO 3-FT DISTANCE)

CODE	HP OR TONS	COMPR. RPM	MANUFACTURER, MODEL AND DRIVE UNIT	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
RC-1	15		(H)	--	88	79	79	82	83	83	74	70
RC-2	20		(H)	--	85	84	78	82	82	82	74	68
RC-3	25		(H)	--	83	84	86	86	86	84	80	73
RC-4	25	1750	Dunham-Bush PC-25 (M)*	--	71	66	72	74	75	68	63	60
RC-5	27	1750	Carrier 06EW033 (M)*	--	--	--	62	76	78	76	67	70
RC-6	50	800	Vilter (E)	79	83	79	83	81	80	76	72	69
RC-7	50			--	77	79	84	78	77	78	79	69
RC-8	60	2200	Vilter 4412-84 (E)	83	84	86	87	93	90	85	80	77
RC-9	60	1500	Acme GD-40 (E)	76	81	83	86	88	88	85	86	81
RC-10	60		(H)	--	90	82	82	88	90	89	82	75
RC-11	62	1750	Carrier 30HR060 (M)*	--	--	--	73	79	80	70	68	72
RC-12	70			--	79	80	85	76	77	74	69	62
RC-13	75	1750	York 3D75E (M)	79	81	83	82	85	89	82	77	70
RC-14	75	1750	York 3D75E (M)	80	82	86	85	91	91	82	82	78
RC-15	75		(H)	--	89	81	92	94	92	88	79	73
RC-16	80			--	80	79	83	83	85	84	78	71
RC-17	80	1750	Dunham-Bush PC-80 (M)*	--	69	70	74	73	79	79	78	75
RC-18	82	1750	Carrier 5H120 (M)*	--	77	82	79	84	80	80	74	64
RC-19	100	1750	Carrier 30HR100 (M)	--	--	68	73	81	81	73	71	72
RC-20	100	660	Worthington 6JF4-100 (M)	--	80	79	82	83	82	79	78	78
RC-21	100	1750	Carrier 30AA-100 (M)	--	85	80	80	86	86	80	70	65
RC-22	100		(H)	--	91	79	85	86	83	83	80	79
RC-23	100		(H)	--	81	77	91	93	89	86	78	74
RC-24	150		(H)	--	84	83	89	90	93	88	85	81

FOOTNOTES:

(M) Compressor driven by electric motor.

(E) Compressor driven by engine.

(H) Data obtained from paper by R. M. Hoover; manufacturer not specified.

* Data provided by manufacturer.

TABLE RC-2

ESTIMATED SOUND PRESSURE LEVELS OF PACKAGED CHILLERS WITH RECIPROCATING COMPRESSORS (AT 3-FT DISTANCE INDOORS)

COOLING CAPACITY TONS	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
10-50	82	86	84	86	87	86	84	80	75
51-175	85	90	89	92	93	92	90	86	81

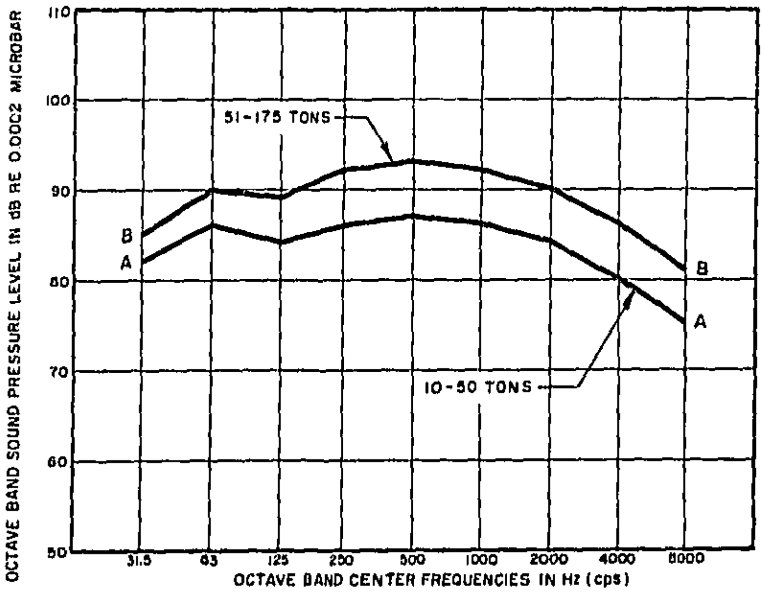

FIG. RC-1 NOISE LEVELS OF RECIPROCATING COMPRESSORS AT 3-FT DISTANCE (Suggested design curves for two ranges of cooling capacity)

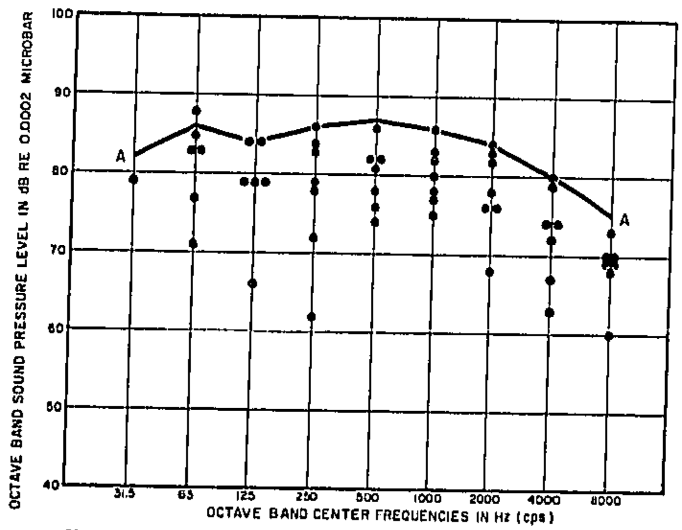

FIG. RC-2 NOISE LEVELS OF RECIPROCATING COMPRESSORS AT 3-FT DISTANCE (Comparison of design curve A with noise data for refrigeration compressors of 15-50 ton cooling capacity)

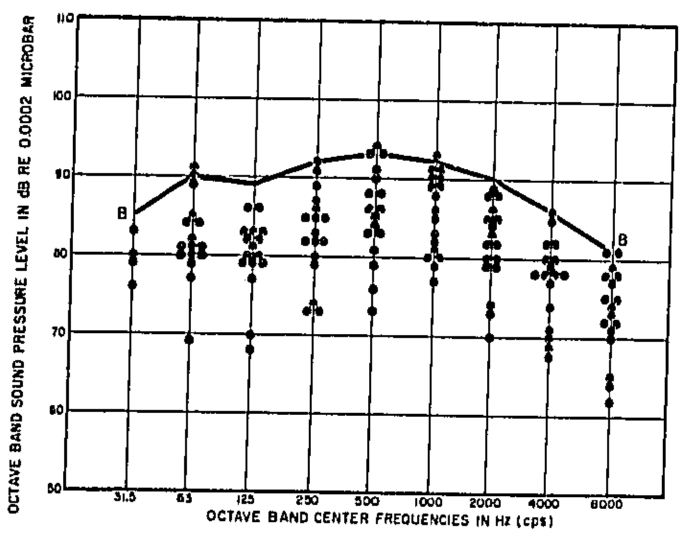

FIG. RC-3 NOISE LEVELS OF RECIPROCATING COMPRESSORS AT 3-FT DISTANCE (Comparison of design curve B with noise data for refrigeration compressors of 51-150 ton cooling capacity) complete analysis of any particular combination, motor data taken from another part of this report can be compared with compressor data. It will be found that some motors in the 1800 RPM group and in the power range of 100 to 200 HP will be slightly noisier than compressors in a few high frequency bands.

SECTION 3.

PACKAGED CHILLERS WITH ROTARY-SCREW COMPRESSORS

In response to our request for noise data on refrigeration equipment, Dunham-Bush, Inc. offered data on packaged chillers equipped with rotary-screw compressors. Apparently this is a relatively new form of compressor application in this country, although a somewhat similar pump by De Laval was encountered several years ago in shipboard applications. We have had no field experience with the Dunham-Bush units, but we include here the noise data offered by them to us. Because of the limited amount of data, no attempt is made to derive a procedure for estimating the noise of similar machines for a variety of sizes, speeds or manufacturers. The accompanying Table RS-1 summarizes the SPLs at 3 ft distance for a 120-ton and two 230-ton units, each operating at 3600 RPM. Dunham-Bush produces five models of this machine, covering the range of 120 to 350 tons.

For purposes of noise control, 3 dB have been added to the highest octave band levels given by the manufacturer, and these higher levels are proposed for use in the Manual. They are shown in the bottom line of Table RS-1. Obviously, this represents an attempt to protect building designs against somewhat noisier equipment than reported by the manufacturer. In view of the shortage of data on this type of equipment, this higher estimate seems justified.

TABLE RS-1

APPROXIMATE SOUND PRESSURE LEVELS AT 3-FT

DISTANCE DUE TO PACKAGED CHILLERS

HAVING ROTARY-SCREW COMPRESSORS

Dunham-Bush Model	OCTAVE BAND CENTER FREQUENCY - HZ								
	31	63	125	250	500	1000	2000	4000	8000
PCX-120H 120 ton, 3600 RPM	-	65	62	77	82	79	77	72	65
PCX-230H 230 ton, 3600 RPM TWO TESTS: (1)	-	68	71	89	86	82	75	72	66
(2)	-	73	77	82	86	82	75	72	70
Suggested Design Noise Levels, 100-300 Tons	70	76	80	92	89	85	80	75	73

SECTION 4.

PACKAGED CHILLERS WITH CENTRIFUGAL COMPRESSORS

In a typical packaged refrigeration machine the compressor is the principal noise source, even though compressor-induced noise may be radiated by the condenser and chiller cylinders and the interconnecting piping. The noise of the drive unit (electric motor, steam turbine or gear) is to be considered separately if the driver is a separate assembly; otherwise, the noise of the drive unit is included with that of the compressor. When the cooling requirement is greater than about 100 to 150 tons, the refrigeration compressor is typically centrifugal. For this study, noise has been obtained or measured for 22 centrifugal type compressors. These measured compressors range in size from 140 tons to 4000 tons and represent five leading manufacturers. The SPLs at the normalized 3 ft distance and other known data of the compressors are included in Table CC-1. These noise levels may be influenced by motors, gears or steam turbines used to drive the compressors, but the measurement positions are generally selected to emphasize the compressor noise. In Table CC-1 most cooling capacity ratings are given in tons, but where this rating was not known, the HP rating of the driver has been used. These two quantities are very nearly equal for many refrigeration machines.

The SPLs of these 22 units are plotted in Figures CC-1 to CC-3, separated into three groups on the basis of rated cooling capacity: 140-350 tons, 400-700 tons and 1200-4000 tons. These are somewhat arbitrarily selected groups, but they illustrate the large range of noise levels found within relatively small intervals of size.

There is no significant correlation between noise and (a) ton rating (or HP of the drive system), (b) compressor speed or (c) manufacturer. Within close groups, based on any one of these three possible parameters, large variations in noise are found. Thus it is believed impractical to try to set up a noise prediction scheme closely tied to individual design or operational factors. The apparent oddity in the data that upsets any simple correlation of noise output with size of machine is the fact that the noise levels of the medium-size group (400-700 tons, Figure CC-2) are generally above those for both the smaller and the larger groups. Within that noisier group there does not appear to be any basic reason for the greater noise: different manufacturers, different drive systems and different speeds are all represented in this group. Due to the limited number of machines represented (only three units are outstandingly noisy: CC-8, CC-11 and CC-12 in Table CC-1), it does not seem justified to construct a special noise relationship around this medium-size group; yet there may be a trend here that deserves more study in the future. Based on some comments from

TABLE CC-1

MEASURED SOUND PRESSURE LEVELS OF PACKAGED CHILLERS WITH CENTRIFUGAL COMPRESSORS
(NORMALIZED TO 3-FT DISTANCE INDOORS)

CODE	HP OR TONS	COMPR. RPM	MANUFACTURER, MODEL AND DRIVE INFORMATION	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
CC-1	140	18000	AMER-STD HI-SPEED TONRAC 601B, HERM.	--	75	83	88	87	83	81	79	73
CC-2	150		TRANE CENTRAVAC, HERMETIC	--	84	73	71	73	78	86	77	69
CC-3	180		YORK TURBOPAK HT-20, HERMETIC	80	73	85	82	82	78	78	78	71
CC-4	225	3600	TRANE PCV-2C	--	80	73	79	81	84	81	79	73
CC-5	240	3600	AMER-STD TONRAC 18-L-135, HERM.	--	77	81	79	77	77	75	71	61
CC-6	350	3600	TRANE CENTRAVAC, HERMETIC	--	83	76	72	71	75	82	74	61
CC-7	350	3600	SIMILAR TO CC-6, SAME INSTALLATION	--	77	79	73	72	73	74	70	59
CC-8	400	3600	CARRIER (17-M?)	84	95	91	88	92	89	87	87	80
CC-9	450	3600	AMER-STD TONRAC 450-701, 2-ST. HERM.	87	78	79	80	81	88	85	77	65
CC-10	480		YORK	86	79	84	80	80	83	89	91	82
CC-11	650		WORTHINGTON, HERMETIC	82	83	86	81	88	97	96	91	86
CC-12	700	6600	YORK 226-A-6 (GEAR DRIVEN)	78	83	87	91	91	93	100	93	88
CC-13	1200		WORTHINGTON 42 EH-12-12 (STEAM TURB.)	88	80	83	80	80	80	88	76	70
CC-14	1250	5894	YORK TURBOMASTER 238-B-8, 2-STAGE (GEAR DRIVE)	--	81	82	82	86	86	95	86	81
CC-15	1500	4600	YORK 238-A-4 (STEAM TURBINE) (MEASURED AT 1/3 LOAD, 3200 RPM)	79	74	75	75	74	79	79	72	72
CC-16	1500	5000	YORK 238-B-8 (MOTOR DRIVEN)	76	79	80	82	86	85	88	74	69
CC-17	1500	3600	CARRIER 19-C-1512, HERMETIC	80	74	75	76	74	77	76	70	69
CC-18	1500	3600	CARRIER 19-C-8x5	84	86	84	84	86	92	85	77	69
CC-19	1500	3600	SIMILAR TO CC-18, SAME INSTALLATION	84	84	84	82	83	87	84	75	66
CC-20	1500	3600	CARRIER, 19 DA 160	--	79	85	85	86	84	90	90	81
CC-21	3500	5610	CARRIER, 17 DA (STEAM TURBINE)	--	70	79	83	86	89	92	84	78
CC-22	4000	4150	CARRIER, 17 DA (STEAM TURBINE)	--	--	87	90	92	96	88	82	78

TABLE CC-2

ESTIMATED SOUND PRESSURE LEVELS AT 3-FT DISTANCE
DUE TO PACKAGED CHILLERS HAVING CENTRIFUGAL COMPRESSORS

	OCTAVE BAND CENTER FREQUENCY - HZ								
Cooling Capacity	31	63	125	250	500	1000	2000	4000	8000
Under 500 tons	87	88	89	90	90	91	92	87	80
500 tons or more	89	90	91	92	93	97	99	94	87

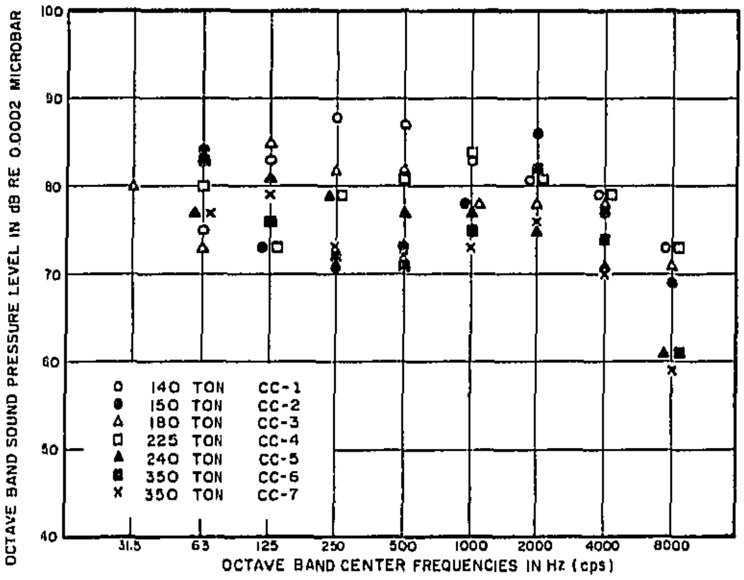

FIG.CC-1 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS, 140-350 TONS
(Measured Data Normalized to 3 ft Distance)

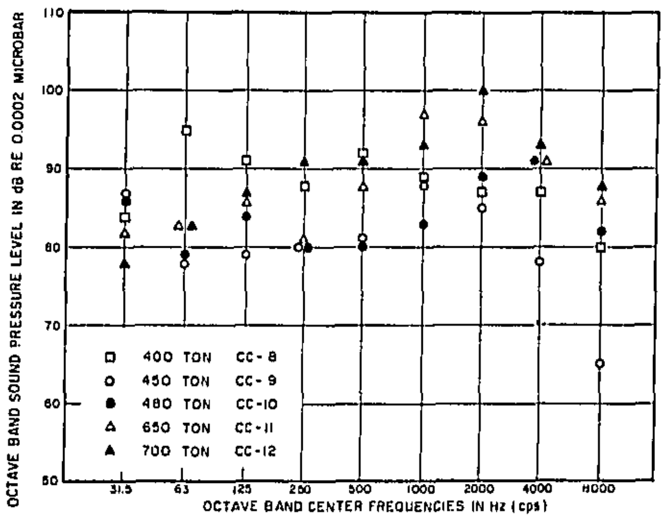

FIG. CC-2 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS, 400-700 TONS
(Measured data normalized to 3-ft distance)

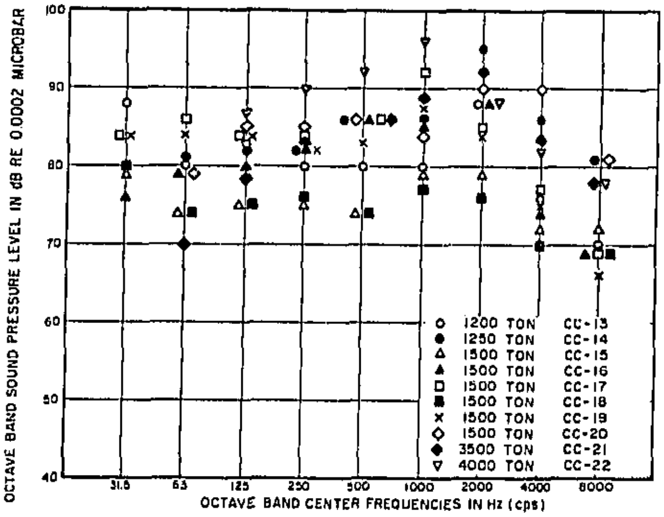

FIG. CC-3 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS, 1200-4000 TONS
(Measured data normalized to 3-ft distance)

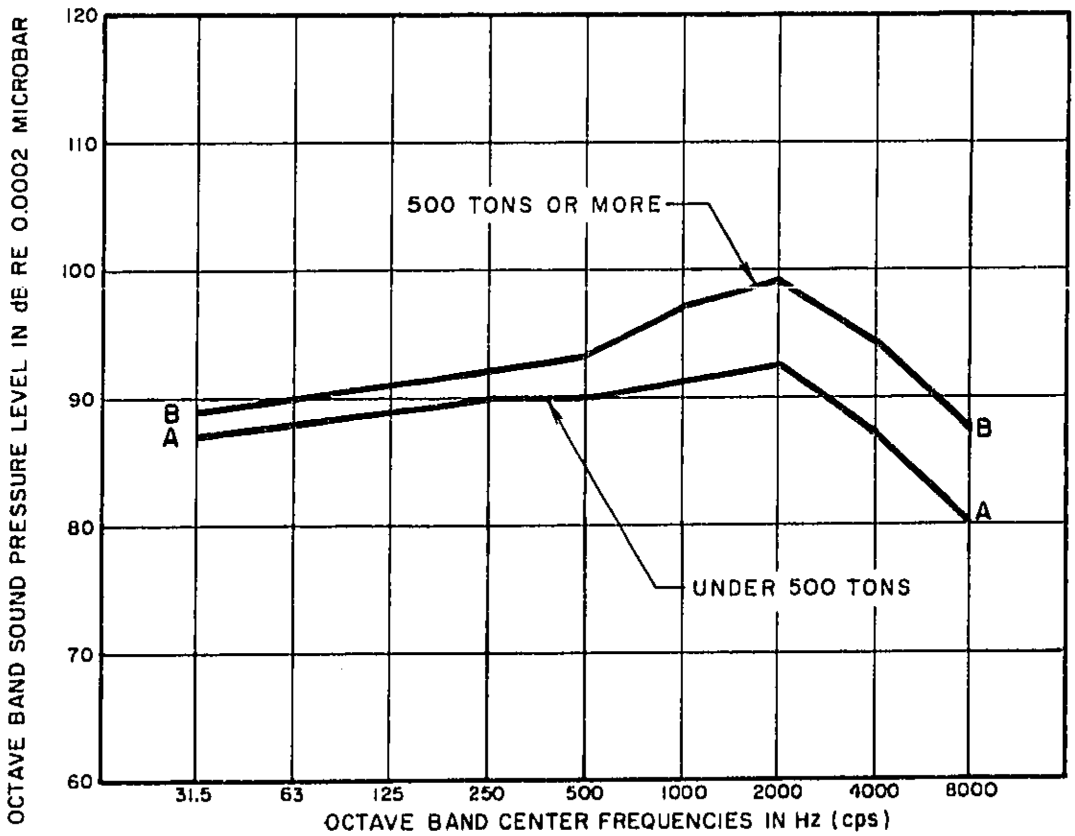

FIG.CC-4 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS AT 3-ft DISTANCE
(Suggested design curves for two ranges of cooling capacity)

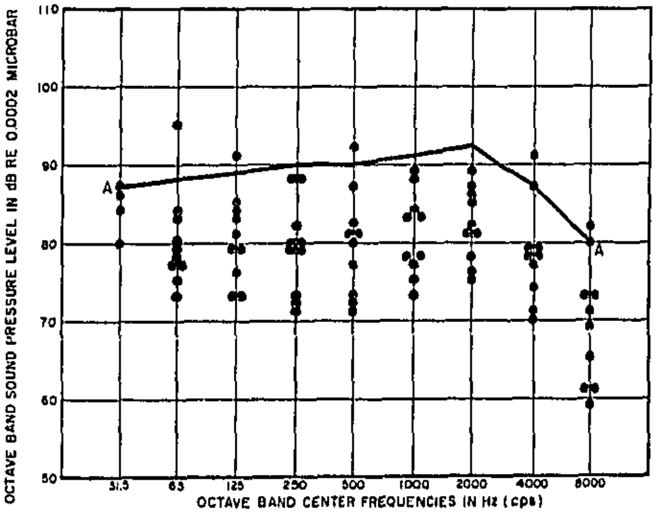

FIG.CC-5 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS AT 3-ft DISTANCE
(Comparison of design curve A with measured noise data for refrigeration machines of under 500 tons cooling capacity)

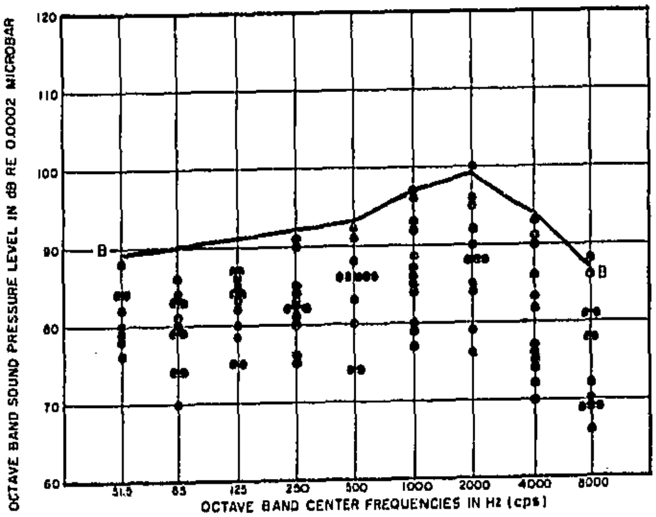

FIG.CC-6 NOISE LEVELS OF CENTRIFUGAL COMPRESSORS AT 3-ft DISTANCE
(Comparison of design curve B with measured noise data for refrigeration machines of 500 tons or more cooling capacity)

Peter Baade of Carrier Corporation it is possible that different units had differing amounts of wrapping or covering or acoustic treatment at or near the discharge line of the machine, which he describes as generally being the noisiest part of the machine.

Two design curves for noise of centrifugal compressors are suggested in Figure CC-4: one curve is proposed to represent units of under 500 tons cooling capacity and one curve is used to represent units of 500 or more tons cooling capacity*. These curves represent approximately the 90 to 100 percentile noise levels in the octave bands above 500 Hz. The design curves are intentionally a few decibles higher than most of the measured noise in the 63-250 Hz frequency bands because sometimes during periods of low cooling loads on a centrifugal compressor an additional amount of low frequency noise is produced. Figures CC-5 and CC-6 compare the two design curves with the measured noise levels for compressors in the "under 500 tons" and "500 tons or more" ratings.

The design curves of Figure CC-4 are proposed for use in the Manual, and the noise level values for these curves are tabulated in Table CC-2. These values are believed to cover a very large portion (possibly 90-95%) of all refrigeration compressors of the centrifugal type. For a less conservative approach, these values could be reduced 3 dB with a slightly reduced probability (possibly 75-90%) of adequate coverage.

SECTION 5. BUILT-UP REFRIGERATION MACHINES

The noise of packaged chillers, as presented in the preceding sections of this report, generally includes the noise of both the compressor and the drive unit. If a refrigeration system is to be made up of separate pieces, then the noise level estimate should include the noise of each component making up the assembly. The noise levels of the components should not be added together, but the noise of the combined equipment in each octave band should equal the highest noise level of each component in that octave band.

As an example, suppose a built-up refrigeration machine is to be made up of a steam turbine, a gear and a centrifugal compressor. Assume a 1000-HP

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

steam turbine at 1800 RPM, a 1000-HP gear at 1800 RPM input speed and 3600 RPM output speed and a 1000-ton centrifugal compressor at 3600 PRM. From the other sections in this report, the following 3 ft SPLs are estimated for the nine octave frequency bands from 31 Hz to 8000 Hz respectively:

for the steam turbine (from Table S-2)

88	93	95	91	87	87	88	85	80

for the gear (from Table G-2)

94	97	100	100	100	100	100	100	100

for the centrifugal compressor (from Table CC-2)

89	90	91	92	93	97	99	94	87

From these three rows of values, it is seen that the gear noise dominates all octave bands. The noise levels for the entire system would then be taken as the highest levels of each of the components, i.e.

94	97	100	100	100	100	100	100	100

SECTION 6. ABSORPTION MACHINES

Noise data have been acquired for only two steam absorption machines for this study. More data would be sought if these were notoriously noisy devices, but they are quiet enough that they are usually ignored and any noise survey of a mechanical equipment room. The noise is usually made up of some crackling, popping or hissing noise that sounds somewhat similar to ice cubes being chipped-up by a rotating propeller blade. The machine usually includes one or two small pumps. Steam flow noise or steam valve noise may also be present.

Figure A-1 includes plots of the noise of two measured Carrier absorption machines plus the noise of an under-12-HP motor and pump as taken from Figures M-1 and P-1 in other sections of this report. It is believed that an envelope curve slightly above each of these individual noise contributions will give adequate coverage of most absorption machines used in refrigeration systems for buildings. The noise levels shown by the design curve in Figure A-1 are given in Table A-1 and are planned for use

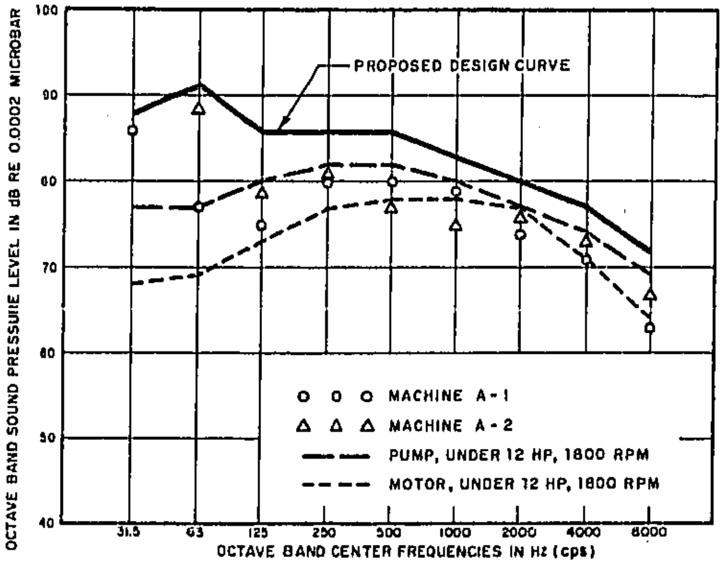

FIG.A-1 NOISE LEVELS AT 3-ft DISTANCE FOR ABSORPTION MACHINES

TABLE A-1

ESTIMATED SOUND PRESSURE LEVELS OF ABSORPTION MACHINE AT 3-FT DISTANCE

OCTAVE FREQUENCY BAND Hz	SOUND PRESSURE LEVEL IN BAND AT 3-FT DISTANCE dB
31	88
63	91
125	86
250	86
500	86
1000	83
2000	80
4000	77
8000	72

in the Manual. There are no data relating these noise levels to the size or capacity of the absorption machines.

SECTION 7. BOILERS

Noise data have been measured or collected for at least 36 boilers; in some installations two or three boilers were measured and the averages are given here as though for a single boiler. In a majority of the cases, neither the manufacturer was known nor was a reliable estimate of the heating capacity of the boiler determined. However, for 13 boilers the heating capacity was known, and an attempt has been made to correlate the noise output with heating capacity. This attempt has been fruitless. Figure B-1 gives the 3-ft noise levels (usually in front of the boiler where combustion and blower noise are a maximum) for 8 boilers in the size range of 50-300 BHP (boiler HP), while Figure B-2 gives the noise levels of 5 boilers in the size range of 600-2000 BHP. There is no clear trend of noise vs boiler rating. In Figure B-1, boilers 4 and 5 are almost equal in size (180 and 200 BHP) while their noise levels represent the largest differences shown. Similarly, in Figure B-2, boilers 2, 3 and 4 are nearly equal in size (1000, 1100 and 1200 BHP), yet they show the largest noise level differences. In one octave band or another, all 12 boilers rated in the 50-1200 BHP range exceed the noise of the largest boiler rated at 2000 BHP; while in one octave band or another the smallest boiler rated at 50 BHP exceeds the noise of 10 larger boilers in the range of 125-2000 BHP.

Thus, it appears that heating capacity alone is not a very significant factor in determining boiler noise. This is not an unreasonable conclusion when one sees the wide variety of blower assemblies, burners and combustion chambers found on the various boilers included in this accumulation of data. There was insufficient knowledge of whether a given boiler was a water-tube type or a fire-tube type, so it was not possible to check this as a noise-influencing parameter. As the names suggest, in the water-tube boiler, combustion takes place in the space all around the water-filled tubes; and in the fire-tube boiler, combustion gases are fed through the tubes that penetrate the water-filled tank.

Heating capacity is given in at least four different ways and it is desirable to be able to interrelate these. For the current study, all ratings have been reduced to equivalent boiler horsepower, designated "BHP". The four rating terms are:

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

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

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

From Figures B-1 and B-2 it appears that there is no predictable relationship between noise and heating capacity, so one must be content to use a noise curve that will provide reasonably good protection for all possible boilers. Figures B-3 and B-4 give noise levels of two additional groups of boilers, mostly unidentified as to details. Figure B-3 represents a group of noise levels of boilers collected over the years by Robert M. Hoover of BBN, and Figure B-4 represents a group of noise levels reported by D. Kibblewhite* in 1967. The Hoover data generally represent approximately 3 ft distances. The Kibblewhite levels cover a total range of approximately 18-300 BHP and are unspecified as to measurement distance. The data are merely listed as being associated with "boiler-houses". On this basis it has been assumed that these might generally represent reverberant field levels, so the published values have been increased by 3 dB to bring them to the common 3 ft distances used here.

The solid curve shown on each of the four figures represents the boiler noise curve proposed for use in the Manual. On the four figures there are a total of five boilers that exceed the noise curve in the 31, 63 or 125 Hz octave bands. These exceedances range up to 14 dB above the noise curve. It does not seem reasonable to raise the noise curve any farther in order to enclose these points since there is ample evidence that many quieter boilers exist. Thus, it is suggested that the noise curve be used both for the design of the boiler room and as a specification of maximum sound pressure levels for boilers. A few noise exceedances exist in the octave bands above 125 Hz but these are not too significant since the low frequency values will almost entirely control the acoustic design of the room.

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* "Noise Levels in Boiler Houses and Plant Rooms," D. KIBBLEWHITE, JIHVE (the British Journal of the Institution of Heating and Ventilating Engineers), June 1967.

TABLE B-1

ESTIMATED SOUND PRESSURE LEVELS OF BOILERS, AT 3-FT DISTANCE *

OCTAVE FREQUENCY BAND (HZ)	SOUND PRESSURE LEVEL (dB re 0.0002 microbar)
31	92
63	92
125	92
250	89
500	86
1000	83
2000	80
4000	77
8000	74

* Distances should be measured from the <u>front</u> surface of the boiler.

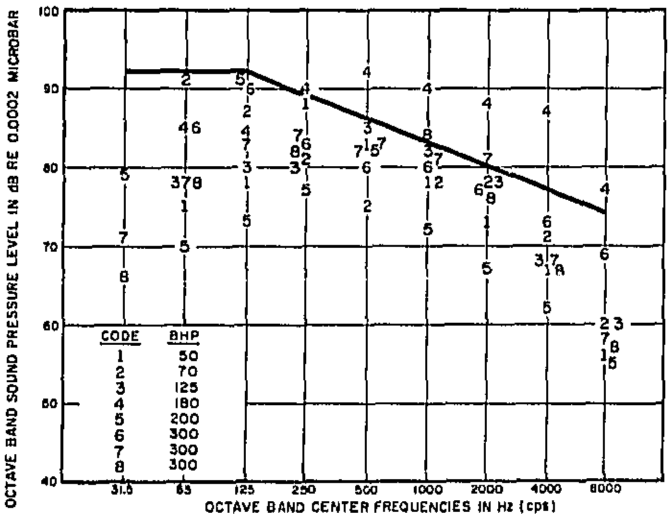

FIG. B-1 NOISE LEVELS OF BOILERS AT 3 FT DISTANCE

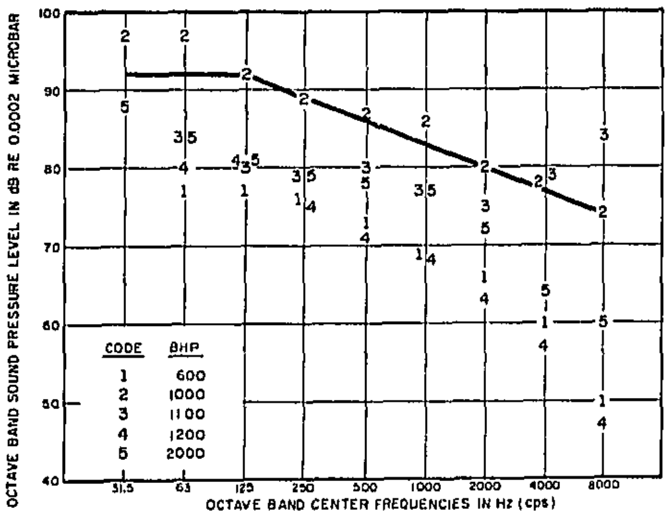

FIG. B-2 NOISE LEVELS OF BOILERS AT 3-ft DISTANCE

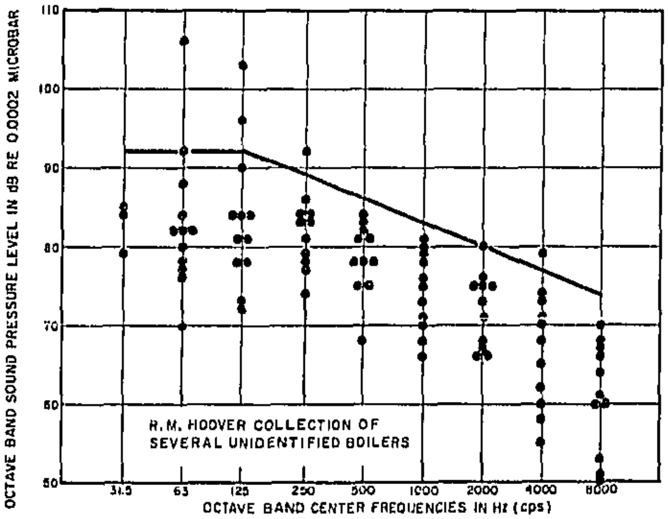

FIG. B-3 NOISE LEVELS OF BOILERS AT 3-FT DISTANCE

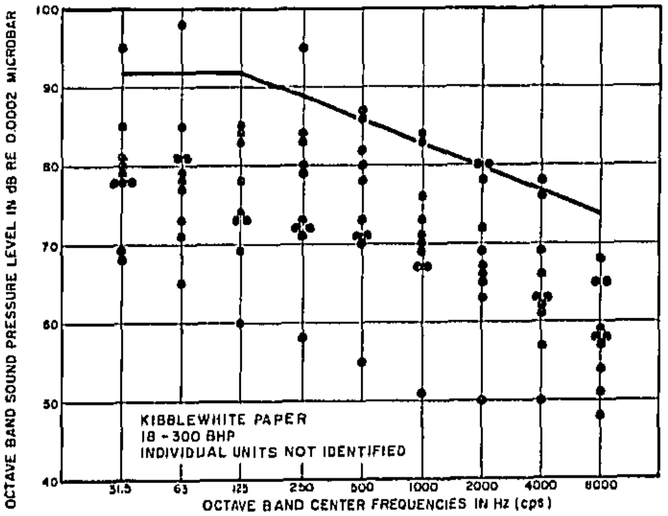

FIG. B-4 NOISE LEVELS OF BOILERS AT 3-FT DISTANCE

The noise levels shown by the solid line in the four figures are given in Table B-1. These noise levels should be associated with the front face of the boiler, where most of the noise occurs.

SECTION 8. STEAM VALVES

Three examples of steam valve noise have been measured. All three are plotted in Figure V-1. In one room of a large building, the steam line and valve serving that building was measured; this example is shown by the thin solid line in the figure. In another mechanical equipment area, three steam lines and valves were located together, so that one noise measurement position combined the noise of three valves. The pressure drop across the valves and the steam flow through the valves was not known, but according to the labels on the pipes, two valves separated "high pressure" from "medium pressure" and one valve separated "medium pressure" from "low pressure" steam. The noise from the valves seemed typical; the measured noise levels are shown by the dashed curve in the figure. The low frequency noise levels are attributed to other equipment in the room. The steam pipes of the above examples were covered with thermal insulation and the conventional spiral-wrapped cloth tape. In the third example, what appeared to be two valves in series were located close-together in one large steam pipe (approximately 20 in. outside diameter, including thermal insulation). The noise levels are shown by the dotted line of Figure V-1. In this example, the thermal insulation was covered with a sheet aluminum cover. The lower noise levels in the high frequency region for this third example presumably can be attributed to the use of two valves in series or to the aluminum wrapping. Either step would reduce the steam noise. Even though the noise is generated at and near the orifice of the valve, the pipes on either side of the valve radiate a large part of the total noise energy that is radiated. Hence, a good pipe wrapping (acoustic as well as thermal) is capable of reducing steam valve noise radiation.

The spectrum of the high frequency valve noise is presumed to be a function of the jet velocity and the size of the valve opening, and the intensity of the noise is assumed to be related to the total mass flow and the velocity of the steam. Because none of these dimensions is known for any of the valves measured, no attempt has been made to study the possible parameters. A very approximate calculation was carried out in accordance with the

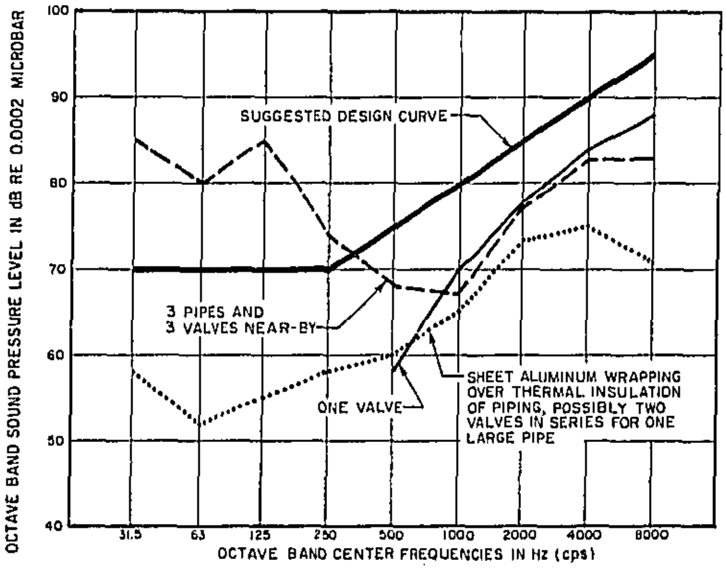

FIG. V-1 NOISE LEVEL OF STEAM VALVE NOISE AT 3-FT DISTANCE

TABLE V-1

ESTIMATED SOUND PRESSURE LEVELS OF STEAM VALVE NOISE, AT 3-FT DISTANCE

OCTAVE FREQUENCY BAND (Hz)	SOUND PRESSURE LEVEL (dB re 0.0002 microbar)
31	70
63	70
125	70
250	70
500	75
1000	80
2000	85
4000	90
8000	95

Note: Assumes simple, lightweight thermal wrapping of pipe but no metal or heavy cover around thermal wrapping. Both the valve and the connected piping radiate noise.

procedure offered by Heitner in his paper, and the agreement was at least reasonable. The calculations are not suggested, however, for an architect or mechanical engineer.

The heavy line shown in Figure V-1 is suggested as a design curve for steam valve noise. The noise level values are listed in Table V-1 and are proposed for use in the Manual. This curve assumes a normal thermal insulation on the piping, but no acoustic cover over the wrapping.

SECTION 9. COOLING TOWERS

Over the years BBN personnel have collected large quantities of noise data on cooling towers, and much of this has been reported in the literature.* Noise estimation procedures originated by BBN have been used in the ASHRAE Guide and have appeared in other publications. Tables 17 and 18 of the Power Plant Acoustics Manual present sound power level summaries of propeller-type and centrifugal-type cooling towers. These values are still considered to be generally applicable, although they may vary from manufacturer to

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* 1. "Cooling Tower Noise," Ira Dyer and Laymon N. Miller, NOISE CONTROL, May 1959.

2. "Noise Characteristics of Several Types of Cooling Tower Installations," R. M. Hoover, presented orally at the Acoustical Society Meeting, Philadelphia, May 1961.

3. "The Noise of Cooling Towers," Laymon N. Miller, Bulletin of International Institute of Refrigeration, 1962 Supplement.

4. "Acceptable Noise Levels of Cooling Towers," Laymon N. Miller, presented orally at ASHRAE Symposium, New York City, February 1963.

5. "The Noise of Cooling Towers," Engineering Bulletin No. 250 of The Baltimore Aircoil Company, prepared by BBN and BAC, March 1962 (reprinted June 1964 and to be brought up-to-date and reprinted in 1970).

manufacturer and from model to model as specific design changes take place. Table 17, for propeller-type cooling towers, has been changed slightly, by adding 3 dB to all PWL values in the 63 and 125 Hz bands. The change is intended to provide for an increase in noise output at the blade passage frequency. Since the blade passage frequency may not be known at the time of the preliminary design, this noise increase is provided in both the low frequency octave bands that might contain the propeller blade passage frequency. Table 18 and the modified Table 17 are reproduced here as Tables CT-1 and CT-2. The <u>average</u> outdoor sound pressure levels at any distance from an unobstructed cooling tower can then be determined by applying the distance term from Tables 46 or 47 of the Power Plant Acoustics Manual to the PWL values of Tables CT-1 and CT-2.

9-01. DIRECTIVITY EFFECT. For the Manual a procedure is offered for estimating the directional effects of the noise of some of these cooling towers in an outdoor situation. This is based on a study of fairly complete data provided by the Baltimore Aircoil Company ("BAC") on a large number of their centrifugal-fan blow-through cooling towers and their axial-flow blow-through cooling towers. The BAC noise data are obtained from free-field sound pressure level readings taken at 5-ft and 50-ft distances from the top and from each of the four sides of each tested tower. From these five sets of readings on a number of towers, generalizations have been drawn for this present study on the directivity of some cooling tower noise. It is obvious, of course, that the noise differs for different radiating surfaces of a typical tower. It is to be recognized that the generalizations drawn here from a rather detailed study of one manufacturer's data will not apply exactly to all other cooling towers, but it is believed that these generalizations are one step closer toward the useful data frequently required by the architect or engineer in laying out cooling towers and cooling tower noise control treatments in any given acoustic environment. It is still desirable to try to obtain from the manufacturer actual measured noise levels for all directions of interest, but if these data are not forthcoming, it is essential to be able to construct approximately the directional pattern of the cooling tower noise.

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

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

Table CT-3 gives the suggested corrections to be applied to the average SPLs of a cooling tower for each of the principal directions (i.e., in a direction perpendicular to each surface of the cooling tower). The directional corrections for centifugal-fan blow-through units and for axial-flow blow-through units are based on the BAC measured data. No detailed data on directionality are available on the other two types of propeller cooling towers, so judgement estimates are given for the induced-draft propeller-type (with the propeller on top of the tower) and the "underflow" forced-draft cooling tower. Directional estimates for the "underflow" cooling tower appear reasonable, although there have been no field measurements to test these estimates. They are considered more realistic, however, than the average SPL without any directional correction. In Table CT-3, "front" designates the surface that contains the air intake (when two surfaces contain air intakes, both surfaces should be treated as "fronts"), "side" designates the solid surface (next to the "front") that has no air intake, and "rear" designates the back surface (opposite to the "front") if it has no air intake opening (otherwise it would be another "front"). If it is necessary to estimate the SPL at some direction other than the principal directions, one should feel free to interpolate between the values given for the principal directions.

The directional corrections and the calculated average SPL (calculated from the PWL values and the distance effects) would not hold for very close-in distances to the cooling tower or for enclosed spaces that modify the sound pattern radiated by the tower. Calculations would be reasonably reliable for distances beyond about twice the largest dimension of the tower. Inside that distance, sound levels may differ from what the "inverse-square law" would predict, and some localized noise sources may produce high local noise levels that do not propagate to the greater distances. Thus, close-in levels are not always predictable. In this project, it is assumed that cooling towers will be used in outdoor locations. If they are located inside enclosed mechanical equipment rooms or within courts formed by several solid walls, the sound patterns will be distorted. In such instances, the PWL of the tower (or appropriate portions of the total PWL) can be placed in that setting, and the enclosed or partially enclosed space can be likened to a room having certain estimated amounts of reflecting and absorbing surfaces. Because of the limitless number of possible arrangements, this is not simply handled in a general way, so the problem of partially enclosed cooling towers will not be treated in detail in the Manual. General guidelines only will be offered.

9-02. EXAMPLES. It is of interest to check the data of Tables CT-1 through CT-3 against some actual measurements. Several comparisons are given below.

a. Centrifugal-fan Blow-through Unit with Fans on only one "Front" Surface. Assume that the fans are driven by a motor having a name-plate rating of 25 HP. According to Table CT-2, the sound power level of this cooling tower would have the following values for the nine octave bands from 31 Hz to 8000 Hz (given in dB re 10^{-12} watt):

91	92	92	90	89	87	88	82	75

Then, according to Table 46 of the Power Plant Acoustics Manual, the average SPL of this cooling tower at 50-ft distance would be (PWL-32 dB), or

59	60	60	58	57	55	56	50	43

Table CT-3 now offers approximate corrections to this average SPL in order to obtain the estimated SPL at various directions from the cooling tower. The estimated SPLs are as follows:

Front								
62	63	62	61	61	58	59	54	44
Sides								
59	60	60	56	54	51	51	45	38
Rear								
59	60	59	56	54	51	51	44	37
Top								
56	57	58	58	58	57	59	54	48

The measured and published SPLs at 50 ft distance for one of the 25-HP BAC centrifugal cooling towers are listed as follows for these same directions:

Front								
62	62	63	60	61	54	57	51	41
Sides								
58	58	60	56	52	48	46	41	34
Rear								
58	58	59	55	53	49	46	41	34
Top								
56	56	58	58	57	55	56	51	45

b. Centrifugal-fan Blow-through Unit with Fans on only one "Front" Surface. Assume that the fans are driven by a motor having a name-plate of 10 HP. According to Table CT-2, the PWL values are 3 dB below those used immediately above. Therefore, the SPLs will also be 3 dB below the estimated SPLs above:

Front								
59	60	59	58	58	55	56	51	41
Sides								
56	57	57	53	51	48	48	42	35
Rear								
56	57	56	53	51	48	48	41	34
Top								
53	54	55	55	55	54	56	51	45

The published data for an earlier 1960-1962 version of a BAC cooling tower of this type are as follows:

Front								
58	58	61	60	58	51	53	50	40
Sides								
57	57	59	54	50	46	44	42	37
Rear								
55	55	57	53	49	45	43	40	34
Top								
53	53	57	54	55	53	54	53	48

Published data for a more recent BAC 1966-1968 version of a similar tower are as follows:

Front								
61	61	61	60	58	54	53	50	41
Sides								
58	58	61	55	51	48	46	41	36
Rear								
59	59	60	54	52	48	46	39	35
Top								
55	55	59	58	55	53	52	47	44

The two different versions of the same size cooling tower are included to show the applicability of the estimation procedure over at least a small range of tower variations as produced by one manufacturer. All BAC data used here were picked at random; no attempt was made to select towers to give the best agreement with the estimated levels. Admittedly, however, the estimation data were derived from a large quantity of BAC measurements in the first place. We have no Marley data for a similar comparison, but it is recalled from an earlier job that noise levels for a particular Marley centrifugal-fan cooling tower were in very close agreement with the levels for an equivalent BAC unit.

c. Centrifugal-fan Blow-through Unit with Fans on "Front" and "Rear" Surfaces (i.e., two "Front" Surfaces). In the examples given above, the fans are located on only one side of the tower. In the following example, fans are located in both the front and rear sides of the tower. In this example, a 120-HP tower is assumed (in this modular arrangement, the fans are driven by six 20-HP motors). From Table CT-2 and Table 46 of the Power Plant Acoustics Manual, the average SPLs of this tower at 50-ft distances are estimated to be:

65	66	66	64	63	61	62	56	49

The corrected SPLs in the various directions become, using Table CT-3:

Front and Rear								
68	69	68	67	67	64	65	60	50
Sides								
65	66	66	62	60	57	57	51	45
Top								
62	63	64	64	64	63	65	60	54

Actual measured SPLs for a 120-HP BAC centrifugal cooling tower are as follows, for the 50-ft distance:

Front and Rear								
67	67	68	65	65	58	61	54	44
Sides								
61	61	63	59	55	51	50	44	37
Top								
63	63	65	66	65	63	64	59	52

d. <u>Axial-flow Blow-through Unit with Fans on only one "Front" Surface</u>. In this example, assume a 15-HP propeller-type fan. Using Tables CT-1 and CT-3 and Table 46 of the Power Plant Acoustics Manual, the following SPL estimates are made for a 50-ft distance:

Front								
69	74	76	73	70	65	62	59	54
Sides								
68	73	73	65	59	55	52	49	45
Rear								
64	69	68	60	57	53	49	43	41
Top								
62	67	67	62	62	60	57	56	50

Measured levels from a BAC 1966-1968 unit of this size and type are as follows:

Front								
--	72	76	75	72	63	61	55	50
Sides								
--	70	72	66	60	51	50	43	41

Rear								
--	66	67	61	57	49	46	39	37
Top								
--	65	67	64	63	57	54	52	46

e. Induced-draft Propeller-type Units. The examples that follow are based on actual field problems, and in no cases are there detailed SPLs for all directions. These examples show the degree of general agreement between the estimated levels and actual measured levels.

(1) 30-HP unit at 325-ft distance:

Measured SPLs facing front of unit

--	58	53	53	53	46	45	40	30

Estimated SPLs for front of unit

54	59	59	55	53	49	45	42	34

Estimated SPL higher than measured SPL by

--	1	6	2	0	3	0	2	4

(2) 60-HP unit at 400-ft distance:

Measured SPLs facing 45° to front

--	62	58	53	48	47	46	37	25

Estimated SPLs for 45° to front

54	59	59	54	51	46	43	38	30

Estimated SPL higher than measured SPL by

--	-3	1	1	3	-1	-3	1	5

(3) Three 10-HP units at 50-ft distance:

Measured SPLs 45° to front

--	73	68	66	63	60	62	60	54

Estimated SPLs for 45° to front

69	74	74	69	66	62	59	56	51

Estimated SPL higher than measured SPL by

--	1	6	3	3	2	-3	-4	-3

(4) Two 20-HP units on roof deck 40 ft below:

Measured SPLs at window sill 5 floors above tower base

(*Blade passage in this band)

-- 75 76 80* 76 67 60 50 40

Estimated SPLs 40 ft above top

78 83 83 78 74 70 67 63 58

Estimated SPL higher than measured SPL by

-- 8 7 -2 -2 3 7 13 18

(5) Two 20-HP units at 5-ft distance:

Measured SPLs facing front of units

-- 89 91 85 87 82 76 68 63

Estimated SPLs for front of unit

(Assume 5 ft from front equals approximately 15 ft from geometric center of total noise. Caution: estimated close-in SPL values not expected to be reliable.)

83 88 88 84 82 78 75 73 68

Estimated SPL higher than measured SPL by

-- -1 -3 -1 -5 -4 -1 +5 +5

(6) Two 25-HP units at 100 ft distance:

Marley data, facing front of units

-- 73 69 68 63 58 48 37 26

Estimated SPLs for front of unit

67 72 72 68 66 62 59 57 52

Estimated SPL higher than Marley SPL by

-- -1 3 0 3 4 11 20 26

(7) Two 20-HP units at 100 ft distance:

Marley data, facing front of units

-- 73 64 60 55 50 43 31 24

Estimated SPLs for front of unit

67 72 72 68 66 62 59 57 52

Estimated SPL higher than Marley SPL by

-- -1 8 8 11 12 16 26 28

(8) Ten 100-HP industrial units at 200-ft distance:

Measured SPLs, facing front of unit

--	79	73	67	65	62	60	59	56

Estimated SPLs for front of unit

73	78	78	74	72	68	64	61	55

Estimated SPL higher than measured SPL by

--	-1	5	7	7	6	4	2	-1

(9) Two 20-HP units at 200-ft distance:

Measured SPLs, orientation unknown (*Blade passage in this band.)

--	72	61	70*	63	54	46	39	--

Estimated average SPL, since orientation is unknown

61	66	66	61	58	54	50	46	40

Estimated SPL higher than measured SPL by

--	-6	5	-9	-5	0	4	7	--

When taken at random, the above examples do not seem to show impressive agreement between measured and estimated data. Many of these field installations, however, do not provide ideal geometrical conditions for measuring the true free-field radiation from a cooling tower, and the various manufacturers insert their own design variations which influence noise production and noise radiation. It is not within the scope of this work to attempt to explore and justify all these factors, although many of them have known causes for not fitting the general pattern. The noise data of Table CT-1 generally enjoys the confidence of many manufacturers and engineers and these data afford a reasonable and useful estimate of the noise output of the many varied propeller-type cooling towers.

9-03. CLOSE-IN NOISE LEVELS. The noise data given in the preceding discussion are most useful in estimating the noise levels of cooling towers as heard at some distance away. Although the sound power level data can be used to estimate approximately the close-in noise levels, in this study considerable close-in data have been collected. These close-in noise levels are used mostly for determining the type of wall or floor required to separate the cooling towers from quiet parts of the building. The accompanying Figures CT-2 through CT-8 summarize all the data collected in this study.

Figures CT-2 presents the noise levels measured at 3-ft to 5-ft distances from the fan discharge of propeller-type induced-draft cooling towers

(Type C in Figure CT-1) for a range of 15 to 75 HP. Because of the wide variety of designs (different speeds, diameters, blade designs, number of blades, tip speeds, clearances to the shroud, and manufacturers), there is no simple obvious relationship between noise and certain noise-influencing parameters, not even HP rating of the fan. The "Suggested Design Curve" would appear to give a reasonably safe working SPL for the close-in positions. The peak at 63 Hz reflects the energy peak at the blade passage frequency that usually occurs in the 63 or 125 Hz band for the large fan blades (typically ranging 6 ft to 20 ft in diameter). The black dots on Figure CT-2 represent one example of an "underflow" tower (Type D on Figure CT-1). Since there are no other data for this type of tower, the fan intake position of this tower is treated as being somewhat comparable to the fan outlet position of all the other towers shown on Figure CT-2. A noticeable difference is the absence of a strong peak at one of the low frequency bands. The triangular point on this plot represents data from an odd problem that included a strong pure tone signal from one particular unit. The pure tone frequency was the third harmonic of the blade passage frequency. An array of stator blades may have helped cause the unusual signal.

Figure CT-3 summarizes the close-in noise levels of the air intake to the propeller-type induced-draft towers. Seven conventional cooling towers, ranging in size from 3 HP to 60 HP, are shown along with the special problem unit from Figure CT-2 and one unit of a group of six 125-HP industrial towers. The cause of the unusually high noise levels of the 125-HP unit is not known, but this tower is excluded from consideration as a representative tower for office building applications. The "Suggested Design Curve" is drawn along the upper range of the remaining eight towers. Note that the low frequency peak is not as pronounced as at the discharge. The two Design Curves are compared in Figure CT-4. These levels are suggested for use in the Manual when known noise levels are not available from the manufacturer.

Figures CT-5 and CT-6 summarize the collected data for the intake and discharge openings of the axial-flow blow-through cooling tower. Most of the data are taken from BAC towers, which are mostly made up of small modules assembled in various configurations. Because the fans are usually not very large, it is expected that some larger towers with larger fans would produce higher noise levels. Hence, in Figure CT-5 the Suggested Design Curve is drawn much higher than the range of BAC data and is given a slight peak at 63 Hz. The noise levels from the "underflow" tower of Figure CT-2 are also shown here because the fan of the "underflow" tower is functionally similar to the fans of these blow-through towers (the principal difference is in the relative location of the fans).

Figures CT-7 and CT-8 summarize the close-in noise levels for the intake and

TABLE CT-1

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

OCTAVE FREQUENCY BAND (HZ)	4 to 8 HP	9 to 16 HP	17 to 32 HP	33 to 64 HP	65 to 128 HP	129 to 256 HP
31	96	99	102	105	108	111
63	101	104	107	110	113	116
125	101	104	107	110	113	116
250	96	99	102	105	108	111
500	93	96	99	102	105	108
1000	89	92	95	98	101	104
2000	86	89	92	95	98	101
4000	83	86	89	92	95	98
8000	78	81	84	87	90	93

TABLE CT-2

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

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

TABLE CT-3

APPROXIMATE CORRECTIONS TO AVERAGE SPLs FOR DIRECTIONAL EFFECTS OF COOLING TOWERS

(Add these decibel corrections to the average SPL calculated for a given distance from the tower. Do not apply these corrections for close-in positions, such as less than 10 to 20 ft. Also, these corrections apply when there are no reflecting or obstructing surfaces that would modify the normal radiation of sound from the tower.)

OCTAVE BAND (Hz)	31	63	125	250	500	1000	2000	4000	8000
For Centrifugal-fan blow-through type									
Front	+3	+3	+2	+3	+4	+3	+3	+4	+4
Side	0	0	0	-2	-3	-4	-5	-5	-5
Rear	0	0	-1	-2	-3	-4	-5	-6	-6
Top	-3	-3	-2	0	+1	+2	+3	+4	+5
For Axial-flow blow-through type									
Front	+2	+2	+4	+6	+6	+5	+5	+5	+5
Side	+1	+1	+1	-2	-5	-5	-5	-5	-4
Rear	-3	-3	-4	-7	-7	-7	-8	-11	-8
Top	-5	-5	-5	-5	-2	0	0	+2	+1
For Induced-draft propeller-type									
Front	0	0	0	+1	+2	+2	+2	+3	+3
Side	-2	-2	-2	-3	-4	-4	-5	-6	-6
Top	+3	+3	+3	+3	+2	+2	+2	+1	+1
For "Underflow" forced-draft propeller-type									
Any side	-1	-1	-1	-2	-2	-3	-3	-4	-4
Top	+2	+2	+2	+3	+3	+4	+4	+5	+5

TABLE CT-4

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

OCTAVE FREQUENCY BAND-HZ

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

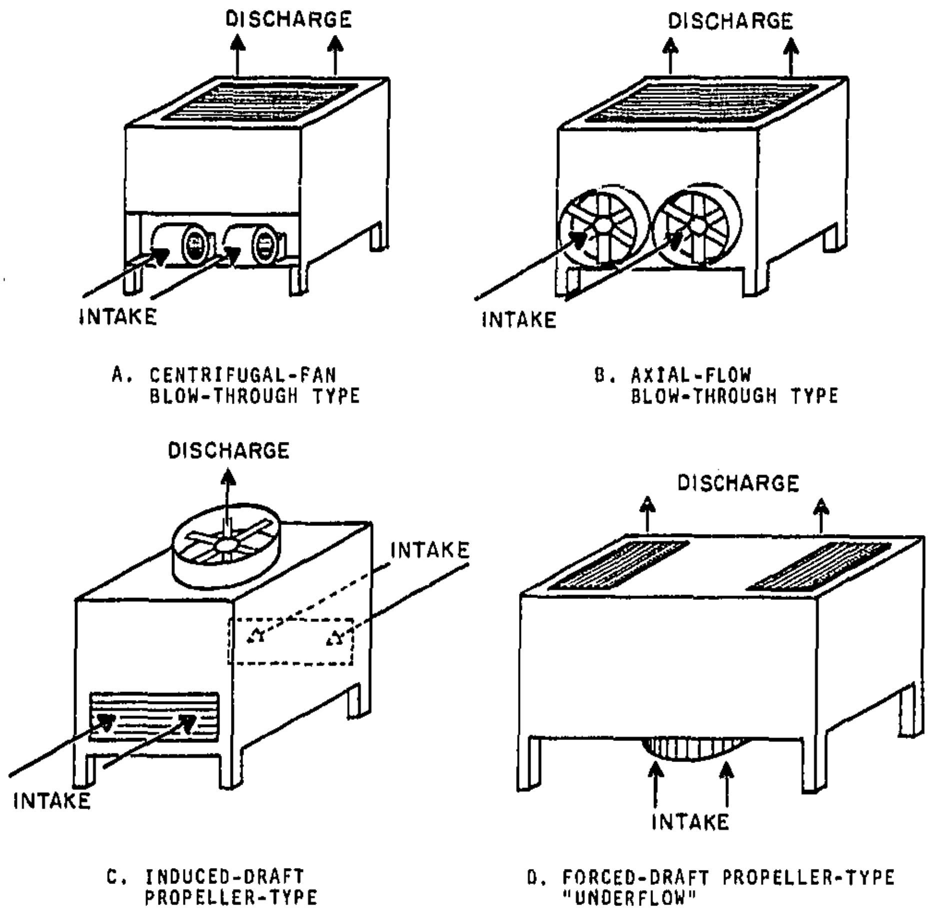

FIG.CT-1 PRINCIPAL TYPES OF COOLING TOWERS

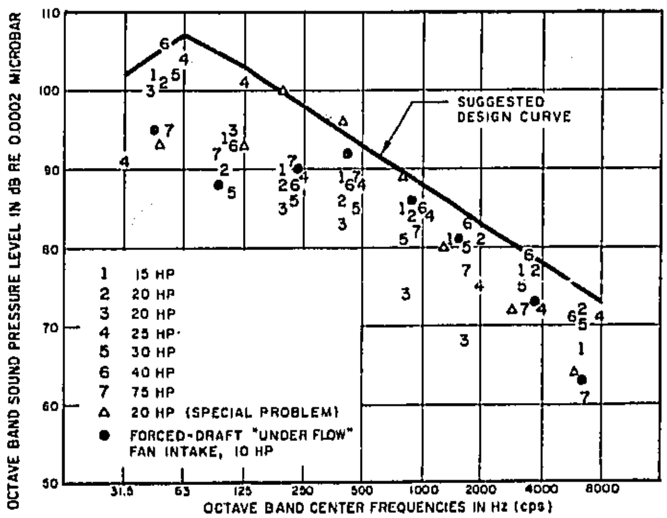

FIG.CT-2 NOISE LEVELS 3-5 FT FROM FAN DISCHARGE OF PROPELLER-TYPE INDUCED-DRAFT COOLING TOWERS

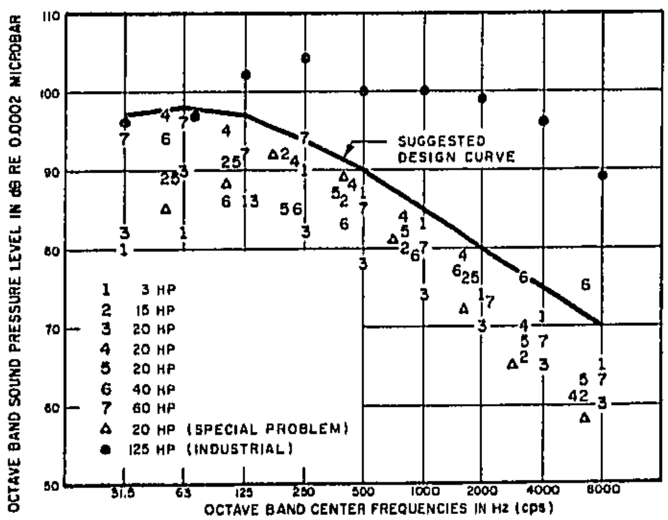

FIG.CT-3 NOISE LEVELS 3-5 FT FROM AIR INTAKE OF PROPELLER-TYPE INDUCED-DRAFT COOLING TOWERS

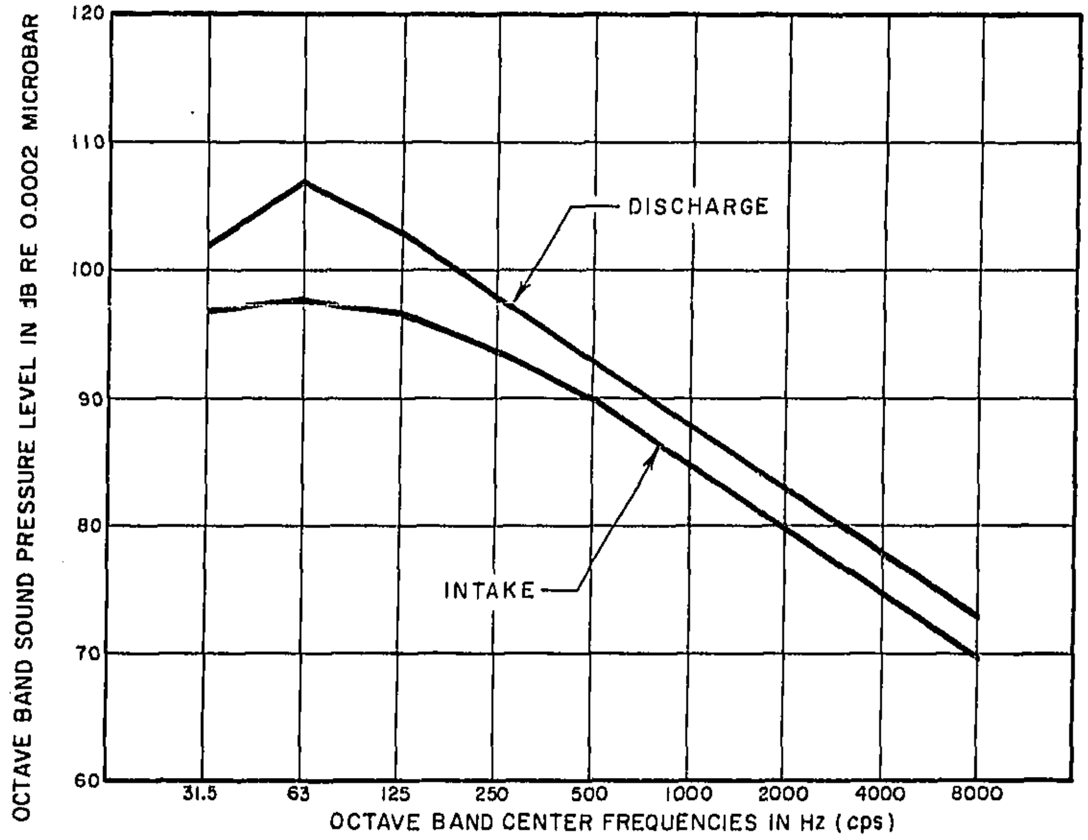

FIG.CT-4 NOISE LEVELS 3-5 FT FROM INTAKE AND DISCHARGE OPENINGS OF PROPELLER-TYPE INDUCED DRAFT COOLING TOWERS

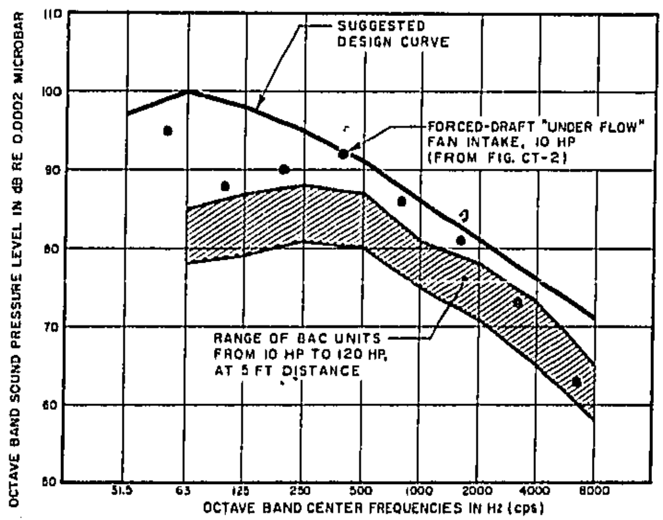

FIG. CT-5 NOISE LEVELS 3-5 FT FROM INTAKE PROPELLER FAN OF AXIAL-FLOW BLOW-THROUGH COOLING TOWER

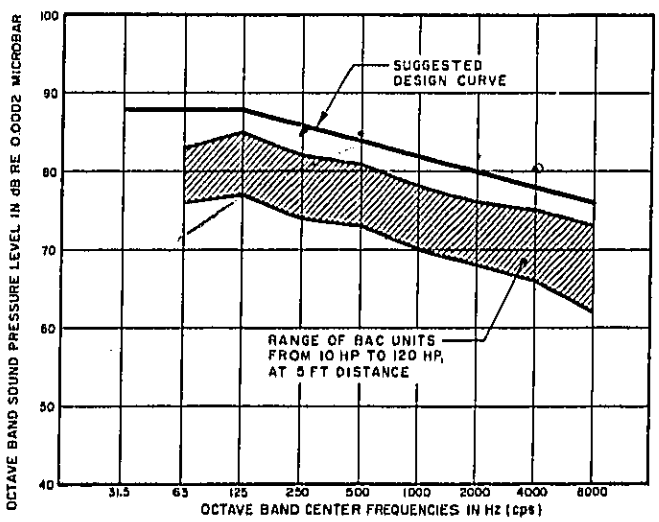

FIG. CT-6 NOISE LEVELS 3-5 FT FROM DISCHARGE OPENING OF AXIAL-FLOW BLOW-THROUGH COOLING TOWER

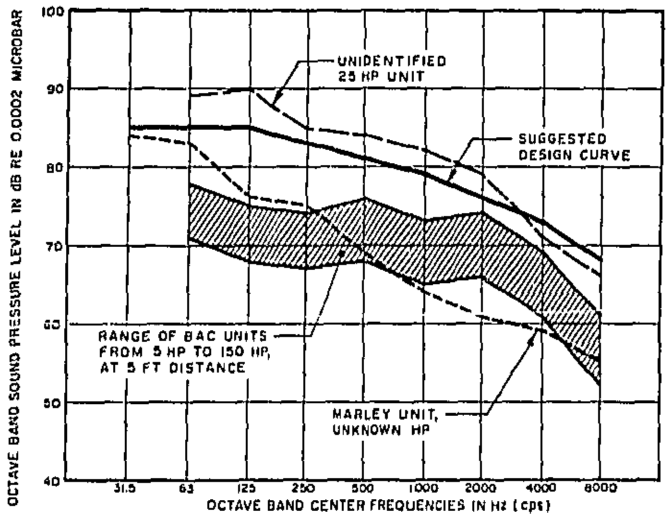

FIG.CT-7 NOISE LEVELS 3-5 FT FROM INTAKE FANS OF CENTRIFUGAL-TYPE BLOW-THROUGH COOLING TOWERS

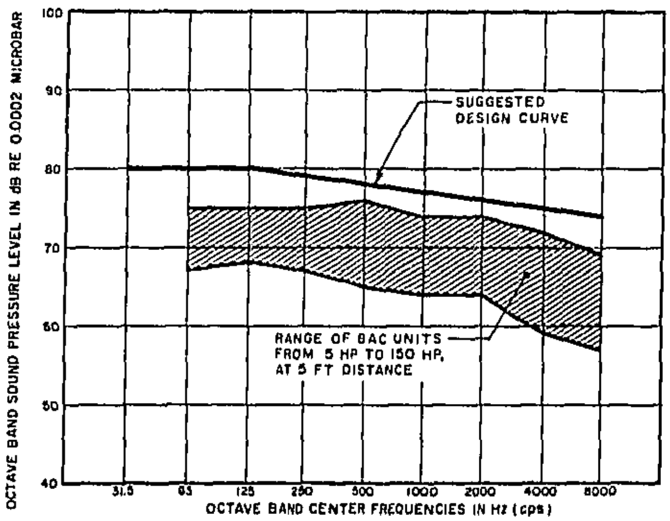

FIG.CT-8 NOISE LEVELS 3-5 FT FROM DISCHARGE OPENING OF CENTRIFUGAL-FAN BLOW-THROUGH COOLING TOWERS discharge openings of centrifugal-type blow-through cooling towers. The data are for the most part taken from BAC measurements. The Suggested Design Curves are drawn several dB above the BAC data to protect somewhat noisier towers. In the low frequency region, the intake fan positions (Figure CT-7) are noisier (due to the fans), but at the highest frequency region the splashing of the falling water produces higher noise levels at the discharge opening. This effect was also present in Figures CT-5 and CT-6.

The noise levels shown by the design curves of Figures CT-4 through CT-8 are listed in Table CT-4 and are recommended for use in the Manual.

SECTION 10. PUMPS

Noise data have been collected and studied for nineteen pumps ranging in size from 3 HP to approximately 2000 HP. All but the 2000-HP pump were used to pump hot or cold water in various typical building applications. The 2000-HP pump was used in fuel oil pipeline transmission and was located out-of-doors. This type of pump is not usually located inside occupied buildings but its noise data are included here to represent a very large size pump. Thirteen of the nineteen pumps operated in the speed range of 1680 to 1800 RPM, four pumps operated at 3500-3600 RPM, one pump operated at 1180 RPM, and one pump operated at 450 RPM. All pumps were loaded but not necessarily at full rated load. The nameplate horsepower of the drive motor or turbine has usually been used to rate the pump power.

All noise data have been normalized to the reference distance of 3 ft in an indoor situation. All known data on these pumps are given in Table P-1. Because pumps are usually located very close to their drive motors or turbines, some of the noise attributed to the pumps may actually be due to the drive unit. In most cases, however, measurement positions were selected to favor the pump noise. The data represent pumps that had both "isolated" and "unisolated" vibration mountings but it is believed that the airborne sound radiated by a pump is not influenced by its base mounting, at least not for large pumps that would be supported on thick, massive concrete floor slabs.

Figure P-1 presents the suggested "Design Curves" of noise levels vs pump size in HP. These curves are based primarily on the 1700-1800 RPM speed group, but corrections for other speeds are offered. There are not enough pumps in the 3600 RPM speed region to justify a correction at this time for this higher speed. One might assume that the higher speed would shift the frequency of the noise upward by one octave, but data from the four examples here do not show this trend. For lower speeds, the corrections shown on Figure P-1 are suggested. Actually, the shortage of the data makes these somewhat unproven corrections. There were insufficient data on fluid flow (GPM) and operating pressure (head in ft or psi) to support any study of these possible effects on noise.

As with electric motors, pump manufacturers point out that sleeve bearings help produce quieter pumps in the low power sizes. There were not enough pumps in this study, however, to include an evaluation of this possible effect.

The noise levels of Table P-1 are presented individually in Figures P-2 to P-10 where they are compared with their suggested Design Curves. All pumps are identified by their code number (see Table P-1) and HP; the speed is also given if it is not in the 1680-1800 RPM range.

The design curves of Figure P-1 overestimate the noise of most of the pumps; yet two of these nineteen pumps produce noise levels in single octave bands (probably impeller blade frequency or a harmonic of impeller blade frequency) that exceed the design curves by 7 and 10 dB. The 10 dB exceedance is associated with the 2000 HP pump. Three pumps have octave band levels that just equal the appropriate design curves in one to four octave bands, and three pumps come up to within 1 dB of their design curves in one or two octave bands. Four pumps come up to within 3 or 4 dB of their design curves.

On the basis of this sampling, the design curves would represent approximately the 80 to 90 percentile noise curves. In the 125-1000 Hz bands, the 75 percentile curve would be about 3 to 6 dB below the design curves and the 50 percentile curve would be about 6 to 10 dB below the design curves. It is cautioned, however, based on the data, that occasional pumps may exceed the design curves by large amounts, i.e., 5 to 10 dB.

Heitner offers in his paper a formula for estimating overall pump noise at a 3-ft distance:

$$SPL = 71 + 10 \log HP \quad (1 - E/2)$$

where HP is hydraulic horsepower and E is pump efficiency. For a pump efficiency of 60% (a reasonably good efficiency for overall operation) and assuming hydraulic horsepower is very nearly the same as the horsepower of the drive unit, the Heitner estimate has been made and is shown on each figure for the largest pump in each grouping. Heitner makes no

TABLE P-1

MEASURED SOUND PRESSURE LEVELS OF

PUMPS NORMALIZED TO 3-FT DISTANCE

(All centrifugal pumps pumping fluid;
may include some motor noise.
*indicates assumed values)

CODE	APPROX. RATED HP	APPROX. RPM	FLOW GPM	HEAD FT.	MANUFACTURER	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
P-1	7.5	1750*				-	76	60	62	66	70	76	61	53
P-2	10	1750*	175	220	Synchroflo	-	59	58	65	68	66	70	63	60
P-3	25	1750			Taco	-	71	60	69	63	64	68	64	54
P-4	30	1750			Wilson Snyder	83	80	76	79	82	86	79	75	66
P-5	40	1750*	600	220		-	70	72	95	80	78	76	67	66
P-6	50*	1750*				79	81	86	84	82	82	82	70	65
P-7	75	1750*			Weil	76	76	84	85	91	89	86	83	78
P-8	75	1750*			Weil	72	71	82	83	83	85	78	74	66
P-9	80	1700	1500		Hi-Temp	78	86	80	81	84	87	85	81	76
P-10	100	1770				-	84	81	86	85	87	88	85	79
P-11	150	1775	4500		Worthington	81	82	84	84	84	82	79	73	74
P-12	250	1750			Allis-Chalmers	84	85	85	85	80	87	82	75	67
P-13	2000*	1790			Bingham	87	88	88	89	99	108	95	83	76
P-14	400	450	24,000		De Laval	77	84	86	84	83	81	79	72	64
P-15	200*	1180	5100		Ingersoll-Rand	75	83	85	85	85	84	81	75	58
P-16	3	3500	90	41	Allis-Chalmers	61	57	54	64	64	73	69	68	63
P-17	15	3520			Worthington	76	75	75	76	79	74	70	63	61
P-18	500*	3600*	14,000*		De Laval	76	86	90	81	82	81	80	73	65
P-19	800*	3600*	21,000*		De Laval	77	84	85	81	82	82	79	72	65

TABLE P-2

ESTIMATED SOUND PRESSURE LEVELS OF PUMPS (AT 3-FT DISTANCE INDOORS) AS A FUNCTION OF POWER AND SPEED
(Note: Noise levels may include some noise from drive motors or turbines)

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

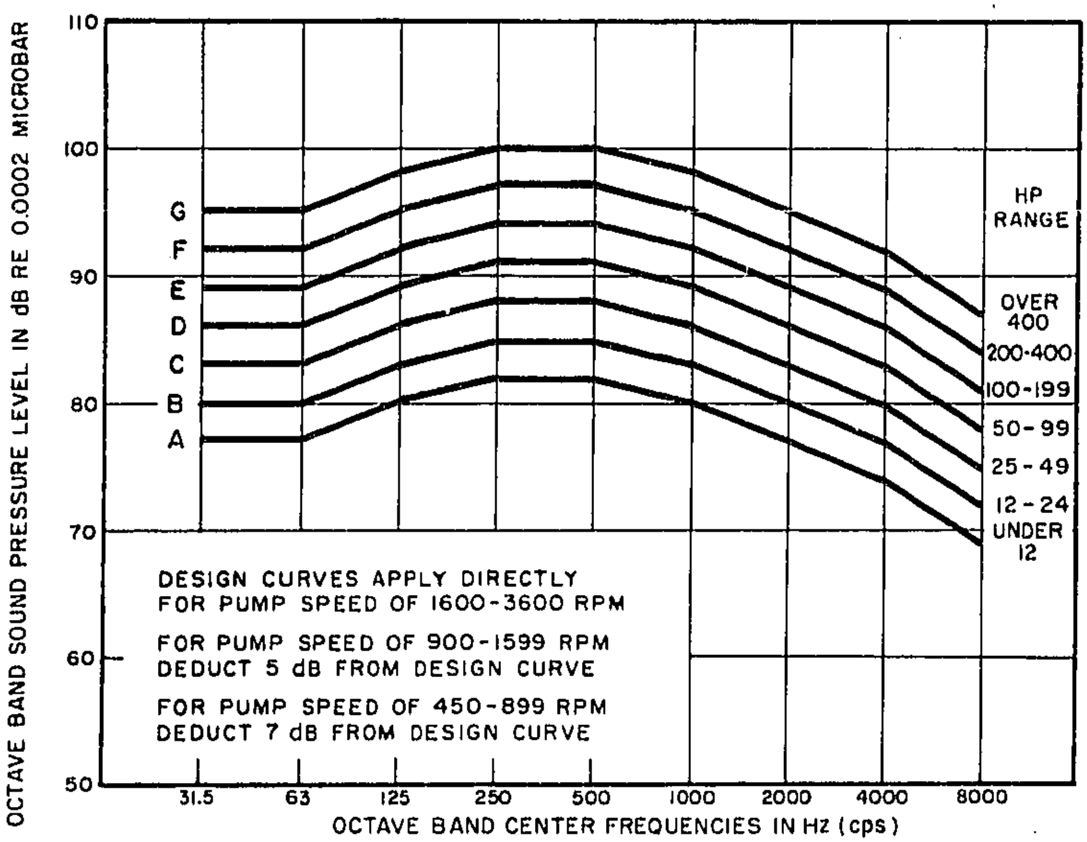

FIG.P-1 PUMP NOISE LEVELS AT 3-ft DISTANCE
(SUGGESTED DESIGN CURVES FOR VARIOUS POWERS AND SPEEDS)

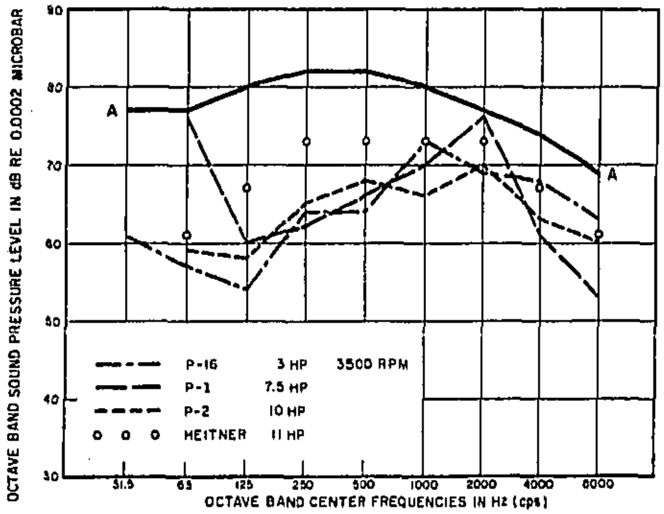

FIG. P-2 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve A compared with noise data for pumps under 12 HP and in 1600-3600 RPM range)

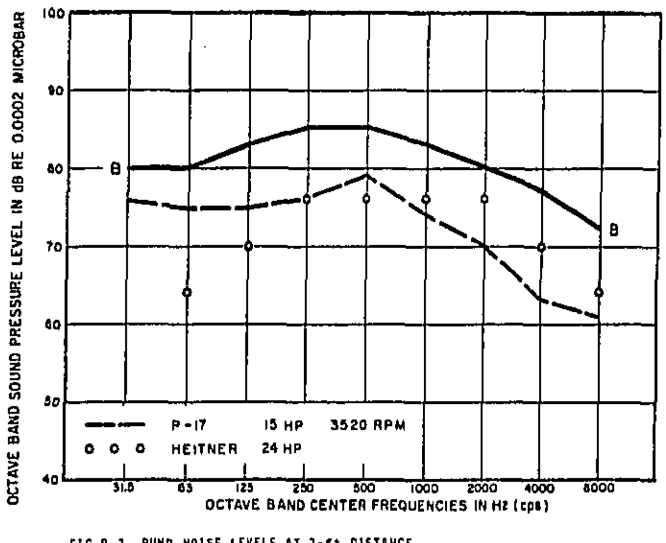

FIG. P-3 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve B compared with noise data for pumps in 12-24 HP and 1600-3600 RPM range)

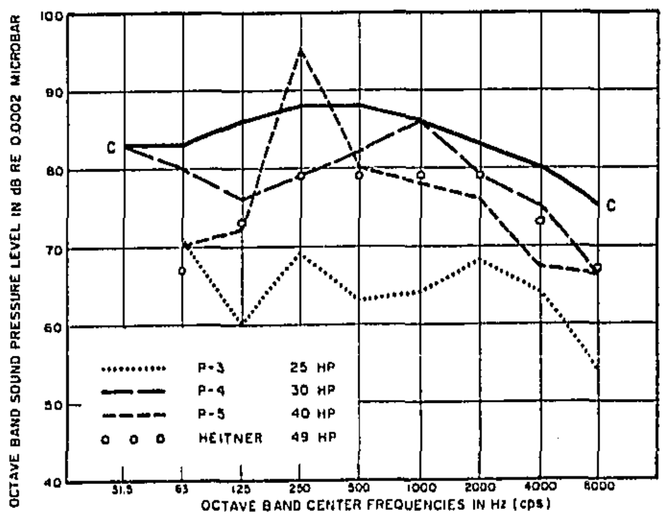

FIG.P-4 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve C compared with noise data for pumps in 25-49 HP and 1600-3600 RPM range)

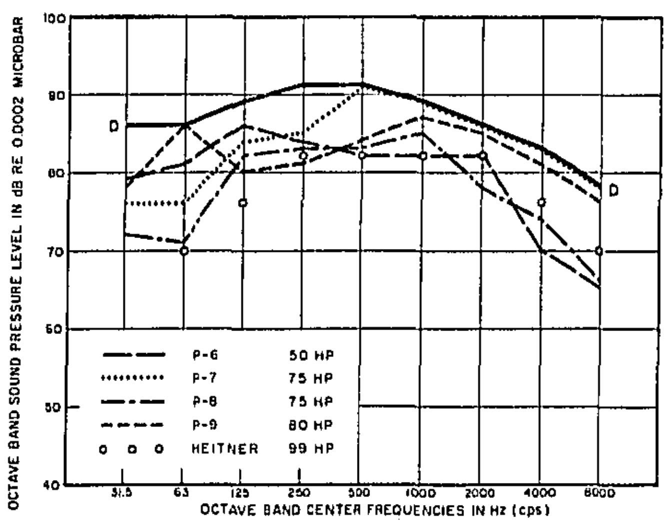

FIG.P-5 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve D compared with noise data for pumps in 50-99 HP and 1600-3600 RPM range)

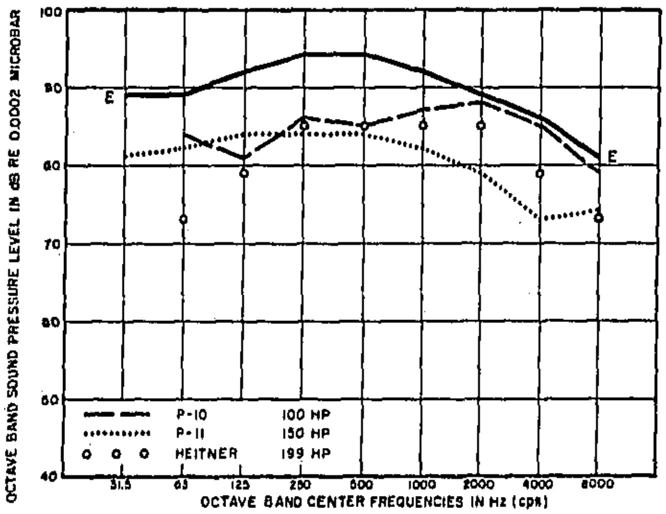

FIG.P-6 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve E compared with noise data for pumps in 100-199 HP and 1600-3600 RPM range)

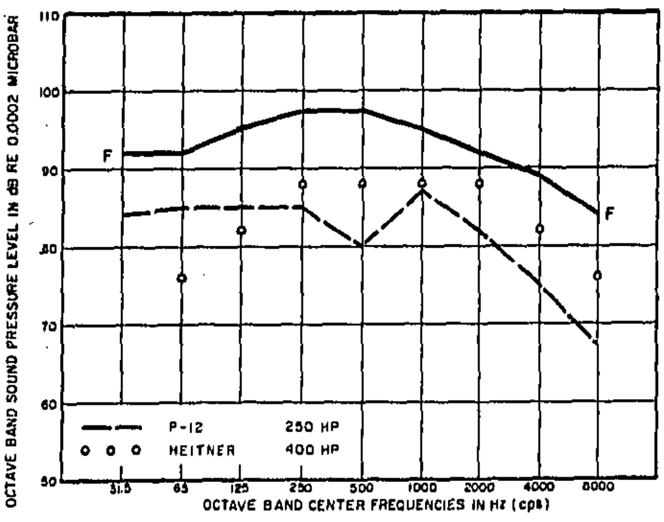

FIG.P-7 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve F compared with noise data for pump in 200-400 HP and 1600-3600 RPM range)

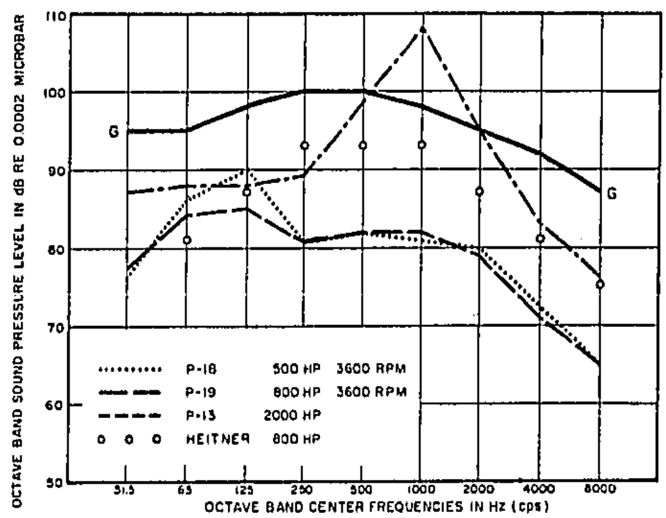

FIG.P-8 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve G compared with noise data for pumps over 400 HP and in 1600-3600 RPM range)

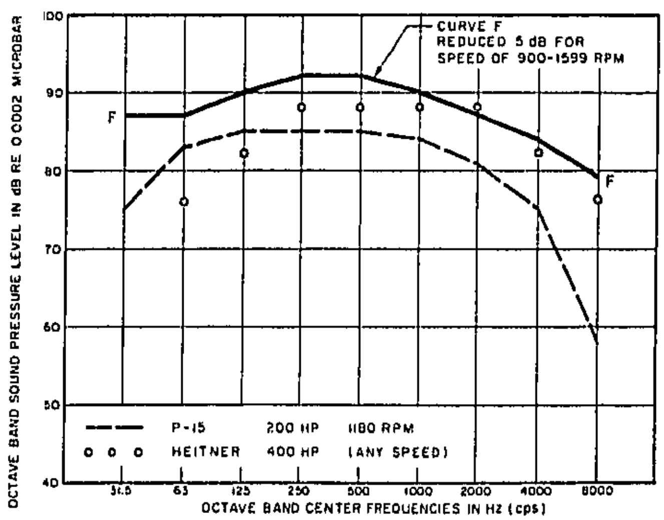

FIG.P-9 PUMP NOISE LEVELS AT 3-ft DISTANCE
(Design curve F compared with noise data for pumps in 200-400 HP and 900-1599 RPM range)

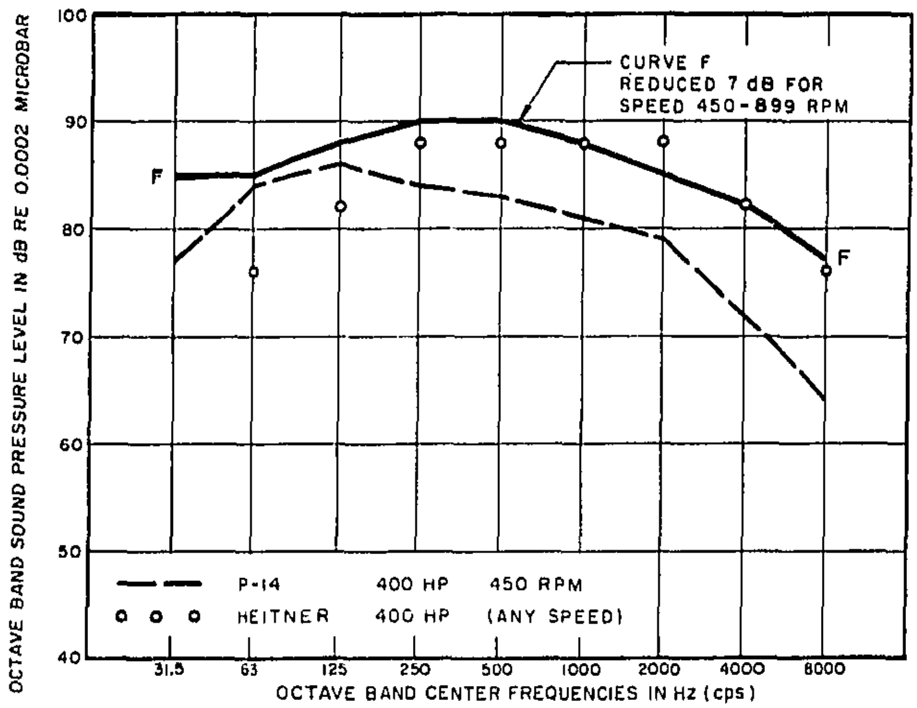

FIG.P-10 PUMP NOISE LEVELS AT 3-Ft DISTANCE
(Design curve F compared with noise data for pumps in 200-400 HP and 450-899 RPM range)

provision for speed differences. Because both the Heitner estimate and the suggested Design Curves follow a 10 log HP relationship, the Heitner curves and the Design Curves bear the same relationship to one another for pumps of 12 - 400 HP and for speeds of 1600 - 3600 RPM. Within this power and speed range, the curves are 4 dB apart at their point of closest approach in the 2000 Hz band. Outside this power and speed range, the curves may move differently with respect to each other. The Heitner estimate assumes no noise change for differences in pump speed. The noise data show a speed effect for lower speeds, hence the Design Curves in Figures P-9 and P-10 drop below the Heitner estimate by a few decibles in one octave band.

The Heitner method does not appear to recognize the low frequency noise radiated by a pump. Almost all the pumps measured have low frequency noise levels considerably higher than the Heitner estimate would predict. Admittedly, some (but not all) of the measured low frequency noise may be contributed by other sources in the pump rooms.

In summary, the Heitner estimates are useful for comparison purposes, but it is believed the proposed Design Curves provide more realistic noise estimates. Thus, the Design Curves are suggested for use in the Manual, and the noise levels from those curves are reproduced in tabular form in Table P-2.

SECTION 11. AIR COMPRESSORS

Two types of air compressors are frequently used in buildings: one is a relatively small compressor (usually only 1 or 2 HP) used to provide a high pressure air supply for operating the controls of the ventilation system, and the other is a medium size compressor (possibly up to 100 HP) used to provide "shop air" to maintenance shops, machine shops or some laboratory spaces, or to provide ventilation system control pressure for large buildings. Larger compressors are used for special industrial processes or special facilities, but these are not considered within the scope of this study.

The noise of four small and five medium-size air compressors have been measured or collected for this project. The noise levels of four reciprocating compressors in the 1-2 HP size range are summarized in Figure AC-2, while the noise levels of two centrifugal and three reciprocating compressors in the 10-75 HP size range are given in Figure AC-3.

The small sample-number of compressors does not justify any detailed data study. It is seen that the noise levels range widely for the relatively small size differences for some of the units. It is considered adequate to draw noise design curves that form envelopes over the top of the various compressor curves. The heavy curves shown at the top of Figures AC-2 and AC-3 are two of the noise design curves arrived at in this manner. The suggested family of three design curves for air compressor noise is shown in Figure AC-1. These noise levels are listed in Table AC-1 and are proposed for use in the Manual.

The one- and two-cylinder low-speed reciprocating compressors are capable

TABLE AC-1

ESTIMATED SOUND PRESSURE LEVELS OF AIR COMPRESSORS, AT 3-FT DISTANCE

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

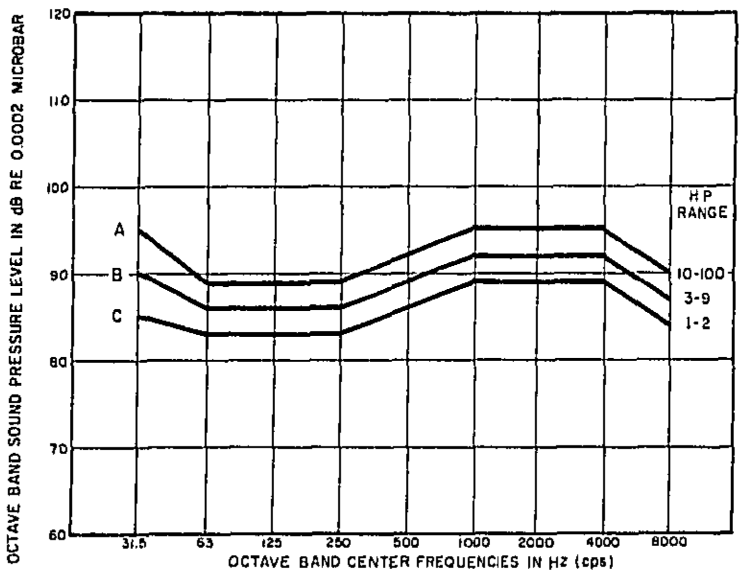

FIG.AC-1 NOISE LEVELS 3 FT FROM AIR COMPRESSORS
(Suggested design curves for either reciprocating or centrifugal type in power range of 1-100 HP)

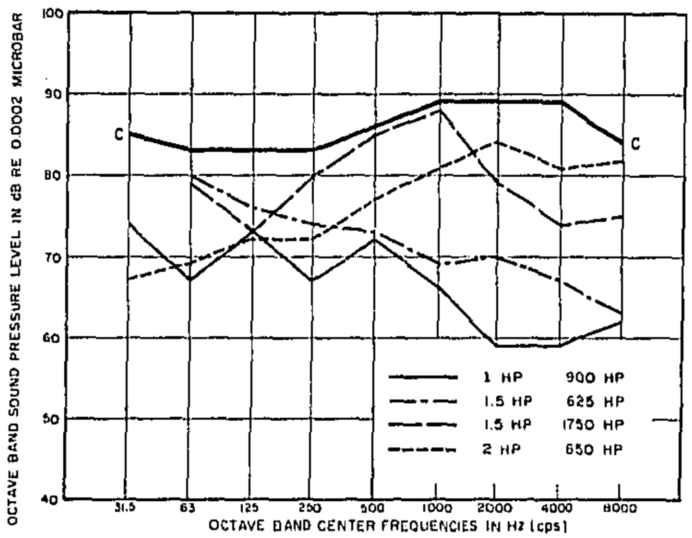

FIG.AC-2 NOISE LEVELS 3 FT FROM AIR COMPRESSORS
(1-2 HP reciprocating air compressors)

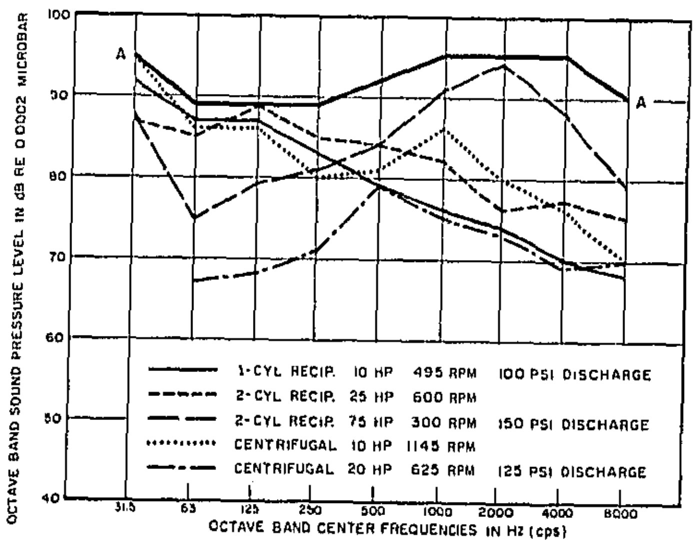

FIG.AC-3 NOISE LEVELS 3 FT FROM AIR COMPRESSORS
(10-75 HP reciprocating and centrifugal)

of producing considerable vibration in a building if they are not properly vibration isolated. Proven vibration isolation mountings will be described in the Manual.

SECTION 12. ELECTRIC MOTORS

A large supply of noise data (94 entries) for electric motors has been accumulated for this study. Surveys of recent BBN jobs and earlier BBN literature have produced noise levels for 26 motors, and material supplied by motor and pump manufacturers have provided noise data for at least 32 motors. In addition, a table of motor noise levels given in the IEEE Publication No. 85* lists probable maximum octave band sound power levels for a range of sizes from 1 to 150 HP for 1800 and 3600 RPM speed and for both "drip-proof" and "totally-enclosed fan-cooled" configurations. In his paper, Heitner also lists "a condensation of one manufacturer's noise levels". An article** on "Electric-Motor Noise" gives a general but helpful discussion of sources of motor noise and a description of the noise-influencing differences between various types of motors.

Because of the difficulties in applying full load conditions to large motors, most published noise data on motors are for no-load operation. Also, because much of the higher frequency noise of a motor is associated with the movement of air within the motor, this higher frequency noise does not change appreciably with load and probably represents full-load as well as no-load operation. The no-load test conditions, however, probably do not reflect adequately the noise of a loaded a-c motor at 60 and 120 Hz and at the harmonics of these basic drive frequencies. Bearing noise also may vary with applied loads, especially for different connections of the motor to its load (i.e., whether directly shaft-coupled or whether connected by belt-drive to a radially offset load).

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* "Test Procedure for Airborne Noise Measurements on Rotating Electric Machinery", February 1965. This table of noise data is repeated in the NEMA Publication MG1-1967 on Motors and Generators. S.F. Henderson of Westinghouse advises that both these documents are under review, but for the present he recommends "use of the values given in the NEMA publication for they do represent the composite of data from a number of companies."

** "Electric-Motor Noise", Jon Campbell (Assistant Editor) Machine Design, August 15, 1963.

Motor noise is made up of magnetic noise (most pronounced at twice the line frequency and at the rotor slot frequency), shaft unbalance noise, bearing noise, brush noise and windage (including ventilation) noise. Various mixtures of these noises occur in the three main types of motors: d-c motors, a-c synchronous motors and a-c induction motors (the latter divided into wound-rotor and squirrel-cage motors). For their low cost and rugged construction, squirrel-cage induction motors are widely used in most industrial applications.

Squirrel-cage motors generally are cooled by one of three types of ventilation. These three types are identified as (1) "drip-proof" (or "splash-proof" or "weather protected"), (2) "totally-enclosed fan-cooled" (TEFC), and (3) "totally-enclosed non-ventilated" (TENV). All three types usually contain internal fan blades in conjunction with the rotor, for circulating air around inside the motor components. In the "drip-proof" model, this circulating air is vented to the outside through openings that provide weather protection to the motor. In the TEFC model, an external fan is mounted on the rotor shaft within a protective fan housing at the end of the motor and this fan helps dissipate motor heat. The fan noise and air flow noise are generally quite high for this type motor. In the TENV model, heat is conducted from the interior to the exterior motor surfaces and radiated or cooled by natural convection.

Sleeve bearings are generally quieter than ball bearings, but sleeve bearings are usually limited to motors of the lower power range, due to lubrication limitations. For quiet operation, motors of low power requirements may be specified to have sleeve bearings.

The attached Tables M-1 to M-5 summarize all data on motors collected under this project. All sound pressure levels have been adjusted to a distance of 3 ft for an indoor location. The data identified by "IEEE" in the tables were taken from the IEEE Publication No. 85 (see earlier footnote) which actually listed sound power levels in the "old octave bands" (i.e., 75-150, 150-300, 300-600 Hz, etc.). The sound power levels (in dB re 10^{-12} watt) were converted to indoor SPLs at 3-ft distance by subtracting 7 dB (this would apply for a room having a room constant R = 500 sq ft, as may be seen from Figure 11 in "Power Plant Acoustics"). Throughout this study, the "old" and the "new" octave bands have been treated as interchangeable without any modification of sound levels. Table M-1 lists the data for "drip-proof" or "weather protected" motors in the 3500-3600 rpm range, while Table M-2 gives the data for TEFC motors ("totally enclosed fan-cooled") in the 3500-3600 rpm range. Tables M-3 and M-4 list data for drip-proof and TEFC motors in the 1700-1800 rpm range, and Table M-5 gives data for 10 motors in the 1200, 900, 600 and 450 rpm groups. In many field measurements, the type of motor or its type of ventilation was not identified. In the accompanying Tables M-1 to M-4, motors that were identified or used as "ourdoor" motors were placed under the group "drip-proof" (or "weather protected"), while those measured and used entirely indoors were placed under the TEFC group.

We especially appreciate the use of data supplied by TACO Inc. (pump manufacturer) and Wagner Electric Company on several series of motors manufactured by Wagner for use with TACO* pumps, and we wish to thank Allis-Chalmers for a set of noise data on a 1250-HP motor that may be connected and run at speeds of 900, 1200, 1800 and 3600 rpm. For all motors measured by BBN in field installations, the motors are assumed to be at or near full-power operation, and there is the possibility that some of the quoted noise may be attributable to other noise sources in the area, although the positions are selected to favor motor noise alone.

The enclosed Figure M-1 summarizes the suggested "Design Curves" for motor noise as a function of rated HP and speed of the motor. Based on the quantity of data included in this study, it would appear that the curves of Figure M-1 will "protect" about 85% to 95% of the motors to be encountered, assuming the data represent a fair sampling of motors. The remaining 5% to 10% may range 0 to 10 dB noisier in some octave bands then the Design Curves. The 75 percentile noise curve falls about 6 dB below the Design Curve; and by proper selection, some motors may range 20 to 30 dB below the Design Curves.

Although one might expect motor noise to increase at the rate of

$$10 \log HP,$$

it appears from the data that unexpected amounts of noise can appear in almost any band and that it might be due to any one of the possible causes of motor noise. This mixture of noise makes it difficult to draw a "tight" design curve over the data or to expect the 10 log HP relationship to hold consistently. No attempt has been made to separate motors into various types or to separate the effects of drip-proof, TEFC, or TENC configurations.

Figures M-2 to M-15 present visual comparisons between various Design Curves and the measured noise data appropriate to those curves. As an example, Figure M-5 illustrates a comparision of data for motors in the speed range of 3500-3600 rpm and in the power range of 50-99 HP. Where appropriate, the

* Because of their use in apartments, hotels, hospitals and other critical locations, the Wagner motors are given special concern for quietness by both Wagner and TACO. This is illustrated by the fact that the Wagner motors are much quieter than the IEEE and Heitner values for motors of the same power.

IEEE and Heitner values are shown along with the values specifically measured or obtained for this study. It is seen in Figure M-5 that Design Curve D is constructed near the higher IEEE values in the upper frequency region. Recall that the IEEE values are the probable maximum noise levels (at no-load) for the motors of several manufacturers, so it is to be expected that few of the motors measured in this study will exceed the noise levels of the IEEE values in the high frequency region. In the lower frequency bands (where magnetic noise and dynamic unbalance can influence the data), it is reasonable to expect many of the measured motors at full load to exceed the noise of the no-load IEEE and Heitner data. At least this appears to be borne out by the data. Figures M-2 to M-7 present data comparisons for motors in the speed group of 3500-3600 rpm. Figure M-7 suggests that the SPL of motors does not increase after motor power reaches about 200 HP; this may be due to the massive construction of the large motors and to the more circuitous (and possibly sound attenuating) path of the air flow through the interior of these motors. Figure M-7 includes noise data for motors covering a power range of 200-4000 HP, yet Curve F is only 3 dB above Curve E (of Figure M-6). Because of the apparent interplay of speed and power, it is not fair to judge each individual Design Curve solely by its own degree of agreement with specific bits of measured data. Rather, the entire family of Design Curves must be appraised in terms of its overall ability to cover adequately the full range of speed and power values.

Figures M-8 to M-13 present data comparisons for 1000-1800 RPM motors over the power range of this study. As seen in Figure M-1, a noise reduction of 5 dB is suggested for this speed range relative to the 3500-3600 rpm speed range. Most of the data in this speed group are for motors in the 1700-1800 RPM range, but the group was expanded to include five motors at or near 1200 RPM, since their noise seemed to fall into general agreement with the noise of the 1700-1800 RPM motors.

Figures M-14 and M-15 complete the sequence of visual comparisons; Figure M-14 presents data for two 200-HP motors in the 450-900 RPM speed group and Figure M-15 presents data for three larger motors in this same speed group. As shown in Figure M-1, a noise reduction of 9 dB is suggested for the speed range of 450-900 RPM, relative to the 3500-3600 RPM group.

It is believed that the purpose of this study is to develop noise estimating procedures and noise control designs that will cover a large percentage of possible noise sources, but it would not seem economically practical to require that noise control be designed to protect against all the noisiest sources that could exist. On this basis, the Design Curves shown in Figures M-2 to M-15 do not envelope the noisiest of the motors. For all data given,

TABLE M-1

SOUND PRESSURE LEVELS OF ELECTRIC MOTORS AT 3-FT DISTANCE (INDOORS)
DRIP-PROOF 3500-3600 RPM

CODE	APPROX. RATED HP	MFGR OR DATA SOURCE	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
M-1	1-3	IEEE	-	-	50	58	59	58	55	52	49
M-2	5-7½	IEEE	-	-	57	63	66	64	60	57	54
M-3	10-15	IEEE	-	-	63	70	72	70	66	63	59
M-4	20-25	IEEE	-	-	69	75	76	74	70	67	62
M-5	30-40	IEEE	-	-	74	79	82	79	74	71	65
M-6	50-60	IEEE	-	-	78	82	86	83	78	74	68
M-7	75-100	IEEE	-	-	79	83	88	85	81	77	71
M-8	125-150	IEEE	-	-	80	84	89	87	83	79	73
M-9	300-500	Heitner	-	83	87	90	88	85	84	80	72
M-10	600-1500	Heitner	-	89	91	90	90	88	88	89	87
M-11	1750-2500	Heitner	-	88	92	95	82	86	89	89	83
M-12	3000-4000	Heitner	-	91	94	87	83	89	92	92	86
M-13	30 #	GE *	-	76	74	79	92	88	84	71	68
M-14	50 #	GE *	-	89	89	97	89	89	89	85	75
M-15	75 #	GE *	-	73	85	92	87	84	80	69	68
M-16	100 #	GE *	-	87	93	98	105	102	96	86	79
M-17	125 #	Continental *	-	90	98	100	99	95	94	91	82
M-18	150 #	GE *	-	89	92	95	98	99	97	89	82
M-19	250 #	GE *	-	78	89	93	94	97	97	92	84
M-20	300 #	GE *	-	90	97	103	98	99	97	90	85
M-21	600	GE	-	-	91	98	95	91	88	82	69
M-22	1250	Allis-Chalmers	-	83	91	90	81	73	78	79	62
M-23	1500	Westinghouse *	91	94	94	90	88	87	94	84	73
M-24	2500	Westinghouse	-	78	93	83	86	84	82	77	72

At or near full load (all other motors assumed unloaded).

* BBN-measured in field condition.

TABLE M-2

SOUND PRESSURE LEVELS OF ELECTRIC MOTORS AT 3-FT DISTANCE (INDOORS)

		TEFC				3500-3600 RPM					
	APPROX.	MFGR OR			OCTAVE BAND CENTER FREQUENCY - HZ						
CODE	RATED HP	DATA SOURCE	31	63	125	250	500	1000	2000	4000	8000
M-25	1-3	IEEE	-	-	57	65	74	77	76	69	61
M-26	5-7½	IEEE	-	-	61	69	78	80	78	71	64
M-27	10-15	IEEE	-	-	65	73	83	85	83	76	69
M-28	20	IEEE	-	-	69	77	88	89	87	81	74
M-29	25-30	IEEE	-	-	73	81	92	93	91	85	78
M-30	40-50	IEEE	-	-	76	84	94	95	94	88	81
M-31	60	IEEE	-	-	79	87	97	97	97	91	83
M-32	75-100	IEEE	-	-	81	89	99	99	99	93	85
M-33	20-50	Heitner	-	72	72	75	84	86	82	78	70
M-34	60-100	Heitner	-	79	78	83	88	90	87	82	74
M-35	125-250	Heitner	-	82	81	87	93	93	90	86	76
M-36	1	Wagner	-	53	46	39	44	47	38	31	30
M-37	800 #	De Laval *	84	92	92	94	90	92	88	84	80

At or near full load (all other motors assumed unloaded).

* BBN-measured in field condition.

TABLE M-3

SOUND PRESSURE LEVELS OF ELECTRIC MOTORS AT 3-FT DISTANCE (INDOORS)

DRIP-PROOF 1700-1800 RPM

CODE	APPROX. RATED HP	MFGR OR DATA SOURCE	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
M-38	1-2	IEEE	-	-	44	48	48	50	51	52	44
M-39	3-5	IEEE	-	-	49	54	54	55	56	54	46
M-40	7½-10	IEEE	-	-	56	60	60	60	61	57	49
M-41	15-20	IEEE	-	-	62	65	65	65	65	60	52
M-42	25-30	IEEE	-	-	67	69	69	69	69	64	56
M-43	40-50	IEEE	-	-	71	72	72	72	72	67	59
M-44	60-75	IEEE	-	-	73	75	75	75	75	70	62
M-45	100-125	IEEE	-	-	74	77	77	77	77	72	64
M-46	300-500	Heitner	-	80	83	78	75	70	66	62	59
M-47	600-1500	Heitner	-	83	84	84	84	85	88	77	74
M-48	1750-2500	Heitner	-	88	89	85	84	85	90	87	77
M-49	3000-4000	Heitner	-	91	93	89	87	89	94	90	82
M-50	40 #	Westinghouse *	-	84	87	88	86	84	96	86	76
M-51	75 #	Westinghouse *	-	79	79	87	82	81	78	71	66
M-52	1250 #	Allis-Chalmers*	-	77	82	86	86	81	78	76	69
M-53	2500 #	Louis Allis *	80	84	85	82	86	83	90	81	73
M-54	4000	G.E.	-	-	94	95	92	92	100	99	90

At or near full load (all other motors assumed unloaded).

* BBN-measured in field condition.

TABLE M-4

SOUND PRESSURE LEVELS OF ELECTRIC MOTORS AT 3-FT DISTANCE (INDOORS)

TEFC — 1700-1800 RPM

CODE	APPROX RATED HP	MFGR OR DATA SOURCE	31	63	125	250	500	1000	2000	4000	8000
			OCTAVE BAND CENTER FREQUENCY - HZ								
M-55	1-2	IEEE	-	-	48	55	62	62	58	53	46
M-56	3-5	IEEE	-	-	55	62	69	69	65	59	51
M-57	7½-10	IEEE	-	-	61	69	75	75	71	64	57
M-58	15-20	IEEE	-	-	65	73	80	80	76	67	60
M-59	25-30	IEEE	-	-	70	78	84	84	80	71	64
M-60	40-50	IEEE	-	-	74	81	87	87	83	74	67
M-61	60	IEEE	-	-	76	82	89	89	85	77	70
M-62	75-100	IEEE	-	-	77	84	91	91	87	79	72
M-63	20-50	Heitner	-	67	63	70	75	76	70	64	58
M-64	60-100	Heitner	-	72	69	76	79	81	77	70	66
M-65	125-250	Heitner	-	74	71	79	86	85	80	75	70
M-66	10 **	Wagner	-	63	53	52	49	46	41	37	30
M-67	10	Wagner	-	58	47	52	56	46	48	39	30
M-68	10 ##	Wagner	-	63	54	52	49	46	42	37	30
N-69	15	Wagner	-	68	56	56	57	55	61	45	36
M-70	15 ##	Wagner	-	59	52	59	58	50	48	44	37
M-71	20 **	Wagner	-	66	58	59	58	52	49	44	39
M-72	20	Wagner	-	72	54	58	60	51	53	45	34
M-73	20 ##	Wagner	-	64	59	60	57	52	50	46	40
M-74	25	Wagner	-	65	61	57	59	53	52	47	39
M-75	25 ##	Wagner	-	67	60	60	59	53	52	47	39
M-76	25	Wagner	-	68	59	65	63	58	57	48	36
M-77	25	G.E.	-	64	59	66	62	60	59	42	36
M-78	30 **	Wagner	-	65	59	61	61	53	50	47	40
M-79	50 **	Wagner	-	68	63	62	68	57	55	50	44
M-80	50 #a	- *	-	75	73	74	73	67	64	58	54
M-81	50 #a	- *	-	77	75	75	81	69	65	61	60
M-82	60 **	Wagner	-	71	73	68	74	66	61	56	50
M-83	100 #	- *	-	80	82	88	86	87	85	82	78
M-84	150 #	- *	81	83	85	85	89	85	81	79	69

Believed loaded (all other motors assumed unloaded).
a Estimated, not known
* BBN-measured in field condition

**Average of 3 motors.
##New design relative to unit in line above.

TABLE M-5

SOUND PRESSURE LEVELS OF ELECTRIC MOTORS AT 3-FT DISTANCE (INDOORS)

CODE	APPROX. RATED HP	RPM	MFGR OR DATA SOURCE	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
M-85	250 #	1180	G.E. *	78	85	81	89	91	90	84	77	73
M-86	1250	1200	Allis-Chalmers	-	77	79	85	81	84	93	88	76
M-87	1250	1200	----	-	82	85	87	86	86	93	84	77
M-88	1500 #	1190	G.E. *	80	85	92	86	78	86	84	77	69
M-89	4000 #	1200	G.E. *	-	80	90	88	88	94	96	94	83
M-90	1250	900	Allis-Chalmers	-	83	86	84	79	78	77	69	63
M-91	200 #	600	Fairbands-Morse *	79	78	78	80	83	89	74	74	68
M-92	200 #	600	Fairbanks-Morse *	81	83	75	77	83	96	82	71	61
M-93	250 #	600	Fairbanks-Morse *	-	-	75	76	84	97	80	66	-
M-94	400 #	450	Westinghouse *	76	83	85	82	83	87	81	77	75

At or near full load (all other motors assumed unloaded).

* BBN-measured in field condition.

TABLE M-6

ESTIMATED SOUND PRESSURE LEVELS OF ELECTRIC MOTORS (AT 3-FT DISTANCE INDOORS) FOR VARIOUS SPEEDS AND POWERS

MOTOR RPM	RATED HP	OCTAVE BAND CENTER FREQUENCY - HZ								
		31	63	125	250	500	1000	2000	4000	8000
3400-3600	UNDER 12	73	74	78	82	83	83	82	76	69
	12-24	78	79	83	87	88	88	87	81	74
	25-49	83	84	88	92	93	93	92	86	79
	50-99	87	88	92	96	97	97	96	90	83
	100-200	90	91	95	99	100	100	99	93	86
	OVER 200	93	94	98	102	103	103	102	96	89
1000-1800	UNDER 12	68	69	73	77	78	78	77	71	64
	12-24	73	74	78	82	83	83	82	76	69
	25-49	78	79	83	87	88	88	87	81	74
	50-99	82	83	87	91	92	92	91	85	78
	100-200	85	86	90	94	95	95	94	88	81
	OVER 200	88	89	93	97	98	98	97	91	84
450-900	UNDER 12	64	65	69	73	74	74	73	67	60
	12-24	69	70	74	78	79	79	78	72	65
	25-49	74	75	79	83	84	84	83	77	70
	50-99	78	79	83	87	88	88	87	81	74
	100-200	81	82	86	90	91	91	90	84	77
	OVER 200	84	85	89	93	94	94	93	87	80

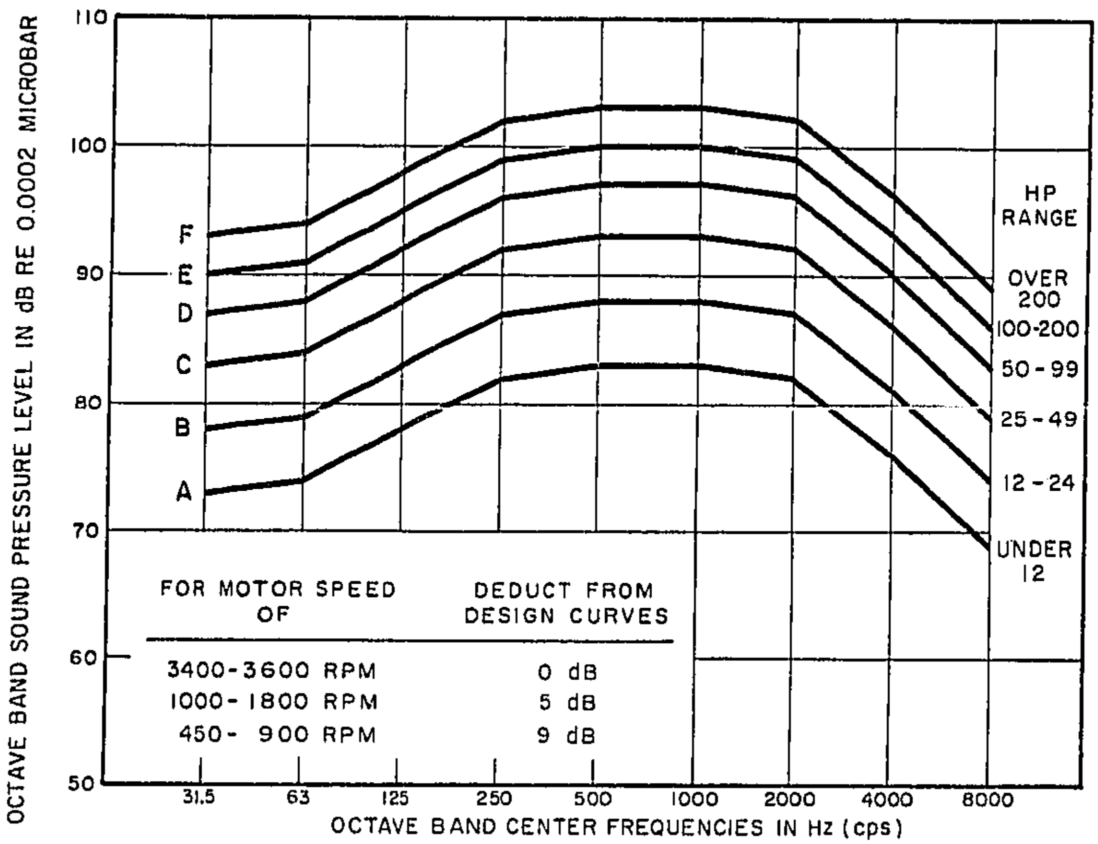

FIG.M-1 NOISE LEVELS OF ELECTRIC MOTORS AT 3-ft DISTANCE (SUGGESTED DESIGN CURVES FOR VARIOUS POWER AND SPEED RANGES)

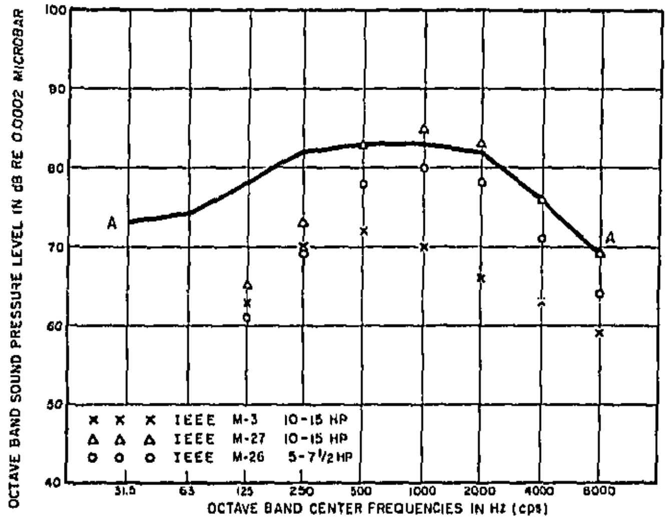

FIG.M-2 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve A compared with noise data for Motors in 3500-3600 RPM and under 12 HP Range)

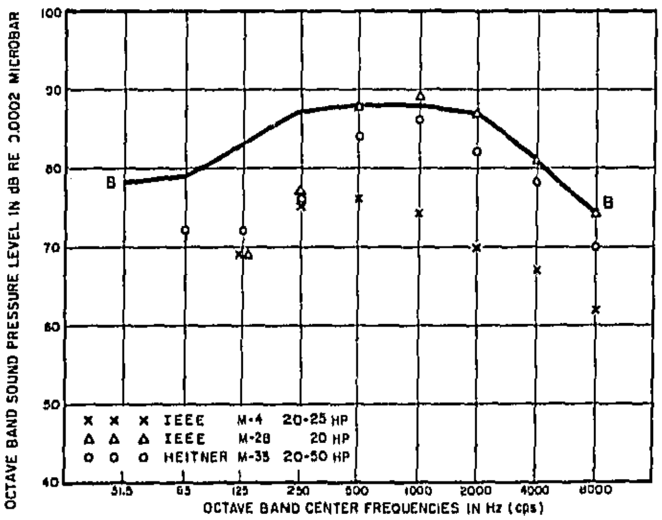

FIG.M-3 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve B compared with noise data for Motors in 3500-3600 RPM and 12-24 HP Range)

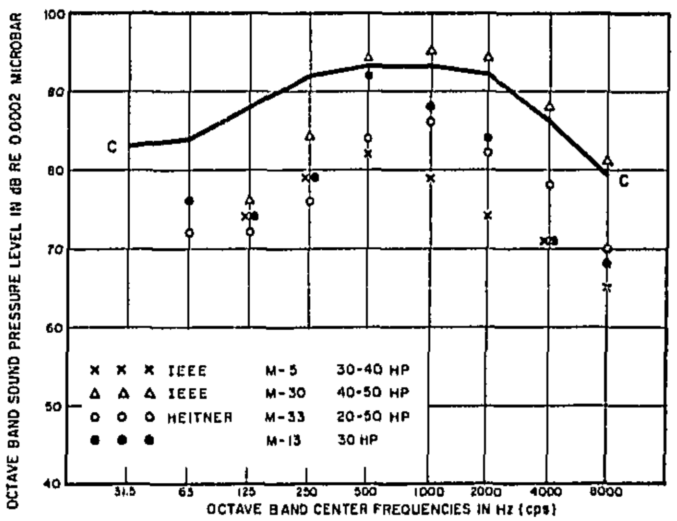

FIG.H-4 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve C compared with noise data for Motors in 3500-3600 RPM and 25-49 HP Range)

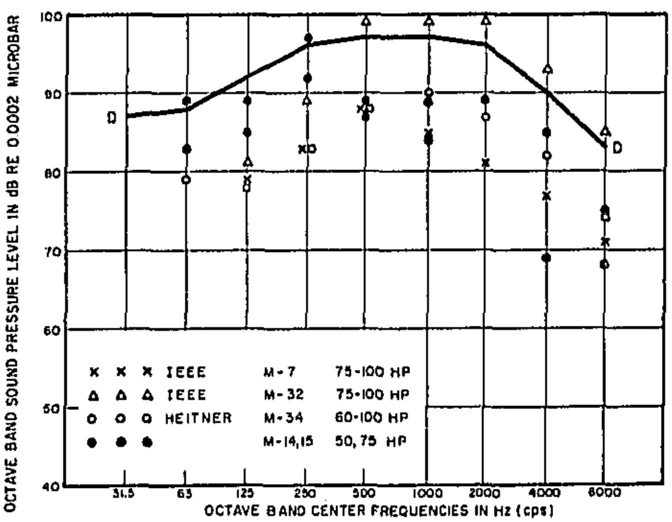

FIG.H-5 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve D compared with noise data for Motors in 3500-3600 RPM and 50-99 HP Range

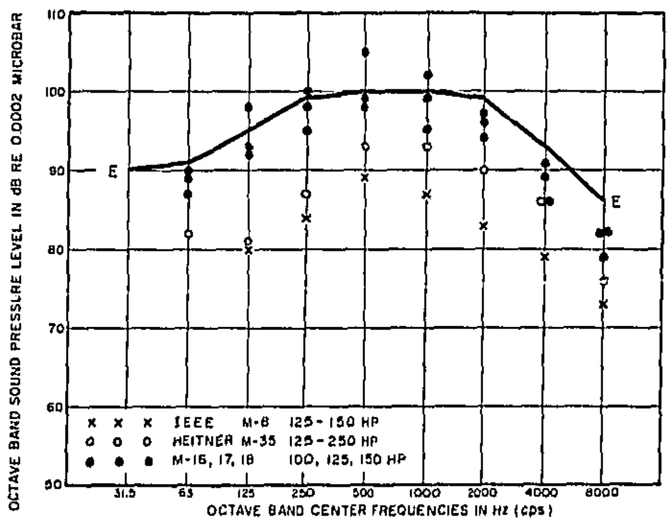

FIG. M-6 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve E compared with noise data for Motors in 3500-3600 RPM and 100-200 HP Range)

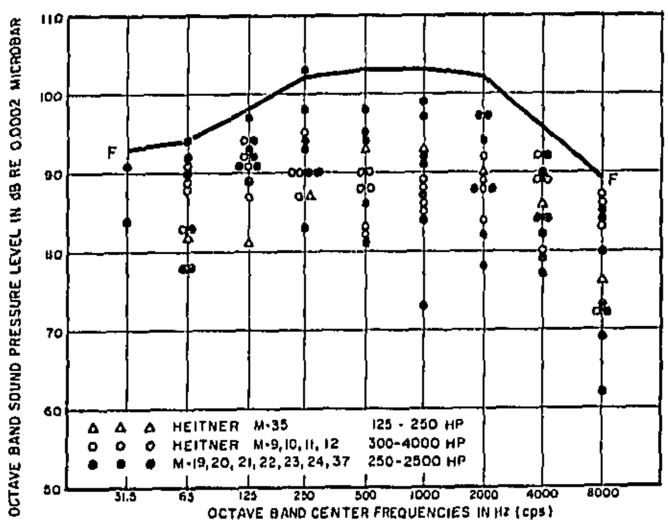

FIG. M-7 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve F compared with noise data for Motors in 3500-3600 RPM and over 200 HP Range)

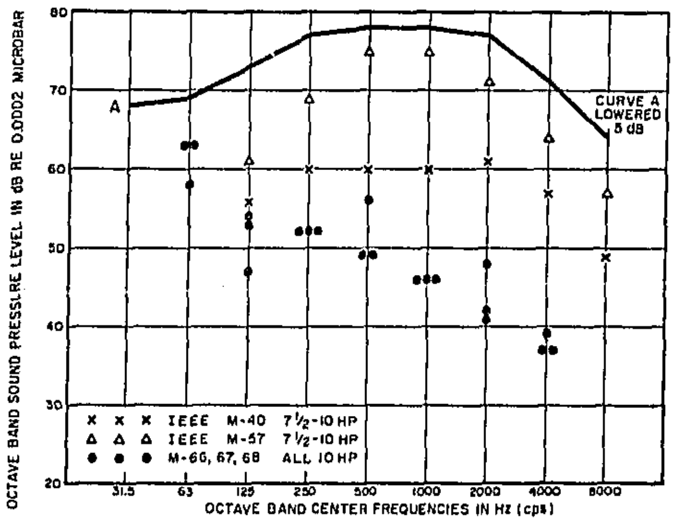

FIG. M-8 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve A, reduced 5 dB, compared with noise data for Motors in 1000-1800 RPM and under 12 HP Range)

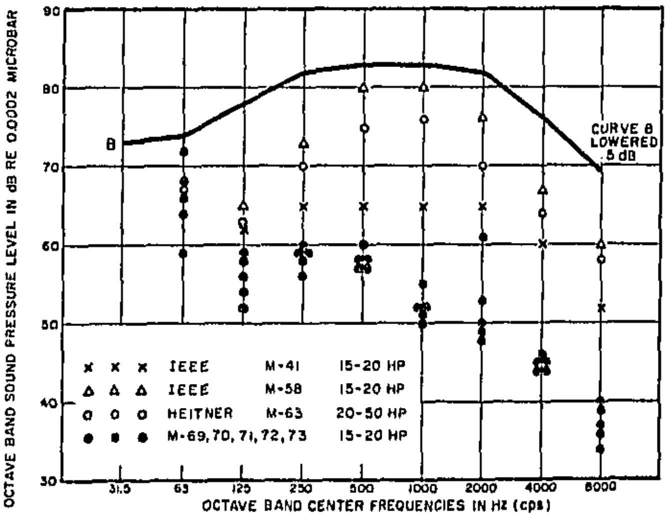

FIG. M-9 ELECTRIC MOTOR NOISE LEVELS AT 3-FT DISTANCE
(Design curve B, reduced 5 dB, compared with noise data for Motors in 1000-1800 RPM and 12-24 HP range)

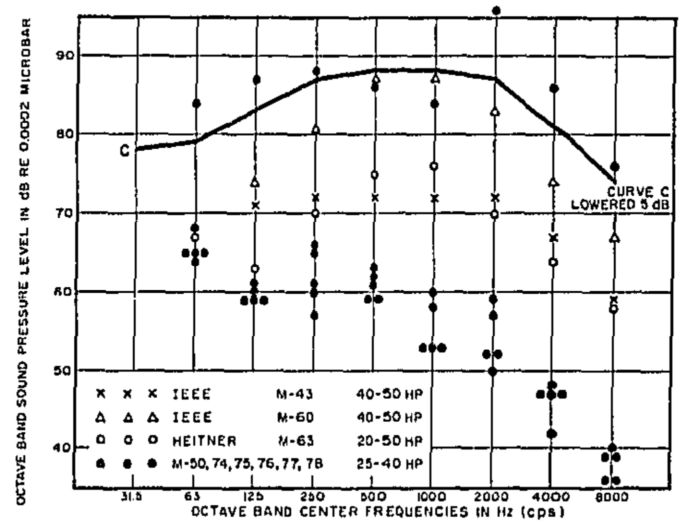

FIG.M-10 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve C, reduced 5 dB, compared with noise data for Motors in 1000-1800 RPM and 25-49 HP Range)

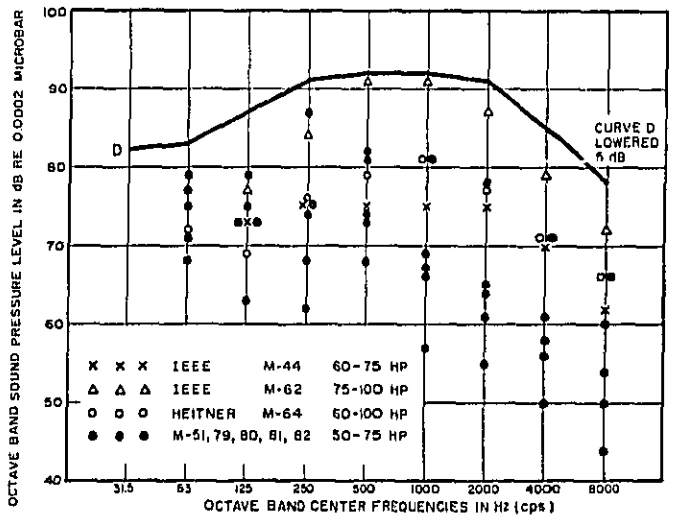

FIG.M-11 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve D, reduced 5 dB, compared with noise data for Motors in 1000-1800 RPM and 50-99 HP range)

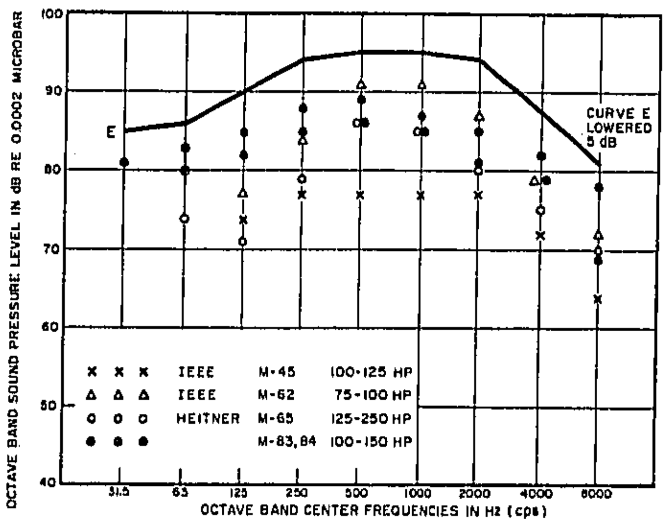

FIG. M-12 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve E, reduced 5 dB, compared with noise data for Motors in 1000-1800 RPM and 100-200 HP Range)

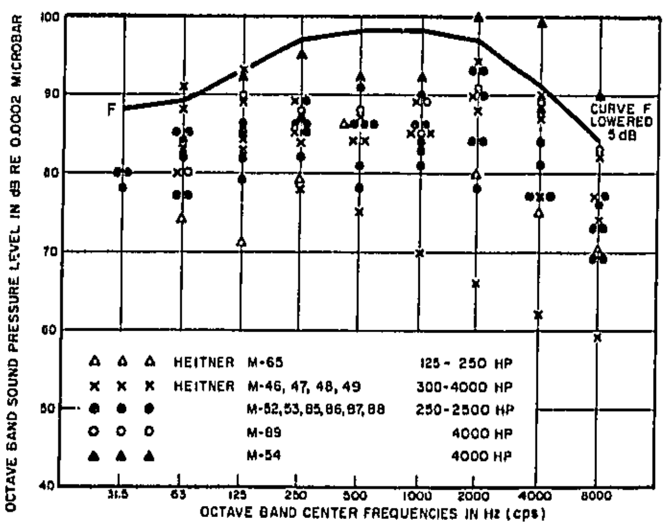

FIG. M-13 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve F, reduced 5dB, compared with noise data for Motors in 1000-1800 RPM and over 200 HP Range)

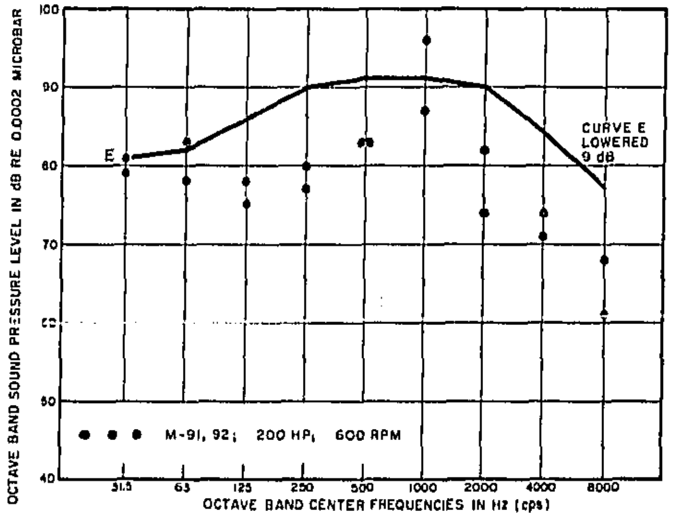

FIG.M-14 ELECTRIC MOTOR NOISE LEVELS AT 3-ft DISTANCE
(Design Curve E, reduced 9 dB, compared with noise data for Motors in 450-900 RPM and 100-200 HP Range)

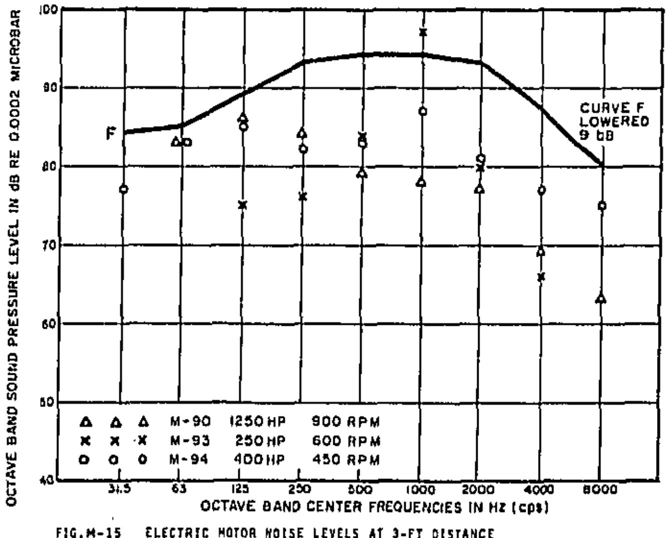

FIG.M-15 ELECTRIC MOTOR NOISE LEVELS AT 3-FT DISTANCE
(Design curve F, reduced 9 dB; compared with noise data for Motors in 450-900 RPM and over 200 HP range)

there are thirteen entries that show greater noise levels than their appropriate Design Curves would predict. Four of these nine entries exceed the Design Curves by 1, 2 or 3 dB, and these four entries represent IEEE values, which are quoted as probable maximum levels for the motors of several manufacturers. Thus, they represent possibly some of the noisiest motors produced within the industry. Of the remaining nine entries of the thirteen exceedances, one is a 3000-4000 HP motor included in Heitner's collection and listed here in the "over 200 HP" group and the other eight are motors measured or collected for this study. The two highest exceedances are 8 and 9 dB in one octave band each, the next two highest are 5 dB each in one octave band, and the four other exceedances are in the range of 1 to 3 dB in one or more octave bands. In view of this distribution (and the possible but unknown influences of other room noise and of no-load vs loaded operation), the Design Curves of Figure M-1 are suggested for this project and are believed to provide realistic estimates of noise for a high proportion (possibly 85-95%) of motors that might be used. Since there seems to be no lack of motors that make less noise than the Design Curves would predict, it would even be possible to specify that a motor for any particular location not exceed the SPL values suggested by the appropriate Design Curve.

No concern has been given here for a more detailed study of motors under 12 HP, as they generally will be quieter than most other equipment in a mechanical space.

The noise level values of Figure M-1 are proposed for use in the Manual and are summarized in tabular form in Table M-6.

The SPL data derived here are considered to be applicable also to electric motor-generator sets.

SECTION 13. STEAM TURBINES

Noise data for eight steam turbines have been collected under this study. An attempt has been made to separate the noise of the turbine from the noise of the pump, compressor, gear or generator driven by the turbine, but, as in any complex assembly of components, it is possible that in some octave bands the quoted noise levels for the turbines are influenced by other nearby attached equipment. Noise levels were originally measured at distances as close as 1 to 5 ft for seven of the turbines or as far as 30 ft for the case of one of the turbines located in a large room. All sound pressure levels have been extrapolated to the "standardized 3-ft distance." The extrapolated noise levels and the known data of the turbines are summarized in Table S-1.

Details of the various attempts to correlate noise with the known performance variables are not repeated here. With the limited number of turbines, there appears to be no strong correlation of sound pressure level with power rating of the turbine, although one might generally expect noise levels to grow as a function of 10 log HP. Also, from the measured data, there is no clear tendency for noise to peak in a predictable high frequency band, although some tendency in this direction might be found if the number of blades on various turbine wheels were known. Available time on this job would not permit an in-depth probing of such details. There is undeniable evidence, however, that the low frequency noise levels (obviously below turbine blade passage frequencies) sometimes exceed the higher frequency noise levels.

Because of the small number of turbines in the study, it is suggested that the noise prediction scheme be tailored to cover the upper limit of the noise of all the turbines given here. This may overestimate the noise of some turbines, but it will also underestimate the noise of any turbine that is actually noisier in any frequency band than the noisiest of the eight turbines studied. Figure S-1 gives the suggested noise level "Design Curves" for the range of power covered by the eight turbines. To check these curves, the measured noise levels of the eight turbines are compared with the Design Curves on the basis of turbine power, in the following Figures S-2 to S-4. Note that the curves of Figure S-1 are separated by 2-5 dB intervals, while representing differences in turbine power of a factor of three. In Figures S-2 to S-4 each Design Curve represents a noise level equal to or higher than the noise levels of all turbines in that particular power range.

The Heitner paper, mentioned earlier, offers a prediction scheme for turbine noise. The overall SPL at a 3-ft distance is estimated by either of two relationships

$$SPL = 58 + 10 \log HP \text{ or}$$

$$SPL = 82 + 10 \log G,$$

where G is the steam flow in lb/sec. The peak of the spectrum is set in the 1200-2400 Hz band (now the 2000 Hz band) for turbine speeds under 9000 RPM, and in that band the SPL is taken to be 4 dB below the overall. Either side of the peak, the SPL drops off at the rate of 3 dB per octave. The Heitner method makes no provision for low frequency noise from the turbine. In the current study, steam flow is not known, so only the relationship involving HP can be tested. In Figures S-2 to S-4, estimates based on the Heitner method are included.

TABLE S-1

SOUND PRESSURE LEVELS OF EIGHT STEAM TURBINES NORMALIZED TO 3-FT DISTANCE

CODE	RATED HP	APPROX RPM	MANUFACTURER	OCTAVE BAND CENTER FREQUENCY - HZ 31	63	125	250	500	1000	2000	4000	8000
S-1	500	--	De Laval	75	80	81	75	85	84	81	78	73
S-2	600	5000	G.E.	-	79	87	85	83	78	84	79	70
S-3	800	--	De Laval	78	90	94	82	85	84	79	76	70
S-4	1200	--	Murray	88	83	83	80	85	81	87	78	73
S-5	6000	--	--	88	95	88	92	88	86	81	76	77
S-6	10000	3600	Westinghouse	87	92	90	87	84	84	82	87	80
S-7	10000	3600	Westinghouse	81	82	83	83	80	83	78	83	83
S-8	11000	3600	Westinghouse	-	83	82	78	80	90	80	77	78

TABLE S-2

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

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

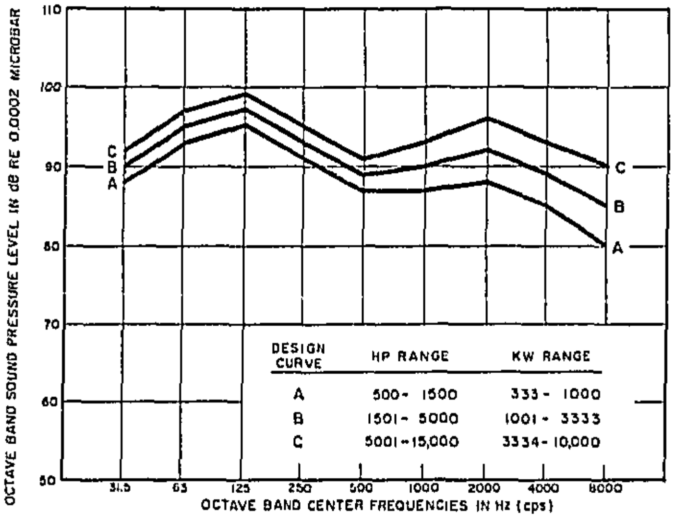

FIG.S-1 STEAM TURBINE NOISE LEVELS AT 3-FT DISTANCE
(Suggested Design Curves for Various Power Ranges)

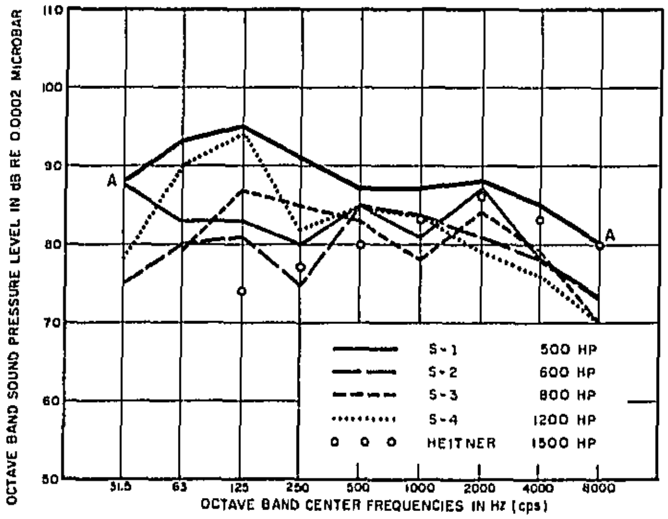

FIG.S-2 STEAM TURBINE NOISE LEVELS AT 3-FT DISTANCE
(Design Curve A compared with noise data for Steam Turbines in 500-1500 HP Range)

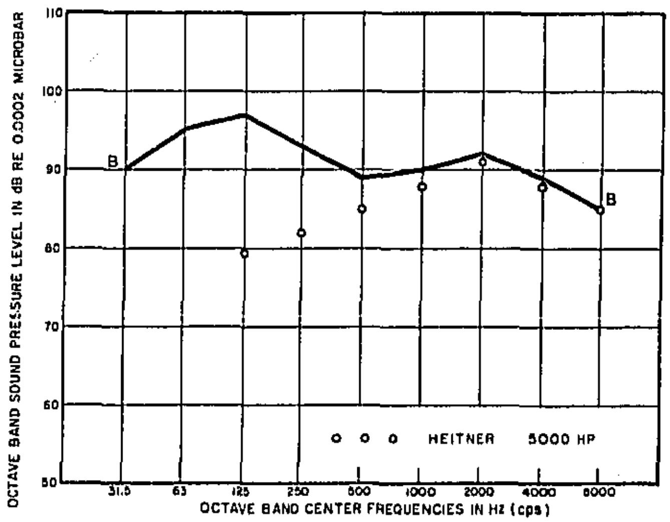

FIG.S-3 STEAM TURBINE NOISE LEVELS AT 3-FT DISTANCE
(Design Curve B compared with noise data for Steam Turbines in 1501-5000 HP Range)

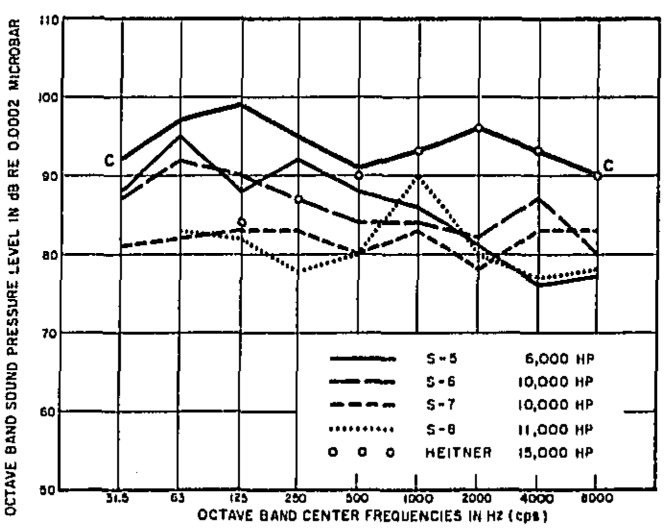

FIG.S-4 STEAM TURBINE NOISE LEVELS AT 3-FT DISTANCE
(Design Curve C compared with noise data for Steam Turbines in 5001-15000 HP Range)

Figure S-2 compares the measured noise levels of turbines S-1 through S-4 with the suggested Design Curve A for turbines in the power range of 500-1500 HP. The Heitner estimate for a 1500-HP turbine is also shown. Figure S-3 compares the Heitner estimate for a 5000-HP turbine with the suggested Design Curve B for the 1501-5000 HP range. Figure S-4 compares the noise levels of turbines S-5 through S-8 and the Heitner estimate for a 15000-HP turbine with Design Curve C for the 5001-15000 HP range.

In the Manual, the noise level values of the three Design Curves of Figure S-1 are suggested for estimating the SPLs at 3-ft distance for steam turbines. These values are given here in tabular form in Table S-2.

SECTION 14. GEARS

Noise data have been measured or collected for nine large gears in the power handling range of 300 to 23,200 HP. Three of these are taken from the G.E. data first given us in conjunction with the gas turbine engine study for the Power Plant Acoustics Manual. The known data for the nine gears are summarized in Table G-1; sound pressure levels have been normalized to a common distance of 3 ft.

Table 15 of the Power Plant Acoustics Manual offered a schedule for estimating the sound power level of gears. That schedule was derived from the G.E. data on three gears plus an earlier BBN generalization that sound power of a gear grows in proportion to the power transferred through the gear and that sound power generally increases with increasing speed of a gear. Because of the usually unknown design of the gear, it has been impossible to try to associate peak frequencies with gear-tooth-contact rates or with "ringing" frequencies due to the bell-like structure of some gear components. In Table 15 of the Power Plant Acoustics Manual, the schedule of sound power level was constructed around the noise of the three G.E. gears and the sound power level change was taken to be 3 dB for each halving or doubling of power and speed of the gear. The speed of the lower speed shaft was taken as the speed parameter.

Study of the data of the nine gears in the present program reveals that the sound putput of gears does not vary as widely as given in the Table 15 (PPA) schedule. A change of 1 dB for each halving or doubling of power and speed appears to be a more realistic rate of change. Even so, it is cautioned that

TABLE G-1

NOISE AND OPERATIONAL DATA ON NINE GEARS

CODE	HP	RPM INPUT SHAFT	RPM OUTPUT SHAFT	MANUFACTURER	SOUND PRESSURE LEVEL IN OCTAVE BAND NORMALIZED TO 3-FT DISTANCE 31 Hz	63 Hz	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz	8000 Hz
G-1	300	1780	8062, 13371	ALLIS-CHALMERS	82	86	92	88	93	98	94	81	71
G-2	450	1200	3600	TERRY	86	89	92	91	96	92	89	89	82
G-3	600	1200	5000	G.E.		82	87	89	87	85	86	79	71
G-4	1250	1200	5894			85	82	86	93	91	99	89	84
G-5	4000	1200	5834	G.E.		82	88	89	91	97	106	94	83
G-6	9300	6000	3600	G.E. #	88	92	99	94	94	93	91	88	83
G-7	11000	1500	3600	WESTINGHOUSE*		84	75	80	84	94	76	74	70
G-8	23200	5100	3600	G.E. #	95	100	95	91	94	103	87	91	103
G-9	23200	5100	3600	G.E. #	101	100	104	99	100	103	92	98	104

* This gear was known to be of "herringbone" design in order to achieve a relatively quiet installation

Taken from the study leading to the "Power Plant Acoustics Manual."

TABLE G-2

ESTIMATED SOUND PRESSURE LEVELS OF GEAR AT 3-FT DISTANCE, IN dB

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

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

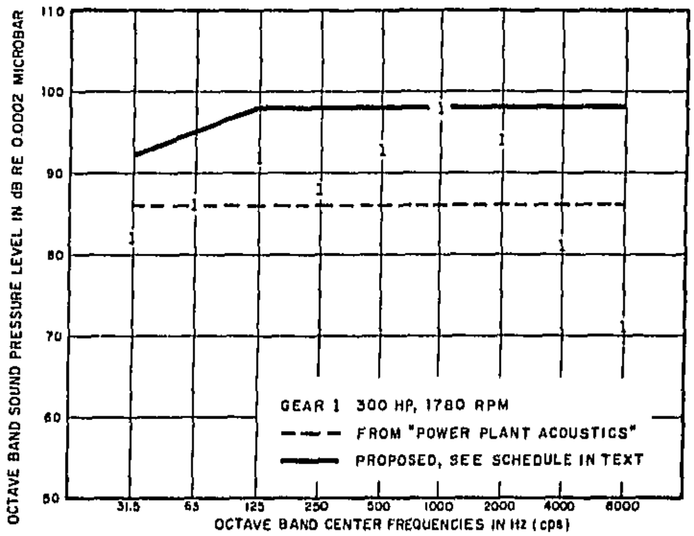

FIG. G-1 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

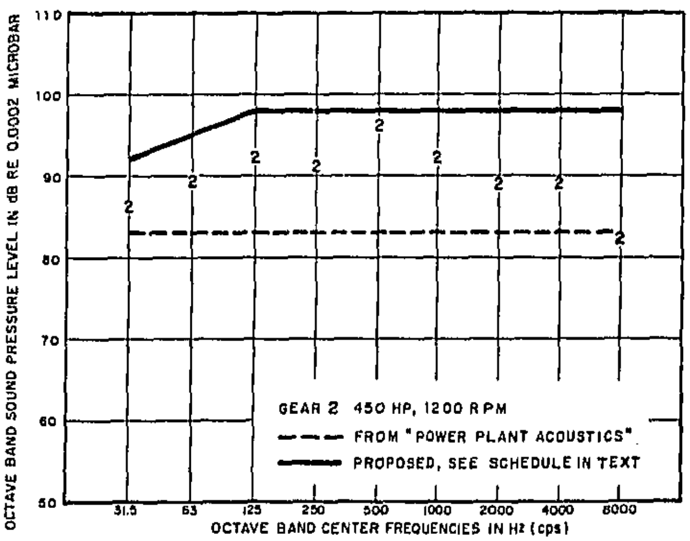

FIG. G-2 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

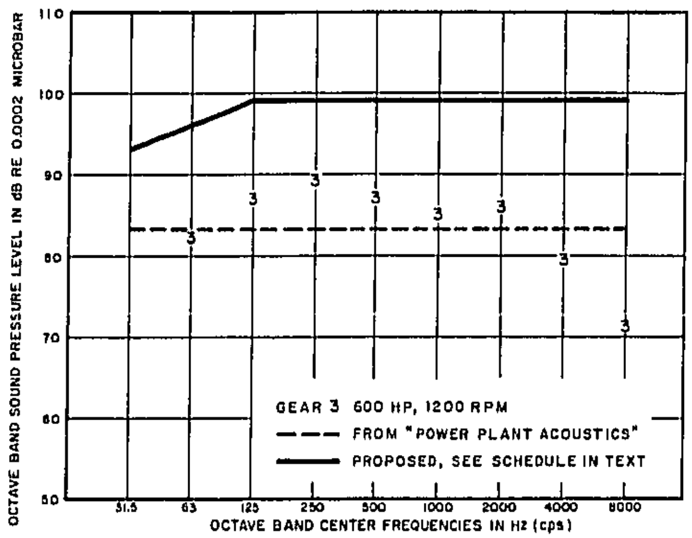

FIG.G-3 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

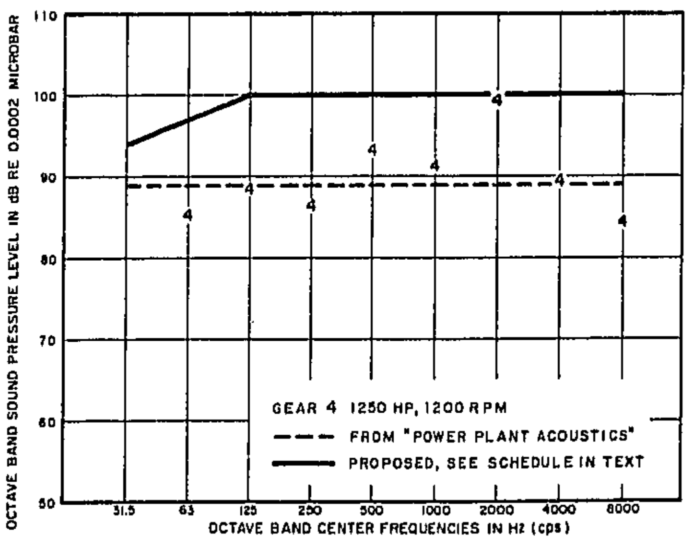

FIG.G-4 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

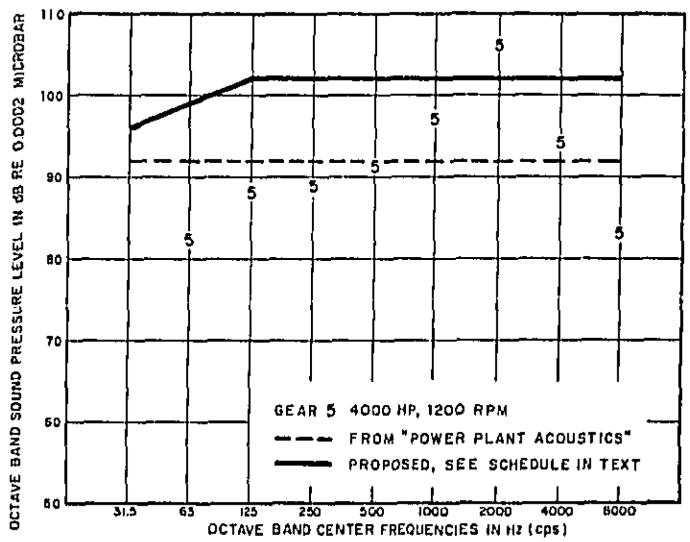

FIG.G-5 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

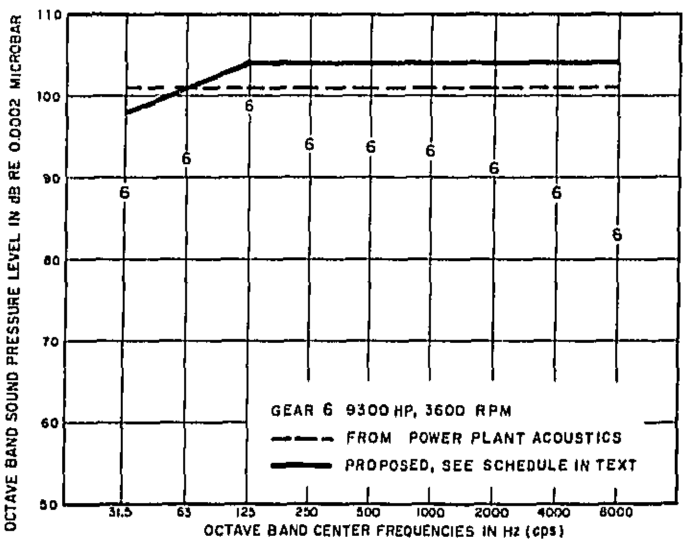

FIG.G-6 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

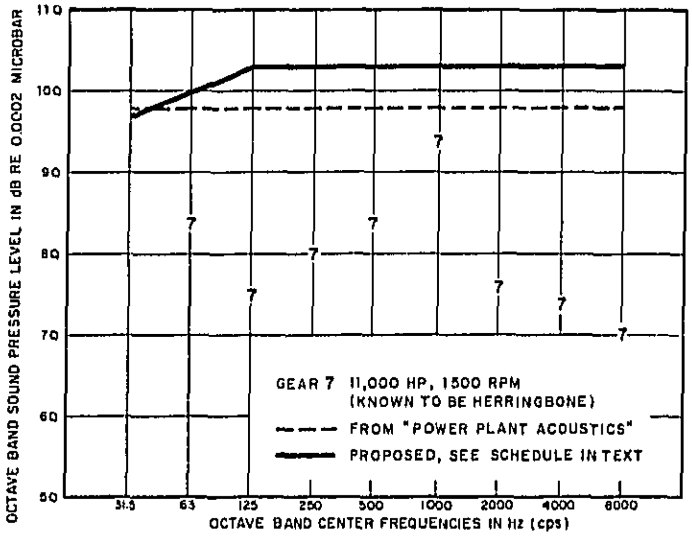

FIG.G-7 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

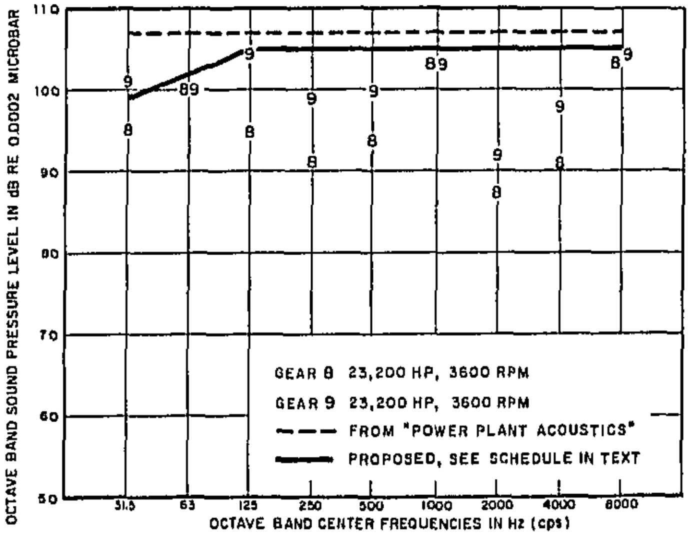

FIG.G-8 NOISE LEVELS OF GEARS AT 3-FT DISTANCE

a group of only nine gears does not constitute a very large sampling, but it is believed that the currently proposed schedule is an improvement over the earlier Table 15. The proposed schedule of sound pressure levels of gears at 3-ft distance is given in Table G-2. This schedule is suggested for use in the new manual, and it is proposed that this schedule take precedence over Table 15.

The noise levels of the nine gears are shown in Tables G-1 through G-8, where each gear is compared with the estimated noise level derived from Table 15 of the Power Plant Acoustics Manual and with the noise level given by the new schedule of Table G-2. A flat spectrum for the 125-8000 Hz octave bands is assumed in the new schedule in order to cover any octave bands in which gear-tooth-contact frequencies or "ringing" frequencies might occur. A slight drop-off of noise in the two lowest octave bands seems justified by the data.

Comparison of the plots of Figures G-1 through G-8 shows that only one gear (G-5) exceeds the proposed estimated value (by 4 dB at 2000 Hz), if we ignore the 31 cps exceedance of gear G-9 (which could have been partially due to noise from the nearby 15.5 megawatt gas turbine exhaust). Four gears fall 0 to 2 dB below the new estimate and four gears fall 5 to 10 dB below the new estimate. At least one of these quieter gears (G-7) was known to be a "herringbone" gear and was specially installed as a quieter replacement for an earlier noisier gear. On the basis of a sample of only nine gears, it is conjectured that the estimating procedure will give adequate coverage of gear noise in most situations.

SECTION 15. TRANSFORMERS

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

The current NEMA Standards Publication No. TR 1-1968 specifies the method for measuring and calculating the sound level rating for a transformer. In effect, the procedure consists of averaging a large number of A-scale sound level meter readings taken all around the transformer (at suitably specified positions) at distances of 1 ft from various surfaces of the transformer (or at 6-ft distances from fan-cooled radiating surfaces). The reader is referred to the NEMA publications for more detailed discussions of the procedure.

It is important to understand the significance of the NEMA "audible sound level," as it is called in the specification. Interest here is limited to 60 Hz power. Due to the magnetostrictive action of the transformer core material, the core goes through a complete cycle of oscillation for each half-cycle of voltage change. Thus, for 60 Hz operation, maximum sound output from the core occurs at 120 Hz and its harmonics (240, 360, 480 Hz and so on). Of course, there are relatively small amounts of sound radiation at 60 Hz and its odd-numbered harmonics but these are not significant in the present discussion.

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

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

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

Many manufacturers produce and sell transformers that are quieter than the NEMA standard for transformers. In fact, one might conjecture that the NEMA sound level standard is high enough that only the most unreasonably noisy transformers would be rejected. Even so, occasional noise problems are produced by transformers, and it is the purpose of this study to protect a building against excessive noise due to a transformer that (1) possibly does not meet the NEMA standard when installed, or (2) becomes noisier with use, or (3) becomes noisier when under load (the NEMA rating is taken under no-load conditions), or (4) is simply too noisy for the quiet environment in which it is to be used, regardless of its NEMA rating. In any event, a reasonable safety factor is derived here so that excessive noisiness can be anticipated and estimated.

First, experience shows that one important manifestation of a "noisy" transformer is that it produces an extra-ordinary amount of 240 or 360 Hz sound components, compared to the output at 120 Hz. Second, because discrete frequency sound signals can produce standing waves in enclosed spaces, provision must be made for sound level build-up. Also, of course, the NEMA data for 1-ft distance must be extrapolated to the normalized 3-ft distance used in this report.

A step-by-step development is given here. First, assume that the measured average <u>sound pressure level</u> at 1-ft distance from a sample transformer is as follows for the specific frequency components:

120 Hz	50 dB
250 Hz	55 dB
360 Hz	55 dB
480 Hz	50 dB

This assumed combination of sound pressure levels reflects the fact that a "noisy" transformer has high signal components at 240 and 360 Hz. It is just such a marginally "noisy" transformer that is to be protected by the building design. If the transformer were much noisier, it would be subject to rejection and return to the manufacturer, if the building specifications contain provisions for such action. Now, for this particular combination of frequency components, the A-scale reading probably would approximate the decibel summation of the component signals after each component passes through the filter action of the A-scale network; i.e.

at 120 Hz:	50 dB-16 dB = 34 dBA
at 240 Hz:	55 dB- 9 dB = 46 dBA
at 360 Hz:	55 dB- 5 dB = 50 dBA
at 480 Hz:	50 dB- 4 dB = <u>46 dBA</u>
Sum (A-scale)	= 53 dBA

It is conjectured here that this particular transformer would have met a 53 dBA NEMA sound level rating.

It is next desired to estimate the sound pressure level that might occur in a transformer room when such a transformer is brought in and installed. In the usually highly reverberant room that houses a transformer, and considering the near-field effect of a transformer that is probably large compared to the 1-ft NEMA measurement distance, a 2-dB reduction is allowed for extrapolating from the NEMA 1-ft distance to the normalized 3-ft distance used in this study (refer to Figure 11 in the Power Plant Acoustics Manual for support of this point--there are a few subtle facts involved which will not be elaborated upon here). There is also the strong possibility that standing waves may occur within the room at one of the transformer frequencies. For this possibility, an increase in sound pressure level of 10 dB is assumed; this is not at all unreasonable. Finally, for purposes of protecting the building design, the actual sound pressure levels of a transformer are assumed to range 5 dB higher than the values considered here. These various adjustments total

-2 dB	for 3-ft distance
+10 dB	for standing waves
+5 dB	for added protection
+13 dB	

In the transformer example started above, a NEMA rating of 53 dBA was found for an assumed sound pressure level of 50 dB at 120 and 480 Hz and 55 dB at 240 and 360 Hz. A 13 dB increase of these values is now suggested. Thus, when designing a room to contain a 53 dBA NEMA-rated transformer, the following assumed sound pressure levels would be considered:

at 120 Hz	63 dB
240 Hz	68 dB
360 Hz	68 dB
480 Hz	63 dB

Note that these values are numerically 10 and 15 dB, respectively, above the NEMA rating of 53 dBA.

When converting the 63 dB and 68 dB discrete frequency components into the octave frequency bands used in this work, the 120, 240 and 480 Hz components clearly fall into the 125, 250 and 500 Hz octave bands, and the 360 Hz band falls just at the cross-over between the 240 and 480 Hz octave bands. Hence, the 360 Hz signal is assumed to be shared equally between the two octave bands. This distribution then leads to the following octave band sound pressure levels:

125 Hz band	63 dB
250 Hz band	70 dB
500 Hz band	67 dB

These values are now seen to be 10, 17 and 14 dB, respectively, above the 53 dBA NEMA rating. In this analysis it is assumed that approximately these same differences would apply for any transformer installed indoors. In Figure T-1, these data points are plotted in such a way that the estimated sound pressure level can be determined for any transformer, provided its NEMA sound level rating is known. For example, if a transformer had a NEMA sound level of 70 dBA, its sound pressure levels would be estimated to be

80 dB	in the 125 Hz band,
87 dB	in the 250 Hz band, and
84 dB	in the 500 Hz band

Below and above these three frequency bands, the noise is taken to drop off at the rate of 5 dB per octave.

Tables T-1 and T-2 summarize some of the sound level data from the NEMA Standards Publication No. TR1-1968. Table T-1 gives a somewhat abbreviated version of NEMA data for dry-type transformers and Table T-2 gives a very abbreviated version of data for oil-immersed transformers. The data given in these tables are offered here only to provide a general view of the overall type of information given in the NEMA publication. The data in Tables T-1 and T-2 are lacking many of the technical qualifications contained in the NEMA standard. Do not use Tables T-1 and T-2 to determine specific sound level ratings for specific transformers! Instead, refer to the actual Applicable NEMA Standard for the appropriate NEMA ratings. However, once that rating has been determined, it is believed tha application of the Figure T-1 adjustments will provide approximately the maximum sound pressure levels required for designing the transformer room. The data of Figure T-1 are tabulated in Table T-3, and this estimation procedure is proposed for use in the Manual.

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

1. Where a manufacturer is willing to guarantee that his product

TABLE T-1

NEMA "AUDIBLE SOUND LEVELS" FOR DRY-TYPE TRANSFORMERS

1500-VOLT INSULATION CLASS AND BELOW

(Refer to latest applicable NEMA Standard for details and qualifications. Several details omitted here.)

EQUIVALENT TWO-WINDING KVA	AVERAGE DBA READING AT 1-FT DISTANCE (IN ACCORDANCE WITH NEMA PROCEDURE) SELF-COOLED SEALED	SELF-COOLED VENTILATED	FORCED-AIR VENTILATED
0-300	57	58	67
301-500	59	60	67
501-700	61	62	67
701-1000	63	64	67
1001-1500	64	65	68 *
1501-2000	65	66	69 *
2001-3000	66	68	71
3001-4000	68	70	73 **
4001-5000	69	71	73
5001-6000	70	72	74
6001-7500	71	73	75 *
7501-8333	--	--	75
8334-10000	--	--	76

* May be 1 dBA lower value for some units in this KVA range.

** May be 2 dBA lower value for some units in this KVA range.

TABLE T-2

NEMA "AUDIBLE SOUND LEVELS" FOR OIL-IMMERSED POWER TRANSFORMERS (CLASS OA, OW AND FOW RATINGS)

(Refer to latest Applicable NEMA Standard for details and qualifications. Several details omitted here.)

NEMA AVERAGE SOUND LEVEL (dBA)	EQUIVALENT TWO-WINDING KVA RATING 125 KV INSULATION CLASS	350 KV INSULATION CLASS	750 KV INSULATION CLASS	1300 KV INSULATION CLASS
48	0-50	--	--	--
51	51-100	--	--	--
55	101-300	--	--	--
56	301-500	--	--	--
57	--	700	--	--
58	--	1000	--	--
59	--	--	--	--
60	--	1500	--	--
61	--	2000	--	--
62	--	2500	--	--
63	--	3000	--	--
64	--	4000	--	--
65	--	5000	--	--
66	--	6000	3000	--
67	--	7500	4000	--
68	--	10000	5000	--
69	--	12500	6000	--
70	--	15000	7500	--
71	--	20000	10000	--
72	--	25000	12500	--
73	--	30000	15000	--
74	--	40000	20000	--
75	--	50000	25000	12500
76	--	60000	30000	15000
77	--	80000	40000	20000
78	--	100000	50000	25000
79	--	--	60000	30000
80	--	--	80000	40000
81	--	--	100000	50000
82	--	--	--	60000
83	--	--	--	80000
84	--	--	--	100000

TABLE T-3

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

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

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

* For Indoor Installations

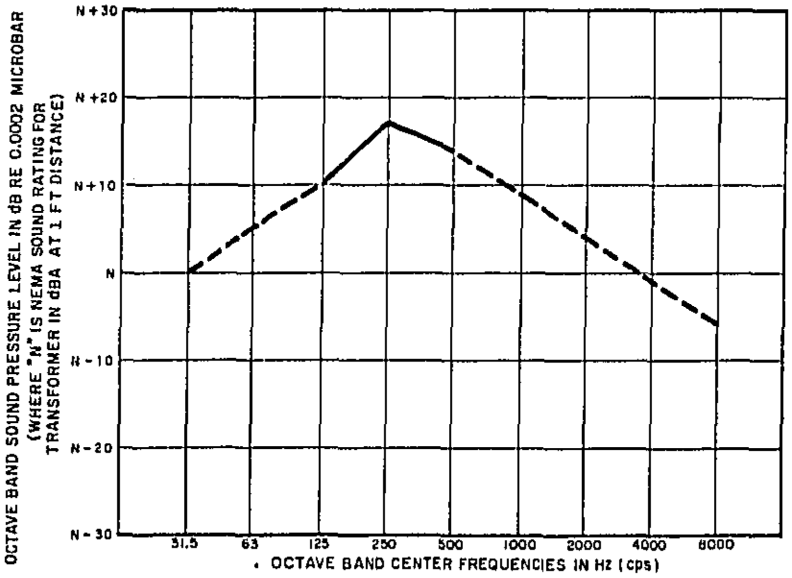

FIG.T-1 ESTIMATED SOUND PRESSURE LEVEL INSIDE A TRANSFORMER ROOM AT A 3-FT DISTANCE FROM A TRANSFORMER HAVING NEMA SOUND LEVEL RATING OF "N" dBA will produce a lower sound level rating than the otherwise-applicable NEMA rating, the manufacturer's sound level value (the average dBA reading taken at 1 ft distance in accordance with the NEMA method) may be used as the value of "N" when entering Table T-3 or Figure T-1.

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

3. Although the procedure developed here is based on transformer noise rather than cooling fan noise, it is believed that the envelope of the noise curve of Figure T-1 will protect against a reasonable amount of fan noise for any large forced-air cooled transformer. One form of this protection is the addition of 10 dB to the estimate in order to cover possible standing wave build-up of the transformer frequencies. Fan noise would not likely have such a large standing wave effect, if indeed any effect at all. In a somewhat similar manner, if the original NEMA ratings were largely influenced by fan noise (in this case measured at a 6-ft distance, instead of a 1-ft distance), the combination of adjustments will still give adequate coverage, since broad-band fan noise would not be subject to standing wave build-up to the same extent as is possible for transformer frequencies.

4. The procedure outlined here has not been tested against specific case histories, so it must be considered somewhat hypothetical. The values and the suppositions are all reasonable, however. Many transformers are quieter than the NEMA standard, many transformers do not produce unusally high 240, 360 and 480 Hz noise components, and for many installations there will be no standing wave build-up, so this procedure will appear to yield high sound pressure levels when tested against many existing situations. As implied at the outset, however, this procedure is designed to protect a room against the marginally "noisy" transformer in which each of these effects may be somewhat pronounced.

ACKNOWLEDGEMENT

This has been an excellent opportunity to collect noise data on much of the equipment encountered in many present-day buildings. The noise data will be of value, obviously, in preparing the Mechanical Equipment Noise Control Manual. In addition, the summaries of this material, with appropriate deletion of references to manufacturers and their products, would make valuable contributions to the noise literature where it would be available to the many people having need of this kind of data.

It is a tribute to the Corps of Engineers that this need has been recognized and that this Manual has been undertaken.

I wish to thank those persons who have already been mentioned in the equipment sections for their assistance and/or the use of their data, and I also want to thank the many BBN staff members in all our offices for their work in collecting data or measuring the noise of much of the equipment. A special thanks is extended to the BBN Illustration Department for the preparation of all the figures, to the Printing Department for the printing of this and the following Manual, and to our secretaries who graciously absorb the task of typing the various drafts and final versions of these reports. These are truly joint efforts.

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NOISE

How to Estimate Plant Noises

Designers can now anticipate how much noise will come from the more noiser pieces of equipment in an HPI plant

Irving Heitner
The M. W. Kellogg Co., New York, N.Y.

NOISE IS BECOMING increasingly important as a result of legislation, insurance requirements, community reaction and union regulations. This report will help you obtain an estimate of expected noise levels so that adequate funds and design can be provided to ensure satisfactory noise levels.

THE NATURE OF NOISE

Noise is defined as unwanted sound. Sound as a vibration is characterized by frequency and amplitude. The following describes some of the ways noise is measured.

Frequency Band. Some noise sources emit a single frequency but most are broad band. Thus bands are used to describe the noise frequency. These bands are generally octaves so that the upper frequency band is double the next lower frequency band. The frequency bands shown in the first column of Table 1 are commonly used in the United States. Outside of the United States the International Organization for Standardization (ISO) preferred frequencies are used. These preferred frequencies are the geometrical center frequencies of the band and are also given in Table 1. The ISO preferred frequencies are already in use in the United States.

TABLE 1—Frequency Bands

U.S. FREQ. BAND		I.S.O. FREQ. BAND
Cycles/Sec. Hz	Approx. Geom. Mean, Hz	Geom. Mean of Band, Hz
20/75	40	16
		31.5
		63
75/150	105	125
150/300	210	250
300/600	425	500
600/1,200	850	1,000
1,200/2,400	1,700	2,000
2,400/4,800	3,400	4,000
4,800/10,000	6,000	8,000

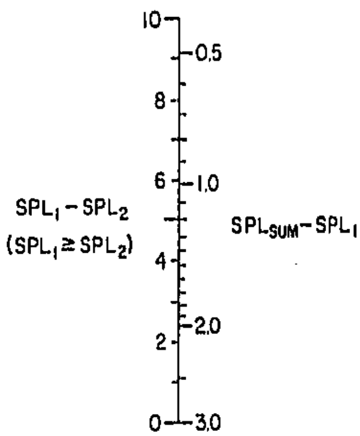

Fig. 1—To add noise from two sources.

Decibels. The strength of a noise source is expressed in terms of the acoustical power it emits. Because of the very wide range of acoustical powers measured it is customary to use a log scale, i.e.,

$$\text{decibels, dbs} = 10 \log_{10} (\text{measured/reference})$$
$$\text{or } PWL = 10 \log_{10} (W/W_o) \qquad (1)$$

where

PWL = logarithmic unit of acoustical power, dbs

W = acoustical power, watts

W_o = reference power, 10^{-13} watts

(Some[1] use $W_o = 10^{-12}$ watts)

It is not convenient to measure acoustical power but it is comparatively easy to measure acoustical root-mean-square pressure. It can be shown[2] that sound power is proportional to the square of sound pressure. Thus

$$SPL = 20 \log_{10} (P/P_o) \tag{2}$$

where

SPL = sound pressure level, dbs

P = rms pressure, dynes/sq. cm.

P_o = reference pressure, 0.0002 dynes/sq. cm.

Noise Addition. To obtain the sum of a number of sources the logarithmic basis for defining decibels must be considered. The energy of the total noise is the sum of the individual energy sources. Thus

$$W_{sum} = W_1 + W_2 + \cdots$$

This is equivalent to

$$P_{sum}^2 = P_1^2 + P_2^2 + \cdots$$

Since

$$SPL_{sum} = 10 \log (P_{sum}^2/P_o^2)$$

and

$$SPL_1 = 10 \log (P_1^2/P_o^2)$$

then

$$SPL_{sum} = 10 \log [\text{antilog } (SPL_1/10) + \text{antilog } (SPL_2/10) + \ldots]$$

A convenient method for obtaining the sum of two noise sources[3] is possible with the scales given in Fig. 1. Repeated addition of pairs can be used for adding any number of sources.

For example, adding two noise sources of equal intensity increases the level by 3 decibels. If one noise source is 10 decibels greater than a second the total level is relatively unchanged by adding the smaller noise source to the larger.

Noise Radiation. The SPL on a surface receiving noise radiation can be related to the PWL of the noise source as follows:[3]

$$SPL = PWL - 10 \log (S/S_o) + 0.5 \tag{3}$$

where

SPL = decibels relative to 0.0002 dynes/sq. cm.

PWL = decibels relative to 10^{-13} watts

S_o = unit area, one sq. ft.

S = total surface receiving noise, sq. ft.

For a noise source in free space and a receiving point free of reflecting surfaces we have spherical radiation and $S = 4\pi r^2$. A receiving point near the ground is approximated by hemispherical radiation for which $S = 2\pi r^2$. In either case S is proportional to r^2 where r is the distance from the source to the area receiving the radiation. Thus

$$\Delta(SPL) = 20 \log (r_2/r_1)$$

or a doubling of distance decreases the SPL by 6 decibels.

In the immediate vicinity of a noise source called the "near field" this relationship does not hold. Measurements show that

$$\Delta(SPL) = K \log (r_2/r_1)$$

where K may be 5 and increase to 20 as r_2 becomes a larger multiple of the physical size of the source.[4]

Noise in Enclosures. There are many areas in processing plants that have considerable reflection and some, such as pipe racks and compressor shacks, are essentially large enclosures. It can be shown[3] that Equation 3 can be extended to give

$$SPL = PWL + 10 \log \left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) \tag{4}$$

$$Q = \text{antilog } [(SPL_\theta - SPL_{av})/10] \tag{5}$$

$$R = \frac{S\bar{\alpha}}{(1 - \bar{\alpha})} \tag{6}$$

$$\bar{\alpha} = \Sigma\, \alpha_i S_i / \Sigma\, S_i \tag{7}$$

where

SPL_θ = SPL at angle θ

SPL_{av} = space average SPL

$\bar{\alpha}$ = average absorption coefficient

α_i = absorption coefficient for element S_i

S_i = element of area; $\Sigma\, S_i = S$, total area

Note that the 0.5 term of Equation 3 has been neglected. The term Q allows for any unequal radiation in different directions. Some directivity corrections will be given later. The term $4/R$ accounts for the reflected noise when equilibrium is established. The value of α_i varies from 0 to 1 and may be taken as 0.03 for steel and concrete, and 1.0 for empty space.

The Source of Noise

Noise production in hydrocarbon plants may be divided into the following categories. First, the effect of gas streams on stationary air—vents and air intakes. Second, the effect of moving fluids on confining metal surfaces—control valves and pipe lines. Third, the effect of moving metal surfaces on stationary or flowing fluids—fans, pumps, compressors. Finally, the effect of periodically contacting metal surfaces—gearing.

Vents. The noise generated by the action of gas flow into the atmosphere will be considered first since vents generally constitute major noise sources. Defining M, the Mach number of the jet as the exit velocity divided by the speed of sound in the surrounding atmosphere, there are the following three classifications:

First, subsonic ($M < 1$) for which the total acoustical power is proportional to V^8. Second supersonic ($M > 1$) where the acoustical power is proportional to V to some power less than 8. Finally, choked flow where the gas reaches its sonic velocity and where its pressure is reduced to atmospheric by a shock or series of shocks.

Aerodynamic noise was investigated by M. J. Lighthill,[5] who showed that the total acoustical power generated by a subsonic jet may be given by:

$$W_a = K_a \frac{\rho_t^2 V^8}{\rho_o c_o^5} d^2 \quad (8)$$

and $W_a = \eta W_m$ (9)

Since

$$W_m = \rho_t V^3 \pi d^2 / 8 \quad (10)$$

then

$$\eta = (8/\pi)(\rho_t/\rho_o) K_a M^5 \quad (11)$$

where

W_a = acoustical power
K_a = acoustical power coefficient
ρ_t = density of flowing gas
ρ_o = density of atmosphere
V = jet exit velocity
d = jet nozzle diameter
c_o = sound velocity in atmosphere
η = acoustical efficiency
W_m = mechanical power
M = Mach number, V/c_o

Experimental results and Lighthill's analysis[6] have shown for subsonic jets that η is of the order of $10^{-4} M^5$ and is proportional to ρ_t/ρ_o and $(T_t/T_o)^2$ where T_t is the absolute temperature of the jet and T_o is the absolute temperature of the atmosphere. In extending Equation 9 to supersonic jets there is evidence that the total acoustical power is proportional to V^n with n decreasing with increasing M. Further, as M increases the acoustical power becomes proportional to ρ_t, instead of ρ_t^2. The temperature correction deceases and eventually η becomes constant independent of M. These variations giving acoustical efficiencies for both subsonic and supersonic jets are shown in Fig. 2.

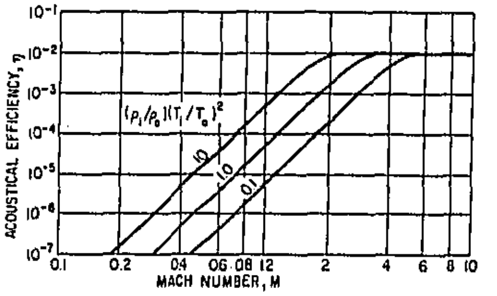

Fig. 2—Acoustical efficiences for jets.[6]

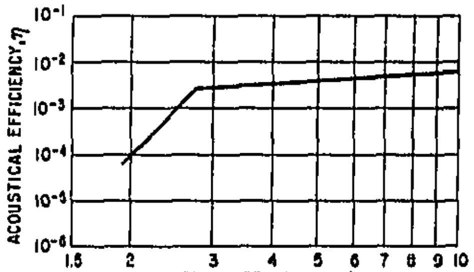

Fig. 3—Choked jets exhibit shock formation.[6]

Choked Jets. A typical jet is cone shaped (angle about 16 degrees) and extends about 17 to 20 diameters[7] with a highly turbulent boundary which is the source of the observed noise. Choked jets in addition to this turbulent boundary noise form a cellular structure[8] as a result of shock formation, which structure is an additional intense source of noise. The acoustical efficiencies shown in Fig. 3 are used to calculate the noise produced by this phenomenon, which noise is in addition to the turbulent boundary noise as calculated from the acoustical efficiency given in Fig. 2 ($\Sigma \eta \leq 10^{-2}$).

Directional Correction. Having obtained the acoustical power in watts the *PWL* is obtained from Equation 1 and the *SPL* from Equation 3. The *SPL* thus calculated is the space average *SPL* and should be corrected for the angle from the jet axis to the point of measurement. The directivity pattern depends on the frequency of the emitted noise but only corrections for the over-all *SPL* will be noted herein. For non-choked vents the corrections[3] to be applied to the aveage over-all *SPL* to obtain the over-all *SPL* in free space at the given angle are given in Table 2.

For choked jets the noise contributed by the shock formation produces strong radiation normal to the jet.[9] For these jets the correction[10] to be applied to the over-all space average *SPL* can be taken from the last column of Table 2.

The corrections for choked jets apply to the total noise, both turbulent and shock. As noted previously, directivity corrections will differ for different frequencies and will differ from the over-all corrections given. However, an individual band correction is not warranted for this sort of estimate.

In a highly reflective environment such as exists at grade in the average chemical plant, the above corrections will not be measured. The corrections given enable one

TABLE 2—Directional Corrections for Jets

Angle from Jet Axis, Degrees	SPL CORRECTION, DBS	
	Non-Choked	Choked
0	0	− 3
20	+ 1	+ 1
40	+ 8	+ 0
60	+ 2	+ 3
80	− 4	− 1
100	− 8	− 1
120	−11	− 4
140	−13	− 6
160	−15	− 8
180	−17	−10

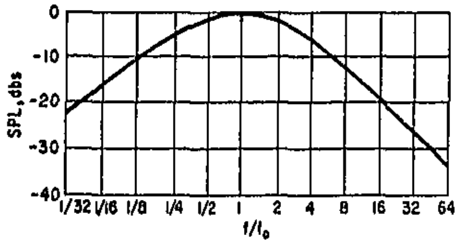

Fig. 4—Noise spectrum for jets.

to calculate Q in Equation 4 and then to estimate or calculate the $4/R$ term. From these calculations the measured noise can be estimated.

Noise Spectrum. To check conformity with plant noise specifications an estimate of the SPL in the different frequency bands is required. For a jet a typical plot[11] of noise intensity versus frequency is shown in Fig. 4. To estimate f_o, the frequency of maximum noise production, a dimensionless number, the peak Strouhal number, N_s, is used:

$$N_s = f_o d/V$$

The value of N_s is generally taken[3] as 0.2. The SPL in the octave containing f_o is taken as 5 decibels less than the over-all SPL and the SPL in the remaining octaves are estimated from Fig. 4. For choked jets the shock noise portion of the total noise will have a Strouhal number which is dependent on the pressure upstream of the shock[3] but for most estimates it suffices to use Equation 12 for choked flow.

Steam Vents. It is a common plant observation that steam vents are especially noisy, possibly as a result of eventual condensation in the atmosphere. There are few theoretical or experimental investigations of steam jets.[14] On the basis of personal measurments it is recommended that the SPL for steam vents as estimated by methods noted previously be increased by 3 decibles if $\eta < .005$.

Control Valve Noise. The important problem of control valve noise for gas flow has been the subject of little theoretical or experimental investigation. One mechanism considers the control valve as forming a jet which is confined by the pipe wall and then applies the previously noted vent formulas.[12] There is disagreement as to the validity of this mechanism[13] but it has proved useful for estimates.

Considering the valve as single seated, calculate the port diameter, velocity and gas density under operating conditions. From velocity, pressure and temperature the acoustical efficiencies for subsonic and choked flow can be estimated. Multiplying Equation 10 by the acoustical efficiency and using Equation 2 gives the PWL. The SPL is calculated by calculating the loss through the valve wall and applying a correction for distance.

To estimate the loss through the valve wall recourse is made to the results of studies of waves in solid structures, chiefly panels.[4] The following is recommended to estimate the transmission loss through a pipe wall:

$$TL = 17 \log (m f) - 36 \qquad (13)$$

where

TL = transmission loss, dbs ($TL \geqslant 0$)
m = weight of pipe wall, lb./sq. ft.
f = frequency, cycles per second, Hz

This formula is intended as a conservative approximate answer to a complex problem.[4] It assumes adequate structural rigidity and thus for large diameter, thin-walled, inadequately supported lines the loss may be less than given above.

A frequency estimate is required to solve Equation 13. Based on calculated single port diameter and calculated single port velocity a peak frequency can be obtained using a Strouhal number of 0.5. This frequency is used in Equation 13 to estimate the over-all transmission loss. Since the transmission loss is greater in the higher frequencies the spectrum of control valve noise will shift to lower frequencies external to the valve. Here the peak frequency is estimated on the basis of a Strouhal number of 0.2 using port size and velocity and the spectrum is obtained from Fig. 4. The transmission loss could be calculated on an octave by actave basis but the mechanism does not warrant this refinement. Further, no directivity corrections are made.

Pipe Lines For Gases. The noise produced by long large high velocity gas lines can be of importance. Again it is useful to use the mechanism of a mechanical power source and an acoustical efficiency even though the mechanism used for vents is not directly applicable since there is no jet. Nevertheless, as friction drop can be expressed in terms of velocity head, a possible mechanism is to consider this drop as the source of mechanical power. The acoustical efficiency would be calculated on the basis of the line velocity and diameter as well as the gas density and temperature. Thus we would have

$$SPL = 10 \log \left(\frac{10^{13} \eta \, W_m \, (\Delta L)}{4 \, \pi r^2} \right) - (TL) \qquad (14$$

$$W_m = 1.36 \, (\Delta P/L) \, (\pi d^2/4) V$$

where

η = acoustical efficiency, Fig. 2
W_m = mechanical power, watts/ft. of length
ΔL = differential pipe length, ft.
r = point of measurement distance to ΔL
TL = transmission loss, dbs
ΔP = total pressure drop, incl. fittings and valves lb./sq. ft.
L = total length of pipe, ft.
V = average line velocity, ft./sec.
d = pipe diameter, ft.

The Strouhal number for calculating the TL is taken as 0.5; external to the pipe, it is taken as 0.2. Equation 14 requires integration. For an infinite line, use

$$SPL = 10 \log \left(\frac{10^{13} \eta\ W_m}{4\ r}\right) - (TL) \qquad (15)$$

where

r = perpendicular distance from the observer to the pipe wall.

The model being imperfect requires correction. For gases whose density is approximately that of the ambient air it underestimates the noise level by 2 to 4 decibels. For high density gases (about 30 times that for atmospheric air) it overestimates the noise level by 2 to 4 decibels. For vacuum lines the mechanism is completely inoperative. For high vacuum systems calculations show the mechanical power in both the friction and in the velocity head are insufficient to account for the acoustical levels measured. But it has been observed that the following term will give results consistent with measured values:

$$W_m = 1.36\ (\rho/2g)\ (\pi\ d^2/4)\ V^3 \qquad (16)$$

where ρ is the density of atmospheric air in lb/cu ft and g is the gravitation constant. This mechanism would suggest that the atmosphere furnishes the power for the noise in vacuum lines.

Pipe Line For Liquids. Because of lower velocities, noise from control valves and pipe lines carrying liquids are of lower intensity and frequency compared to lines carrying gases. Control valve noise for lines carrying water may be approximated[10] by

$$SPL = 38.5 \log V + 53 - 22 \log (d_{pipe}/d_{port}) \qquad (17)$$

where

SPL = sound pressure level at pipe wall

V = fluid velocity at valve port, ft./sec.

We can check Equation 17 against the previous mechanism proposed for flow through pipes by starting with Equation 3 as follows:

$$SPL = PWL - 10 \log S$$

where

$$S = 4\pi r^2$$

Also

$$PWL = 10 \log \frac{10^{13} \eta\ W_m}{4\pi r^2}$$

Applying Equation 16 for W_m and separating terms gives

$$PWL = 130 + 10 \log \left[\left(\frac{\eta}{4\pi r^2}\right)\left(\frac{1.36\rho}{2g}\right)\left(\frac{\pi d^2}{4}\right) V^3\right]$$

where

d = port diameter, ft.

V = fluid velocity through port, ft./sec.

ρ = water density, 62.4 lb./cu. ft.

r = distance to port; i.e., $d_{pipe}/2$, ft.

g = gravitational constant, 32.2

This will give units equal to Equation 17 if

$$\eta = 6 \times 10^{-8} V^{0.85}$$

and a coefficient of 20, instead of 22, is used for the last term of Equation 17.

The result is the SPL at the inside of the control valve and must be corrected both for distance from and transmission loss through the valve wall. A Strouhal number of 0.05 based on port diameter and velocity can be used to calculate the wall loss and the spectrum external to the pipe can be approximated similar to that for gas flow. The noise produced by liquid flow in pipes can generally be neglected. If an estimate is required an acoustic efficiency of $6 \times 10^{-8}\ V^{0.85}$ can be used with the mechanical source taken as the friction drop.

Rotating Blades. The action of rotating solid members in producing noise is second in importance only to jet noise. The subject is complex since there are several noise sources[15]: thickness noise (blade thickness), thrust noise, vortex shedding noise, wake noise and noise generated by the interaction with solid stationary surfaces. In general the PWL generated is a function of a typical velocity to a high power (5 to 10) and the spectrum generated is generally a function of the rotating speed.

Fan noise can be estimated by the following formula which is based on reported propeller noise data.[4]

$$PWL = 116.5 + 0.027\ U + 10 \log (HP/25\ B)^{1.83} + 10 \log (B/3) \qquad (18)$$

where

U = tip speed, ft./sec. ($100 < U < 700$)

HP = horsepower

B = number of blades

The principal frequency with the greatest noise power is estimated by

$$f_o = B\ (RPM)/60$$

The octave containing this frequency has a noise power of about 3 dbs less than the over-all noise level. The noise power decreases about 8 dbs per octave for frequencies below the principal frequency and decreases about 4 dbs per octave for frequencies above the principal frequency.

Air coolers can be estimated by using Equation 18. To select fans for minimum noise, apply the fan equations of

$$q \propto D^2 UB$$

$$P \propto U^2$$

$$HP \propto QP$$

where

q = air volume per unit time

D = blade diameter

P = static pressure

Another source[1] gives the following:

$$PWL = 100 + 10 \log HP \qquad (19)$$

with a correction added of 10 log P when P exceeds unity as measured in inches of water.

For cooling towers, Koppers Co., Inc. gives the following procedure:

Step 1. Compute over-all acoustical power by

TABLE 3—Cooling Tower Noise Corrections

Octave, Hz	Octave Band Correction, dbs	DIRECTIVITY CORRECTION, DBS — For Outlet at Given Angle from Vertical: 0°	45°	60°	90°	135°	180°	For Inlet
37.5/75	−5	+3	0	−1	−5	−5	−5	0
75/150	−6	+4	0	−3	−9	−9	−9	−3
150/300	−8	+5	+1	−5	−10	−12	−12	−6
300/600	−10	+5	+3	−5	−11	−14	−14	−4
600/1,200	−12	+5	+5	−5	−12	−16	−16	+5
1,200/2,400	−17	+5	+5	−5	−13	−18	−18	+5
2,400/4,800	−21	+5	+5	−5	−14	−20	−20	+5
4,800/9,600	−25	+5	+5	−5	−14	−21	−21	+5

$$PWL = 105 + 10 \log HP \qquad (20)$$

Step 2. Apply octave band correction as shown in one of the columns of Table 3 for propeller fans.

Step 3. Compute fan frequency by

$$f_e = B\,(RPM)/60$$

and add 5 dbs for the octave containing this frequency.

Step 4. Apply directivity corrections as shown in Table 3 which are based on a cooling tower having a 64-square foot outlet.

Motors. Sources of noise in electric motors are mechanical (bearings and rotors), aerodynamic, and magnetic. For fan cooled motors the major noise source is the fan and preceeding formulas may be used to estimate the noise level. Induction motor design allows 100 to 150 cubic feet per minute of cooling air per kilowatt of loss which varies from 10 percent for motors about 10 hp to 6 percent for motors exceeding 1,000 hp.[3]

Uniquestionably the most accurate way to obtain motor noise is from manufacturer's data. Motor noise levels are a function of horsepower, speed and type of construction.

Table 4 is a condensation of one manufacturer's noise levels giving maximum *SPL* at 3 feet from the motor. Totally enclosed fan-cooled motors give levels 2 to 3 dbs above those for weather protected motors. Special enclosures for higher horsepower motors can reduce the noise levels significantly.

Compressors. Modern hydrocarbon processing plants frequently have large compressor installations which are major noise producers. For centrifugal and axial flow compressors the noise sources are turbulence, separated flow and unsteady flow over the vanes. For reciprocating compressors the source is turbulence, pressure fluctuation and non-uniform mechanical operation.

For centrifugal compressors the following formula are recommended:[4]

$$PWL = 20 \log HP + 81 + 50 \log (U/800) \qquad (21)$$

with the reading being taken at the exit piping.

The frequency of maximum noise production is taken as

$$f_o = 1.25\,U \qquad (22)$$

The *SPL* in the octave range containing f_o is taken as 4 decibels less than the over-all *SPL* and the slope is taken as 3 decibels per octave above and below the maximum noise octave.

In calculating the *SPL* using Equation 21, allowance must be made for the transmission loss through the pipe wall at the exit piping.

For axial compressors, the following is a procedure[4] for estimating noise:

$$\text{Over-all } PWL = 76 + 20 \log HP \qquad (23)$$

The frequency of maximum noise output is the second harmonic, or

$$f_o = 2\,B\,(RPM)/60 \qquad (24)$$

The spectrum is obtained from the following equations: For the 37.5-75 Hz octave

$$PWL = 85 + 10 \log HP \qquad (25)$$

For the 300-600 Hz octave

$$PWL = 80 + 13.5 \log HP \qquad (26)$$

For the octave containing f_o

$$PWL = 74 + 20 \log HP \qquad (27)$$

For the octave containing f_h

$$PWL = 80 + 13.5 \log HP \qquad (28)$$

where

$$f_h = f_o^2/400 \qquad (29)$$

TABLE 4—Approximate Noise Levels for Electric Motors

Horsepower	Enclosure*	Speed, RPM	SPL FOR GIVEN OCTAVE: 37.5—75	75—150	150—300	300—600	600—1,200	1,200—2,400	2,400—4,800	4,800—9,600
20 To 50........	TEFC	3,600	72	72	76	84	80	82	78	70
20 To 50........	TEFC	1,800	67	63	70	75	76	70	64	58
60 To 100.......	TEFC	3,600	70	78	83	88	90	87	82	74
60 To 100.......	TEFC	1,800	72	69	70	79	81	77	70	66
125 To 250.......	TEFC	3,600	82	81	87	93	93	90	86	76
125 To 250.......	TEFC	1,800	74	71	70	86	85	80	75	70
300 To 500.......	WP	3,600	83	87	90	88	85	84	80	72
300 To 500.......	WP	1,800	80	83	78	75	70	66	62	59
600 To 1,500......	WP	3,600	89	91	90	90	88	88	80	87
600 To 1,500......	WP	1,800	83	84	84	84	85	88	77	74
1,750 To 2,500......	WP	3,600	88	92	95	82	80	89	89	83
1,750 To 2,500......	WP	1,800	88	89	85	84	85	90	87	77
3,000 To 4,000......	WP	3,600	91	94	87	83	80	92	93	86
3,000 To 4,000......	WP	1,800	91	93	89	87	89	94	90	82

* Enclosure Types:
TEFC—totally enclosed fan-cooled.
WP—weather protected.

To plot the total spectrum, the SPL for the 37.5-75 Hz octave is connected to the SPL for the 300-600 Hz octave by a straight line. A smooth curve is drawn through the SPL values for the 300-600 Hz octave, the octave containing f_o and the octave containing f_h. Beyond the f_h octave, the slope is continued as a straight line.

For reciprocating compressors the following equation can be used as a rough approximation in the absence of published data.

$$\text{Over-all } PWL = 115 + 10 \log HP \tag{30}$$

The fundamental frequency of the machine is calculated and the PWL in that octave is taken as 4 decibels less than the over-all. The levels in higher and lower octaves decrease by 3 decibels per octave.[4] The reading is at the compressor and a correction must be made for the casing loss. At distances close to the compressor the surface for computing the SPL should be similar to the compressor exterior.

Turbines. Turbines, both gas and steam, are major sources of noise in process plants but have received little attention in published literature. The following empirical equations from limited data are offered for estimating turbine noise levels.

$$SPL = 58 + 10 \log HP \tag{31}$$

$$SPL = 82 + 10 \log G \tag{32}$$

where G Steam flow, lbs./sec.

The SPL for above equations are 3 feet from turbine casing.

For calculations involving enclosures where the PWL is required the following equation has been found useful

$$PWL = 109 + 10 \log G \tag{33}$$

and where a transmission lost through the casing (estimated as 28 dbs) has been included.

To estimate the spectrum of turbine noise a rough rule is to select the octave of maximum noise production as 1,200 to 2,400 Hz for turbines operating below 9,000 rpm, and the 2,400 to 4,800 octave for turbines operating above 9,000 rpm. The SPL in the maximum frequency octave may be taken as 4 decibels below the over-all SPL with a slope of 3 decibels per octave above and below the maximum frequency octave.

Pumps. The low horsepower of pumps makes them individually minor noise sources but collectively they serve to raise the general noise level of the plant. In the absence of published data on pump noise the following formulas based on limited data are presented as rough guides.

$$SPL = 71 + 10 \log HP_H \,[1 - (E/2)] \tag{34}$$

where

HP_H = hydraulic horsepower

E = pump efficiency

SPL = sound power level 3 ft. from pump

A rough approximation for the PWL of noise in an enclosed area is

$$PWL = 97 + 10 \log HP_H \,[1 - (E/2)] \tag{35}$$

Casing transmission loss is included in the foregoing equations.

The noise spectrum for pumps is flat and as a rough estimate for pumps below 500 hp one may take an equal noise level in each of the octaves between 150 and 2,400 Hz (7 dbs below the over-all) with a decrease of 6 dbs per octave above and below this region. For pumps 500 hp and above the SPL in the octaves 150 to 300, 300 to 600 and 600 to 1,200 may be taken as 5 dbs below the over-all with a decrease of 6 dbs per octave above and below this region.

Furnaces. Furnace noise represents a combination of several noise producing mechanisms: first, the noise produced by the entering fuel gas which represents a critical drop if above 15 psig; second, the noise produced by the intake of primary and secondary air; third, the noise produced by the combustion process.

To calculate the fuel gas noise the procedure for estimating control valve noise is used. For burners having a high fuel gas pressure this is the dominant noise source. For burners using fuel oil this noise source is negligible. Low pressure drop flow into a furnace is calculated as a vent.

To calculate the noise 3 feet from the burners and produced by the flow of primary and secondary air the following formulas are recommended.

$$SPL = 10 \log V^{4.4} + 10 \log G^{1.7} - 15 \tag{36}$$

$$PWL = 10 \log V^{4.4} + 10 \log G^{1.7} + 6 \tag{37}$$

where

V = air velocity through register, ft./sec.

G = air flow, lb./min.

To estimate the octave of maximum noise a Strouhal number of one is used, i.e.,

$$f_o d/V = 1$$

where d is the smallest dimension of the air opening. The SPL in this octave is taken as 3 dbs below the over-all with a slope of 5 dbs per octave above and below the octave of maximum noise production.

Burner noise must be estimated octave by octave since the fuel gas noise is high frequency noise and the air intake noise is low frequency. Having obtained the noise for the individual burner, the furnace noise is obtained by summing the noise of all the burners.

Combustion noise is not as significant as that produced by air and gas flow. A complete analysis is given in the literature.[17] As a condensation of this analysis we may use

$$W = 1055\eta\, GH \tag{38}$$

where

W = acoustical power, watts

η = acoustical efficiency, use 10^{-6}

G = flow rate, lb./sec.

H = heating value, Btu/lb.

The octave of maximum noise production may be estimated as 300 to 600 Hz and the SPL in this octave may be taken a 3 dbs below the over-all with a slope of 6 dbs per octave above and below this octave.

TABLE 5—California Noise Control Safety Orders

Freq. Band, Hz	Octave Band SPL, dbs
20/75	110
75/150	102
150/300	97
300/600	95
600/1,200	95
1,200/2,400	95
2,400/4,800	95
4,800/10,000	95

TABLE 6—Noise Attenuation in Air

Octave, Hz	Air Absorption, dbs/1,000 ft.
37.5/75	0
75/150	0
150/300	0
300/600	0
600/1,200	1
1,200/2,400	2
2,400/4,800	6
4,800/9,600	20

Mechanical Noise. Under this classification is included noise generated by the contact of metallic surfaces and the vibration of pipes and structures. The magnitude of these noise producers is essentially a function of mechanical design. Careful machining, adequate plate thickness and proper supports can lower these noise sources to levels below those previously considered. Because of this dependence upon factors outside of the mechanical energy available for transformation into acoustical energy no attempt will be made to estimate their magnitudes by formal means. We cannot, however, fail to consider gearing. Larger gear boxes in compressor trains can be designed by special order for quiet operation. However, for the usual process plant installations field measurements indicate readings in the vicinity of gear boxes are generally 2 to 3 decibels higher than the readings taken at the driver or compressor.

Noise Specifications

The procedures presented herein have been applied with success in estimating generated noise levels of hydrocarbon processing plants. The estimates have provided a basis for required silencing to meet specifications.

Typical of noise specifications is the following excerpt from the California Noise Control Safety Orders:

> "If an employe is exposed to noise for five or more hours per normal workday, the levels shown (Table 5) are the levels at and above which the wearing of hearing protectors is mandatory. For employes whose exposure to occupational noise is less than five hours per day, the noise levels may be three decibels higher for each halving of exposure time."

We note the following:

- Longer exposure time lowers permissible noise levels.
- Higher frequencies lower permissible noise levels.
- The specifications are intended to avoid hearing damage. Speech interference or psychological effects are not considered.
- The wording is legal. The failure of an employe to wear hearing protectors may not relieve the employer of responsibility for employe hearing loss and may not bar an employe from receiving workman's compensation. Indeed, insurance and design specifications make no reference to hearing protectors.

Noise and the Atmosphere. For noise radiation to distant points one should include the effect of atmospheric attenuation. One authority[15] gives attenuations in units of decibels per 1,000 feet as shown in Table 6.

Wind and temperature gradients are unimportant within the plant but for communities at a distance they can increase the expected noise levels. These effects are discussed in the literature.[1]

Ground Reflection. In plant measurements the effect of ground reflection is of importance. For enclosed areas this will be considered in the evaluation of the room constant. For open areas near the ground one method is to consider the sound propogation as hemispherical (i.e., $S = 2\pi r^2$) which is equivalent to an increase of 3 decibels over radiation in free space.

About the author

IRVING HEITNER is a senior analytical engineer (Systems Division) for The M. W. Kellogg Co., New York assigned to special projects. He holds a bachelor degree in chemical engineering from City College of New York and a master degree in mechanical engineering from University of Pennsylvania. His past experience includes eight years as a planning engineer for Celanese Corp. of America, and seven, years as process, project and equipment engineer for Scientific Design Co. He has a professional engineering license from the State of New York.

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