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Final Report

# STUDY OF NOISE-LEVEL REDUCTION OF AIRCRAFT GROUND-SUPPORT EQUIPMENT

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Section II: A Review of Nuisance and Hazardous Noise.
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Section III: Experimental Noise Measurements and Noise Reduction.
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Section V: Noise-Reduction Technology Applicable to Aircraft Ground-Support Equipment.
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# ABSTRACT

The program was carried out in several distinct but related, concurrent phases: correlation of nuisance and hazard noise criteria; measurement of the noise characteristic of several types of ground-support equipment; experimental studies to quiet several ground-equipment units; analytical and experimental study of exhaust-muffler attenuation characteristics; and compilation of noise-reduction techniques applicable for quieting ground-support equipment. The results include: a basis for ranking noises of different intensities and frequency composition according to (1) the seriousness of the psychologically disturbing or physically harmful effects which they produce, and (2) the extent to which they mask or interfere with speech communication; information about the sound-pressure levels and the octave-band frequency distribution of the noise of certain typical ground-support units; a demonstration of the application of practical noise-reduction techniques to ground-support equipment; a consideration of the attenuation characteristics of acoustic filter configurations, an examination of the correlation of computed and measured attenuations, and an evaluation of an experimental muffler design; and a compilation of noise-reduction techniques, methods, and measurements.

# OBJECTIVE

The work reported here was directed toward devising effective methods (1) of reducing the potentially disturbing and harmful high radiated noise levels of aircraft ground-support equipment now in service, and (2) toward the establishment of noise-reduction criteria to aid in the future development of this equipment.

# SECTION I

## INTRODUCTION

Present types of aircraft ground-support equipment radiate noise of an intensity which can impair the performance of operating personnel by interfering with speech communication and by creating a psychologically disturbing nuisance condition. Further, the high-level sounds are potentially hazardous to the hearing of individuals subjected to it for long and repeated intervals. Thus the motivation for the research studies described in this report is obvious.

Under the U. S. Air Force Contract No. AF 33(616)-2856, Task No. 60151, members of the staff of The University of Michigan Engineering Research Institute have carried out a program of studies which, consistent with the motivation cited above and with the intent of the original research proposal (see Appendix A), were directed toward reducing the noise radiated by ground-support equipment. The scope of these studies is indicated by an enumeration of the tasks and results described in the several sections of this report:

Section II—A Review of Nuisance and Hazardous Noise.
A survey of literature pertaining to the psychological and physiological effects of noise.

Section II—Experimental Noise Measurement and Noise Reduction.
A series of measurements to determine the noise produced by several types of ground-support equipment and the demonstration of the applicability of palliative noise-reduction techniques to ground-support-equipment noise problems.

Section IV—Exhaust-Muffler Study.
An analytical review and experimental study of exhaust-muffler attenuation characteristics.

Section V—Noise-Reduction Technology Applicable to Aircraft Ground-Support Equipment.
A classification of pertinent information from an existing noise-reduction manual.

It is apparent from this outline that all these studies are related, and could probably be best undertaken in a given sequential order. However, because of the short time of performance permitted under the original contract time period, all studies were begun and carried out concurrently. The original contract period extended from 15 February 1955 to 15 November 1955, but subsequently the time of performance was extended to 28 February 1957, although experimental work was terminated 30 June 1956.

The results of the studies outlined above are presented in this report in textual, graphical, and tabular form. Figures referred to in the text appear interspersed appropriately in the body of the report. Material worthy of inclusion, but not conveniently incorporated in the body of this report, is presented in three appendices: the original research proposal submitted to the Air Force; a description of the noise-measurement instrumentation and calibration procedures; and a tabular presentation of the noise data accumulated in the experiment studies.

## SECTION II

## A REVIEW OF NUISANCE AND HAZARDOUS NOISE

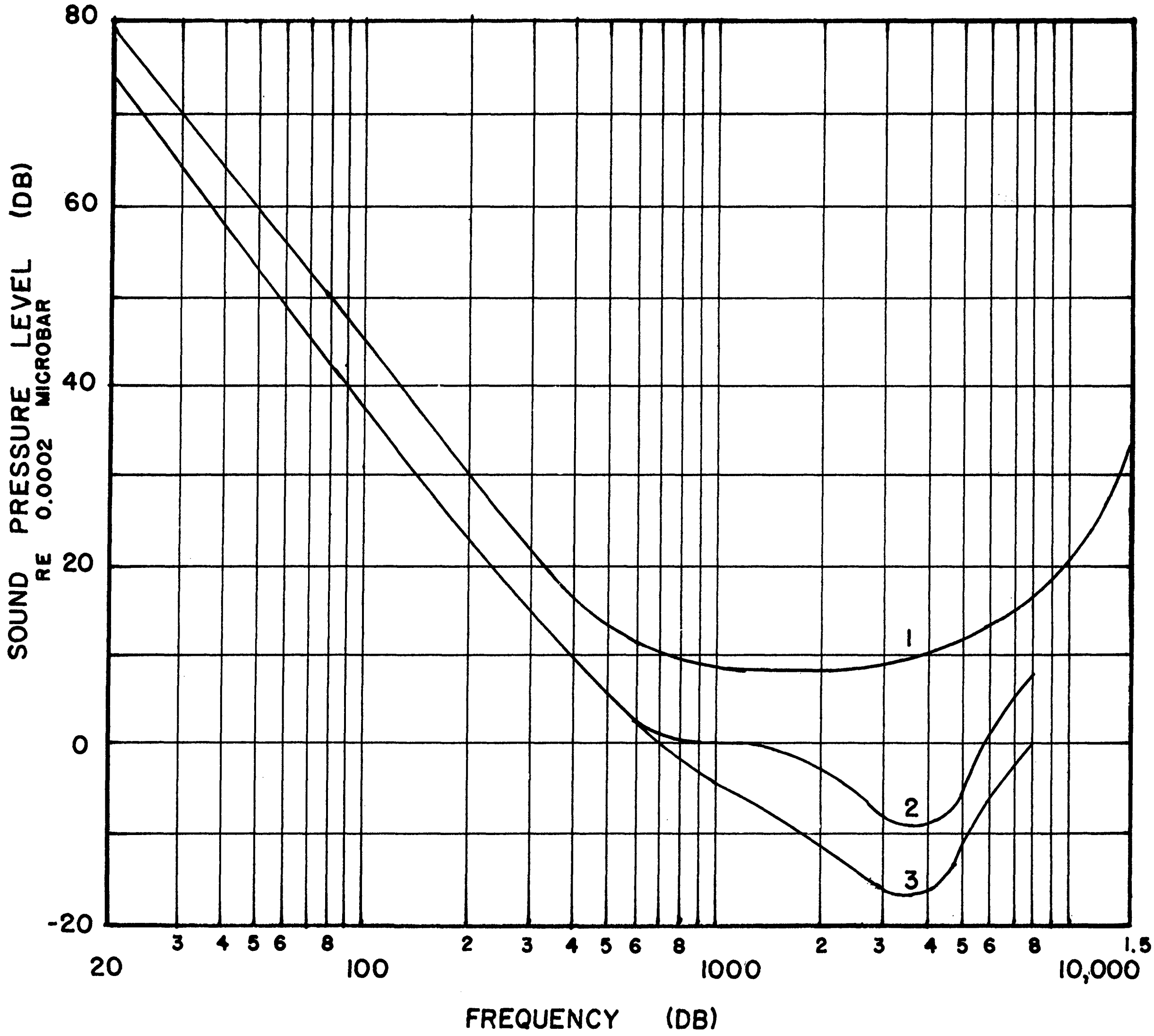
### INTRODUCTION

In a literature review pertaining to nuisance and hazardous noise levels, it is first necessary to define what is meant by nuisance noise and what may be categorized as hazardous noise. As may be expected, there is no clear-cut separation between these two terms. Indeed, the transition is a gradual one with parameters of sound-level spectrum, individual acuity, exposure time, and the like.

In a general sense, nuisance noise may be characterized as noise encompassing a spectrum and range of levels which is irritating and inconveniencing but which produces no serious auditory effect over an extended period of time. This is the usual case of noises of 85 db (re 0.0002 dynes/cm$^2$) and below. On the other hand, hazardous noises are usually of a level greater than 85 db and produce either auditory or nonauditory physiological reactions or both, depending on their character. It should be remembered, however, that the labeling of a sound as merely nuisance noise might be severely questioned by personnel losing a limb due to a masking of the communication channel by a low-level noise. A more quantitative description of a noise relative to hazard and nuisance can be obtained by categorizing the spectrum content and level of the noise with respect to its effect upon the exposed individual. For this reason, this literature review has dealt with the problem of environmental noise and its ensuing effects upon man through a discussion of the following topics: masking effects, temporary and permanent hearing loss (or threshold shift), annoyance, the use of protective devices, and nonauditory levels, and effect of high-level noise on the individual. To provide a basis for better understanding of the subjects which will be discussed, some general basic information on the reaction of the human ear to various sounds follows.

Audible Range.—The audible frequency region is usually considered to be from 50 cps to 15,000 cps, although the limits vary from person to person, and, for a particular individual, depend on time and environment. The sensitivity of the ear is remarkable since in the vicinity of 3000 cps it responds to sound pressures as low as $10^{-4}$ dynes/cm$^2$ (which is -6 db re 0.0002 dynes/cm$^2$). On the other hand, a pressure of 300 dynes/cm$^2$ (or 120 db) corresponds to a noise level which results in only mild discomfort to the listener.

Threshold of Hearing.—For practical purposes it would be of interest to investigate the response of the ear to everyday sounds, that is, to sounds which are complex in character. A useful method of specifying the sensitivity of the ear to these sounds is to determine the lowest sound pressure that is detected by the ear, i.e., to determine the threshold of hearing. Although such determinations are complex, simplified measurement methods have resulted in the establishment of threshold-of-audibility curves as shown in Fig. 2.1.[1] The American Standard pure-tone curve shown is obtained by measuring the sound-pressure level in an anechoic chamber at a point in the center of the head position at a specified distance from the sound source with no




1. Monaural curve (Ref. 52)
2. Binaural curve (Ref. 53)
3. Binaural curve (Ref. 54)

Fig. 2.1. Threshold of audibility curves.

one in the room. The binaural response of the individual is then plotted in db versus frequency. Notice that this curve crosses the 0-db level at 1000 cps and that the most sensitive ear response lies between 1000-6000 cps. Curves obtained by two other threshold detection methods are also shown in Fig. 2.1. Since the speech frequencies are predominant in the middle range of the spectrum, audiometric tests are usually conducted in this range, i.e., between 100-8000 cps.

Threshold of Tolerance.—At the other extremity of level response lies the threshold of tolerance. This is a difficult term to define because of the reaction of the ear. Here, as well as at low sound-pressure levels, exposure of the ear to sound tends to elevate the threshold limits under examination. Attempts to set limits of discomfort, tickling sensations, and pain have shown that sound-pressure levels of pure tones adjusted to correspond to these limits and applied through earphones had to be increased as the experiment progressed. Figure 2.2 shows the results of several experiments along this line.[2] Because of this indefiniteness, the most that can be stated is that for pure tones there exists a threshold for tickling sensations and pain above 140 db while the discomfort threshold varies between 120-130 db. Any attempt to set up a criterion for establishing an overall standard must be tempered by both the physiological and psychological reactions of the human auditory system with respect to body functions, exposure time, sound spectrum, presentation, etc.

Pitch.—Pitch may be defined as the response of the ear which differentiates sounds as they appear on a musical scale, the unit of pitch being the mel. Pitch is not a simple functional response of the ear, and though related to frequency, pitch is also influenced by the spectrum and amplitude of sounds. That is, sounds of the same frequency composition but of different levels do not have the same pitch. As a result, the following arbitrary standard has been established: a 1000-cps tone of 60 db is said to be 1000 mels. Figure 2.3 gives the relation between the subjective pitch in mels versus frequency using a tone with a loudness level of 60 phons.[3]

Loudness.—Loudness, although primarily a function of sound pressure also depends on spectrum content, and is the auditory sensation attribute which scales a sound from "soft" to "loud." The unit of loudness is the sone, which by definition is the loudness of a 1000-cps pure tone at 40 db above the listener's threshold.[4]

Loudness Level.—Loudness level is measured in phons. By definition a phon is the sound-pressure level in db of a 1000-cps tone that seems as loud as the sound under test. Representative equal loudness-level contour curves are shown in Fig. 2.4 for both pure tones and bands of noise. The relationship between loudness in millisones and loudness level in phons is demonstrated graphically in Fig. 2.5.

## MASKING

The ability of the ear to hear two sounds at one time is a physiological phenomenon. However, this function, which provides for the discrimination between sounds, is subject to the effect of imposing one sound on another. This causes one of the sounds to become indistinct, or, as it may be said, auditory masking has occurred. Auditory masking is best defined as "the shift of the threshold of audibility of a masked sound due to the presence of a masking sound."[5] For the purposes of this report the masked sound will be limited to speech and the masking sounds will, therefore refer to any other sounds which may interfere with the speech signals from a communications standpoint. In practice the detection and reduction of masking sounds may be undertaken by either of two general methods. The first is one in which an attempt is made to reduce the entire noise spectrum proceeding from the largest single contributor to the overall noise level, until a favorable reduction of the over-

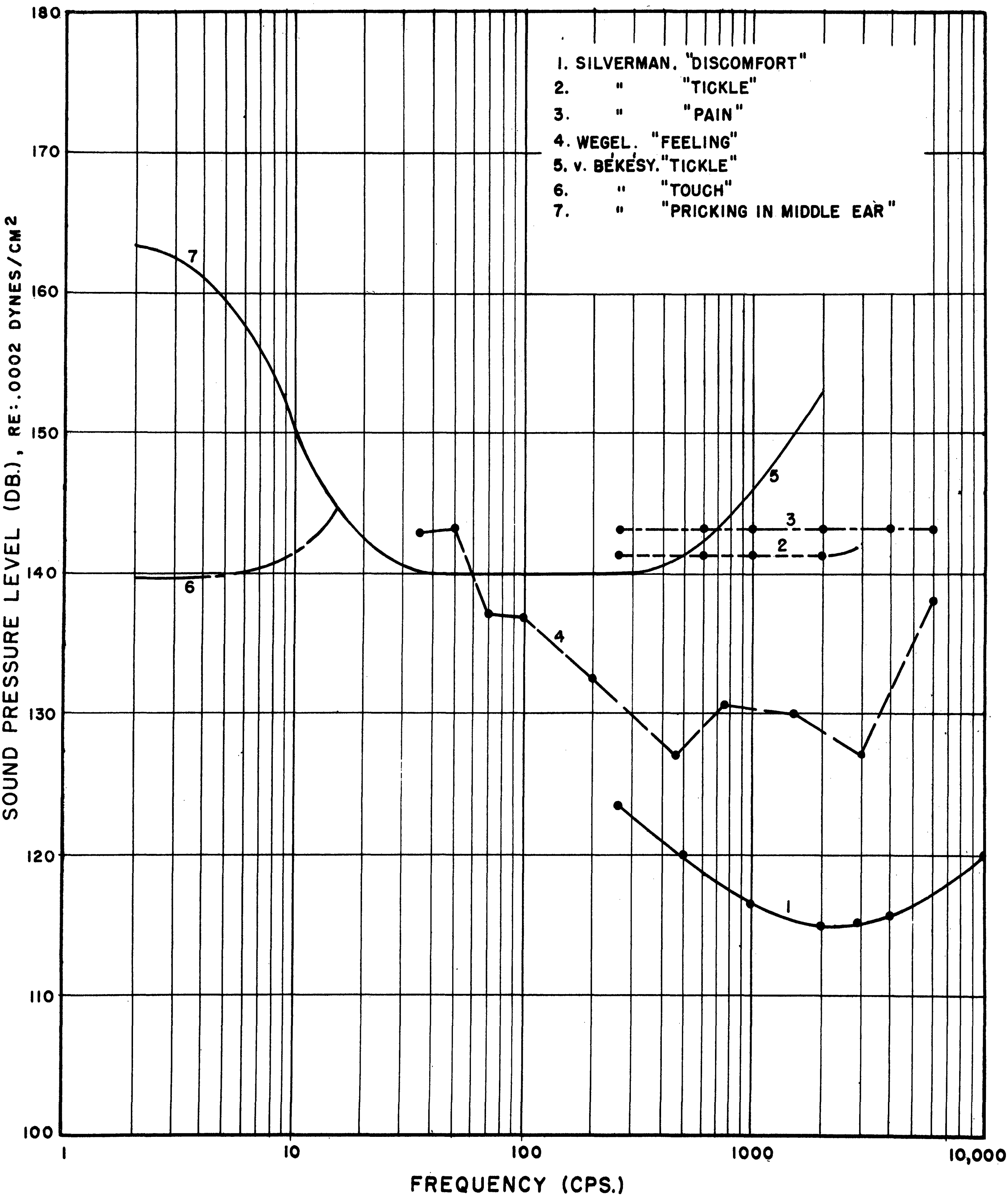



Fig. 2.2. Threshold of tolerance curves.
(Ref. 2)

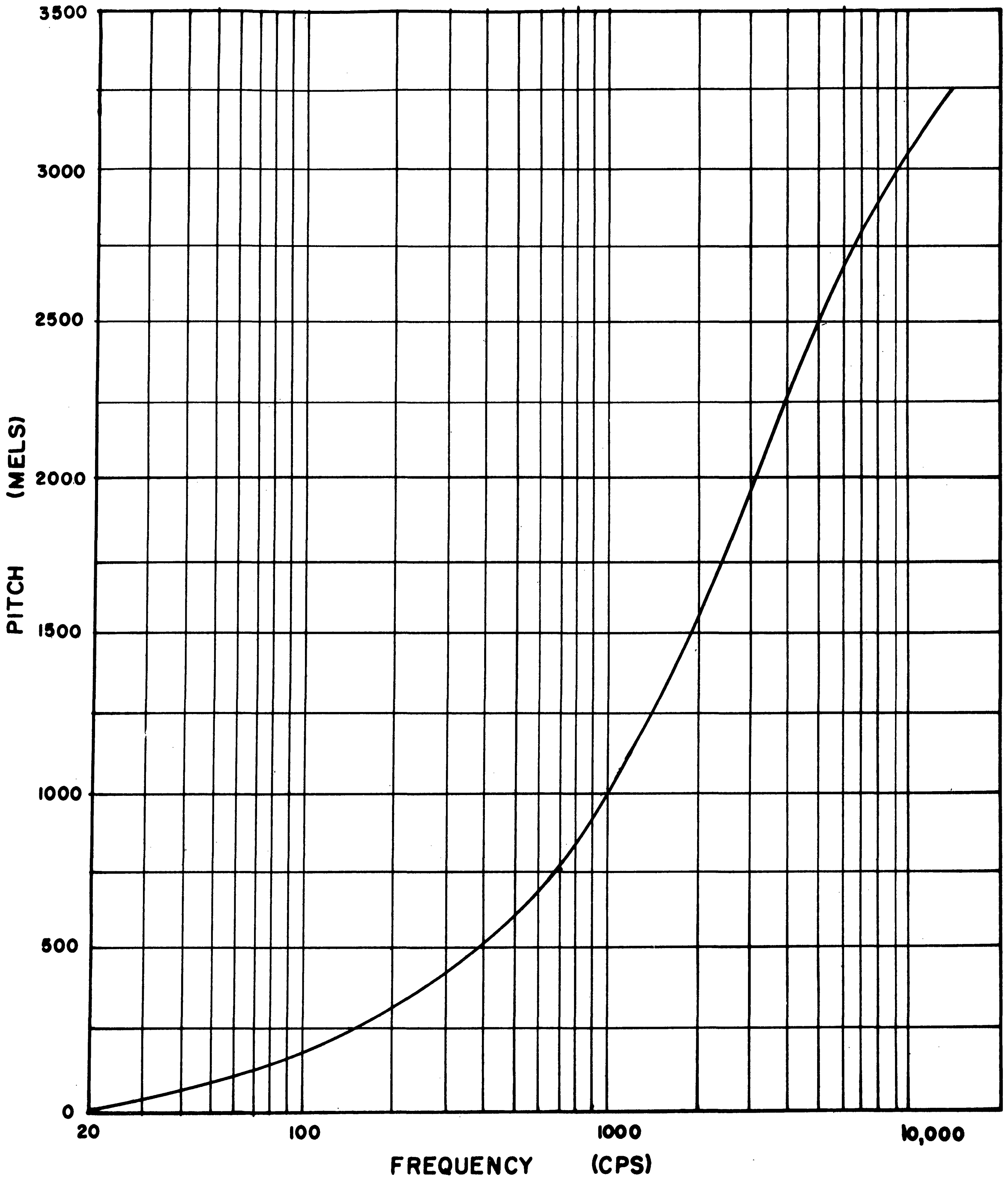



Fig. 2.3. Pitch as a function of frequency.

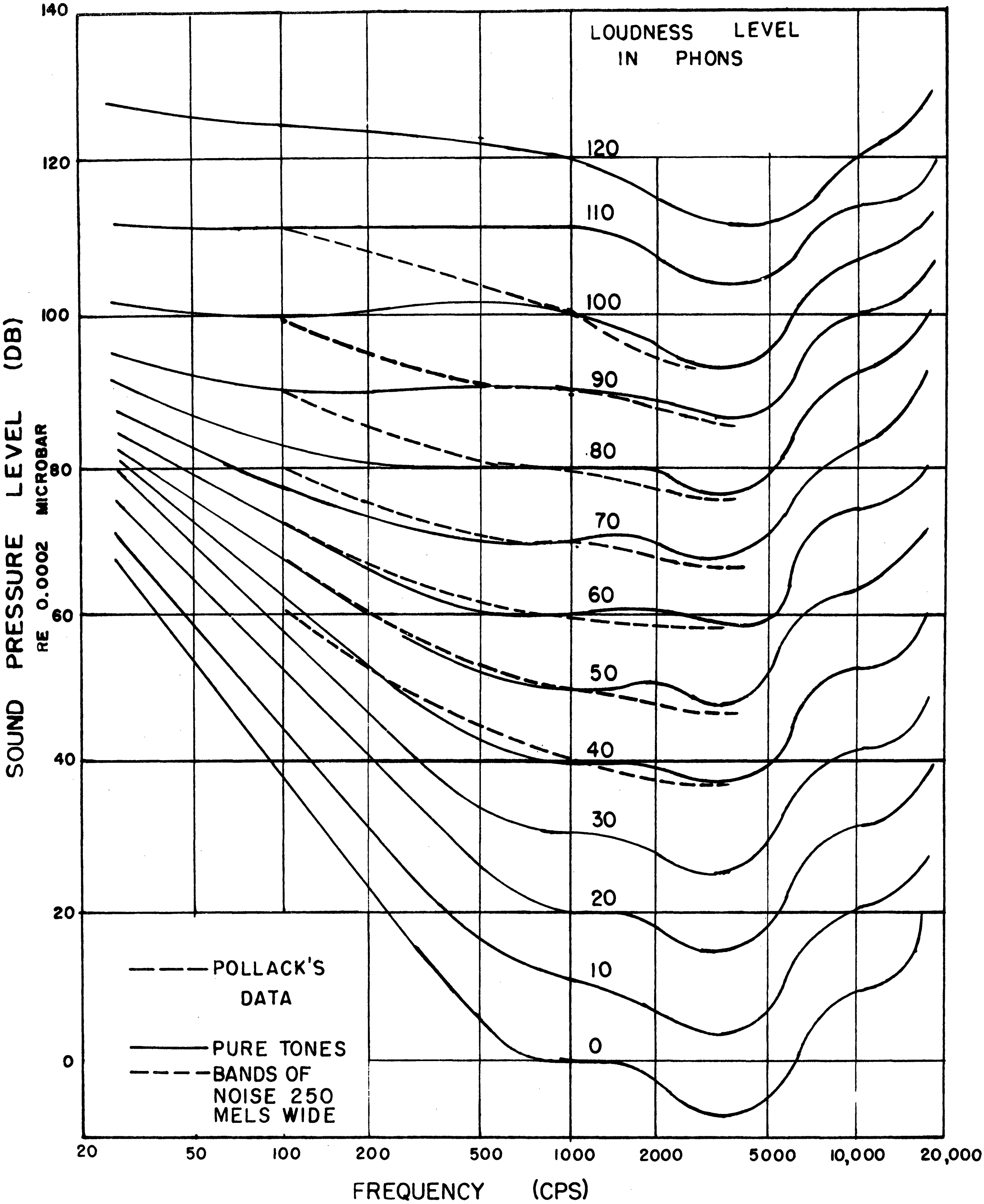



Fig. 2.4. Equal loudness level contours.
(Ref. 1)

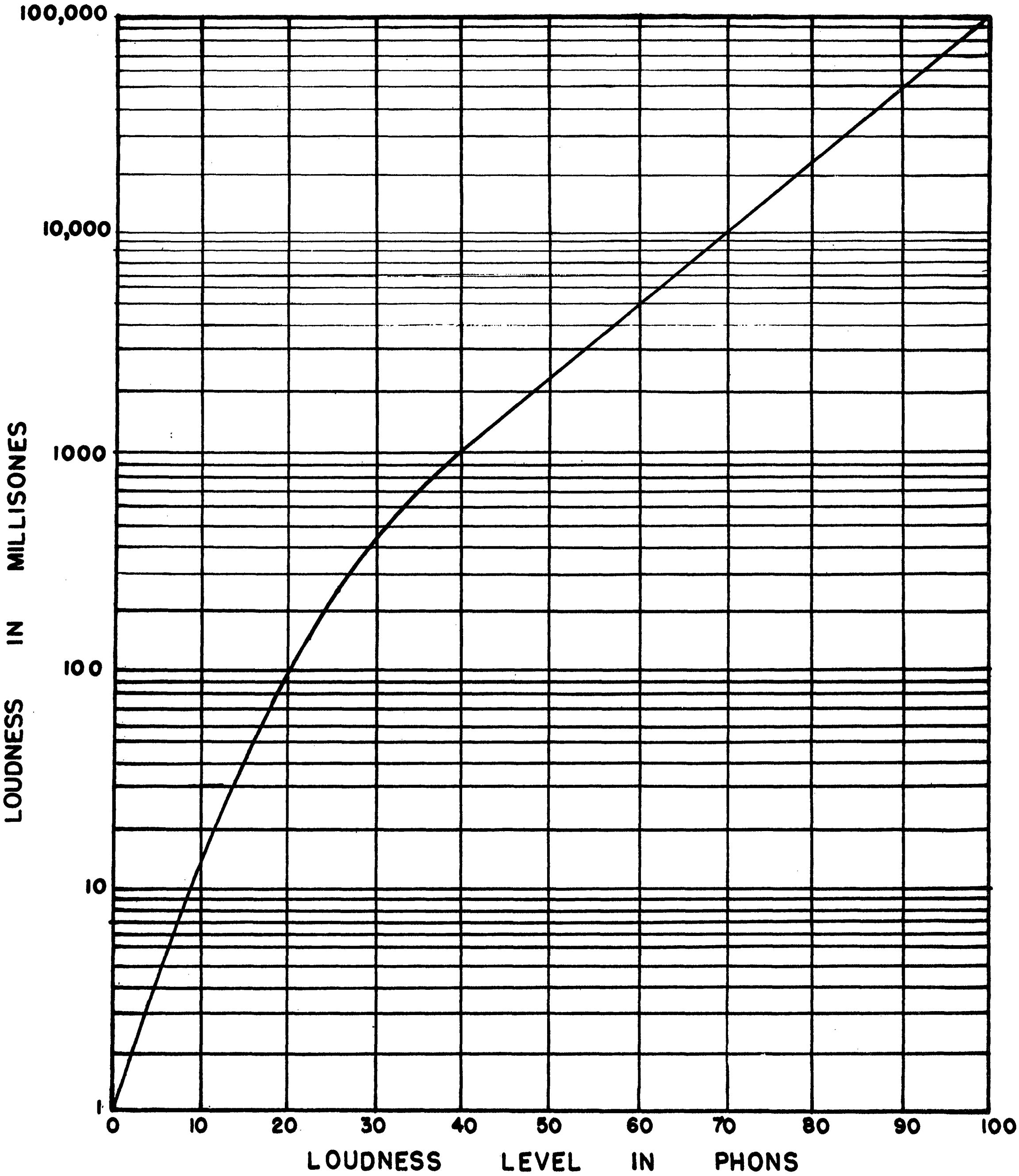



Fig. 2.5. Loudness vs. loudness level.

all noise level has been obtained. The second method of reduction is aimed at those specific masking sounds which perform the masking function with respect to the human speech spectrum.

In some cases of noise-reduction problems, treatment of the noise source using each of the two criteria mentioned might well result in two different final noise spectra, the main difference being that the reduction with respect to specific masking sounds would result in a preferential reduction of the sounds below 5000 cps. In this respect, the noise spectrum generated by most Air Force ground-support equipment encompasses that portion of the audible spectrum in which masking is of primary concern, and fortunately, regardless of which procedural reduction method is employed, the end result is a reduction of the masking properties of the source with respect to human speech.

Speech Spectrum.—To properly evaluate reduction of a masking sound, the speech spectrum should be studied and the effectiveness of the masking of the intrusive sound should be determined. Many different experiments have been undertaken with respect to the masking of speech, and rather than confuse the problem with the wide variety of experimental evidence available, an attempt will be made to use data of a few representative experiments, leaving to the reader the alternative of reading from supplementary references.

In general, the power distribution in the speech spectrum is concentrated, for the average male, between 100 and 5000 cps with the peak lying at approximately 500 cps. Figure 2.6 is the long interval speech spectrum of 7 male voices plotted in rms pressure for a bandwidth of 1 cycle.[6] This was obtained by utilizing a condenser microphone pickup located 18 inches in front of the lips and analyzing the resultant voltage with an audio spectrometer. The overall pressure at this point was 76 db with 1-cycle bandwidth variations from 47 to 16 db.

Filtering Function of the Ear.—Measurements made by Bekesy indicate that different portions of the basilar membrane produce different amplitudes of vibration when exposed to different frequencies.[7,8] This is effectively a filtering action in the electrical sense of the word and allows the ear-response functions to be described in terms of bandwidth levels and response. Indeed, the critical bandwidths have been defined as those bandwidths in which a pure tone can be detected in the presence of white, random noise. With the availability of data on thresholds of audibility and perceptibility and the knowledge of the bandwidth character of the ear, rather thorough investigations of masking have been undertaken.

In his investigation of masking,[6,9] Miller, et al., used the threshold of perceptibility as the identifying level. This threshold is the level at which the gist of continuous speech discourse can just be understood. Using this simple criterion, listeners were instructed to determine the threshold of perceptibility for speech when various sounds were used as maskers. Figure 2.7 shows the masking of speech by pure tones of 300 cps and 1000 cps, and by random noise. Notice that a pure tone of 1000 cps at 100 db raises the threshold 18 db, while a 300-cps tone of 100 db raises the threshold by 42 db, and random noise at 100 db raises the threshold by 68 db.

In an effort to obtain improved accuracy, articulation tests have been used. In this test, a speaker reads a series of words, or a recording of words is run off and the percent of words heard by the listener is called the articulation score, a 50-percent score being considered as the threshold level.

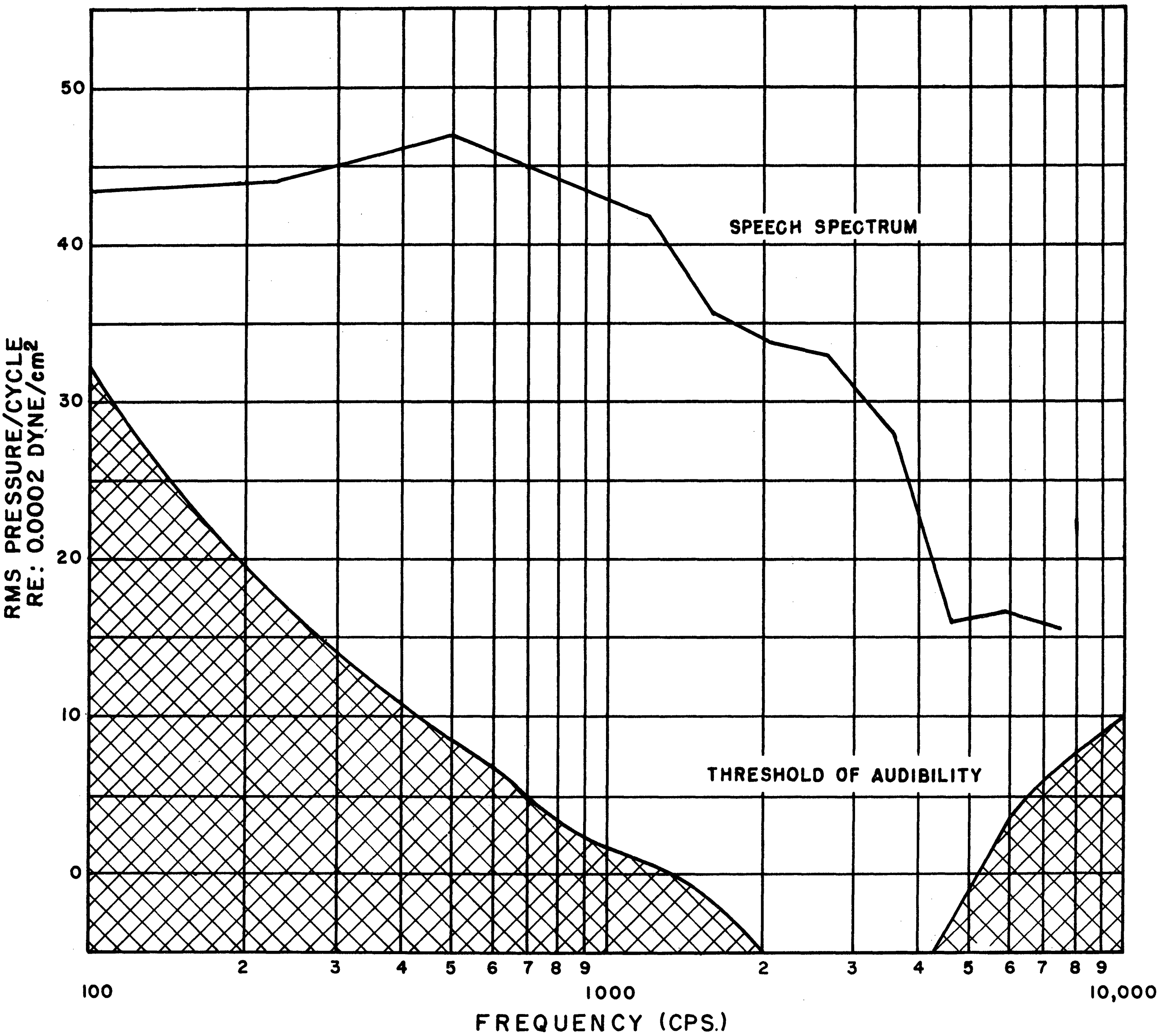



Fig. 2.6. The long-interval speech spectrum for seven male voices.
(Ref. 6)

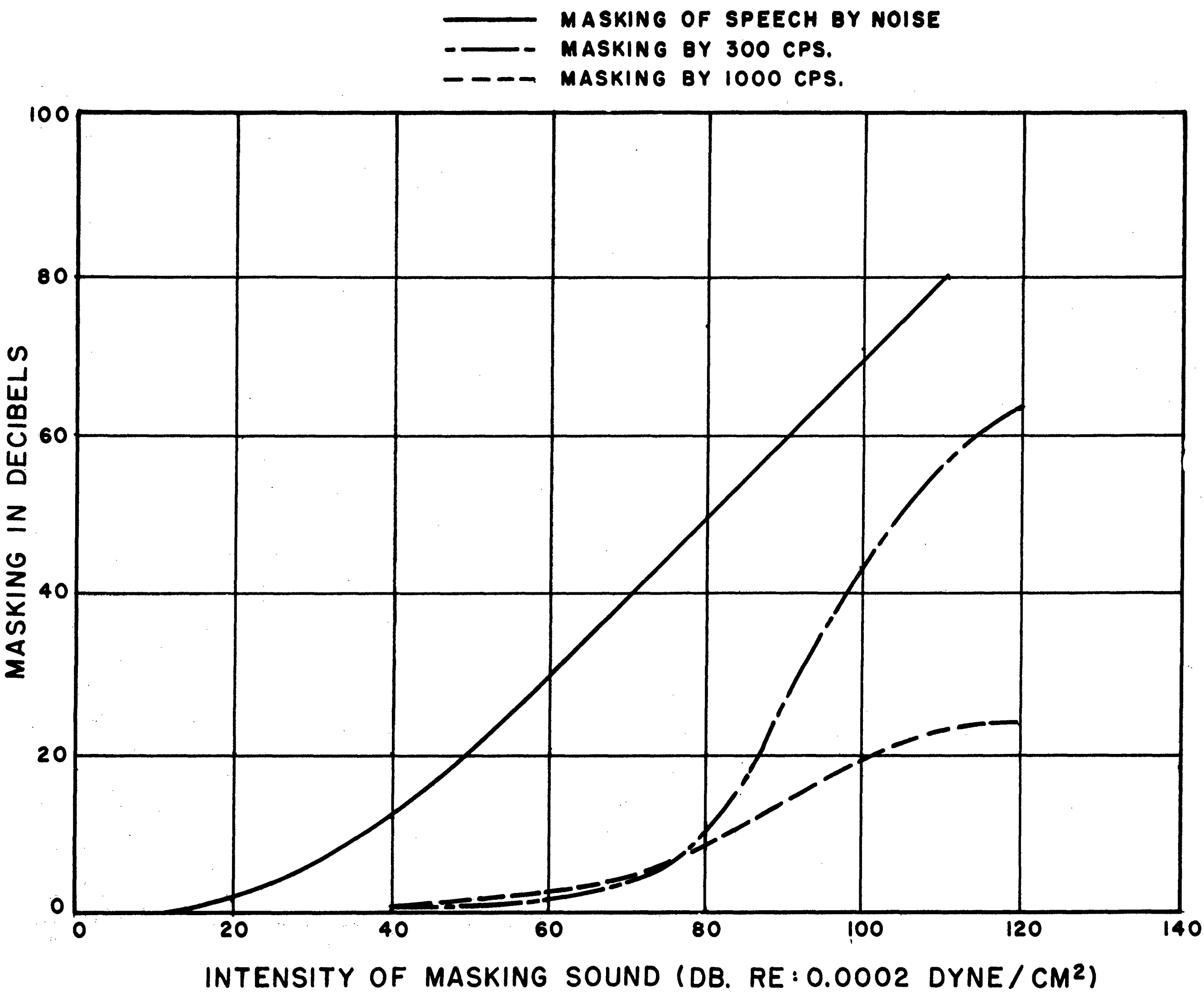



Fig. 2.7. The shift in the threshold of perceptibility for speech vs. the intensity of different masking sounds. (Ref. 6)

Masking of Speech by Pure Tones.—Pure-tone masking was studied quite thoroughly by Steven, Truscott, and Miller.[9] It was found that the greatest masking occurs around 500 cps by sine waves, but at high intensities the largest masking is encountered at about 300 cps. This is due to the function of the ear which results in the upward spreading effect of maskers on the frequency scale. This has been pointed out to be the effect of distortion by harmonics produced in the ear at high levels and by auditory nerve response characteristics.[10,11,12]

Complex Tones.—For practical purposes it is much more valuable to examine the available data on complex tones and noise. By using maskers composed of square waves, it was shown that waves with fundamentals of 80 and 400 cps each have the same approximate masking ability, and are therefore less critical to frequency change than sine waves. Furthermore, there is no apparent optimum masking frequency for square waves as the intensity level of the maskers is increased.

Using 10-microsecond pulses at different repetition-frequency rates, it was found that the greatest masking occurs at approximately 200 pulses per second. Correlation of the above data showed that for equal sound-pressure levels the pulsed masker is the most effective, and is 7 db more effective as a masker for speech than the square wave, which in turn is 7 db more effective than the sine wave.

Further experimentation with complex tones of high harmonic content was carried out, using a warble-tone relaxation oscillator in which the tone rose slowly from the lowest to the highest frequency, and then dropped quickly back. With the speech output maintained at 95 db, the warble tone was raised in 6-db steps under different experimental conditions. Figure 2.8A is a plot of warble-tone frequency band versus percent articulation. Notice that the lowest band of 170 to 220 cps is by far the most effective masker, showing 50-percent articulation at approximately 7 db above the speech level, whereas the 550-750 cps band showed 50-percent articulation at a level of 20 db above the speech output.[6]

Figure 2.8B shows the masking comparison of three warble tones of the same mid-frequency and the same warble rate, but of different bandwidth. There is little difference between the 400- to 500-cps warble and the 250- to 650-cps warble, but the 50- to 850-cps warble shows some decrease in masking ability. Obviously, this is due to the warble spread into the higher frequencies of smaller masking ability.

When the warble rate is varied between three and fifteen warbles per second, there is no significant difference in the masking quality of the warble. This is shown in Fig. 2.8C.

Another experiment was performed in which complex tones ranging in fundamental frequency from 300-600 cps were repeated in an irregular pattern. From Fig. 2.8D an increased masking of 10 db can be observed when a complex tone of 200 cps is added.[6]

From the experiments outlined above it may be observed that, for both pure and complex tones, the low tones are more effective in masking speech, since low-frequency sounds tend to mask upwards on the frequency scale as their intensity is increased, whereas high-frequency maskers do not effectively mask down scale.

Masking of Speech by Noise.—As was shown in Fig. 2.7, random noise is a more effective masker than a pure tone of the same sound-pressure level. Figure 2.9 shows the results of resolving noise into narrow bands and obtaining the masking

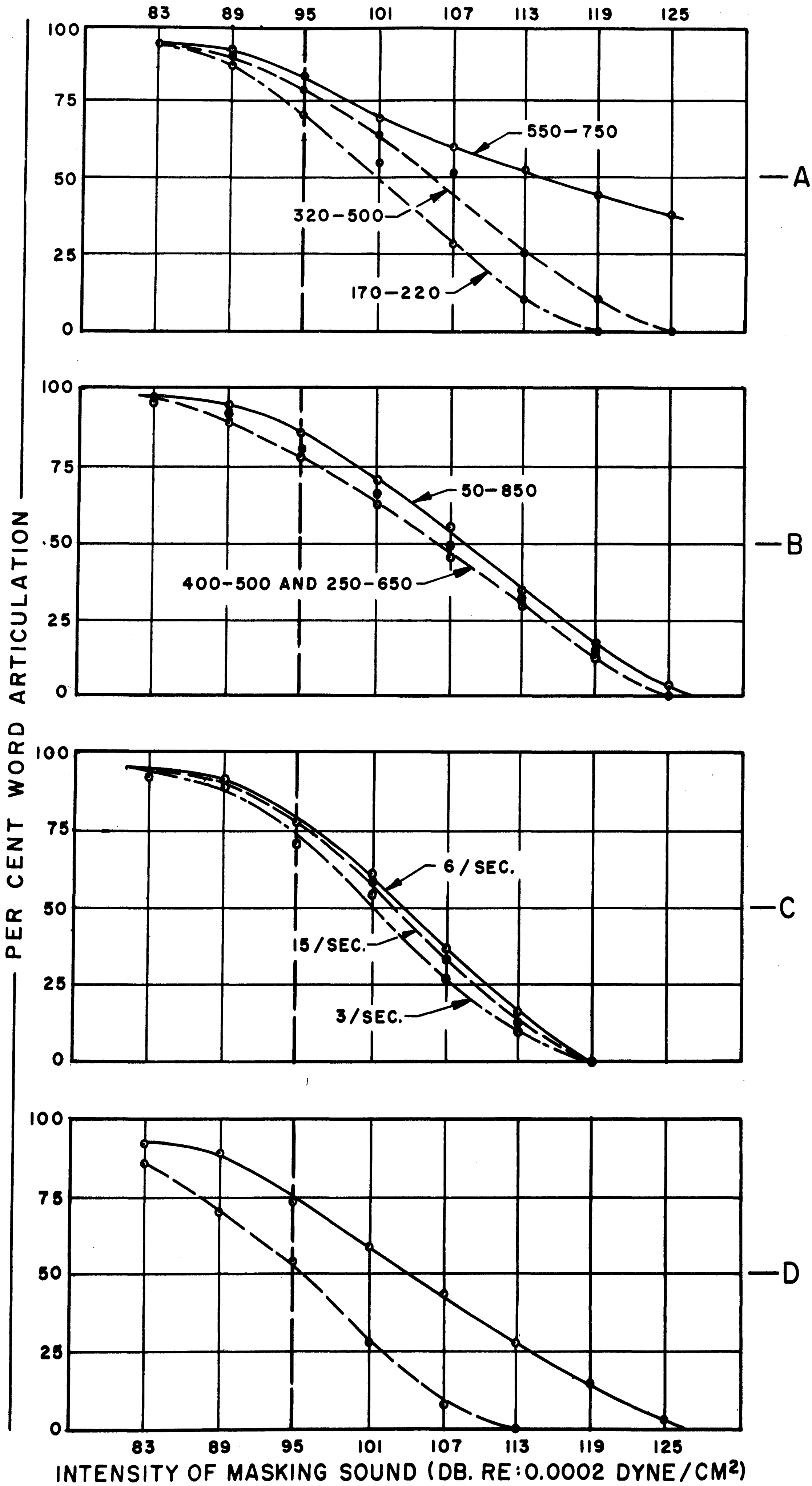



Fig. 2.8. Percent of words correctly heard as a function of the intensity of various tonal masking sounds. (Ref. 6)

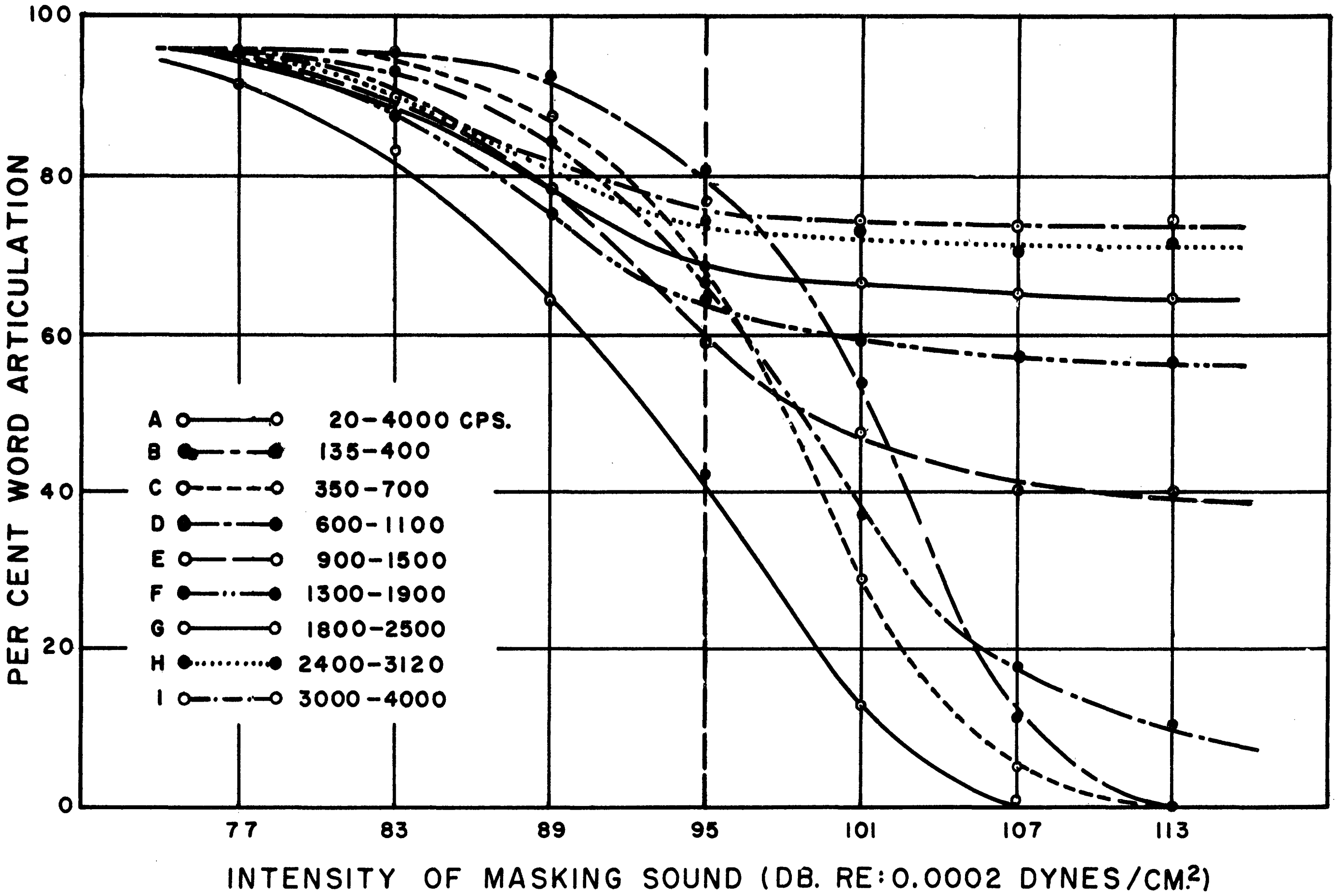



Fig. 2.9. The articulation score as a function of the intensity of the masking noise of various bandwidths on a speech level of 95 db. (Ref. 6)

ability of each band using a 96-db speech-level output. From this plot several facts are evident: (a) a wide band of noise covering the entire speech spectrum masks more effectively than any single noise band within the spectrum; (b) at low noise levels, high-frequency noises are better maskers than low-frequency noises; and (c) at high noise levels, low-frequency noises are better maskers than high-frequency noises. Result (a) is easily understood since it would be expected that the masking of speech would be much more effective by a sound whose frequency range covered the output spectrum of human speech. Results (b) and (c) are explained by two facts which have been previously observed. They are that only low frequencies of high-pressure levels tend to extend their masking into higher frequency ranges, and also that the speech spectrum shows a high output in the low-frequency regions and low output in the high-frequency regions. Thus, for low masking-noise levels, the high-frequency noise masks the upper frequencies of speech, whereas the low-frequency noise is not able to mask effectively the low frequency of speech. However, as the masking-noise intensity increases, the high-frequency noises continue to mask only the high speech frequencies, whereas the low-frequency noises now not only mask the low speech frequencies but also extend the masking into the higher frequencies.

Some other experiments concerning the masking properties of noise versus noise content include modulating a random noise by switching filters in and out, and by comparison of two noises having identical spectra but different wave forms.[6] In the first case the modulated noise did not change the articulation score obtained, and in the second case little variance in masking ability was noted between identical spectra of random noise and frequency-modulated noise.

Among other masking experiments of note, but which do not bear as directly upon the ground-support noise problems, are interaural phase variations in binaural perception of sounds and the interaural phase relations between speech and masking noise at the ear.[13,14]

<u>Time Variation of Noise in the Masking of Speech</u>.—Investigation of the temporal continuity of masking noise has revealed two interesting facts, that the masking function of the noise is related both to the amount of time the noise is present, and to the actual interruption rate of the noise. Figure 2.10 shows that a steady noise is a more effective masker than the same noise present at only a percentage of the steady-state condition. A noise which is on for 65 percent of the time must have a level of better than 120 db to reduce the articulation score to 50 percent with a 95-db speech level, whereas a continuous noise of only 95 db results in the same articulation score.[6]

The interruption rate of noise has also been studied; it has been determined that for noise interrupted at a rate of 1-100 times per second the masking effectiveness is substantially reduced from that obtained using a continuous masker, whereas interrupted noise at rates of 200 times per second and above shows a masking essentially equivalent to that of continuous noise.[15]

<u>Speech Interference Levels</u>.—Since articulation tests are a tedious undertaking and vary with sentence or word presentations, it was found possible to correlate the contribution of the speech spectrum by bands taking into account the speech-to-noise ratio in the band, and using weighting factors to give an articulation index.[16] This presents a computation problem and in itself is a significant task. Indeed, the problems of incorporating narrow band noise and repetitive sounds are yet to be evolved accurately. In lieu of the articulation index a more simplified standard has been suggested. This is the speech interference level. Since approximately 80 percent of the

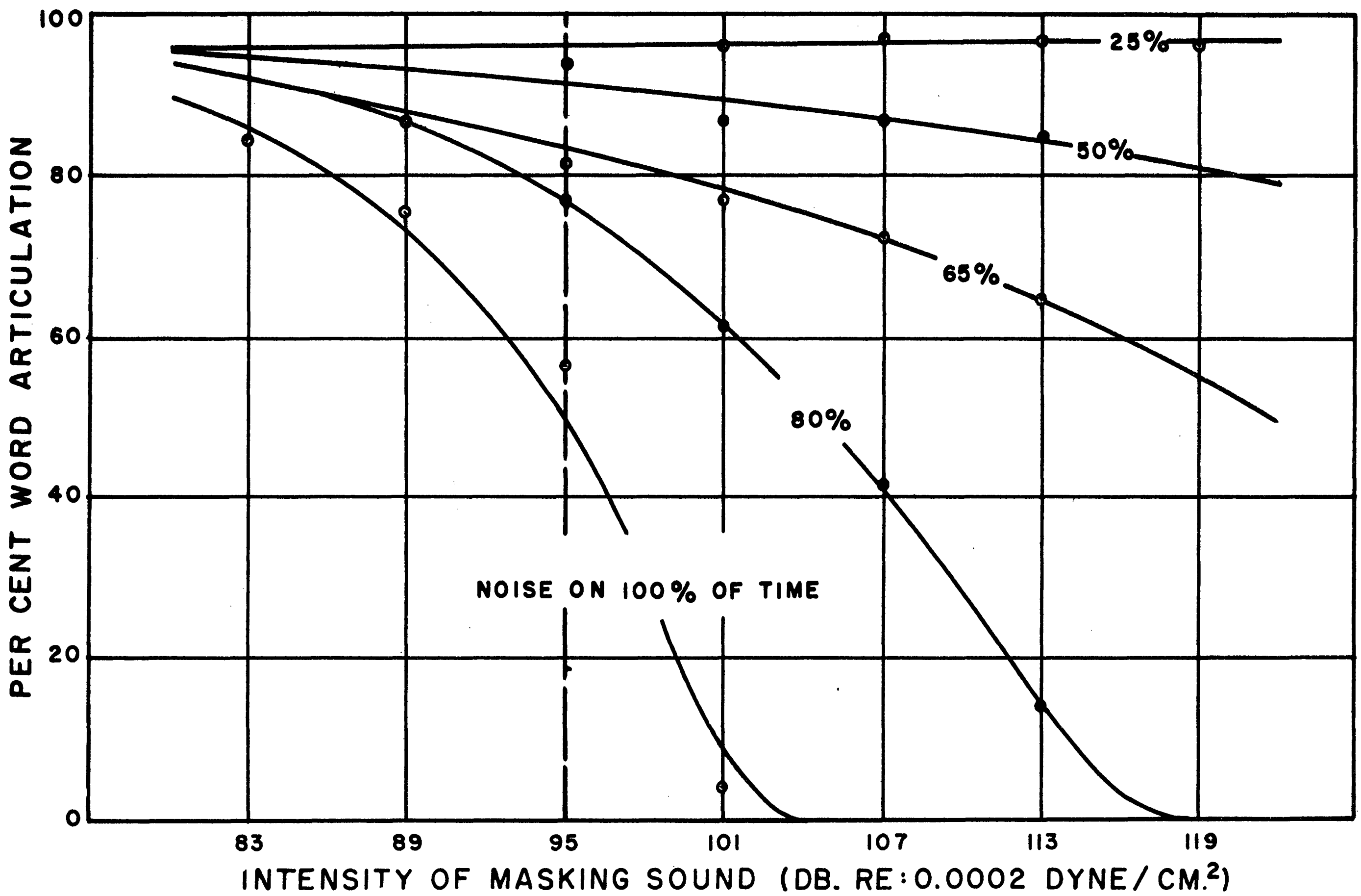



Fig. 2.10. The articulation score as a function of the intensity of interrupted masking noises and percent time of masking for a speech level of 95 db.
(Ref. 6)

most important range of speech frequencies is covered in the range of 600 to 4800 cps, this range has been divided into three octave bands of 600-1200, 1200-2400, and 2400-4800 cps, the speech interference level of a masking noise being defined as the arithmetic average of the sound-pressure levels in these three bands.[17,18,19] Since sound-pressure levels are often measured in octave bands for a rapid and first approximation of the prevailing noise level, the use of speech interference levels may be practical.

From Fig. 2.11 it can be seen how readily the speech interference bands mentioned above cover the speech spectrum.[19,20] By obtaining the masking noise levels in the three bands, it is possible to obtain the articulation index with a fair degree of accuracy. It may also be noted from Fig. 2.11 that if the audible speech spectrum lies in a shaded portion above the threshold of hearing and the ambient noise, but is below the overload area, then the articulation index will be 100 percent. However, if the masking noise covers some of the shaded region, falls below the threshold of hearing, or is above the overload line, then the articulation index will be less than 100 percent. Thus, if the masking noise followed the level of speech minima of Fig. 2.11, its rating would be 43 db, and then for a speech level as shown of approximately 69 db, there would be no effect on the intelligibility of the speech.

By taking into account (1) the speech interference level of the masking noise, (2) the level of the speaker's voice, (3) the distance from the speaker's mouth to the listener's ear, and (4) the type of vocabulary used, Table 1.1 was tabulated for speech interference levels which barely permit reliable conversation at the distances

TABLE 1.1. SPEECH COMMUNICATION CRITERIA[2]

Relation between SC criteria expressed by speech interference levels (SIL) and the communication conditions for a degree of intelligibility that is marginal with conventional vocabulary and good with selected vocabulary.

| SIL in Decibels | Voice Level and Distance | Nature of Possible Communication | Type of Working Area |
|---|---|---|---|
| 45 | Normal at 10 ft | Relaxed conversation | Private offices, conference rooms |
| 55 | Normal at 3 ft | Continuous in work areas | Business, secretarial, control rooms of test cells, etc. |
| | Raised at 6 ft | | |
| | Very loud at 12 ft | | |
| 65 | Raised at 2 ft | Intermittent | |
| | Very loud at 4 ft | | |
| | Shouting at 8 ft | | |
| 75 | Very loud at 1 ft | Minimal (danger signals; restricted preparranged vocabulary desirable) | |
| | Shouting at 2-3 ft | | |

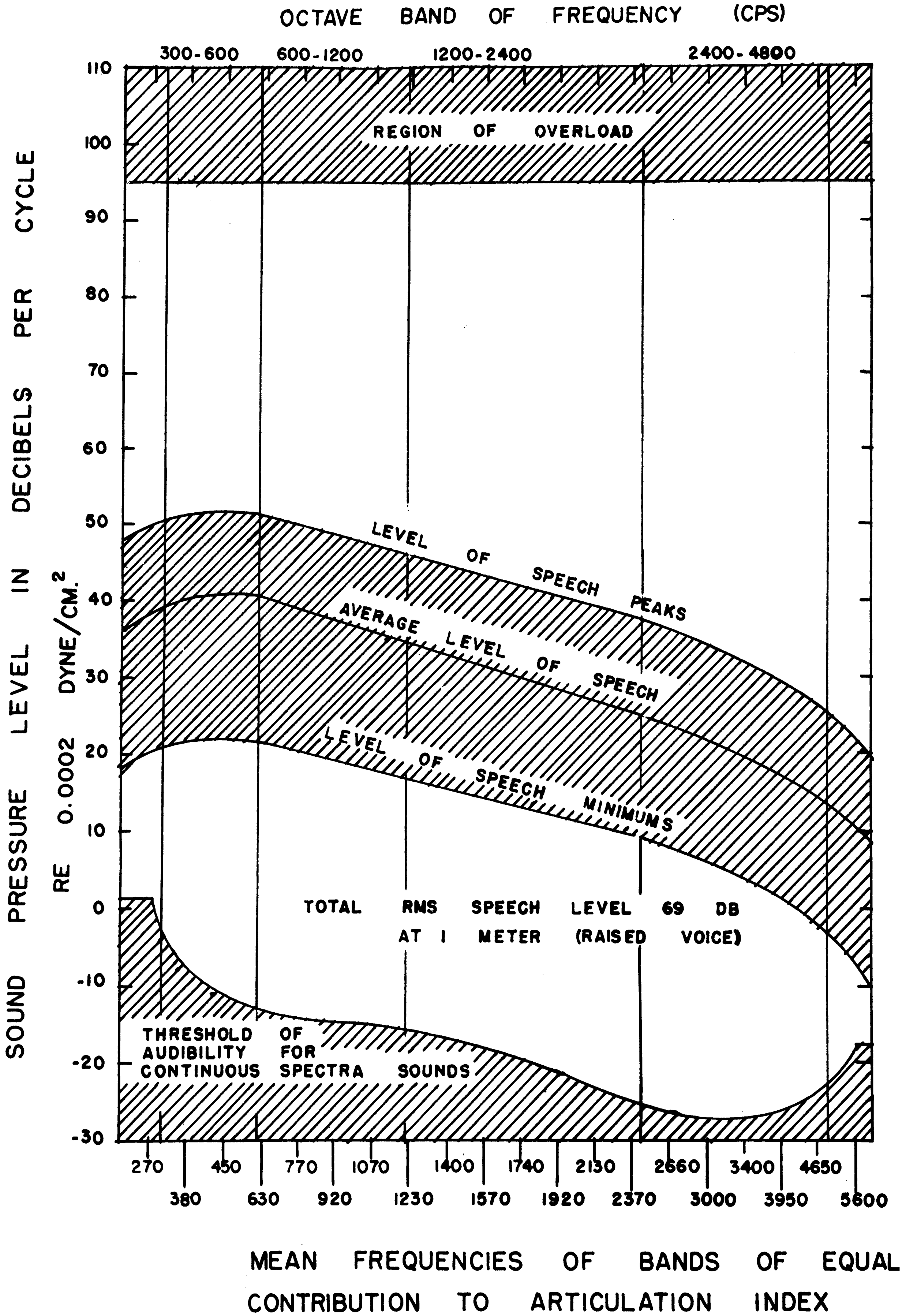



Fig. 2.11. Chart for computing articulation index for speech.
(Ref. 1)

and voice levels noted.[2,19,21] It must be remembered that the lower frequencies, that is, those below 600 cps, are not taken into consideration and, as has been demonstrated, low-frequency high-intensity sounds are excellent maskers. It must also be noted that attempts to employ speech interference levels will be in error if the masking sound is repetitive, pure tone, or shows some peculiar spectrum. To take low-frequency masking components into account, speech communication criterion (SC) curves have been formulated and extrapolated to low frequencies. Figure 2.12 is an SC plot of sound-pressure levels in bands versus frequency. To illustrate the use of the information presented, the SC 45 curve designates the octave-band level permissible if a condition of sound interference level of 45 is to be obtained. From the foregoing discussion one may observe that it is possible to categorize certain sounds as effective maskers. Therefore, information such as given by SC curves has a definite value to the noise-reduction engineer when he is confronted with a particular noise-masking problem.

## TEMPORARY AND PERMANENT HEARING LOSS

In the discussion of masking, the interference effect of a sound during its presence was systematically investigated. If the temporal aspect of the measurement procedure was to be changed so that the auditory response was to be measured following the termination of the interfering sound, the problem under investigation would now be the post exposure response of the auditory system. As such, the measuring procedure becomes somewhat more complex. Since the prime objectives of post exposure auditory measurements are to assay both temporary and permanent hearing losses, the complexity of the measurements is greatly increased, especially with respect to testing for permanent hearing loss, because of the accompanying continuous pathological changes in the auditory system as well as possible psychological changes.

Thus far, there is no absolute time limit accepted as valid for measurement of either temporary or permanent hearing loss. However, many experiments have been carried out and considering the number of variables involved, a certain pattern in information concerning post exposure hearing loss has become apparent. The general statement can be made that both temporary and permanent hearing loss are linked to the intensity and frequency of the noise as well as the duration of exposure. It has been proposed that the minimum sound-pressure level of noise below which no hearing loss occurs is approximately 85 db.[22] This does not mean that any noise above this level is certain to produce a hearing loss. Indeed, the consequences of exposure to noise varies not only with the individual but also with respect to each ear of the individual. Another factor involved in the specification of hearing loss is that of adaptation, meaning here that property of the auditory system which decreases the auditory response in the presence of continuous high-level noise (from an initial peak value). However, adaptation ought not to be accepted as beneficial for ear protection, even though the auditory response is reduced, since it does not necessarily offer any measure of physical protection for the auditory system.

Keeping in mind the diverse number of parameters involved, and the literally hundreds of experiments which have been undertaken, it is to be realized that this survey of hearing loss will attempt only to summarize the results of past efforts and to incorporate the data of more recent reports. Hearing loss and its permanency has been a chief concern of the American Academy of Ophthamology and Otolaryngology. The guide which it has set up as a conservation-of-hearing tool in the industrial field is applicable in many other fields.[23] A recent ASA publication presents an excellent reference to the relationship between high-intensity noise and man.[24]

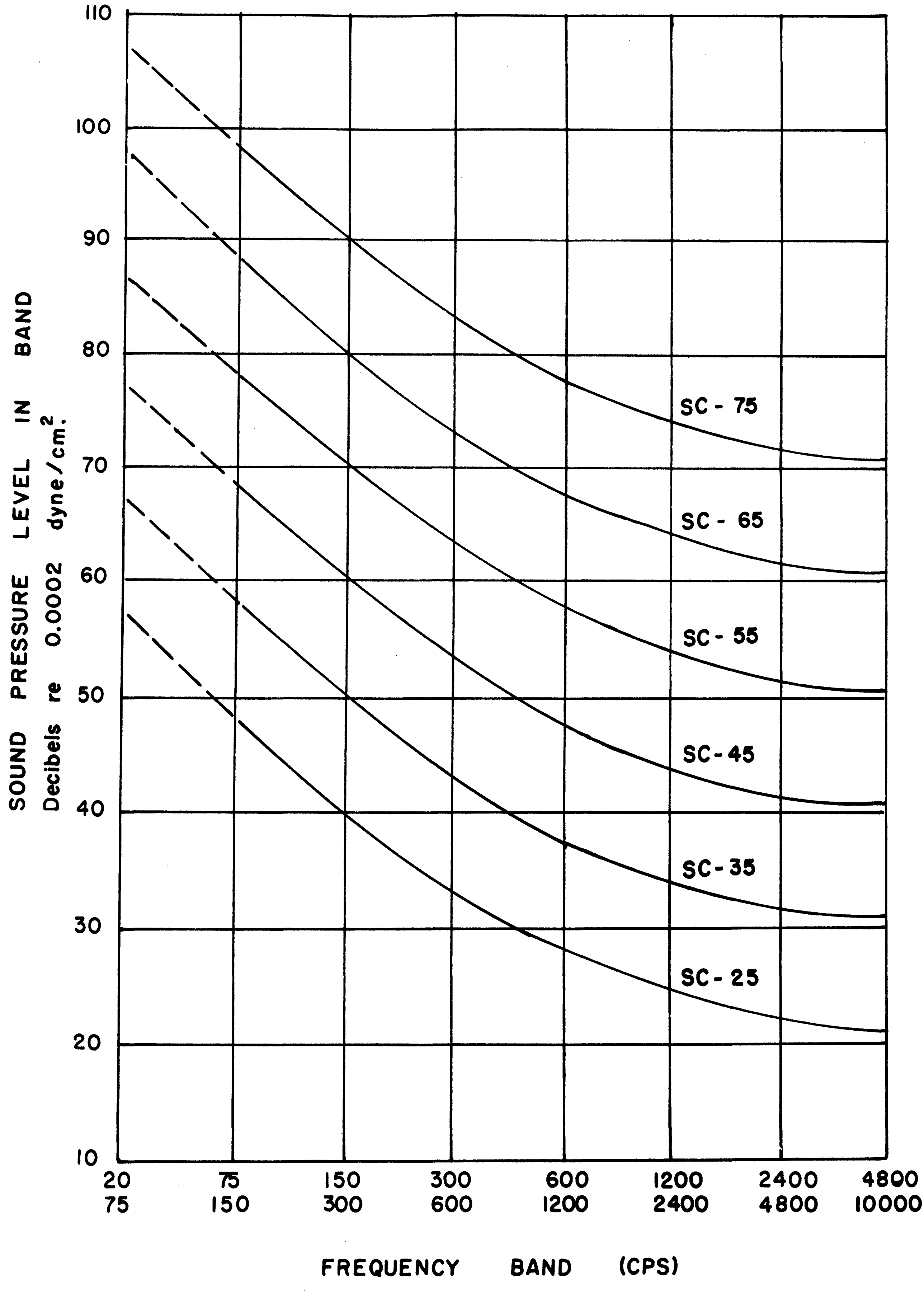



Fig. 2.12. Curves for speech communication criteria.
(Ref. 2)

Temporary Hearing Loss.—Hearing loss may be a reversible phenomenon. If a person is exposed for a time to high-intensity noise, he may suffer a hearing loss; if he is then removed to quiet surroundings, his hearing may return to normal after a period of time. The man has experienced a temporary hearing loss or temporary threshold shift. The same experiment may be repeated and the same results obtained. However, if the conditions are varied so that the man is returned to the noise environment before recovery, an added increment of hearing loss will be incurred, and this loss plus the loss which remained upon re-exposure demands an even longer quiescent recovery period. Upon continuation of these conditions a point of no return is reached and the subject has suffered a permanent threshold shift. It has been suggested that such a shift be considered permanent if the hearing loss is apparent after removal of the subject from the noise environment for a period of six months.[23]

The initial temporary threshold shift usually occurs in the higher range of frequencies (that is, above those frequencies most useful for speech perception), and gradually moves into the lower frequencies. However, the spectrum and the level of the noise to which the subject is exposed has a determining factor on the threshold shift. For instance, Fig. 2.13 gives the temporary hearing loss for a subject exposed for ten minutes to an intense jet-engine noise source of 146 db.[25] Notice that the peak loss, some 68 db, occurs at 1000 cps with the major overall loss apparent between 300 and 3500 cps. Thus the frequency components present in the intense jet noise are capable of producing temporary shifts at speech perception frequencies.

Observation of subjects exposed to noise of lower intensities is also of interest.[23,26] Temporary threshold shift studies were conducted on 31 subjects at their normal industrial occupation. Of these subjects 16 were classified as normal, 15 as having impaired hearing. Four other subjects were used as a control group and were not exposed. Thresholds were measured before the daily exposure, during the day, and immediately after the daily exposure. Actually, the noise environment could be classified as two different spectral distribution categories at two different plant locations. The noise source consisted of nine supercharged diesel engines of 1700 horsepower producing an overall noise level of 105 db on the work floor. This can be compared to a level of approximately 20 db less in the control room and switchboard. By comparison of the two noise levels and the resulting temporary threshold shift of personnel occupying the respective environments, several facts are evident. Figure 2.14 shows a comparison of the noise environments and the corresponding hearing losses. Note that although the highest noise level occurs at the lower frequencies, the threshold shift for the higher noise-level environment (see Fig. 2.14A) tends to increase with frequency, showing a maximum loss of approximately 18 db at 6000 cps (see Fig. 2.14C). On the other hand, the lower noise level, as shown in Fig. 2.14B produces approximately a 2-3 db loss and is for all purposes constant over the frequency range tested. By separating the response curves of those exposed to the high level, as shown in Fig. 2.14A, into temporary shifts for the normal group and for the impaired group, it can be seen from Figs. 2.15A and 2.15C that the overall threshold shift for the normal group is several db greater than that of the impaired group. Notice, however, that the control group subjected to a "quiet" noise environment of approximately 50-60 db (Fig. 2.15B) exhibits a negative threshold shift as shown in Fig. 2.15D, that is, a lowering of the hearing thresholds of approximately 2-4 db. Another important factor is apparent from Fig. 2.16, which is a series of two-hour plots of hearing loss during exposure to the spectrum of Fig. 2.14A. Notice that the maximum shift occurs during the first two hours of exposure, with lesser shifts apparent at the end of four, six, and eight hours, respectively.

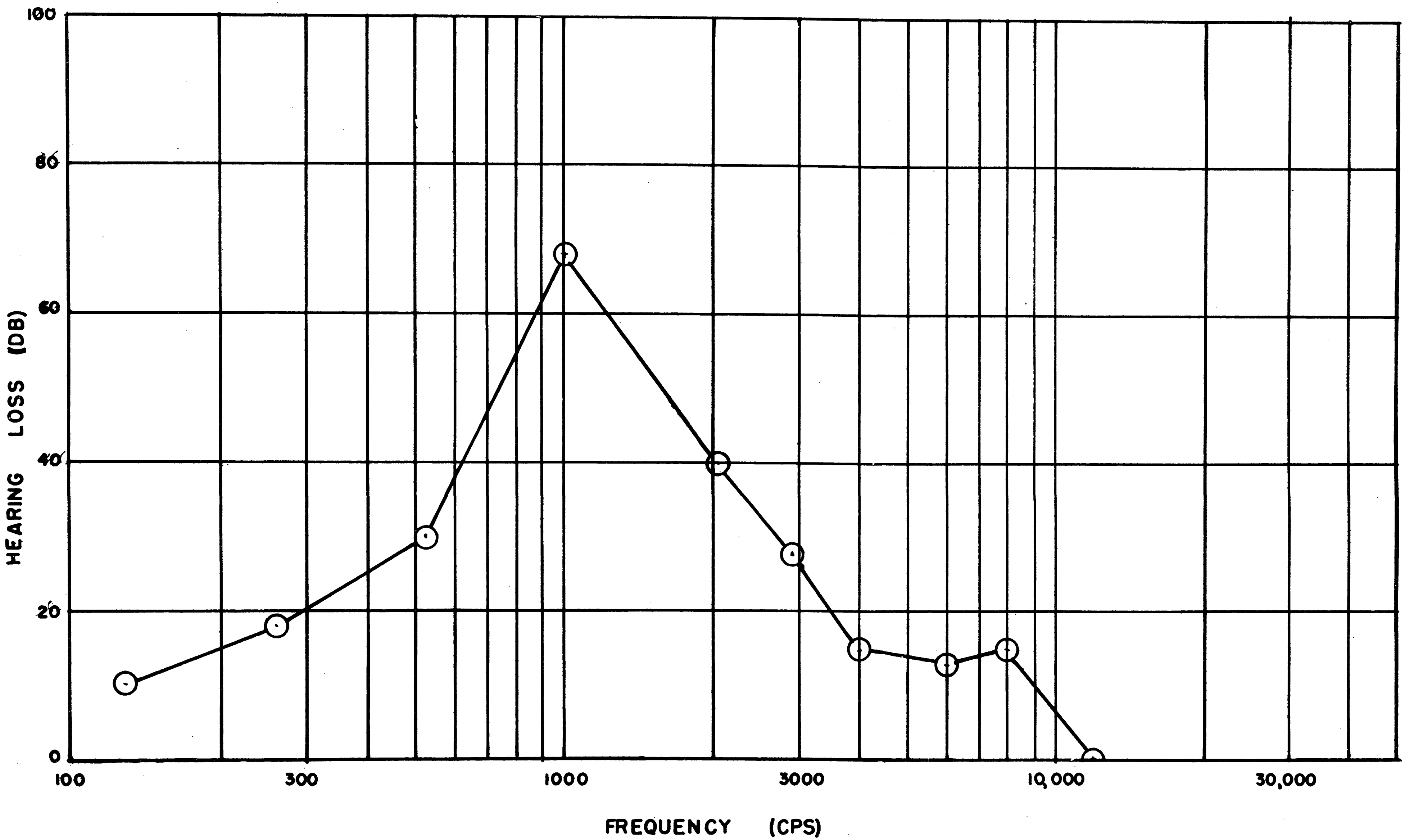



Fig. 2.13. Temporary hearing loss after a 10-minute exposure to a 146 db jet engine noise source. (Ref. 25)

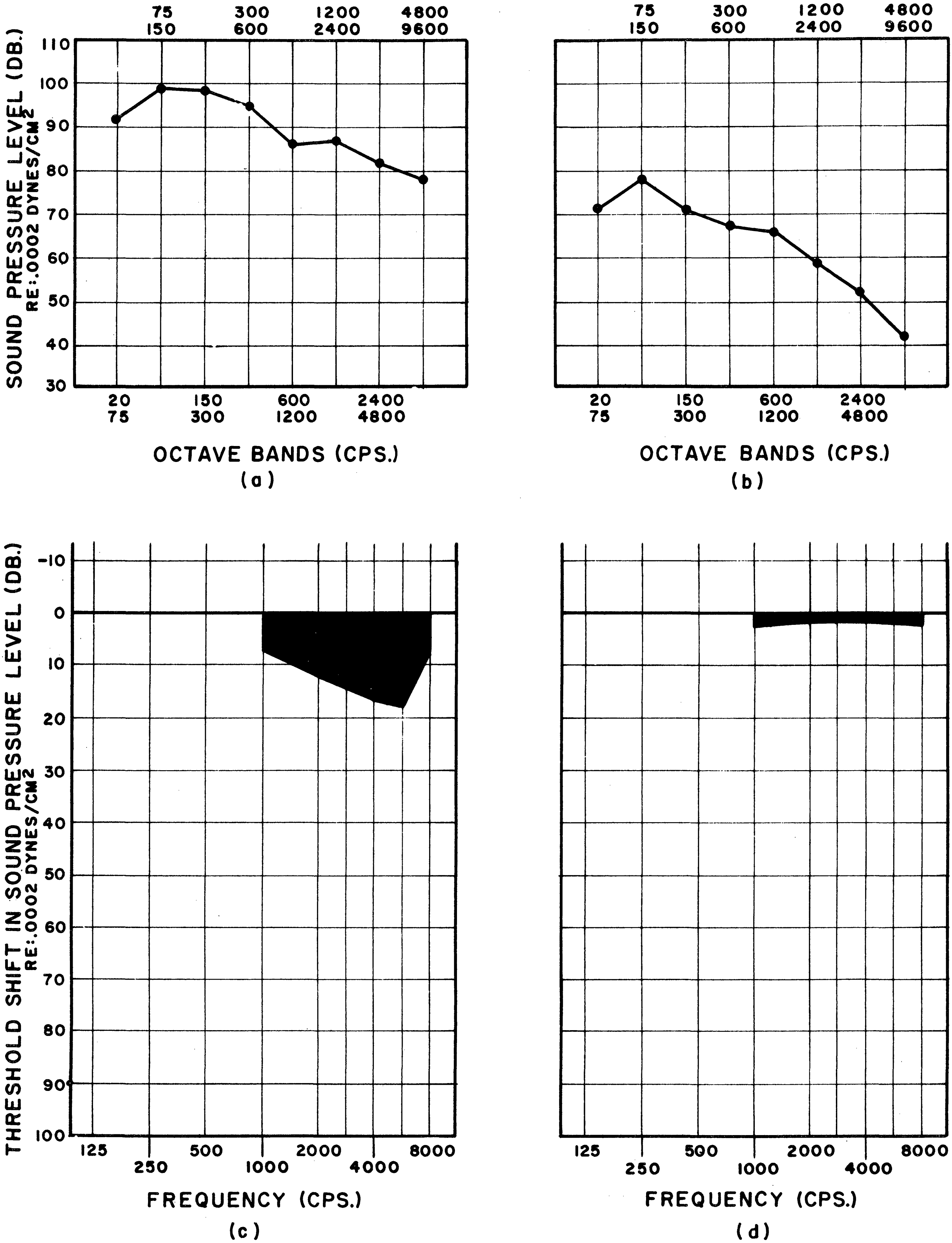



Fig. 2.14. Hearing loss as a function of frequency for two different noise environments. (Ref. 23)

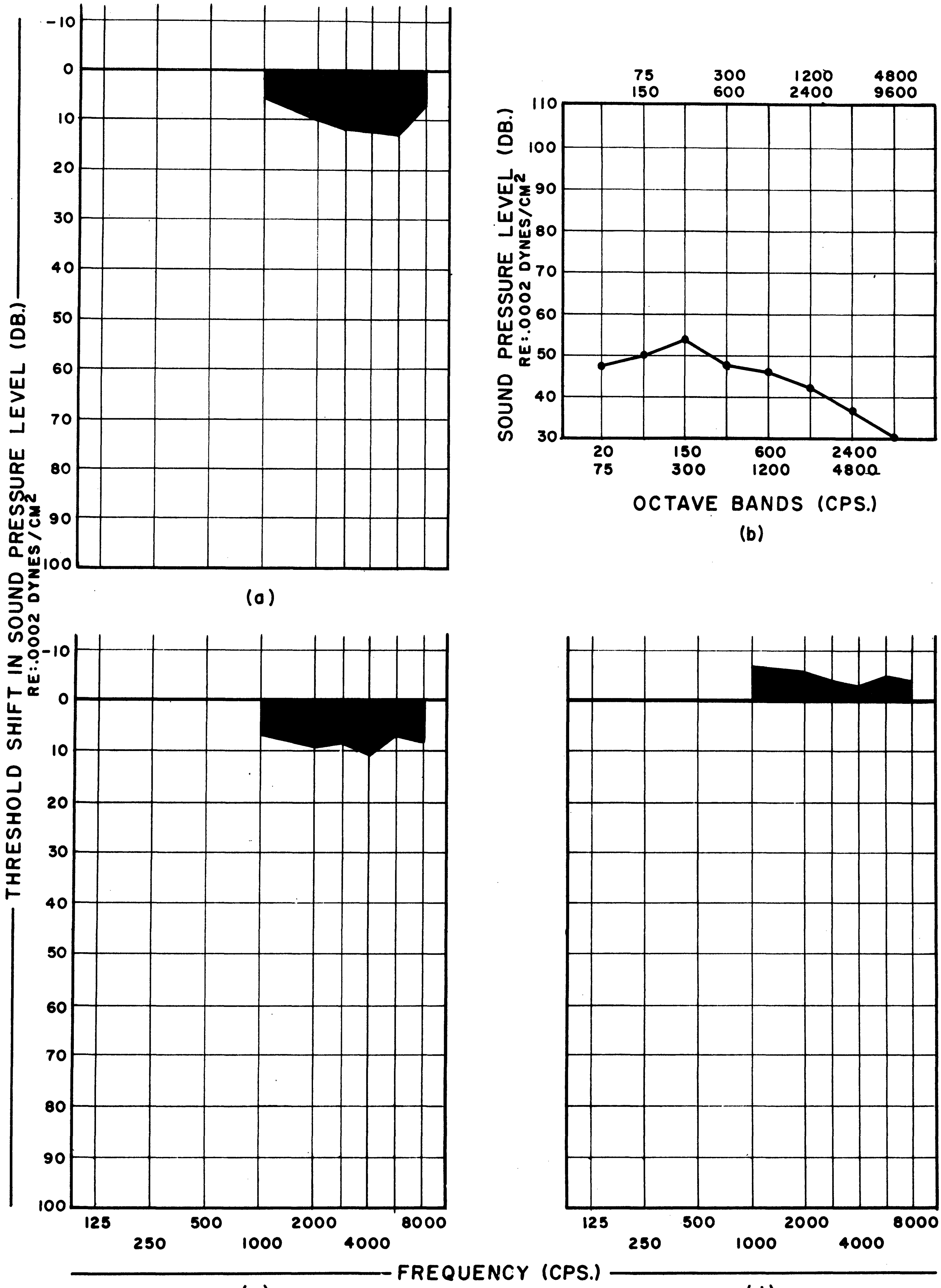



Fig. 2.15. Comparison of temporary hearing losses suffered by normal (a,c) and impaired (b,d) auditory systems. (Ref. 23)

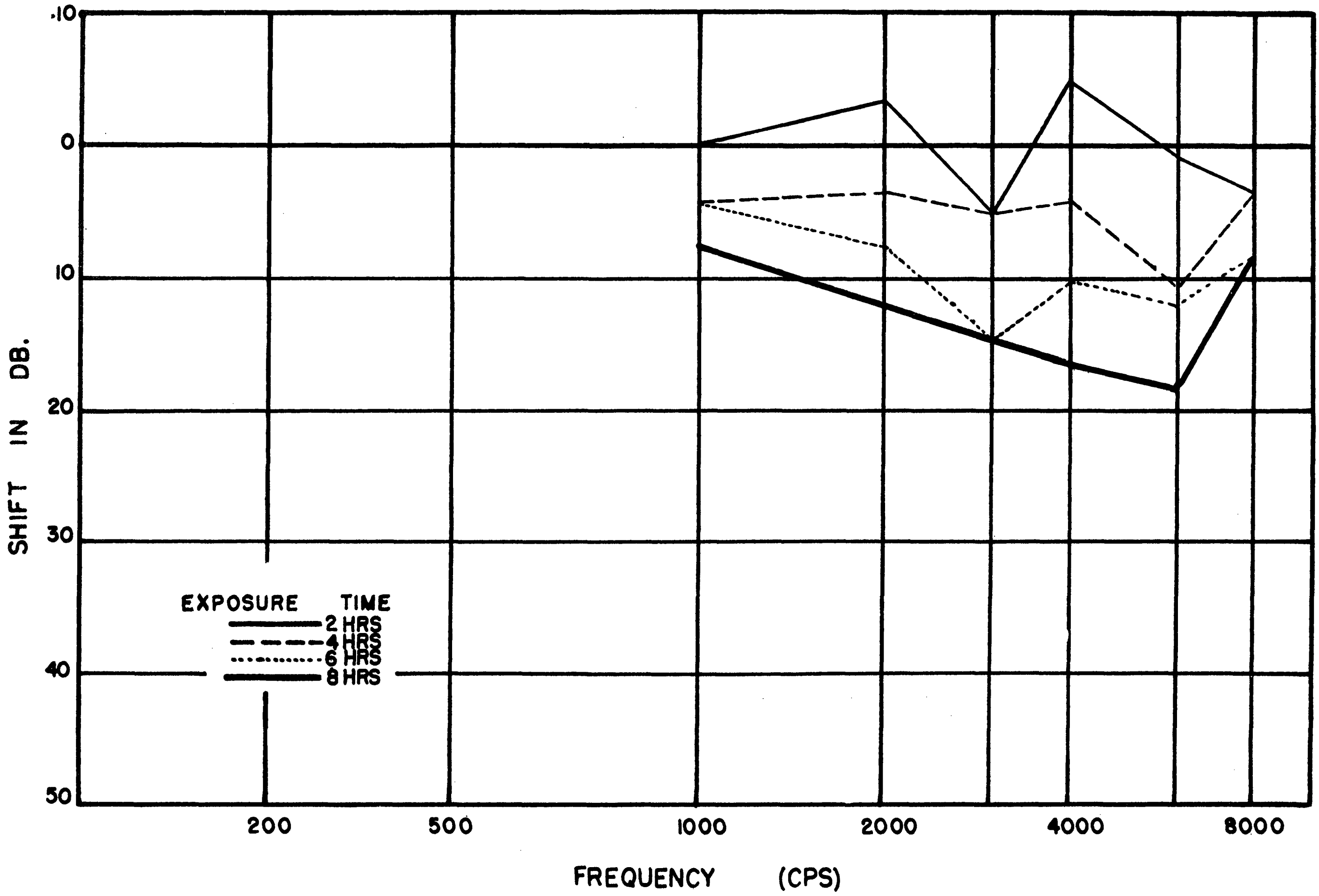



Fig. 2.16. Hearing loss as a function of frequency for exposure times of 2, 4, 6, and 8 hours. (Ref. 23)

These data are interesting from the standpoint of ground-support-equipment noise-reduction effort when one considers the remarkable similarity between the relative octave-band plots of the C-26, MA-1, etc., and that of the data under consideration. The value of such information lies in the fact that here are data on temporary hearing loss which have been scientifically gathered and evaluated, utilizing the ideal laboratory condition of the actual complex noise environment.

Hearing Losses Caused by Exposure to Pure Tones.—Experimentation which has been carried on in the field of threshold shifts, using a pure tone either as the exposure stimulus and/or the recovery testing stimulus, has yielded some significant results. As in the case with pure-tone masking stimuli, a low-level exposure stimulus tends to raise the threshold in a more or less symmetrical manner about the exposure frequency regardless of exposure time. As this level is raised, not only does threshold shift become distributed more to frequencies above the exposure stimulus, but also the recovery period is shorter for lower than for higher frequencies. There is some doubt as to whether the use of pure-tone sounds gives an accurate accounting of threshold shift. Indeed, some evidence has been found that a 5000-cps tone and noise used in pulse-tests do not measure the same responses of the hearing mechanism.[27] Nevertheless much useful information using pure tones has been obtained. Figure 2.17 shows the exposure-time effect of a 500-cps tone relative to a 1000-cps test stimulus tone of 30 milliseconds' duration.[2] Notice that there is no noticeable threshold shift with duration of 500-cps exposure tone until the sensation level reaches 70 db. In other words, for low-level exposure tones, the threshold shift is not a function of the duration of the exposure stimulus. Notice also that these data are for relatively short time exposures (i.e., 0.1-4 seconds). For a longer exposure time of one minute, threshold shifts increase at a rapid rate for a sensation level of approximately 90 db. This is apparent from Fig. 2.18A where both the one-minute exposure stimulus and the test stimulus are a 2048-cps tone.[2] Using the same frequency for both exposure and test stimuli, the threshold shift of four different subjects was examined. In Fig. 2.18B the change of threshold shift is shown for subjects who were exposed to the 2048-cps tone at a sensation level of 100 db for a period up to five minutes.

Hearing Losses Caused by Exposure to Complex Noises.—As indicated by results from measurements of engine noise, the most valuable data on temporary threshold shifts should be those obtained for complex sounds used as the exposure stimuli. The usual types of noise spectra used as stimuli are octave-band spectra, although one-half octave-band noise and controlled shape spectra have also been used. As is the case with pure tones, the reactions to complex sounds used as exposure stimuli are influenced by spectrum content, intensity, and duration. Recent pulse-type tests, using both shaped noise and octave-band noise as exposure stimuli for levels up to 115 db, showed little change in fatiguing effects except over a two-hour exposure period. Even that exposure period produced a shift of only 4-5 db.[28]

It is obvious that high-intensity exposure stimuli must necessarily be used in the laboratory if actual conditions are to be approximated. Figure 2.19 shows the temporary hearing loss incurred by several subjects when exposed to band spectra of 130-db overall level for a period of 32 minutes.[22] It can be observed that the average threshold shift has increased in the frequency bands tested up to approximately 4000 cps, above which there is a decrease in threshold shift. It may also be noted that the greatest losses occur in the speech reception range. Intensities over 150 db have also been utilized for determining temporary threshold shifts. In this case the acoustic level has been obtained by using either a jet engine or a siren as the source. Even at this level threshold-shift levels returned to normal after a maximum of 7 days,

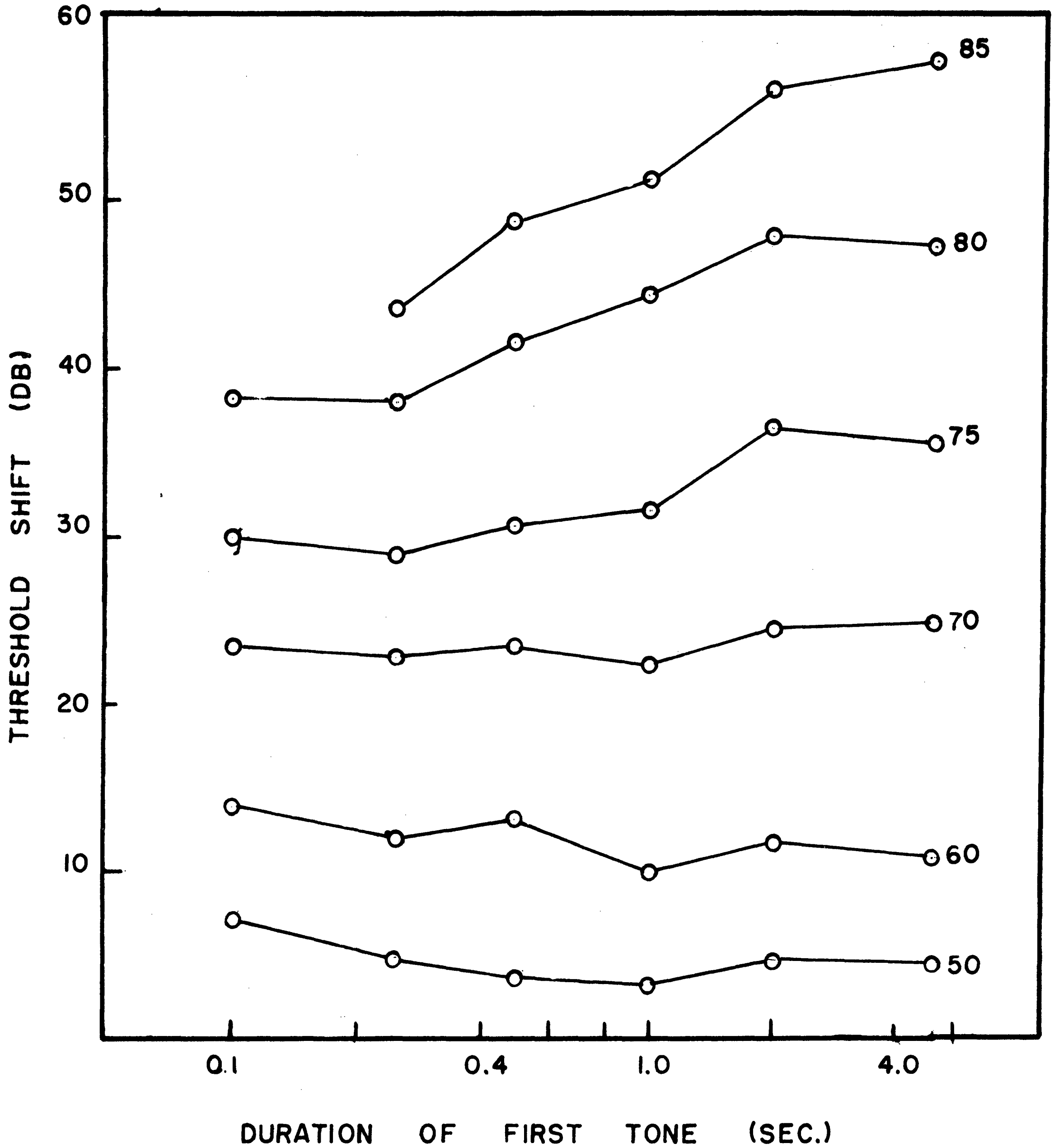



Fig. 2.17. Effect of duration of exposure tone of 500 cps on threshold shift of a 1000 cps tone of 30 milliseconds duration. (Ref. 2)

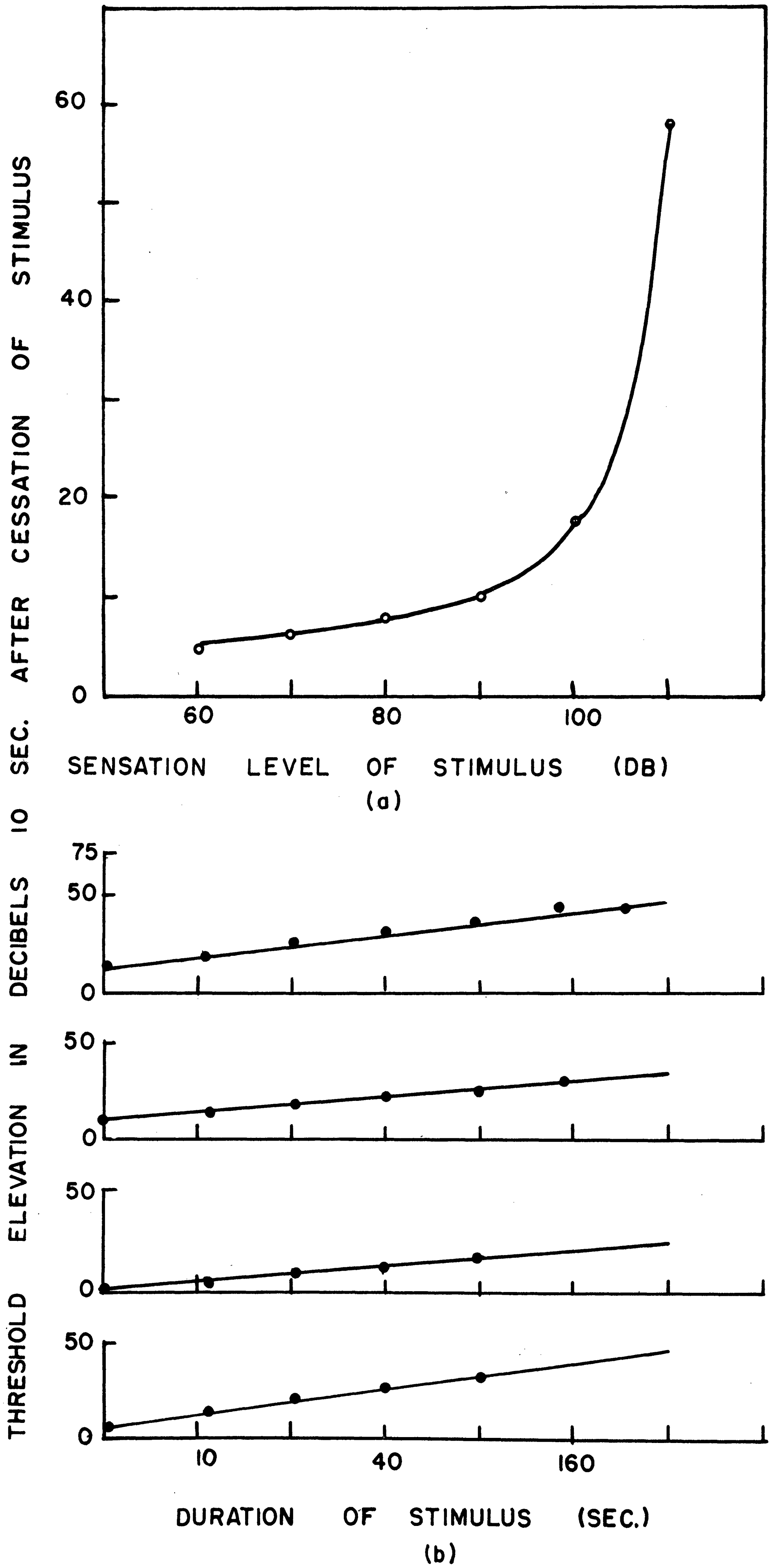



Fig. 2.18. Threshold shifts as a function of stimulus intensity and duration of exposure. (Ref. 2)

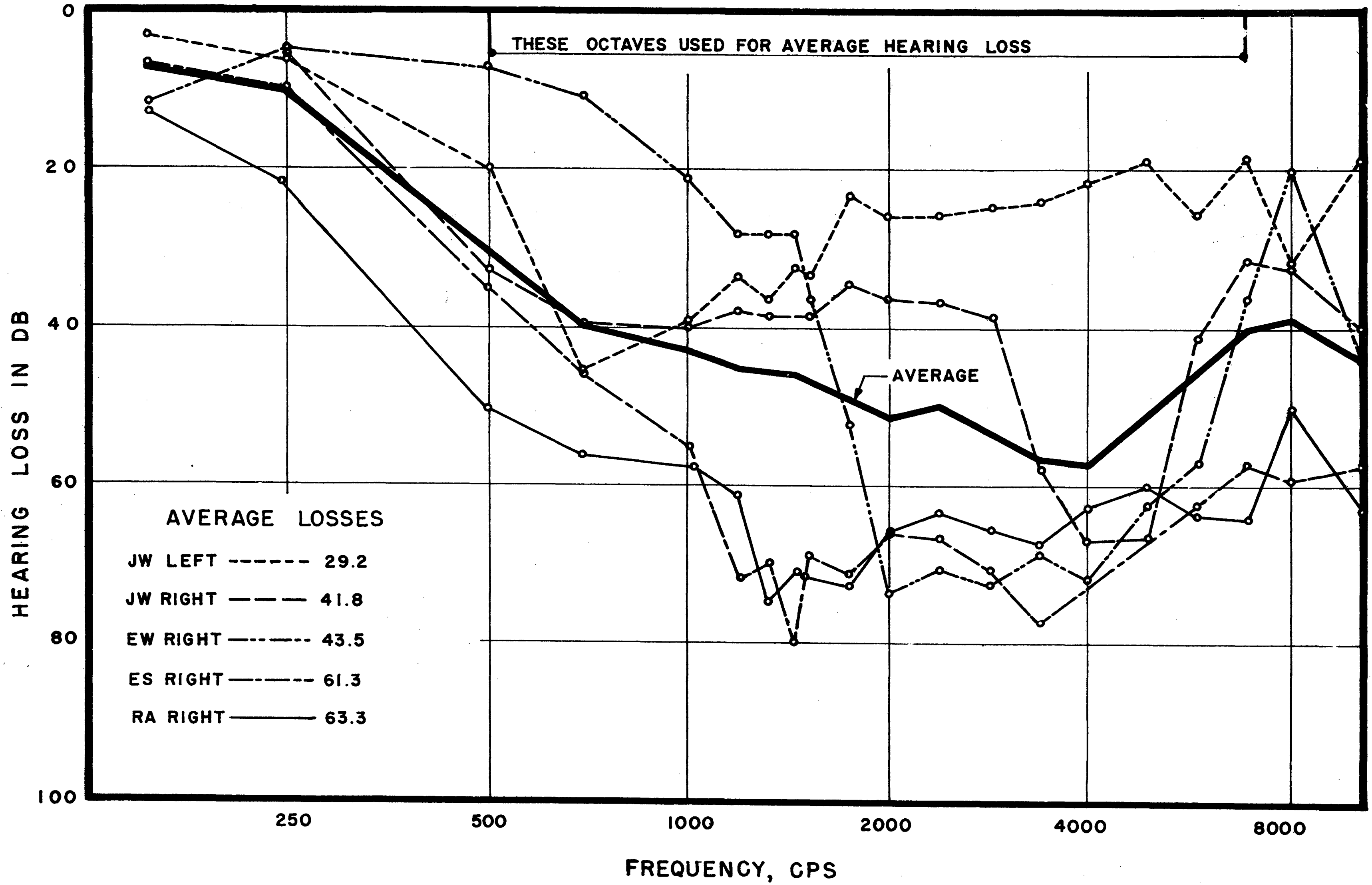



Fig. 2.19. Audiograms for 5 subjects following 32-minute exposure to 130 db band spectrum. (Ref. 22)

although the initial threshold shift occurring within the exposure time of 3-10 minutes was about 60 db. Various other noticeable effects on threshold shift have been observed from high-intensity exposure on diplacusis.[29]

Permanent Hearing Loss.—Experiments to determine the contributing factors to permanent hearing loss have not been extensive for obvious reasons. Instead, attempts have been made to determine hearing-loss criteria to be used in the prevention as well as the detection of such loss.[23,24] Some of these criteria are apparent from the temporary threshold shift data already presented. Preliminary predictive tests have been carried out with varying results. Figure 2.20 shows the threshold shift obtained using a noise exposure stimulus of 105 db for a 30-minute period. Note that the average shift increases with frequency from 2000 cps to 4000 cps and 6000 cps. This suggests a displacement upwards of the effect in magnitude of the exposure noise. Figure 2.21 shows the recovery time at these three frequencies, emphasizing that the recovery time is also a function of initial threshold shift as well as frequency and intensity. To correlate predictive information for prevention of permanent hearing loss a damage-risk (DR) criterion has been proposed.[2] This has been based on numerous field studies and is divided into criteria for both long-term and short-term exposures. Figure 2.22 shows the proposed lifetime and steady-noise DR criterion curves of wide-band noise as well as pure-tone and critical bands of noise which have modified the proposed criterion referred to previously.[22] On examination of the DR curves presented in this figure, it may be noted that long-term exposure to levels lying above these curves is inadvisable. The absolute value for a DR criterion for short-term exposure to very high noise levels has not been determined, and it has been suggested that although some individuals have been exposed to 150-db levels of noise without apparent results, there should be no exposure to levels of 145-150 db or more for a time greater than one minute.

## ANNOYANCE

In the field of acoustics, most parameters such as loudness, pitch, etc., involved in both physical and psycho-acoustic measurements can be described in at least a quasi-quantitative quantity either by straightforward instrument measurement or by jury analysis. Unfortunately, the annoyance factor has thus far resisted so simple a treatment. Everyone with normal hearing has had an occasion to be in an area of annoying sound and it seems that if one is annoyed by sound, it should be simple to define this sound. However, the factors which make a sound annoying are variable, and the annoying aspects of sound change not only with its physical characteristics but also with the environment and attitude of the individual. It is well known that many persons are sensitive to personal and environment factors which affect their physical or mental well-being, and by proper "needling" a desired reaction can be obtained. Thus, if a large group of people were to be subjected to certain particular stimuli, certain general reactions could be expected. This is true also of annoying sounds, and as a result some general annoying properties of sounds may be tabulated. However, a single comprehensive annoyance number has, as yet, not been formulated and accepted as a rigid laboratory standard.

Contributing Factors to Annoyance.—A number of annoyance factors have received considerable attention with respect to their relative contributions to a total annoyance value. These major contributors are intensity, frequency, intermittency, unexpectedness, and inappropriateness.[25] One experiment consisted of presenting to groups of 10 to 20 subjects a pair of sounds with instructions to decide which was more annoy-

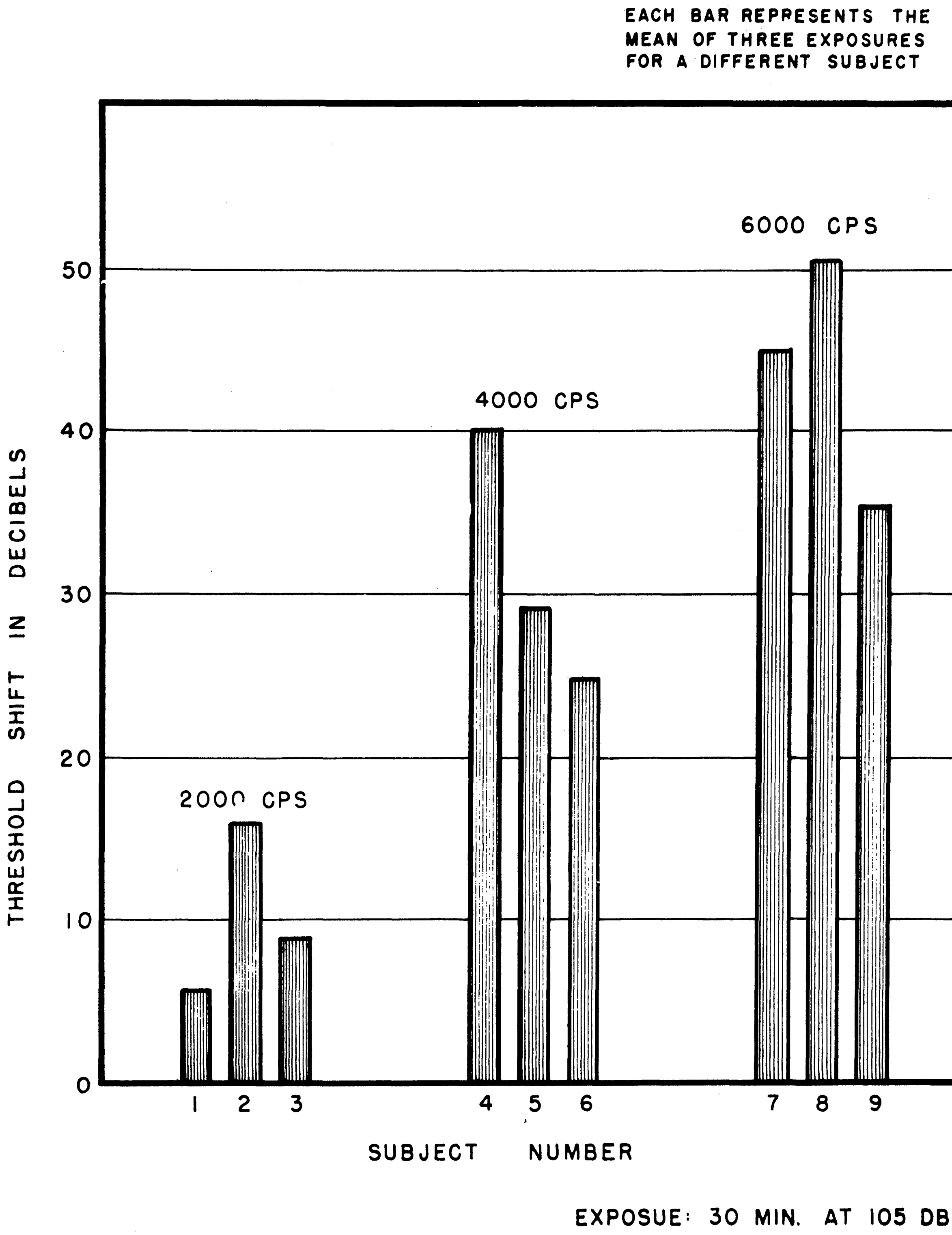



Fig. 2.20. Initial threshold shifts at different frequencies.
(Ref. 2)

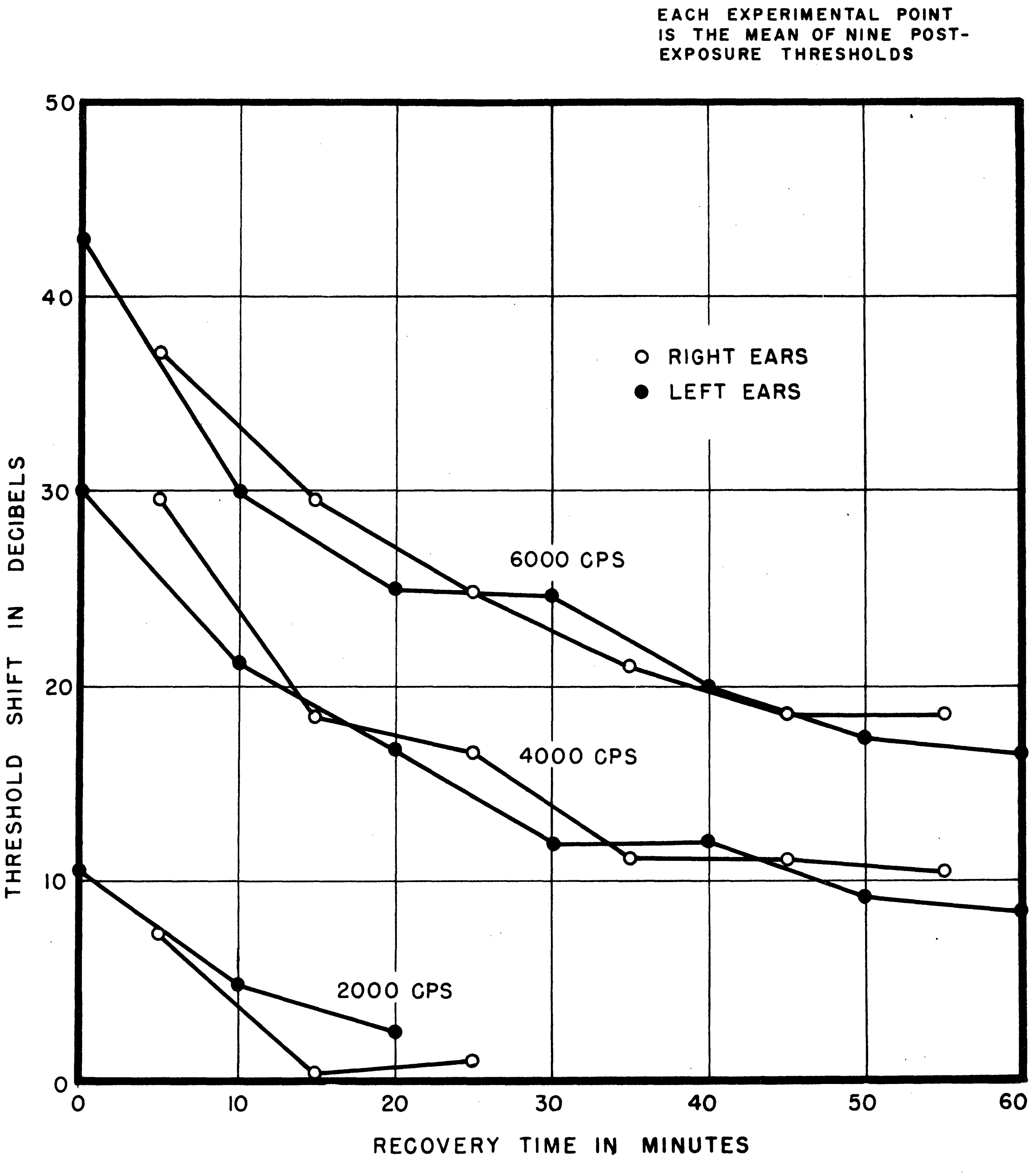



Fig. 2.21. Recovery curves for nine subjects—three at each frequency. (Ref. 2)

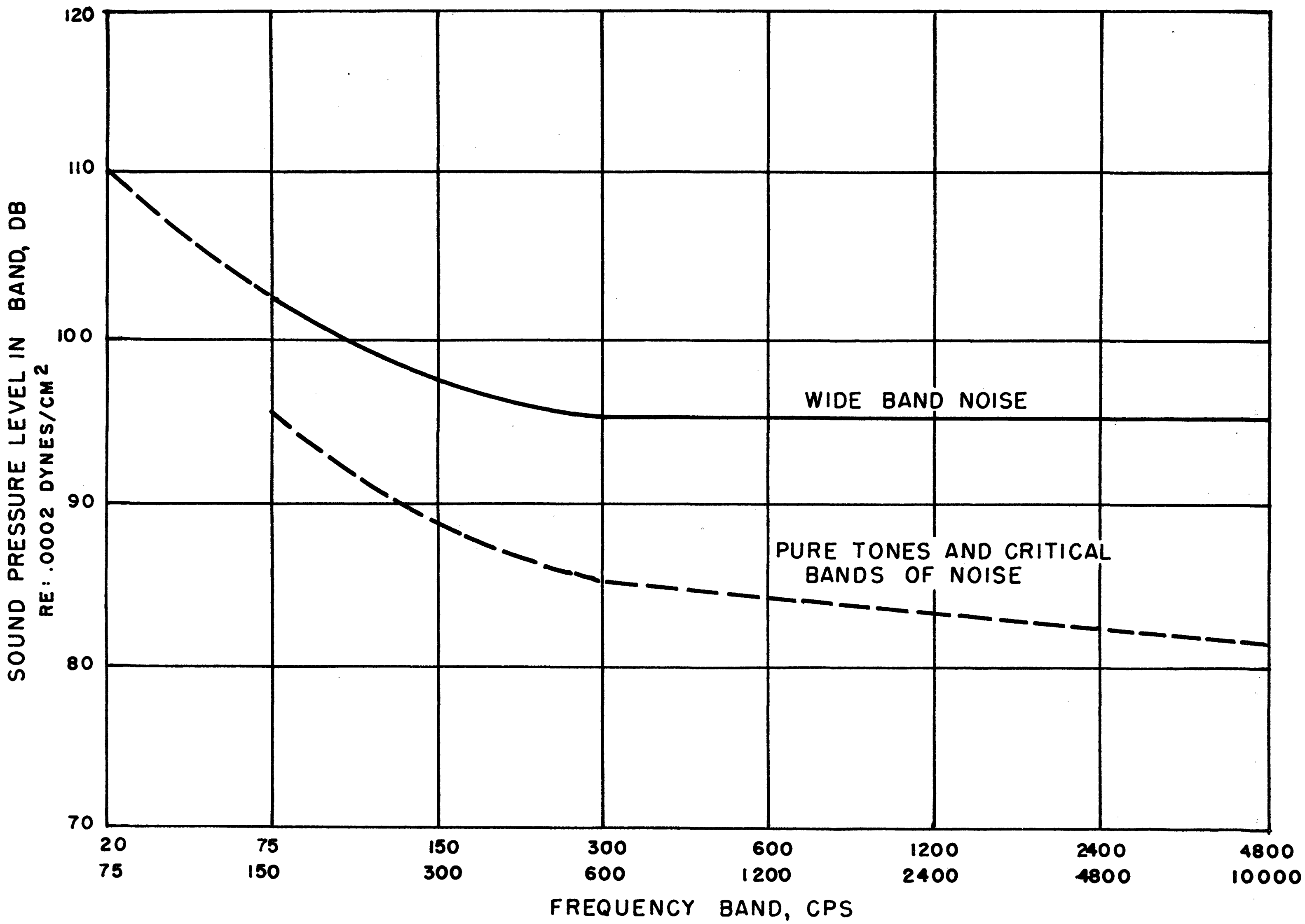



Fig. 2.22. Damage risk (DR) criterion for steady noise and for lifetime exposures. (Ref. 2)

ing, taking into account the probable annoyance involved if a long-term exposure were involved. In one instance, a stepped pattern of tones was utilized with results obtained for eight contributing annoyance factors. The results are quoted below:[6]

"1. The higher the pitch of the component tones, the greater the annoyance-value. The range of frequencies tested was from 200-1500 cycles.

"2. A wide range of frequencies between the highest and lowest steps is more annoying than a restricted range. Listeners reported that the wide range of component frequencies tended to be perceived alternately, first as a complete pattern and then as two patterns, one of high and one of low pitch. This effect is very similar to figure-ground reversals in visual perception.

"3. The addition of continuous tones to the stepped pattern of tones produces complex effects dependent upon the frequency-relation between the tones. Beats give the sound a rough pulsing irregularity which the listeners disliked.

"4. Listeners asked to compare continuous sounds of different wave-shapes found the complex sounds especially brief pulses, more annoying. In general, the sine wave was found to produce little annoyance.

"5. Patterns of 3, 4, 6, and 12 tones were compared but the number of different steps in the complete pattern had little effect on the judgments of annoyance.

"6. If one of the steps of a pattern is slightly longer in duration from the others, a rhythmic quality is added which the listeners judged to be more annoying than tones of equal duration. Even more annoying, however, is the pattern in which all the tonal durations are randomly varying.

"7. A slow rate of repetition for a pattern of tones is considered slightly more annoying than a rapid rate.

"8. Up to a certain limit, the annoyance-value is increased if silent intervals are introduced between the successive steps."

Practically all surveys of annoyance factors agree that intensity is the most important conbributor. This is quite important with respect to ground-support-equipment personnel who, even when not exposed to painful noise, may be irritated by the annoyance contribution of a moderately high-level noise. This level seems to be somewhat in excess of 80 to 90 db. Several experiments have been carried out in which subjects have been exposed to aircraft-type noise. In one instance several subjects were exposed to aircraft noise of 90 db and 115 db (overall).[30] Exposure to the lower level showed no particular annoyance response, whereas continued exposure to the higher (115 db) level tended to produce irritation and tiredness.

Several experimental attempts have been made to correlate frequency and loudness to annoyance. Figure 2.23 shows such an attempt.[4] Here the equal loudness curves have been converted into equal annoyance curves. The annoyance contours follow the equal loudness contours at the lower frequencies but digress at the higher frequencies. Application of these curves can be demonstrated by examination of the following figures. Note that Fig. 2.24A is a plot of the loudness levels of two office-machine rooms.[4] By application of the annoyance-level curves, Fig. 2.24B shows a marked increase of annoyance at the high-frequency sector of the plot.[31]

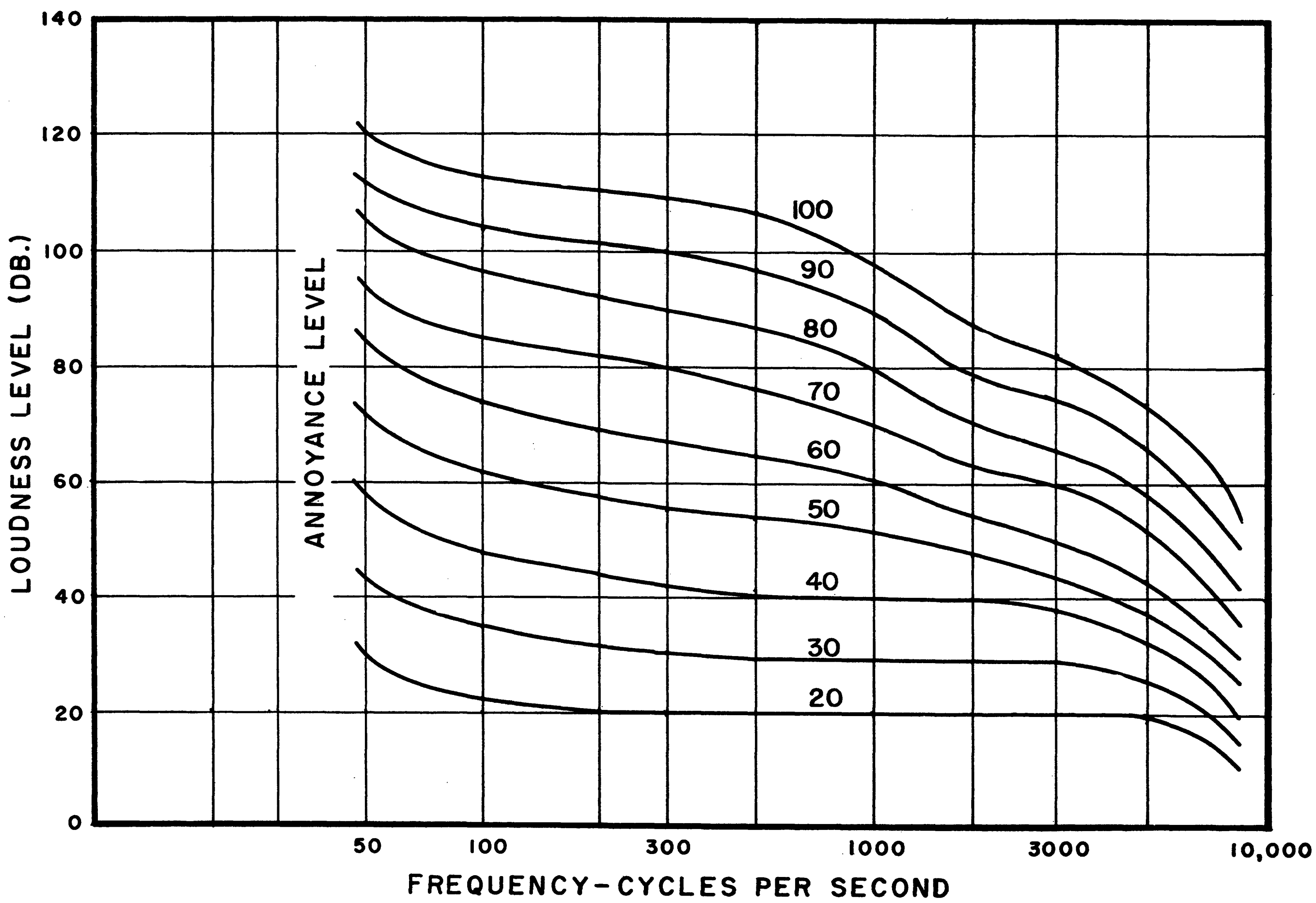
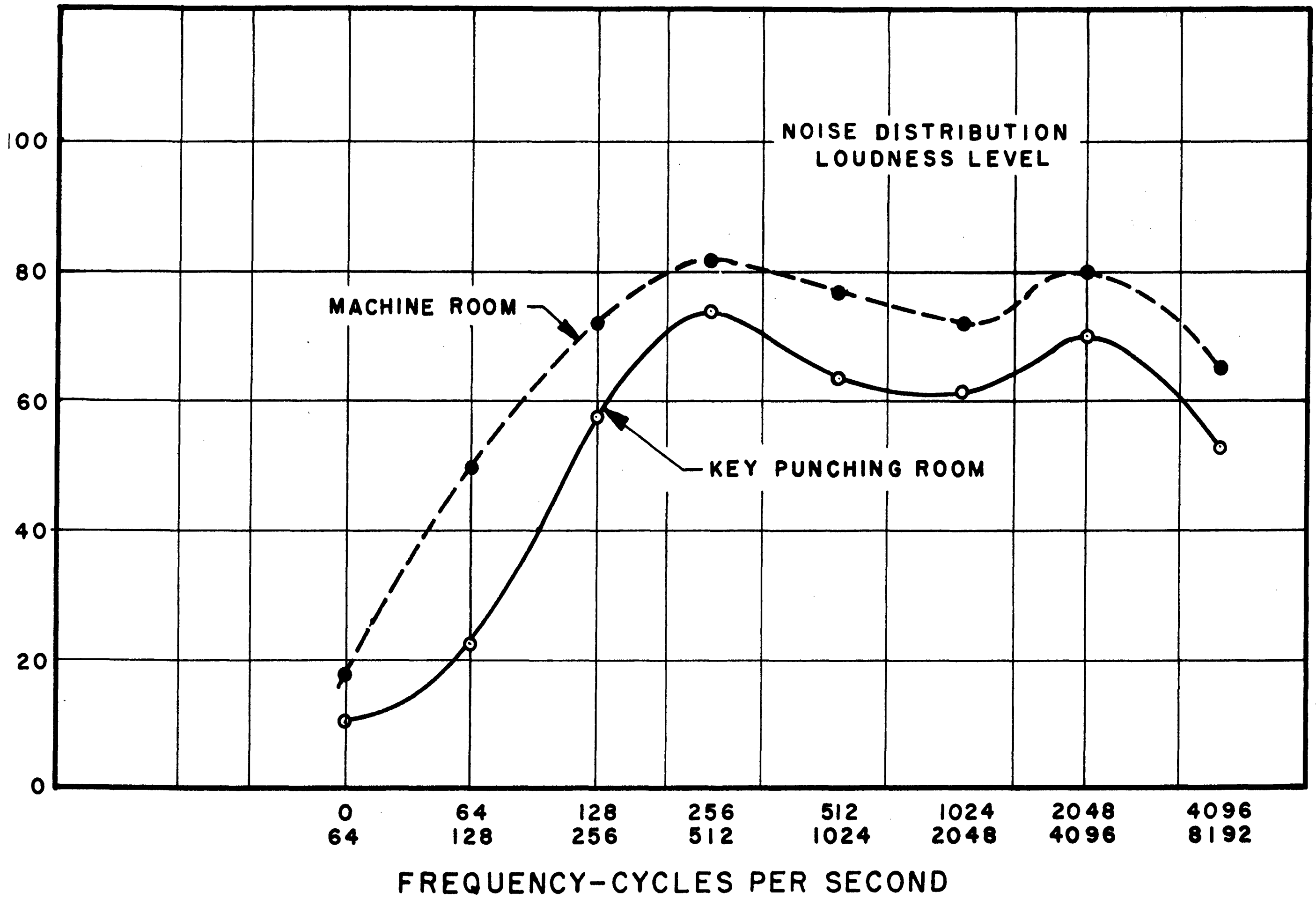
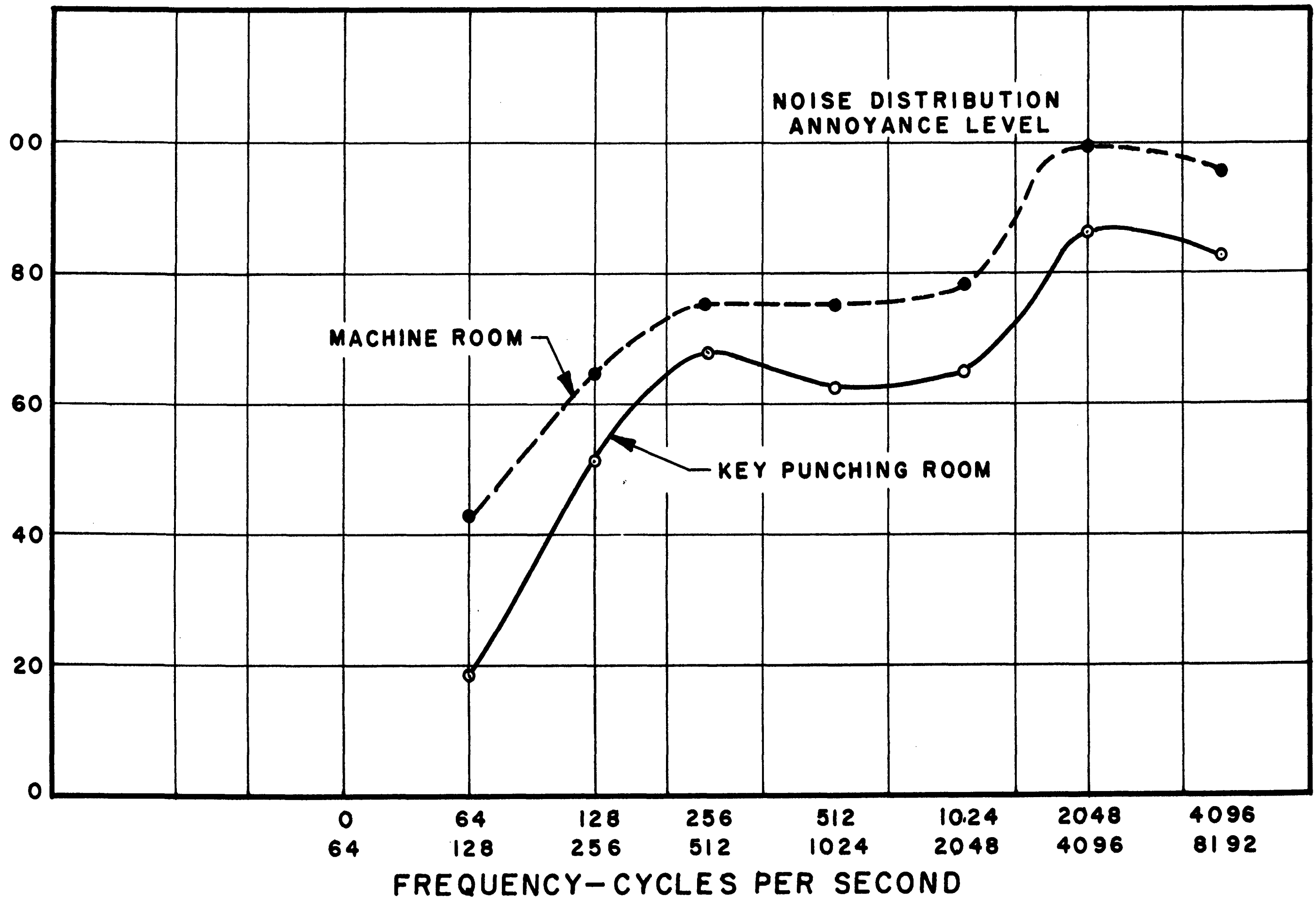



Fig. 2.23. Annoyance levels as a function of frequency. (Ref. 4)




(a)




(b)

Fig. 2.24. Loudness and annoyance levels of noise in two rooms. (Ref. 4)

Another experiment consisted of utilizing bands of noise for evaluation according to annoyance value. A band of noise of 1900 to 2450 cps was chosen as a standard, and other bands were treated for equal loudness and then for equal annoyance to this band. Figure 2.25 shows comparisons with the level or the standard band varying from 94 db to 64 db.[22] In each case, the annoyance contribution shows an increase with frequency. This is significant at the lower intensity levels since this points out the definite effect of frequency where intensity is no longer a major consideration.

The contributions of intermittent, unexpected, and inappropriate noise to annoyance may well be explained by the following: ". . . Accordingly, we apparently find it difficult to ignore any noise or sound unless we can attach some meaning to it and understand the reasons for its presence. This is presumably highly distracting, keeping one forever alerted when unanticipated, novel, unusual, sudden sounds or noises are received."[4]

Thus it can be observed that annoyance in ground-crew environment is something of a predicament as regards noise reduction. Whereas attempts may be made to reduce high-intensity pain-producing noise levels, low-frequency masking-noise levels, and annoyance due to high-frequency spectrum content, it is nearly impossible to remove the factors of intermittency, unexpectedness, and inappropriateness where these must by the nature of the situation become an integral part of the acoustical environment.

Airport Environment Annoyance Problems.—However, certain aspects of annoyance outside the acoustical confines of ground-support operational environment have been investigated. These are the aspects of annoyance with which "outsiders"—that is, the residents of neighborhood areas—are primarily concerned. Indeed, a whole set of criteria has been compiled on the basis of investigation of community reactions to noise problems. From these criteria one may ascertain what the reaction of a particular community will be when confronted with a particular type of intrusive noise. The results of the investigations have led to a single noise-evaluation number designated as the composite noise rating (CNR).[32] This number, as is implied, is of a composite nature taking into account such factors as the intensity of the noise, its loudness, spectrum, repetitive character, duration, and seasonal appearance as well as other factors including the background noise and the previous exposure to noise of a community. From proper evaluation of the above factors and correlation with noted community response, a set of curves can be formulated as is shown in Fig 2.26, in which community response is noted on the ordinate and the composite noise rating in upper case letters is plotted on the abscissa. Thus, when the composite noise rating is ascertained by proper evaluation of the factors noted above, the range of expected reaction of a community may be obtained from the plot of Fig. 2.26. Thus, a CNR of value E may be expected to result most generally in widespread complaints, though variations may occur ranging from reactions of sporadic complaints to threats of community action.

However, assaying community response to annoying noises is only the first step in solving an annoyance problem. That is, the intruding noise has been identified and its consequent reaction on a community has been determined; the problem now is how to lessen the noise. This is a major problem at airport installations since the noise contributors are not only units of ground-support equipment but also engine test stands and operations aircraft.

Ground-support equipment and engine test stands offer less of a problem than do operational aircraft with respect to controlling the noise emitted from each of

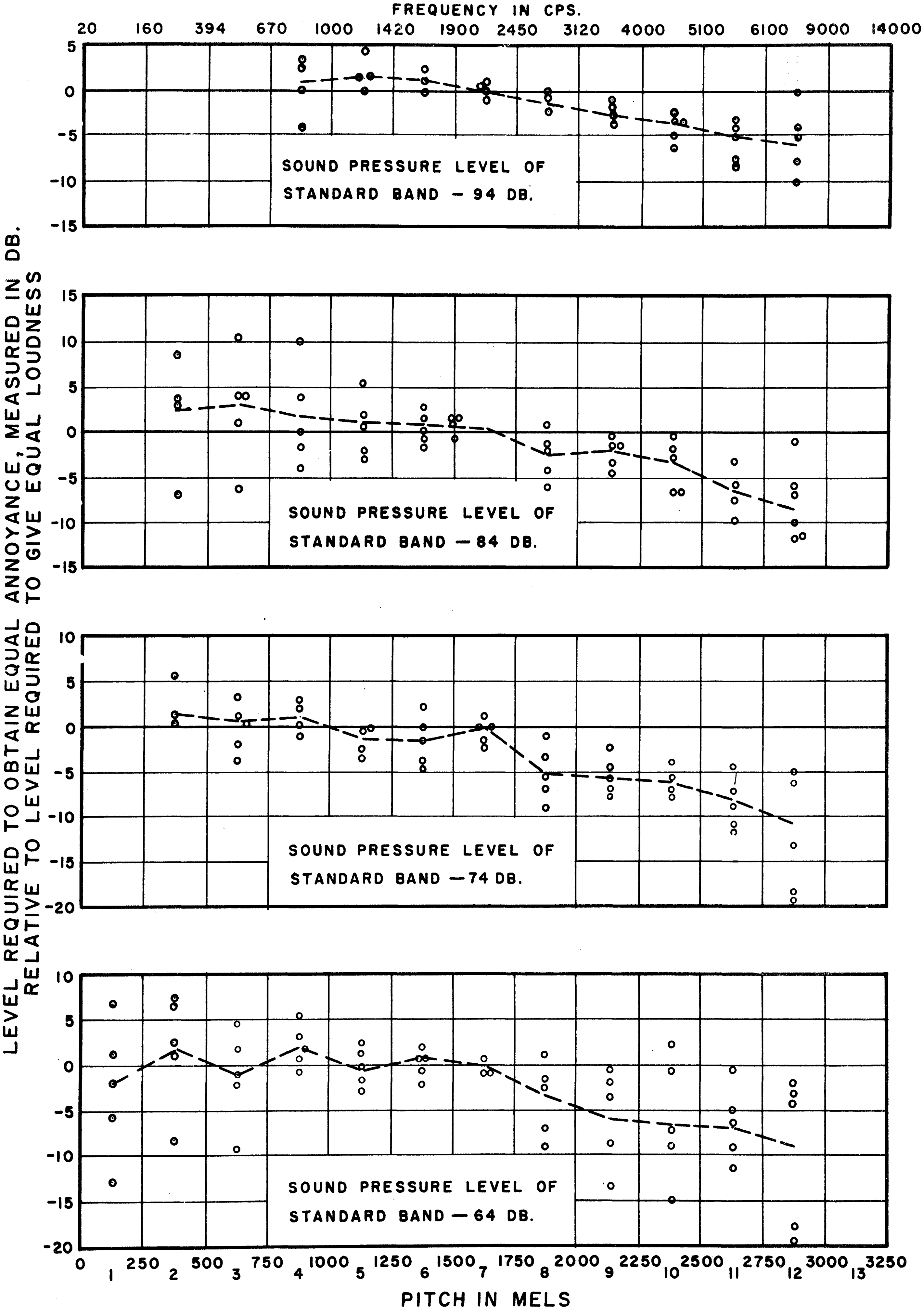



Fig. 2.25. Equal annoyance contours for bands of noise 250 mels wide; band No. 7 taken as standard. (Ref. 22)

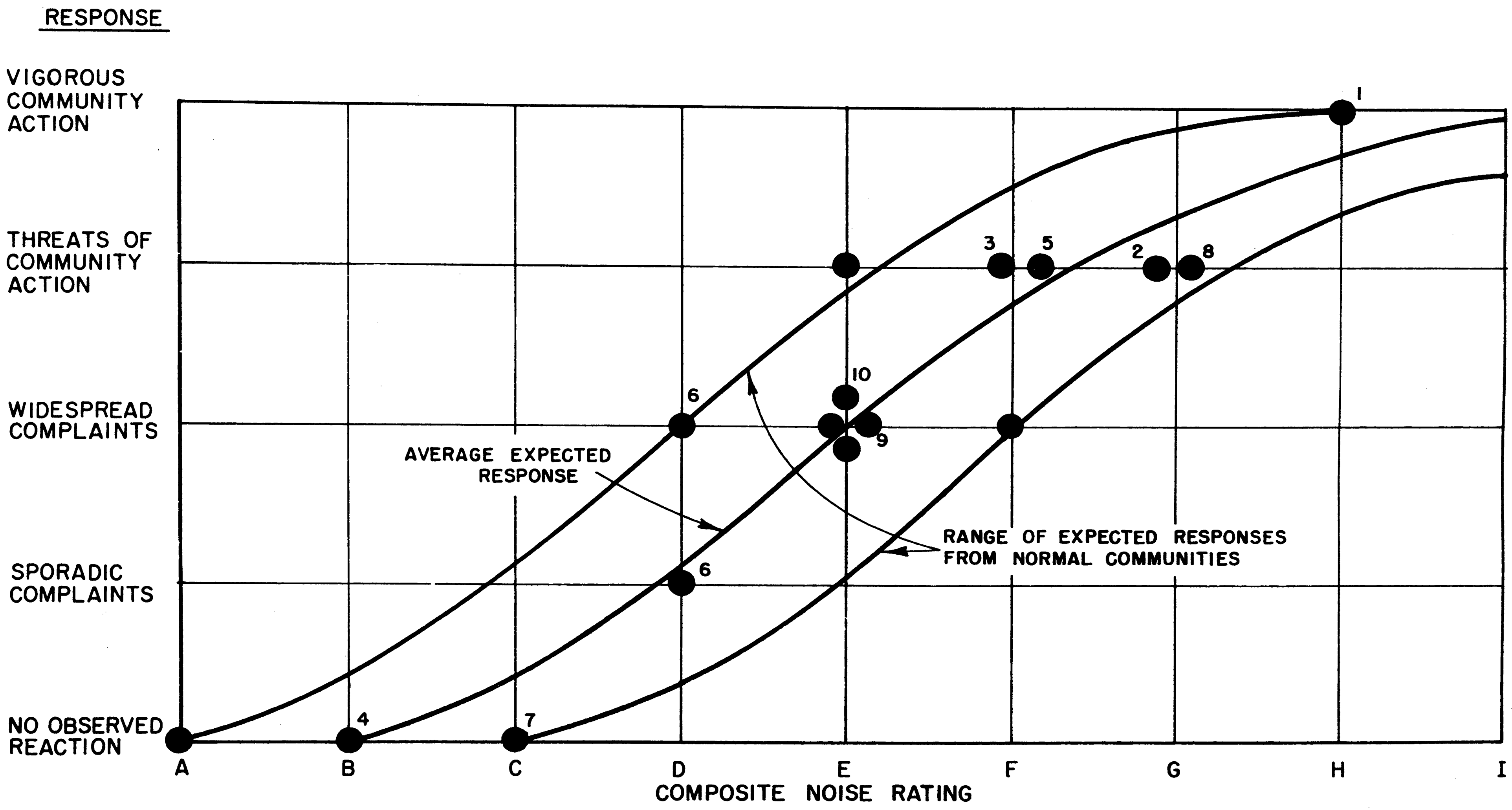



Fig. 2.26. Community response vs. noise severity
(Ref. 32)

these units. Ground-support equipment, though mobile, contains power plants of lesser horsepower and accordingly produces less acoustic energy than do aircraft. Furthermore, ground-support equipment is more amenable to noise-reduction treatments than are aircraft since maximum engine efficiency, weight, and contour design which may be affected in the course of treatment are not nearly as critical on ground-support equipment as they are on aircraft.

Engine test blocks, although sources of large acoustic power output, have the advantage over aircraft in that they are stationary, and therefore acoustic treatment, though costly, can be designed into the permanent installation of the test stand.

On the other hand, operational aircraft must, for economic and military reasons, operate at peak efficiency. Thus the possibility of applying palliative noise-reduction treatments, since these treatments usually result in added bulk or interfere with the original airflow design, is greatly reduced compared to ground equipment. However, this is not of much concern to communities adjacent to airport facilities. To such neighborhoods, a noise which interferes with their normal routine is annoying and this annoyance must somehow be alleviated. The question now arises—what can be done in a situation where operational aircraft are a disturbing influence on communities bordering an airport installation? Eventually, this problem is placed in the lap of the airport management, which may institute one or more of the following corrective measures:[33]

1. Prohibit use of the airport to aircraft which cannot be operated below maximum noise limits.

2. Elimination of unnecessary flights—especially during evening hours.

3. Designate the use of preferential runways so that both take-offs and landings occur at areas which are at a maximum distance and/or downwind from the neighboring community. This also applies to engine run-up areas.

4. It has also been suggested to lengthen the runways sufficiently to provide for a longer take-off run and thus reduce the necessity for full power application before the aircraft is airborne without reducing any safety factor in take-off.

5. Emphasize proper flying technique to aircraft operators to reduce instances of full engine-power application caused by such factors as unfamiliarity with the airport, selection of propeller pitch, flap positions, etc., during landing procedure.

6. Locate future buildings so that they act as an acoustic barrier between the aircraft and the community.

7. Plant thick foliage on airport perimeters and construct earth barriers between the airport and the community either to absorb or reflect some of the acoustic energy, and also to shield aircraft operations from the sight of the community; this latter effect presumably would have some psychological value.

8. Encourage neighboring communities to zone areas adjacent to the airport for industrial use only.

9. Finally, institute a policy of good public relations with surrounding

communities to provide for better understanding by both airport officials and residents of their mutual problems.

In summary, it should be noted that annoyance due to ground-support equipment is an important factor of special concern to all persons including those who are not directly operating such equipment.

## EAR PROTECTION DEVICES

The advent of high-energy noise sources and the realization of the severe damage which these sources can work upon the human body has given great emphasis toward design of protection devices. The experimental work has resulted in the development of at least four "receiver" noise attenuation devices or ear defenders: (1) the ear plug, (2) the cushion, (3) the helmet, and (4) the acoustic barrier. The reasons for such a variety of devices are obvious. In the first place, a combination of two or more of these devices can decrease the noise levels to the sensitive structure of the ear more than each used individually, and secondly, the body must be protected in extreme intensity noise environments since exposure produces extra auditory effects, i.e., nausea from chest-cavity resonances.

In a general consideration of the why and wherefore of ear protectors it must be realized that the air conduction path, though normally the most sensitive path, is not the sole available means of exciting the hearing nerves. Indeed, if total air conduction were eliminated, the hearing level would only be reduced by 45 db.[34] This is due to cranial excitation by the airborne sound energy. If both these means of sound conduction are sufficiently shielded, the sound energy may reach the hearing mechanism by means of the chest in which case the level is reduced by 60 db, and by shielding the entire body above the abdominal cavity from airborne sound one may expect a reduction of 80 db. It seems rather startling that a person using such drastic acoustic shielding should still be hearing a sound of 75-db level in the acoustic environment produced by some types of jet engines.

For practical reasons the ear plug and the ear cushion have been used much more extensively than either acoustic helmets or acoustic barriers. The latter is most practical where groups of co-workers must enjoy reasonable communcation, i.e., jet test sites or flight-deck islands.

Since the initial recommendations in 1890 of the ear plugs, and in 1925 of the ear cushion, much progress has been made with respect to the design of these noise attenuation devices.[35] Whereas the intial progress in formulating a workable attenuator consisted mainly of trial-and-error techniques, recent work has produced quite exacting and realistic analyses in the presentation of attenuator properties.[36,37] For example, the human ear, using both insert and cushion types of noise attenuators, can be analyzed as its electrical analog, with factors of skin compliance, transmission through the attenuator, leakage path, and bone conduction being appropriately defined.

<u>Testing of Ear Defenders</u>.—The actual testing of ear defenders may be accomplished by either of two major methods, (1) the psychophysical method in which the reaction of the individual is an inherent factor in the analysis, or (2) the physical method in which the individual is merely the holder for the attenuator, the measurements being taken across the device to determine its transmission loss. A variation

of the latter method is the artificial method in which a mock head is used as the holder. The most popular method is the psychophysical method since human reaction is desired when various ear protection devices are to be tested. With the general psychophysical method two types of measurements are used: the absolute-threshold-shift method, and the adjustment or loudness-balance method. The absolute-threshold-shift method consists of comparing the hearing threshold of the open ear with that of the defender-protected ear. Many variations are apparent, including earphone testing versus free field tests, monaural versus binaural testing, etc. Errors are likely to be incurred for several reasons.[34] The noise environment may be of a level sufficiently high to confuse the low-level threshold value; there is an apparent increase in physiological noise when the ear opening is blocked; and there is a difference between levels required to give equal apparent response between open and partially covered ears. The loudness-balance method uses levels well above absolute threshold, and is usually only subject to the error incurred by measuring levels between partially covered and open ears. The sound to be identified may be presented by earphones, by a loud speaker, or by utilizing one of each. Obvious problems of leakage, time, etc., are to be expected.

Actually, the results of various attenuation methods using a particular ear defender were not as large as might be expected. Comparison of a binaural free-field threshold shift and monaural psychophysical test, using six different ear defenders of the cushion type, yielded similar results. Figure 2.27 is a typical comparison in which the general attenuation curve varies by a maximum of approximately 5 db.[35] Other experiments show that careful control of the involved experimental parameters provides good reproducibility of results from different attenuation tests.[34]

Attenuation Values of Various Defenders and Combinations Thereof.—The value of an ear defender depends primarily upon its noise-attenuation properties, although other factors such as comfort, fit, hygiene, durability, etc., deserve some consideration when a choice is to be made. The attenuation characteristics of various plugs, cushions, and helmets have been measured by many investigators with favorable agreement.[34,35,38] Typical attenuation curves are shown plotted on Fig. 2.28. An interesting variation is the attenuation curve for a particular experimental plug filled with either mercury or viscous material as shown in Fig. 2.29.[39] As may be seen from the above presentations, there is general increase in attenuation of ear plugs with frequency, ranging from approximately 10 db at 100 cps to approximately 24 to 50 db at 6000-8000 cps. When one considers the acoustic energy output of ground-support equipment in the low-frequency end of the spectrum, the importance of using ear defenders properly to obtain maximum protection is apparent. One method of assuring such protection is by utilizing a combination of ear-defender devices. Such a combination gives added protection should one device become dislodged while the wearer is in a high-level noise environment. This can be shown by comparison of Figs. 2.30A and 2.30B. Figure 2.30B portrays the attenuation curve of each device used singly, while Fig. 2.30A gives the attenuation of several combinations.[35,36,38] It should be noticed that the maximum limits of attenuation for combinations are limited. For instance, used singly, the Mk VI cushion shows an attenuation of approximately 10 db. more than the Mk I cushion. However, the combination of each with the NDRC V51R plug shows no significant difference. This may well be the result which might be obtained with any cushion-type defender used with the NDRC V51R plug. One might assume the same result with the SMR plug and Mk I defender; that is, a combination of the SMR and Mk VI would give the same attenuation as the SMR and Mk I combination. An interesting fact with respect to communication by wearers of the NDRC V51R ear plug may be obtained from examination of Fig. 2.31.[22] Here percent word articulation was meas-

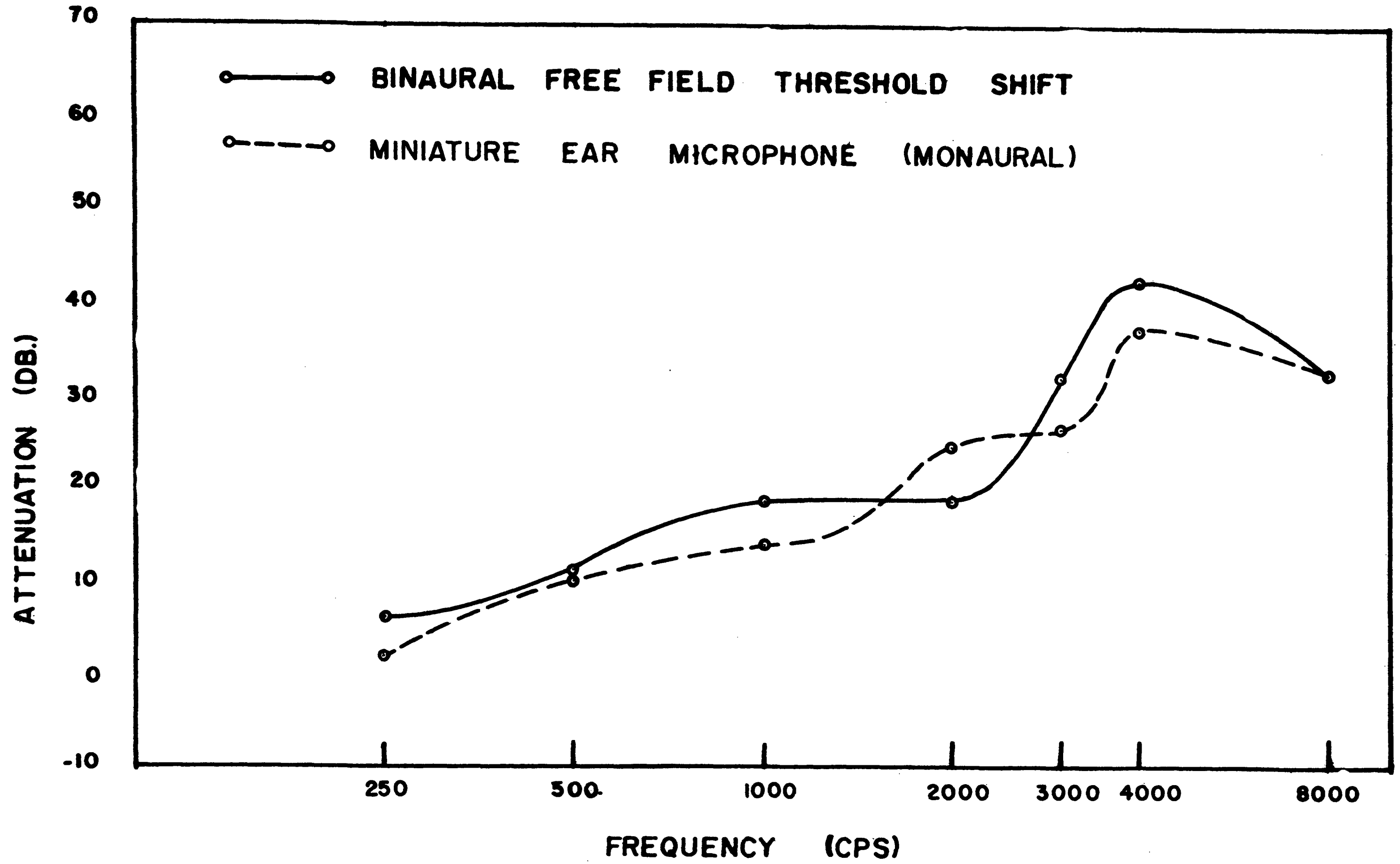
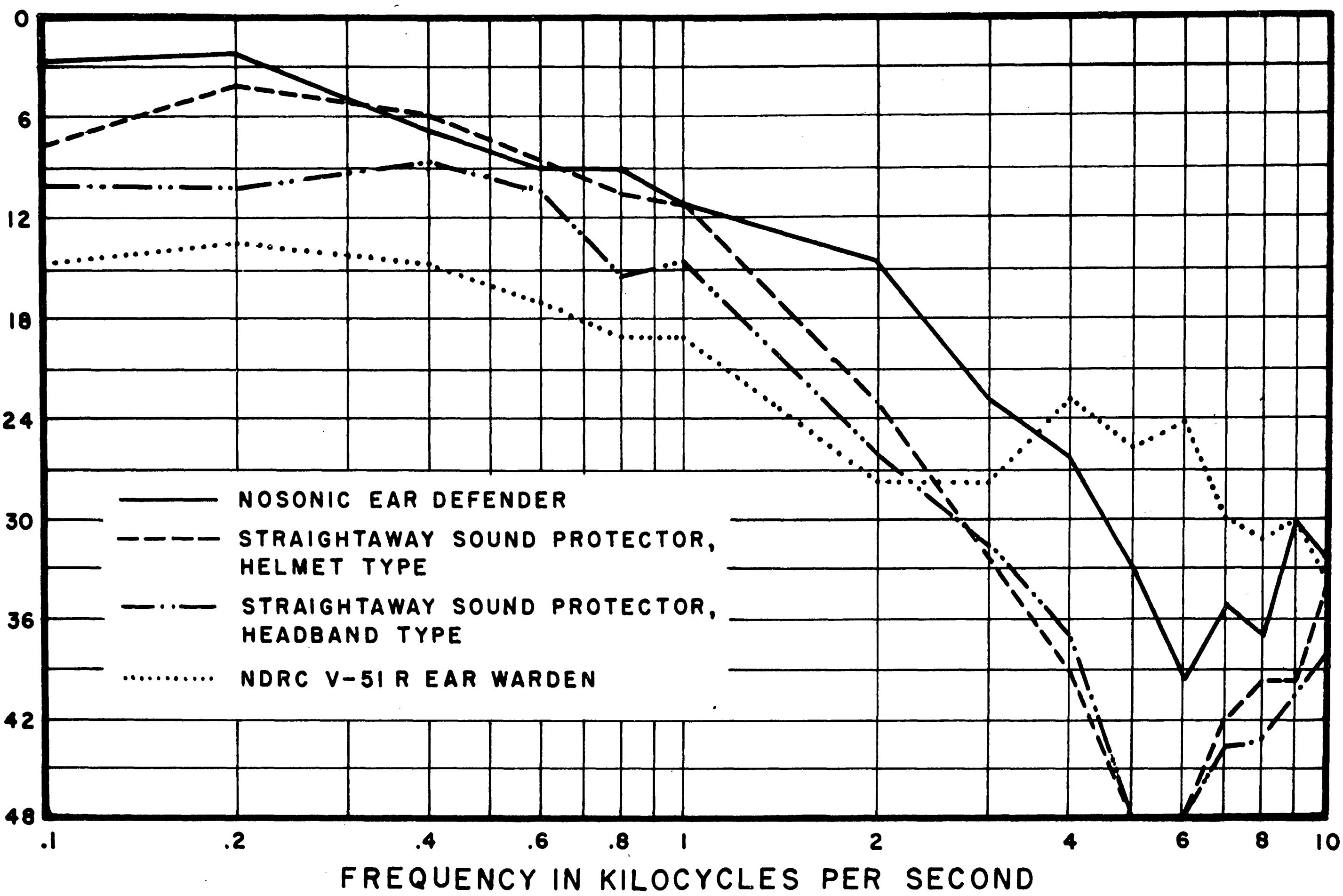
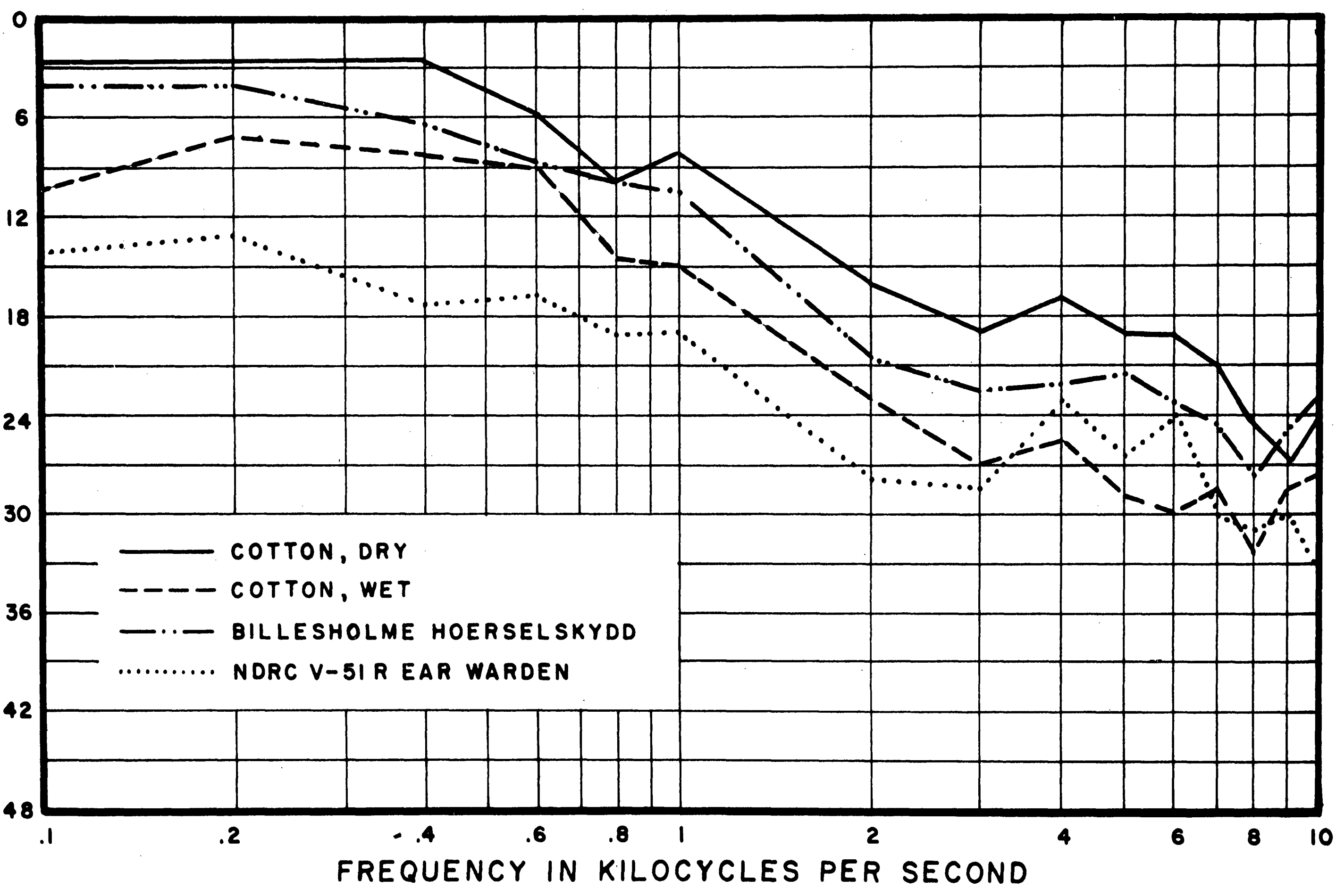



Fig. 2.27. Royal Air Force type 'F' flying helmet attenuation curves.
(Ref. 35)




SOUND ATTENUATION, VARIOUS TYPES OF NOISE PROTECTION DEVICES




SOUND ATTENUATION, VARIOUS TYPES OF NOISE PROTECTION DEVICES

Fig. 2.28. Acoustic properties of various headgear. (Ref. 38)

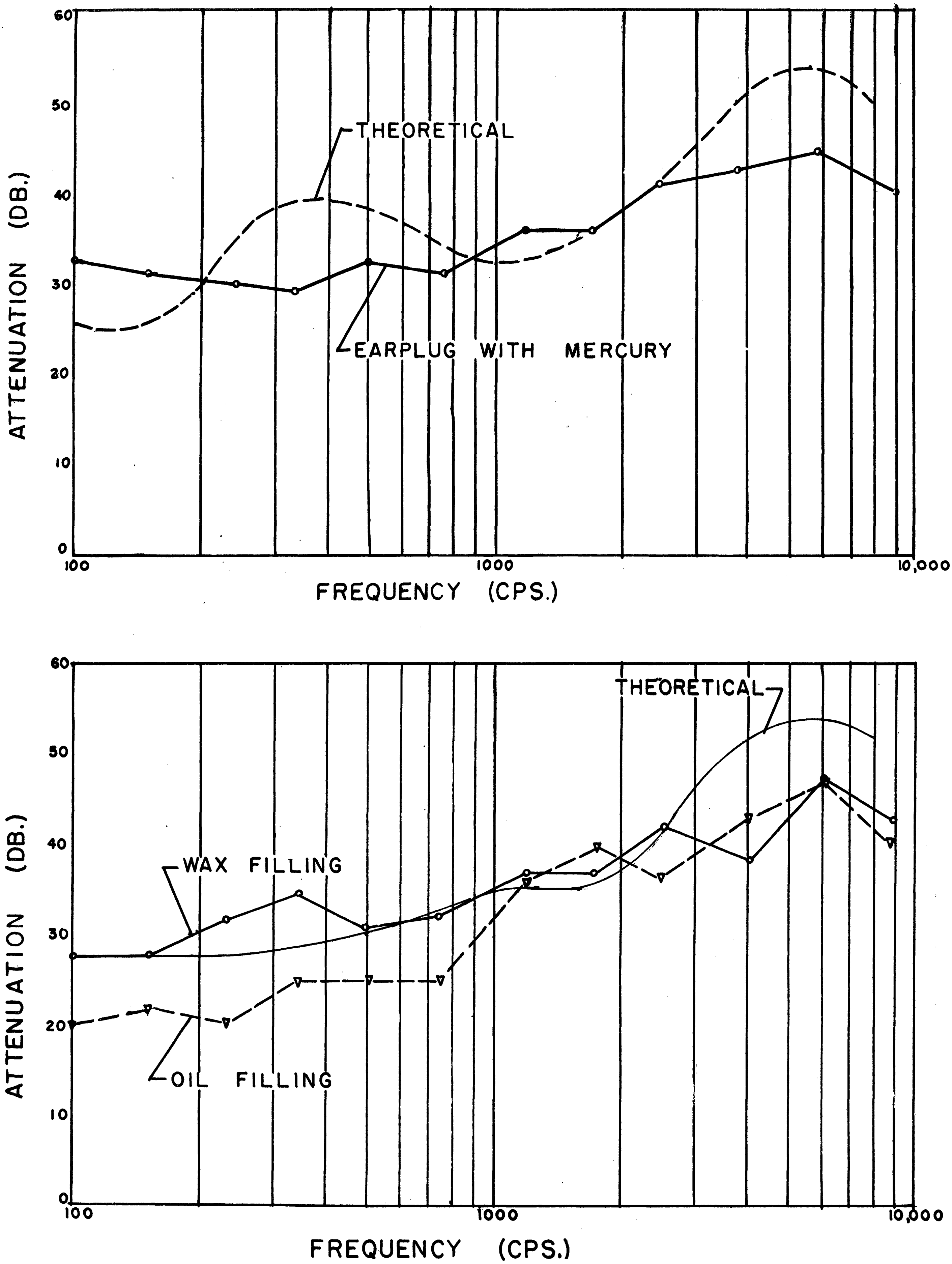
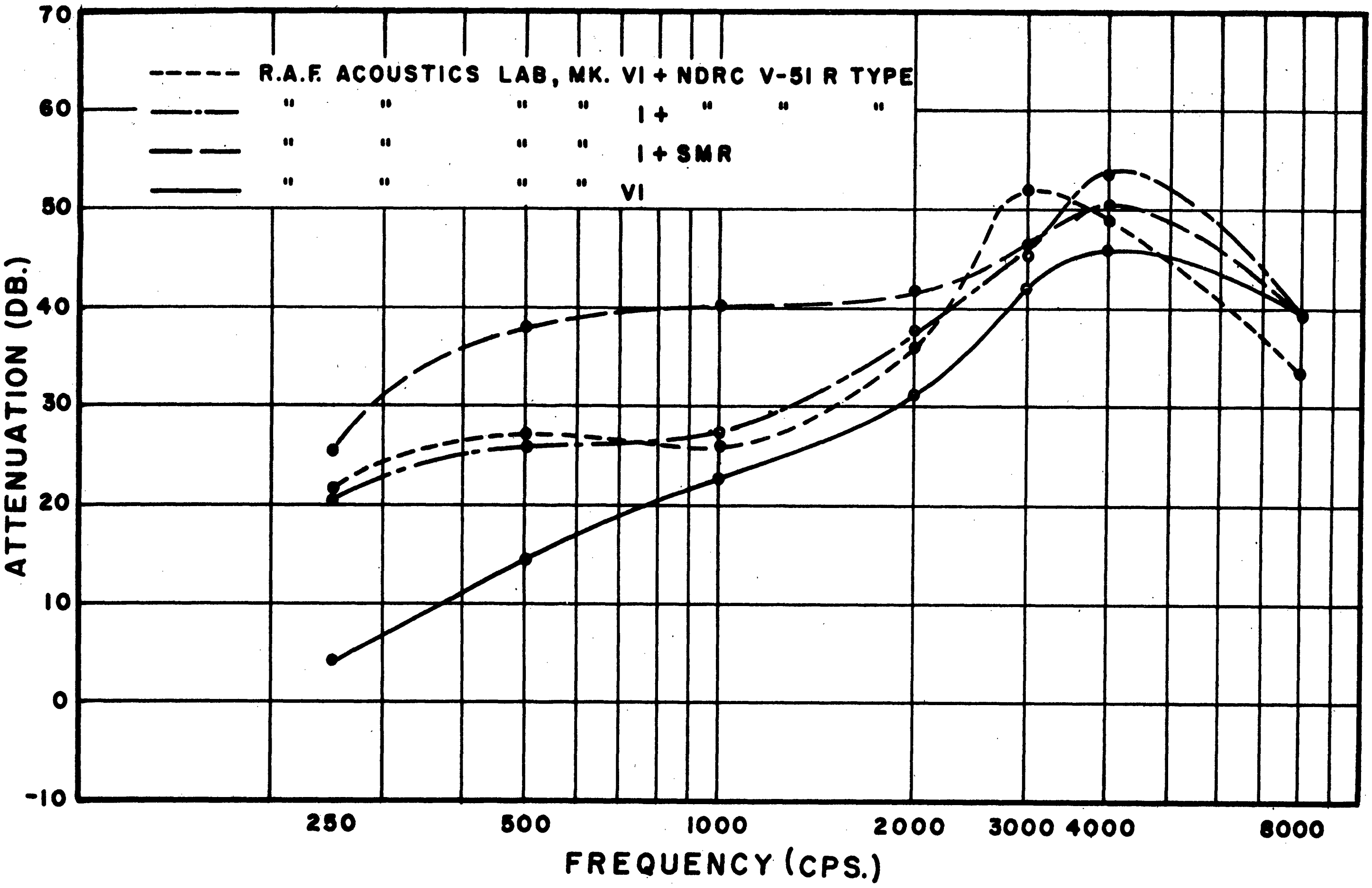
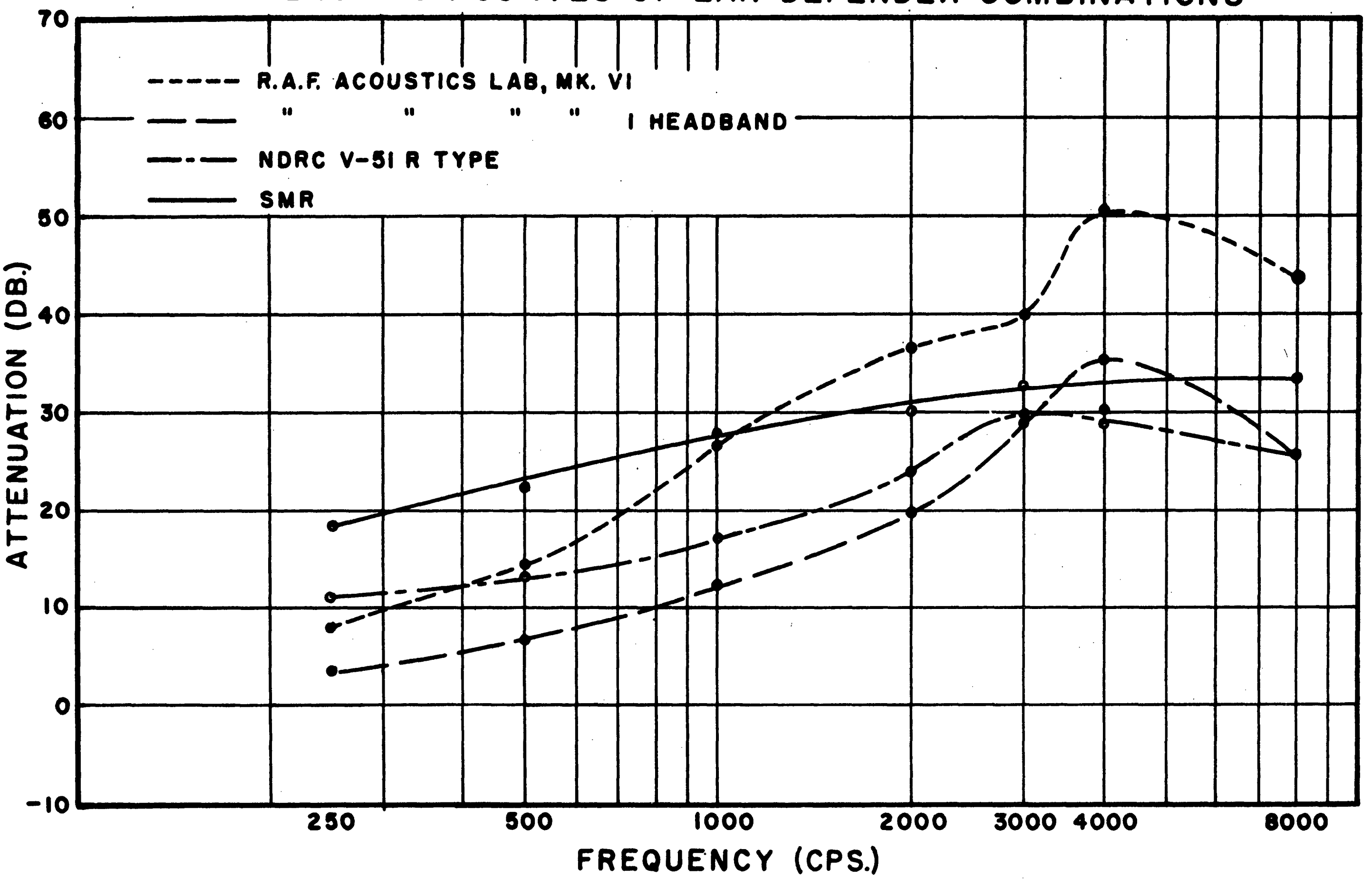



Fig. 2.29. Attenuation curves obtained on a trained listener for various earplugs. (Ref. 39)




(a) ATTENUATION CURVES OF EAR DEFENDER COMBINATIONS




(b) ATTENUATION CURVES AS DETERMINED BY BINAURAL FREE FIELD PURE TONE AUDIOMETRY

Fig. 2.30. Attenuation curves of various ear defenders singly and in combination. (Ref. 35)

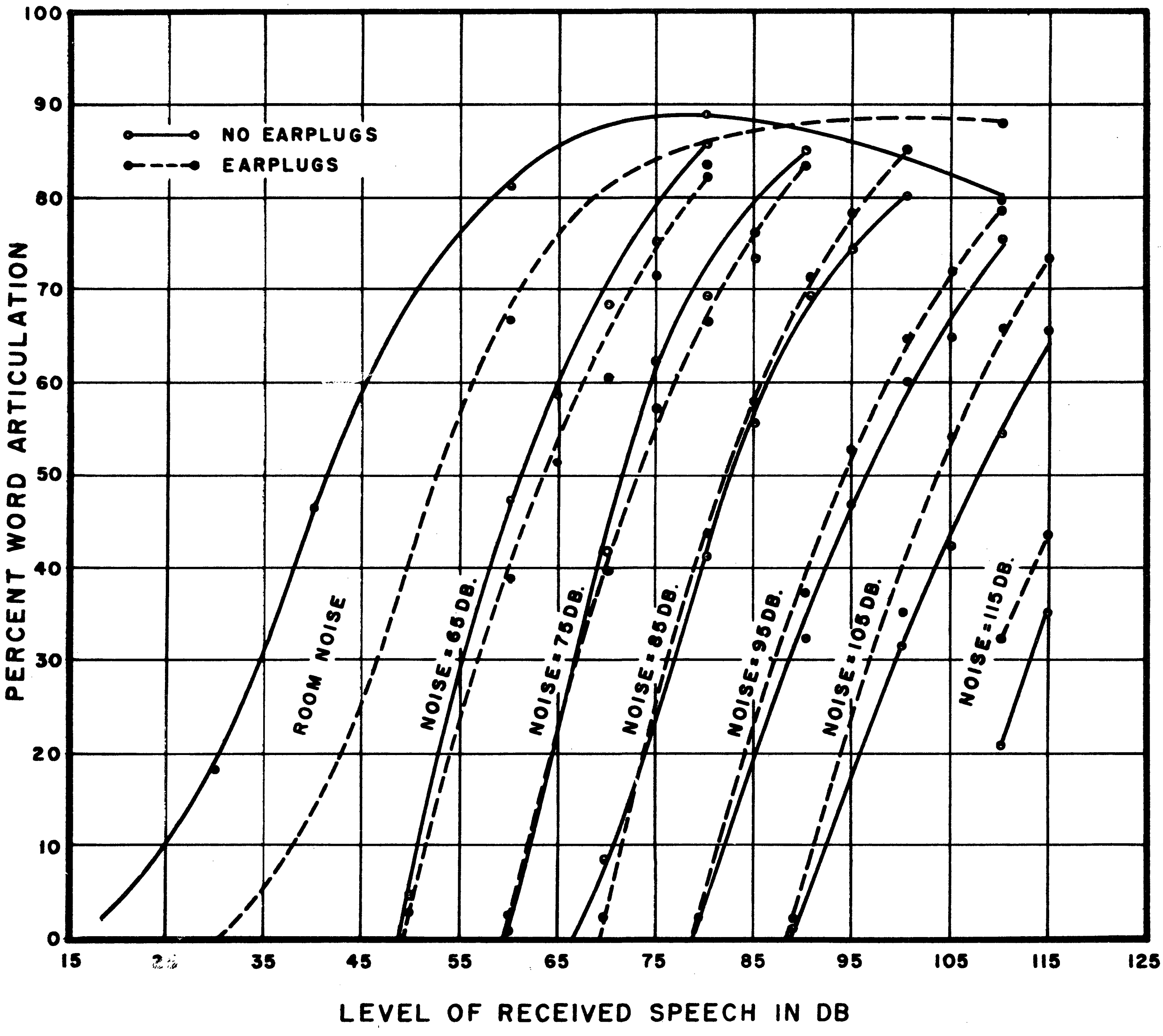



Fig. 2.31. The relation between articulation and speech level with noise level—with and without earplugs. (Ref. 22)

ured with an increasing speech level and at an increasing background-noise level. Notice that as the noise level increases over 75 db the percent word articulation becomes greater for the wearer of ear plugs than for the open ear. In other words, such plugs as the NDRC V51R type may well serve not only as ear protection at high noise levels but also as a means of increasing communication.

Ear Protection Criteria.—Application of ear-defender attenuation data has been applied to various noise criteria.[34,40] Figure 2.32B presents a simplified attenuation plot of NDRC V51R ear plug, a cushion, and the combination of both. By reference to the Bolt, Beranek and Newman Damage Risk Criterion, and to the Western Electro-Acoustic Laboratory Risk Criterion, the resulting environmental acoustic level for no serious ear damage to the wearer is shown in Fig. 2.32A. It must be remembered that these risk criteria are based on so-called lifetime exposure. A proposal for a short-term criterion has been brought forward for utilization by ground-support personnel.[40] Here a sound-pressure-level time factor has been developed so that the hearing mechanism is exposed to no greater sound energy for protected ears than for unprotected ears. The basic noise is assumed to be jet noise with a peak output between 75 and 600 cps. Figure 2.33 presents the various curves based on the "no protection" curve which allows exposures up to approximately 10 seconds for 135-db levels with a continuous drop of 3 db for every doubling of exposure time up to an 8-hour day. Thus by knowing the sound-pressure-level plot surrounding a particular aircraft, and the time required to remain in such a level, the proper protection may be chosen to avoid ear damage. It may be noticed that no criterion is proposed for 150-db levels and up, since exposure to these levels without protection produces permanent hearing loss and nonauditory effects.

## NONAUDITORY EFFECTS OF HIGH-LEVEL NOISE AND VIBRATION

One aspect of the nonauditory effects of sound has been discussed in the section on annoyance. However, there are other nonauditory effects which should be mentioned to complete the discussion. In many respects the field of nonauditory effects has been adequately summarized quite thoroughly elsewhere.[2,22] For the purposes of their relation to ground-support personnel as hazards, these effects will be lightly reviewed with the addition of some relevant data, especially with respect to whole-body vibration. As has been the case in the preceding discussion of the physiological and psychological effects of noise on humans, the main parameters here are the frequency and the sound-pressure level of the acoustic source.

High-Level Noise Sources.—The sound-pressure level in which ground-support personnel normally operate is usually above 90 or 100 db. Therefore, the physiological effects of acoustic stimulus below this level will not be considered. Indeed, a summary of results concerning such physiological reactions at low acoustic levels points out that man adapts quite well to such stimuli and, in some cases, may even exhibit somewhat greater efficiency due to his increased effort under influence of stress.[22] The deleterious effects of supersonic energy radiated by jet engines is also questionable. Though some disagreement exists, it is generally agreed that the supersonic power output of present-day jet noise is well below dangerous levels.[39,40,41,42] However, the general trend in noise reduction of jet engines places an emphasis on shifting the energy in the spectrum from the audible into the supersonic regions. Should such practices become both widely employed and more efficient, a re-evaluation of potential physiological damage by supersonic noise environments may be necessitated.

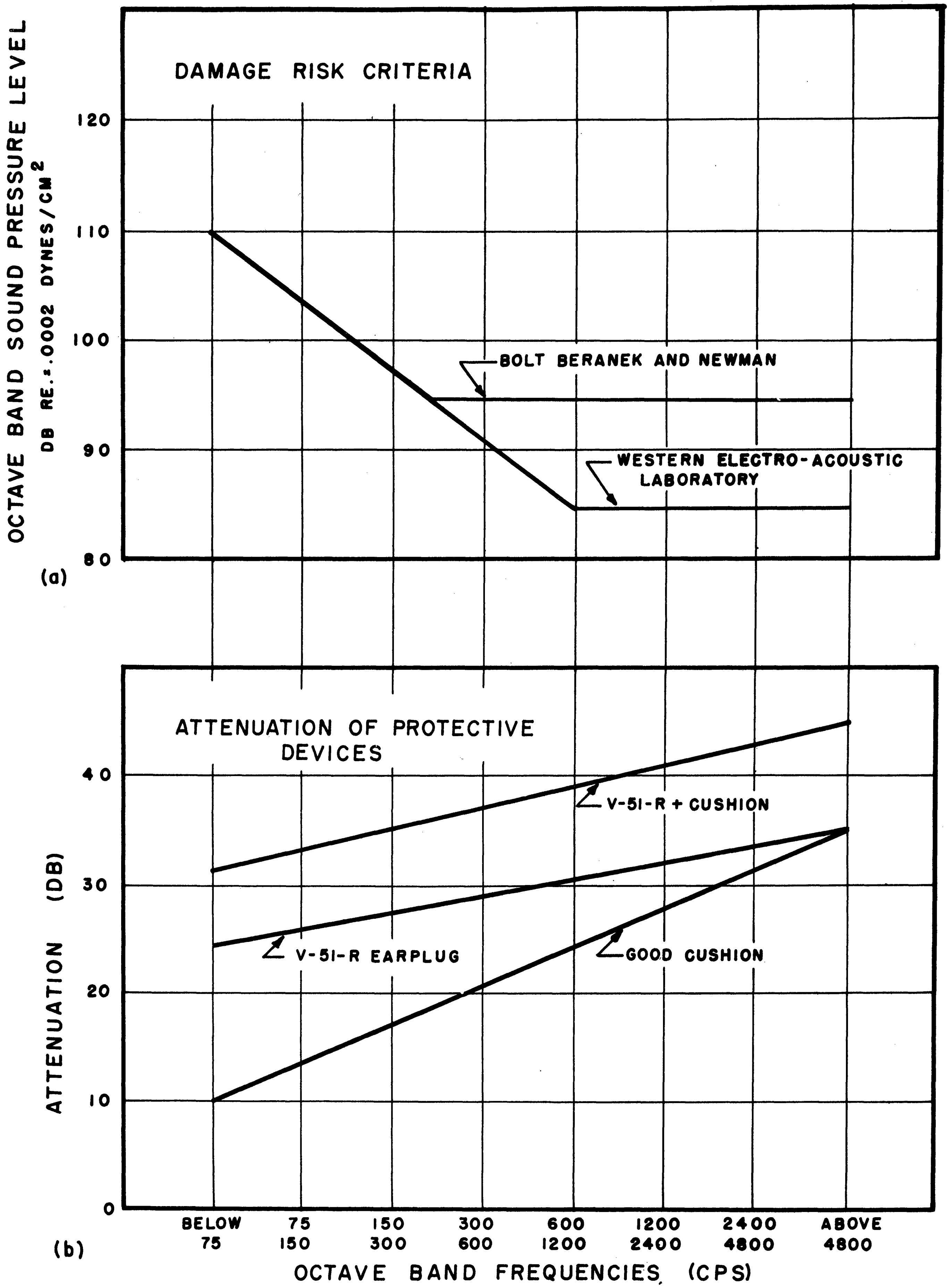



Fig. 2.32. Attenuation available with some protective devices and proposed aural damage risk criteria. (Ref. 34)

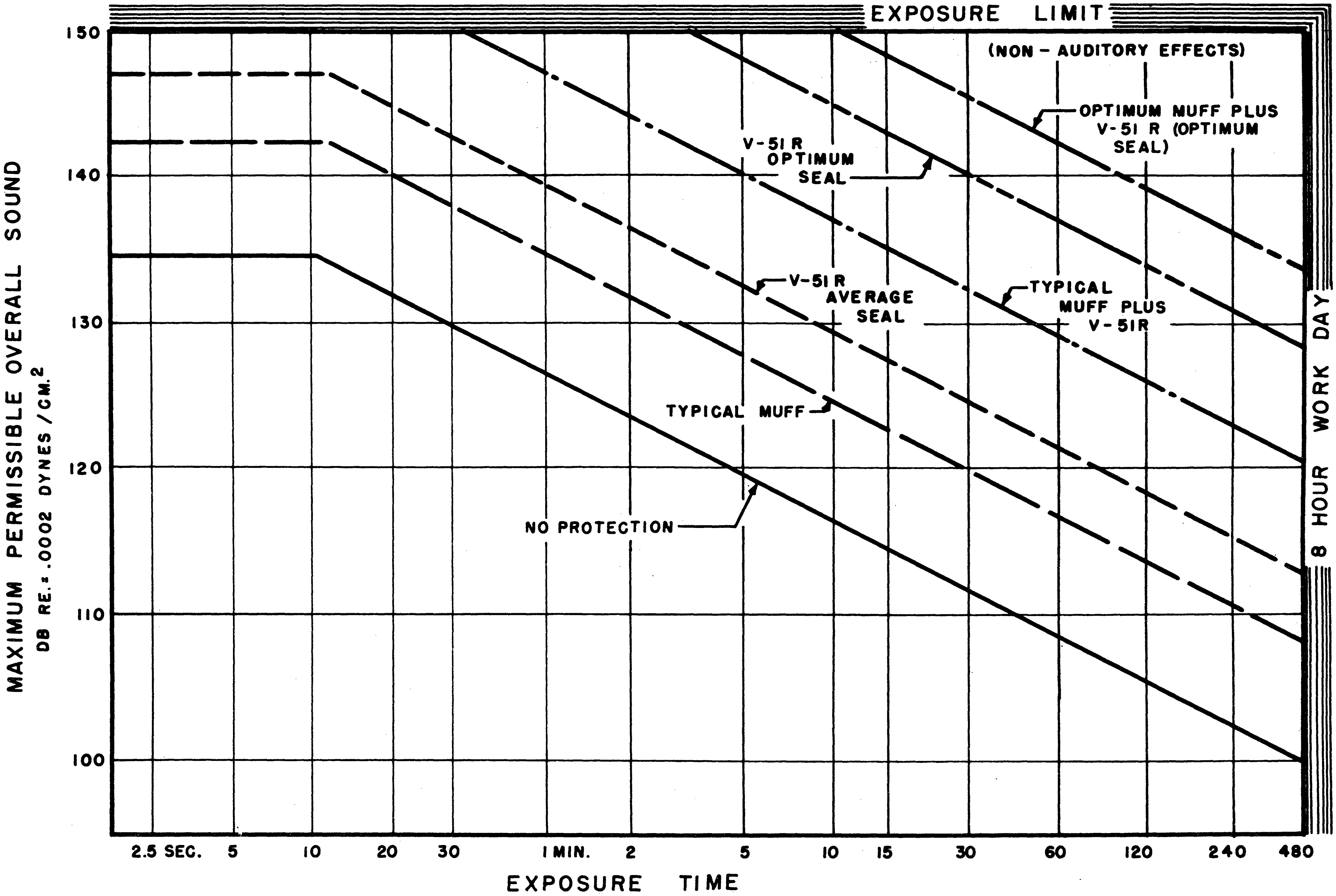



Fig. 2.33. Short time exposure criteria for jet type noise. The maximum permissible overall sound pressure level of jet exhaust noise is given as a function of the average daily exposure time for the protected and unprotected ear. (Ref. 40)

Nonauditory Effects of Airborne Sound.—Reliable information on the physiological reactions of human beings from acoustic energy in the sonic region at sound-pressure levels above 100 db is based on the reports of Finkle and Poppen,[41,43] and H. O. Parrack.[44] In the first experiment, the subjects were exposed to noise levels of 120 db from a jet source for periods of one hour for ten days and then for two hours each day for five days. Very thorough physiological checks were made of bodily functions such as basal metabolism and clotting times. Also, x-ray, electrocardiograph, and other examinations were made. No significant abnormalities were found prior, during, or after the experiments. The Parrack experiment was carried out in an acoustic environment of 150 db in which the subjects experienced such effects as cranial vibration, blurred vision, and an adverse effect on the proprioceptive reflex mechanism resulting in a "weakness in the knees." The cranial vibration was noted within the range of 700-1500 cps. Other experiments have shown this frequency to be about 800 cps.[45,46] Visual tests performed under auditory stimulus have shown little, if any, significant results.[47,48,49] As a result, it might be assumed that the blurred vision occurring in the Parrack experiment was the result of gross eyeball vibration.

Nonauditory Effects of Vibration.—Generally, when the acoustic environment of ground-support personnel is considered, only the airborne energy is considered. This may not be the only source of vibrational energy reaching the subject. In some types of ground-support equipment, such as the Consolidated MA-1, there is a place for an occupant in the equipment, and therefore he may be exposed to vibrational energy transferred from the chassis to his body. This type of nonauditory reaction experiment was carried out using a seat mounted on a platform which was vibrated at frequencies of 5-40 cps in a sinusoidal manner with peak-to-peak amplitudes varying up to one-half inch.[50,51] Certain levels of "annoyance" and "tolerance" were determined on thirteen subjects at 15, 25, and 35 cps. For the whole-body vibration study, eighteen subjects were tested, six each at 15, 25, and 35 cps, using both light and heavy vibrations. The light-vibration levels for each frequency were set at the mean of annoyance levels, while the heavy-vibration levels were set at the mean of the tolerance levels. To separate the nonauditory effects due to the whole-body vibration and those resulting from the accompanying airborne noise, each subject was also exposed to 115-db noise level and a control condition. The program consisted of tests once before exposure, four times during exposure and once after each exposure, including tests under the 115-db noise condition and the control condition. Each subject was tested for visual acuity, tremor in a supported hand, and aiming tremor. Figure 2.34 shows the effects of whole-body vibration on visual acuity. Notice that there is little decrease of acuity under noise conditions and the decrease of acuity is consistent with both an increase in vibration and a decrease in frequency. Figure 2.35 illustrates the results of the manual tremor experiment. The significant differences from the normal control condition appear at both 15 and 25 cps under both conditions of vibration, with startling loss of control apparent at the lower frequency, especially during the heavy-vibration condition. Figure 2.36 depicts the loss of aiming control, with the results again pointing toward greater effect at lower frequencies and heavy vibration.

It may be said that the major contributors of energy which tend to produce nonauditory effects beyond the annoyance stage are sources which emit both high-level sonic and subsonic energy. Should the trend toward increasing supersonic energy output of jet engines continue, these frequencies may also prove injurious.

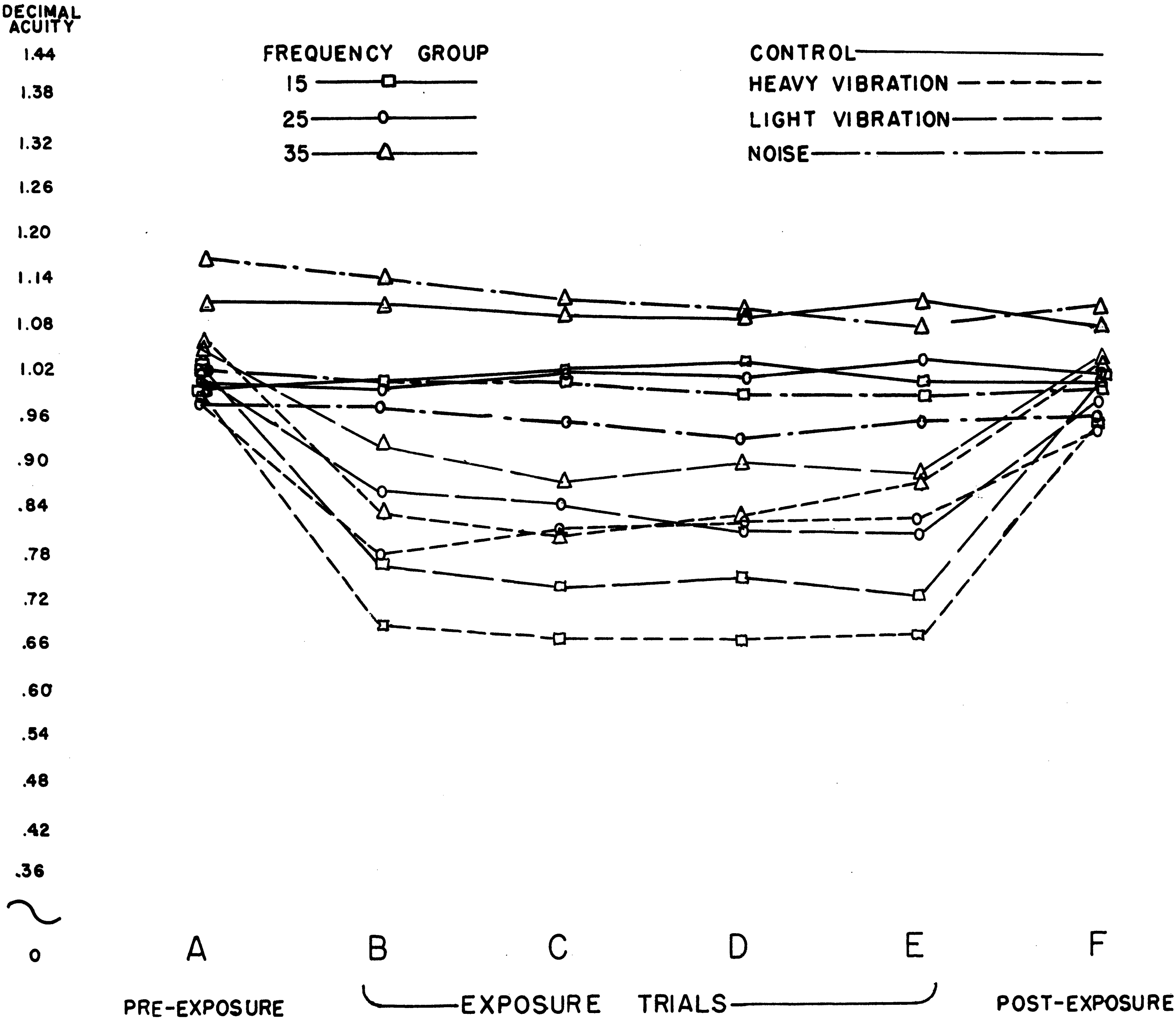



Fig. 2.34. Effects of noise and vibration on visual acuity. (Ref. 51)

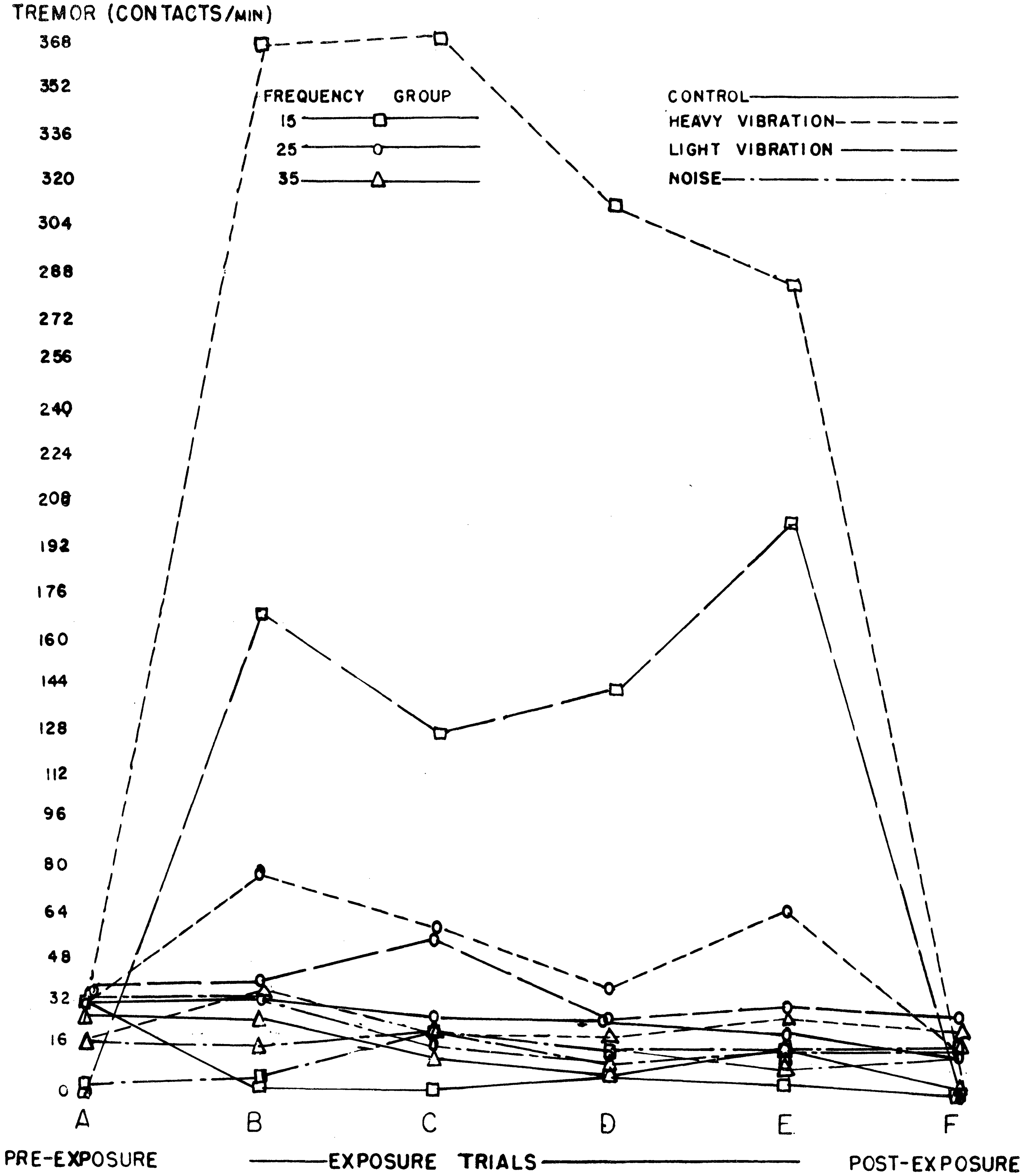



Fig. 2.35. Effects of noise and vibration on manual tremors.
(Ref. 51)

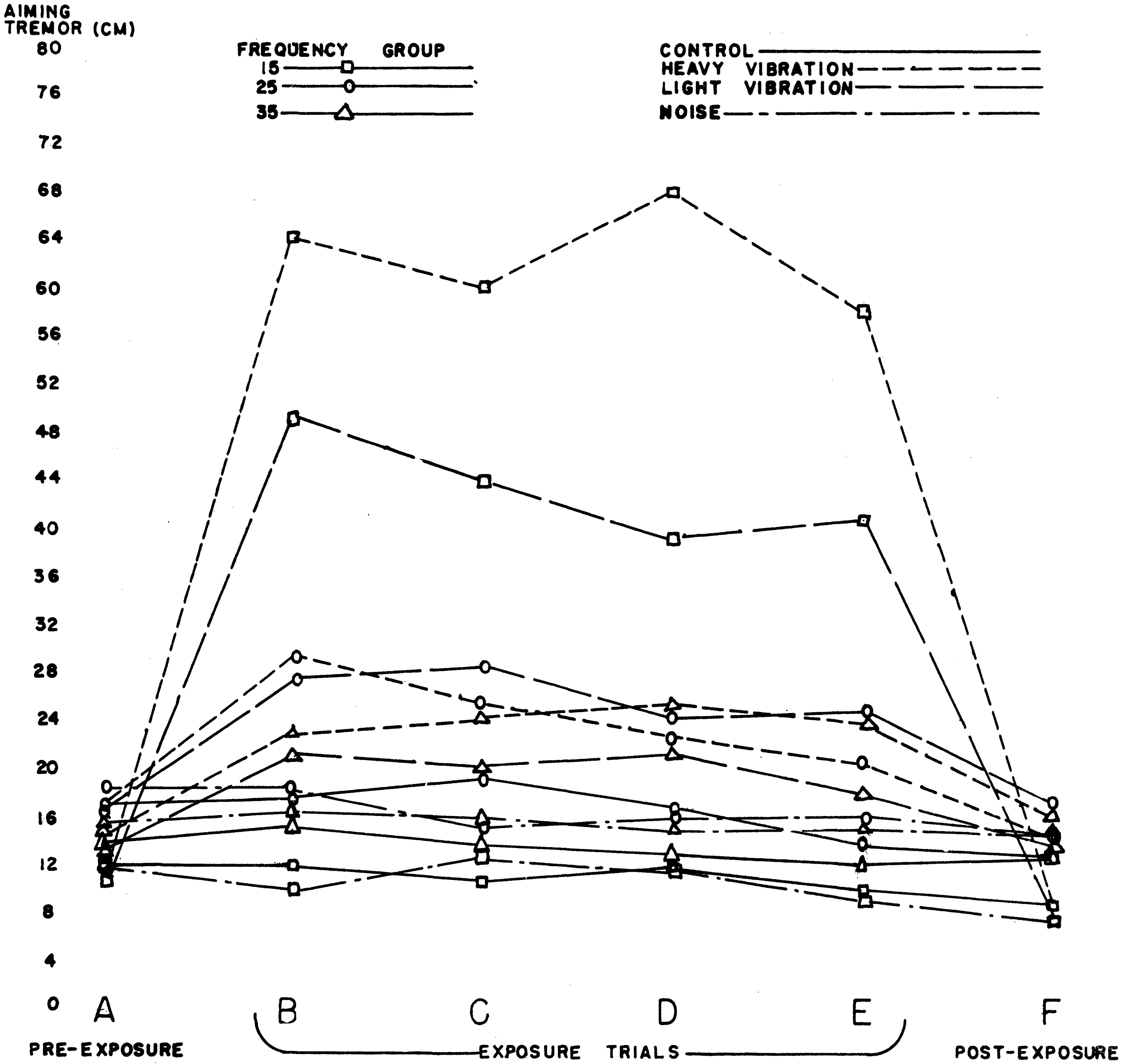



Fig. 2.36. Effects of noise and vibration on downward aiming tremor. (Ref. 51)

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SECTION III

EXPERIMENTAL NOISE MEASUREMENT AND NOISE REDUCTION

## INTRODUCTION

The primary purpose of the experimental studies reported in this section is to demonstrate in a realistic and practical manner how noise reduction may be accomplished on aircraft ground-support equipment. It was explained in some detail in the original research proposal (see Appendix A) that two broad approaches may be taken toward any general machinery noise-reduction problem, namely, palliative noise reduction and noise reduction at the source. While palliative noise reduction involves the containment and harmless dissipation into heat of the acoustic energy generated by the offending machine, noise reduction at the source implies altering the machine itself to render it incapable of generating acoustic energy without seriously impairing the machine's useful functions.

The latter approach, noise reduction at the source, is usually difficult, time-consuming, and expensive. For example, just altering one component, such as a gear or cam shaft, suspected of contributing to the machine's noise may involve considerable high-precision shop work. Furthermore, to evaluate the acoustic importance of various experimental changes, precise and elaborate acoustical measuring programs are usually required.

On the other hand, the palliative approach is often capable of providing rather large noise reductions with only moderate expenditures of time and effort. This is particularly true when low-level machine noise is not an original engineering design criterion and when the offending machines have remained untouched acoustically. Under these conditions, the acoustic evaluation programs are far simpler since only large changes in noise are sought. After the first several stages of palliative noise reduction, however, this approach also increases in complexity and expense. The acoustic improvement corresponding to each successive alteration is generally smaller, entails more effort and expense, and is more difficult to evaluate objectively. Moreover, several simultaneous alterations may be required to accomplish a significant acoustical improvement.

Scope of Experimental Studies.—Consideration of the noise problems posed by aircraft ground-support equipment, the indicated urgency for at least partial relief from such noise, and the size of a suitable noise-reduction program which could be undertaken immediately required the rejection of a noise-reduction-at-the-source approach. Such an approach would take too long a time to reach fulfillment and preferably should be included from the start in the design of new equipment rather than directed toward the alteration of existing equipment. On the other hand, the palliative approach seemed to offer opportunity for immediate demonstrable improvement by techniques which, in principle, are applicable to a wide range of ground-support equipment. Moreover, it was judged from inspections and descriptions of the several pieces of ground-support equipment most urgently requiring noise reduction that they were almost completely untouched acoustically. Hence, the simpler and more dramatic first steps of palliative treatment were assumed applicable and the experimental phases of this research were undertaken on this basis.

According to the original plan, the items of ground-support equipment submitted for study were to fall into two categories. Two items, namely, the C-26 Gasoline Generator Set and the MA-1 Turbine-Driven Air Compressor, were to be studied most extensively, and as much noise reduction as possible demonstrated experimentally. In the second category were an indefinite number of other items of ground-support equipment to be submitted for preliminary study only. These were to furnish background information about the broad problems of noise reduction as applied to ground-support equipment in general.

In carrying out the experimental palliative noise-reduction program, major importance was attached to two criteria in addition to acoustic effectiveness of the treatment; namely, (1) alterations to the ground-support equipment were to be made in such a way that each item could easily be returned to its original untreated condition, and (2) experimental treatments were to be practical in nature and to be indicative of treatments which ultimately could be incorporated into the military hardware and withstand the rigors of environmental qualification tests. Therefore the palliative treatment applied was to be an interim measure since the final complete and detailed specification of the acoustically revised item of ground-support equipment requires the coordinated design-engineering effort of a wide range of specialists, including, of course, competent acoustic specialists. Such a complete specification obviously lies beyond the scope of this research project.

Equipment Studied.—The experimental program, as actually carried out, depended upon the particular items of ground-support equipment which could be made available for study. No items were immediately available at the start of this project. Several months later, the A-1 and the C-26 Gasoline Generator Sets became available on extended loan, and were studied extensively as reported below. Both general similarities and individual differences in the necessary palliative treatments are evident. After considerable noise reduction had been demonstrated on both generator sets, research effort was concentrated on the C-26 Generator Set since it had been designated originally as an item of major interest.

The MA-1 Turbine-Driven Air Compressor did not become available for study until much later; indeed, for a time it looked as if this item would never become available. Numerous measurement problems were encountered with this particular item and the noise-reduction problem as finally defined by objective measurements was found to be quite different from the one anticipated at the inception of this research program. Normal palliative treatments can, at best, provide only slight noise reduction on this machine in its present configuration. Considerable experimental effort was devoted to eliminating interpretational difficulties and to delineating the actual noise-reduction problem.

The Consolidated MA-1 Multipurpose Unit was supplied for informational purposes only. It arrived for an unspecified loan period which was abruptly terminated after a short time with no advance warning. Considerable background information was collected on this item before it had to be returned although certain supplementary tests had to be foregone.

The B10B Generator Set and the MDX Cart were measured, as authorized, at the Wolverine Diesel Power Company under non-free-field conditions when it was determined that these units could not be made available for more extensive free-field surveys. The results obtained from these non-free-field measurements are of very limited value compared to the free-field surveys performed on the other items of ground-support equipment studied.

No additional items of aircraft ground-support equipment were supplied for study during the course of this research contract.

Experimental Approach.—The general experimental procedure employed for the several items studied began with a free-field "as-received" octave-band survey under representative operating conditions to establish a point of departure for further work. Interpretation of the first survey data, based on experience and supplemented by additional tests as necessary, was used to select the appropriate palliative first treatment. When this first treatment had been accomplished, its effectiveness was evaluated by appropriate tests. The achievement of noise reduction with a particular treatment served the two-fold purpose of verifying the original identification of the "problem" and of demonstrating at least one way to accomplish the palliative treatment.

Further systematic cycles of identification of the noise "problem," selection of the treatment, application of the treatment, and evaluation of the result were carried out until a point of diminishing returns was reached. This limiting point was established either by the prohibitive alterations of the item required to proceed further, or by limitations in the effectiveness of the commercial materials used for the palliative treatment. In either case, the objectives of this research program as outlined above were satisfied.

To proceed further with palliative noise reduction would involve longer, more expensive and extensive programs yielding noise reduction in much smaller steps and, except in the case of the MA-1 Turbine-Driven Air Compressor, would probably not be justified for the current items of aircraft ground-support equipment. However, if new models of equipment are anticipated, additional research with a view to influencing the design before it is frozen for production would certainly be justified.

The MA-1 Turbine-Driven Air Compressor presents a unique noise-reduction problem requiring a separate research program. It should be treated with a noise-reduction-at-the-source approach aided by certain palliative techniques after the jet-exhaust noise is brought under control. Because this gas-turbine engine does not fly, there appear to exist several techniques and treatments, which, if developed by appropriate research, give promise of controlling its noise. However, these noise-control measures are also completely beyond the stated scope of this research program, and thus need to be the subject of some future research program.

To carry out successfully the free-field noise surveys and other tests reported below, it was necessary to accomplish a three-fold coincidence. That is, the item of ground-support equipment had to function properly, the acoustical measuring instrumentation had to be fully operative and in accurate calibration, and the weather at the test site and human activities in the vicinity of the test site had to be suitable. For example, winds above a few miles per hour distort the far-field sound patterns and also generate extraneous noise at the microphone, either of which can invalidate the survey measurements. Thus the performance of a free-field survey requires holding both the machine and the instrumentation in readiness until a suitable test-site condition prevails, and then accomplishing the full measurement while these ideal conditions for proper operation of the machine and instruments continue to hold. There is no need to discuss this aspect of the tests further except to state that numerous instances of instrument failure and/or machinery malfunction delayed or invalidated many tests and absorbed much time and effort in addition to that reflected by the successful tests reported below.

Organization of Material.—In the detailed discussions which follow describing the experimental program on each individual item of equipment studied, only those data and graphs directly applicable to the discussion are included with the text. The associated information regarding instrumentation and calibration have been assembled in Appendix B while the more detailed original data are presented without discussion or interpretation in Appendix C. By utilizing this organization of the extensive material, it is possible to retain the emphasis of the following discussions on the primary acoustical aspects of the problems.

Each free-field survey is presented by a group of three graphs: (1) the computed overall-noise-level directionality pattern plotted in polar coordinates at 30-degree intervals; (2) the average octave-band noise-level profile or spectrum averaged over all measuring directions; and (3) a composite plot showing the directional deviations at 30-degree intervals from the averages displayed in the preceding graph. This somewhat unconventional display presents the relevant acoustic features uncovered by each survey in compact form, facilitating interpretation for noise-reduction purposes. Other more conventional displays can be derived easily from these graphs or from the data in Appendix C.

## TYPE A-1 GASOLINE GENERATOR SET

From the acoustical viewpoint, the Type A-1 Gasoline Generator Set consists of a small two-cylinder, four-cycle aircraft engine (Continental Model PC30-2, rated 30 hp at 2400 rpm) geared to a d-c aircraft generator (Westinghouse). Cooling air for both the engine and generator is supplied by a vane-type blower gear-driven by the engine. This assembly of machinery is mounted in a four-wheeled aluminum cart measuring roughly four feet wide by six feet long by four feet high. Thin flexible washers of some type have been inserted between the engine and the cart at the mounting points, but they are too thin and stiff to constitute resilient mounting in the ordinary sense. The maximum recommended steady output is about 280 amperes at 28 volts dc.

The aluminum cart is of integrally stiffened construction. The end adjacent to the towing hitch contains the fuel tank, battery, power-cable storage, and control panel. The opposite end encloses the engine and generator. Cooling air is ducted in through a grill in the machinery end of the cart (opposite the towing hitch) and exhausted through the cart bottom. The engine exhaust is directed vertically through the top of the cart and the internal exhaust stack is surrounded by a duct or augmentor which provides ejection of air circulating within the cart. Figure 3.1 shows a schematic elevation of the A-1's interior arrangement. Figure 3.2 presents a schematic plan view which identifies the approximate location of various important features, and defines the azimuthal directions at which all of the free-field sound-pressure measurements were taken on this particular item of ground-support equipment.

As indicated in Fig. 3.2, the cart has four hinged side-panels, two on each side. During operation, the solid panel covering the control panel must be opened and the

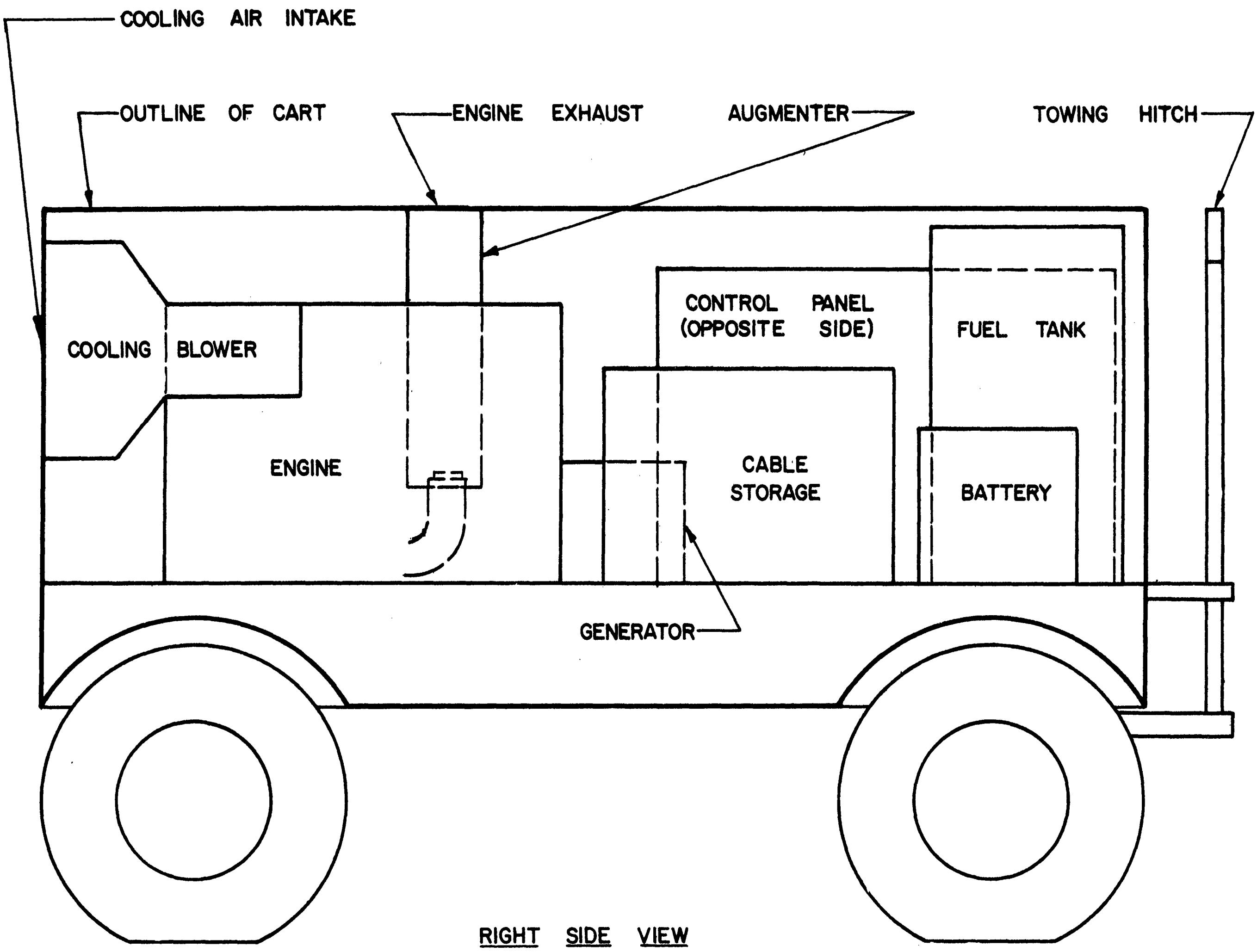



Fig. 3.1. Schematic elevation of A-1 interior arrangement.

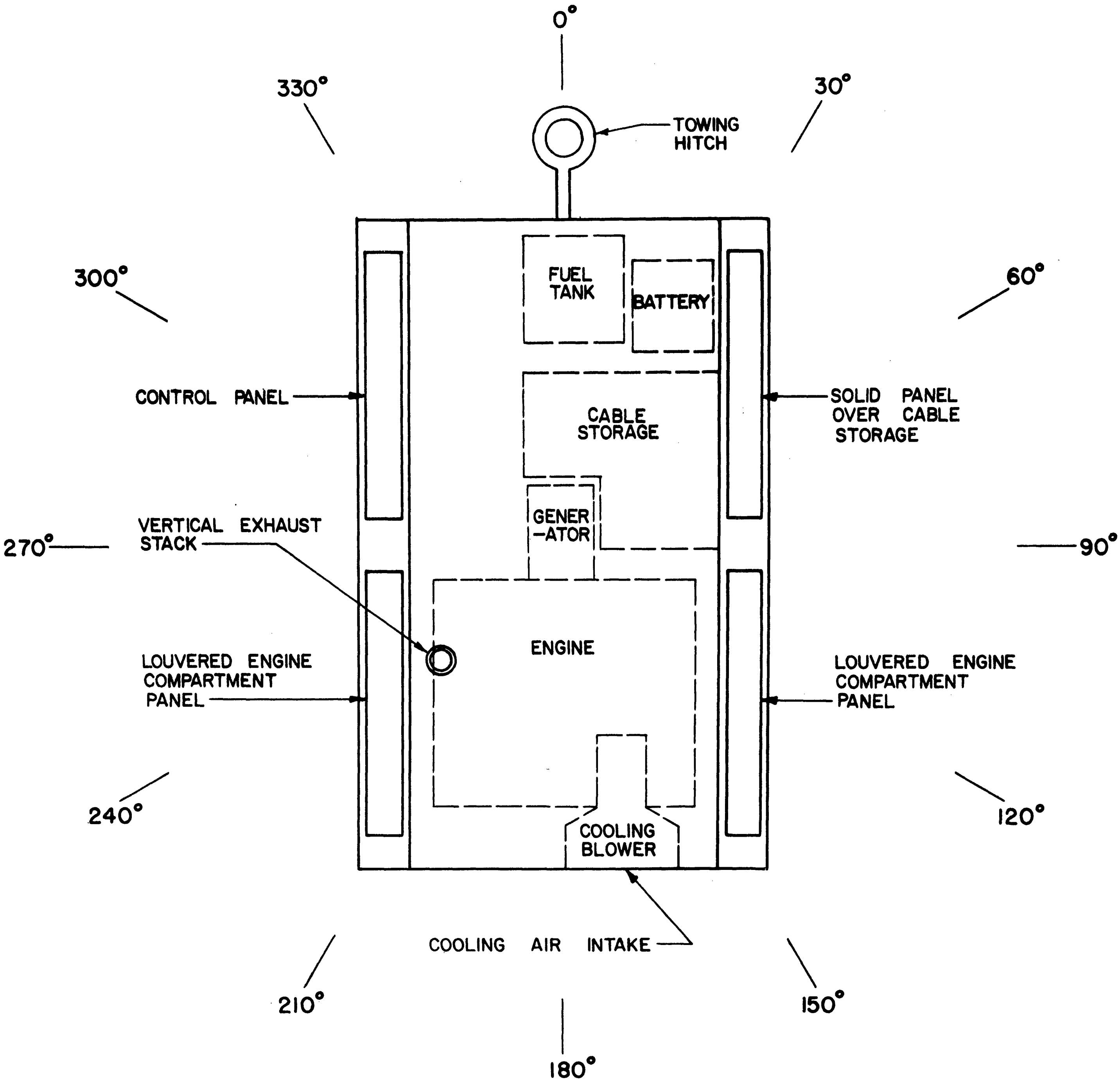



Fig. 3.2. Schematic plan view of A-1 generator set.

control panel itself does not completely fill the resulting aperture. The solid panel opposite, providing access to the battery and cables, may be left closed and the two louvered, hinged panels on opposite sides of the engine compartment normally remain closed.

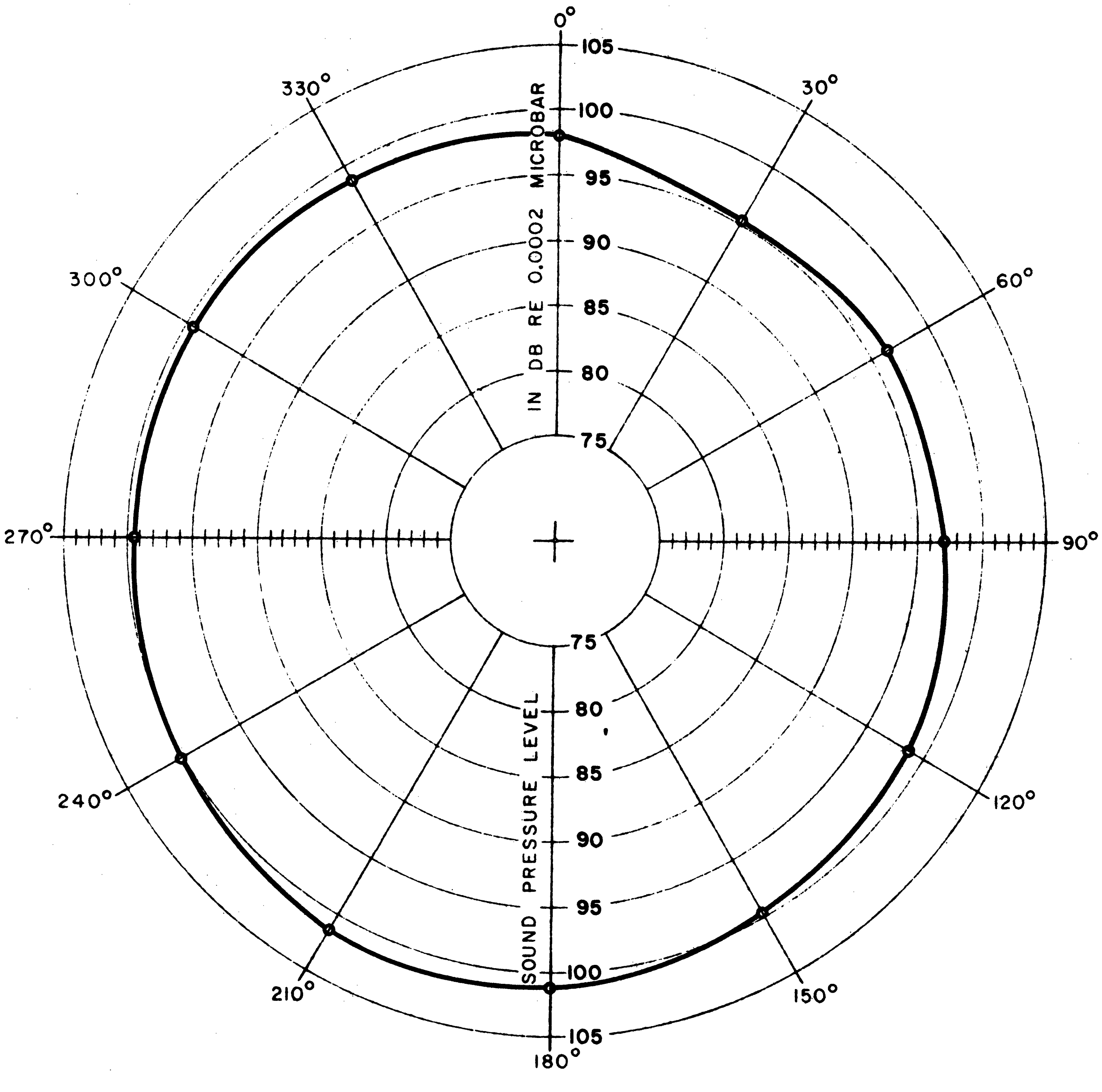
As-Received Free-Field Noise Survey.—As a preliminary step to evaluating the problem of reducing noise radiated by the A-1 Generator Set, a free-field, octave-band noise survey was conducted on the machine in the as-received condition. Measurements were carried out at the free-field site (see Appendix B) with the microphone located at a distance of 30 ft from the center of the A-1 and at a height of 5 ft 5 in. above the ground. These measurements, taken at 30-degree intervals around the machine, are asserted to be a reasonably accurate characterization of the far-field noise radiated by the A-1. Actually two complete octave-band spectra were obtained at each azimuthal position, one taken immediately after the other to check on the short-time variability of the experiment (see Appendix C). The agreement of the two sets of data was good, and consequently the values from both sets of data were averaged for presentation here.

During this free-field survey, the A-1's operation was maintained as constant as possible without involving elaborate control procedures completely unrealistic in terms of normal operation. A convenient electrical configuration of the load banks dissipated a steady current of 195 amperes while the engine operated at 2500 rpm according to the instruments on the A-1's control panel (see Appendix C for additional operating data). This 195-ampere load was selected as a standard condition for all subsequent noise tests on the A-1.

The results of this noise survey are presented concisely in Figs. 3.3, 3.4, and 3.5, which show, respectively, the polar distribution of the computed overall noise, the average octave-band noise profile or spectrum, and the directional deviations from the average of the individual octave-band noise. Several important features are immediately evident. No marked directionality appears in the overall noise presented in Fig. 3.3 and the levels attained are quite comparable to those produced by the much larger C-26 Generator Set. Figure 3.4 shows, that on the average, the spectrum is dominated by noise occurring in the 75-150 and 150-300 cps octave bands but that appreciable high-frequency noise is present also. Figure 3.5 indicates that no obvious directionality occurs in the lower-frequency octave bands, but that marked directionality sets in above about 1000 cps. A general high-frequency maximum occurs at bearings in the neighborhood of 180 degrees and a corresponding minimum in the neighborhood of 0 degrees.

Analysis of Initial Noise-Reduction Problem.—On the basis of the as-received noise survey report above, listening by ear while conducting the survey, a careful visual examination of the machine, and experience with similar machinery items, the initial noise-reduction problem may be interpreted as follows. The dominant low-frequency noise probably consists principally of exhaust tones. These will occur at a frequency spacing corresponding to one-half engine rpm for a four-cycle engine, i.e., at about 21 cps intervals in this case. The average octave-band maximum in the 75- to 300-cps range is typical for an unmuffled internal combustion engine of this type operating at moderate speed. The lack of low-frequency directionality can be attributed to two causes. First, lower frequency sounds usually do not exhibit as pronounced directionality as higher frequency sounds. Second, if these low frequencies issue from the vertical exhaust stack of the A-1, then, from the geometry of the situation, we would not expect to find directionality in a horizontal measuring plane,

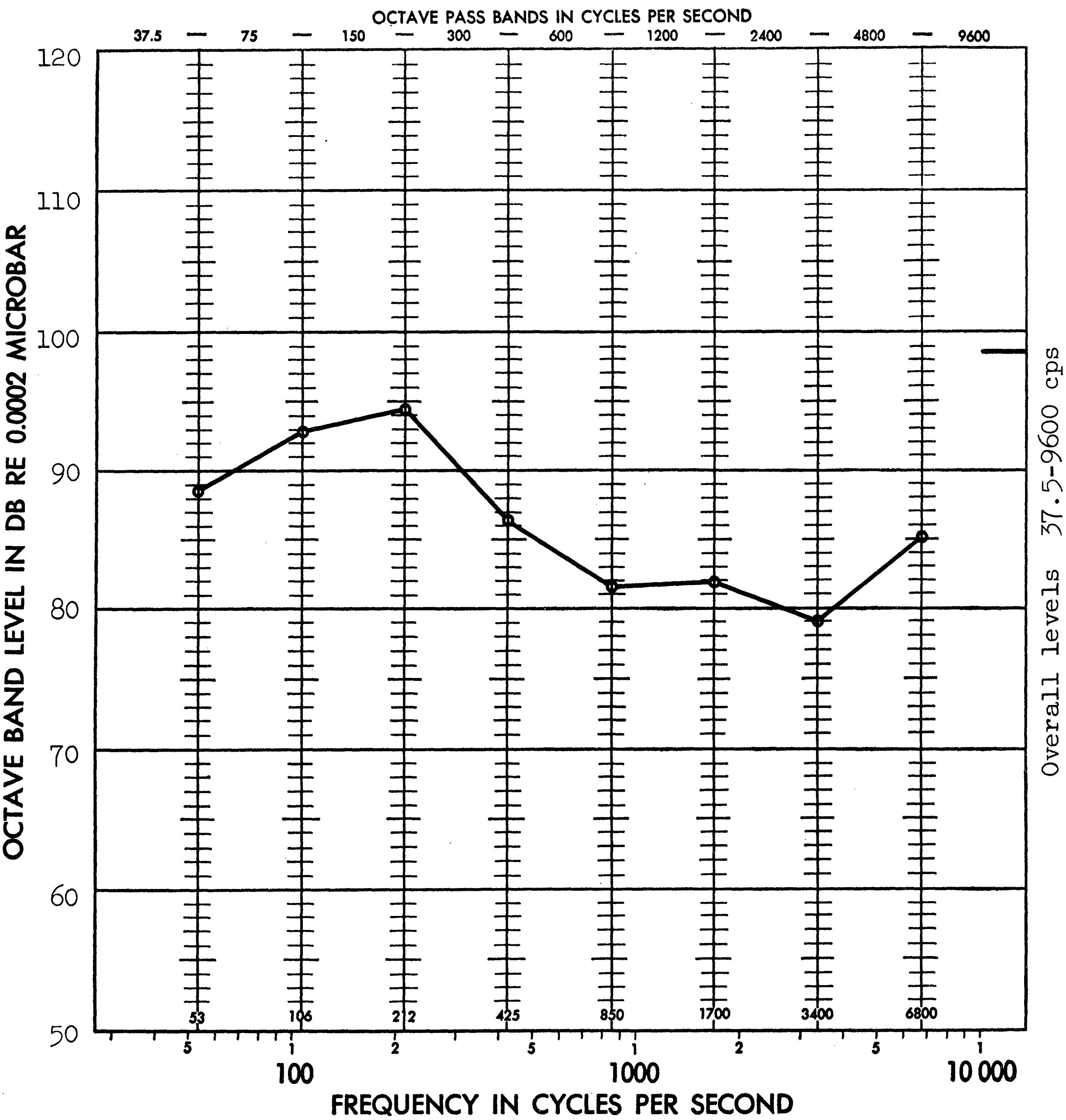
Free-field conditions
Microphone distance 30', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type A-1 gasoline generator set, as-received condition
2450-2650 rpm, 190-204 amps, 28.5-29.2 volts dc
Tested 29 July - 2 August 1955

Fig. 3.3. Polar distribution of overall noise; A-1, as-received.

Free-field conditions
Microphone distance 30', height 5'5"
Average of 12 octave-band sound pressure levels




Type A-1 gasoline generator set, as-received condition
2450-2650 rpm, 190-204 amp, 28.5-29.2 volts dc
Tested 29 July - 2 August 1955

Fig. 3.4. Average octave-band noise profile; A-1, as-received.

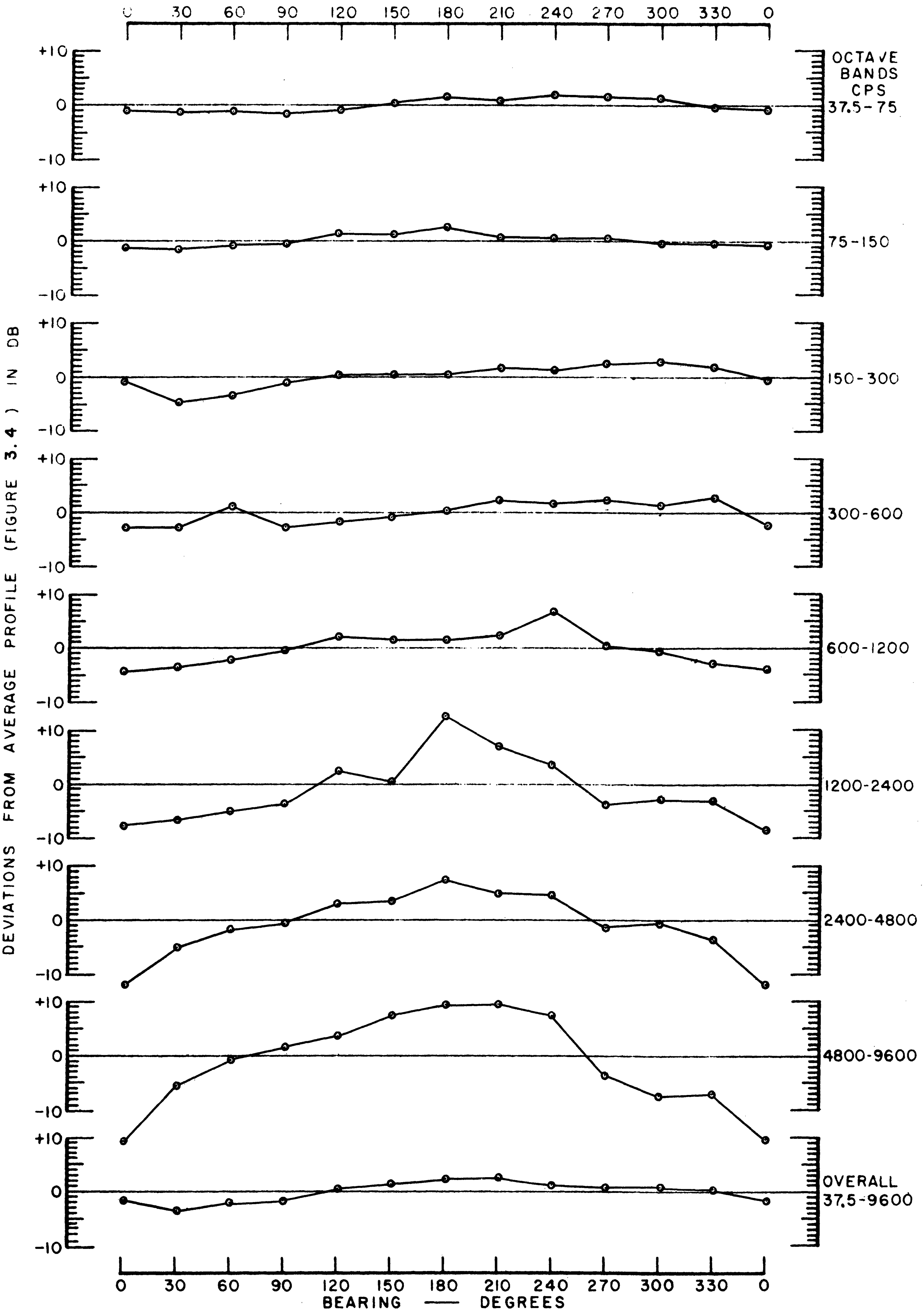



Fig. 3.5. Directional deviations from average profile; A-1,as-received.

at least, not at the standard microphone height.

If, acoustically, the A-1 were just an unmuffled internal-combustion engine, then it would be expected that the noise profile shown in Fig. 3.4 should continue to fall in level above 1000 cps. The relatively high levels of the high-frequency bands indicate additional sources of high-frequency noise. Listening tests suggest that the cooling-air blower may be a major contributor and the high-frequency directionality displayed in Fig. 3.5 strengthens this supposition. The high-frequency noise levels are highest around the end of the cart containing the cooling-air inlet and lowest around the other end of the cart.

There are undoubtedly many additional noise sources of secondary importance. However, before devoting any effort toward uncovering them, it is essential to muffle the exhaust noise and to identify positively and reduce the prominent high-frequency noise, which is probably due to the cooling-air blower.

Muffler Tests.—To proceed with experimental noise-reduction studies, it was expedient to try several commercial automotive-type mufflers. These were mounted one at a time horizontally on top of the A-1 housing above the engine compartment. The exhaust opening pointed in the 180-degree direction and the mufflers were connected to the original exhaust stack by means of flexible metal tubing. For comparison purposes, a section of flexible tubing of approximately equal length replaced the mufflers.

Ducting the exhaust gases to these several mufflers completely eliminated the Venturi action in the augmentor duct surrounding the original exhaust stack, and consequently reduced the air circulation throughout the engine compartment. However, less than a 10°F rise in oil temperature resulted, an amount certainly permissible for laboratory purposes. A static pressure tap was installed in the exhaust line to permit measuring the backpressure caused by the several mufflers (see Appendix B).

Figure 3.6A shows a schematic elevation of the muffler installation on top of the A-1 cart and Fig. 3.6B is a plan view of the acoustic measuring arrangement. The microphone was maintained at a 30-ft radius and 5 ft 5 in. above the ground. However, the measuring site is not free-field and represents only a convenient fixed arbitrary acoustic environment; consequently, only differences in noise level have significance. Further, only three microphone bearings, 170, 180, and 190 degrees were sampled and the results averaged together. This small sampling was justified since only large changes in acoustic level are sought in these special purpose tests.

Figure 3.7 presents the average results of these tests with automotive mufflers. A straight length of 2-in. ID flexible metal hose provides the "no muffler" reference noise levels. The Riker HD-200, a tractor muffler, and the Walker 582, a truck muffler, provide appreciable attenuation in the 300-600 and 600-1200 cps octaves, but provide no worthwhile reduction in the lower 75-150 and 150-300 cps octaves where attenuation is needed most for this application. The Walker 639, a Chevrolet passenger-automobile muffler, provides large attenuations ranging from 3 db to more than 17 db in the octaves between 37.5 and 1200 cps. The first two mufflers tested produced negligible increases in backpressure, while the Walker 639 contributed less than 0.4 in. of mercury backpressure. Because of the success of the Walker 639 muffler, it remained installed on the A-1 for the remainder of the laboratory tests.

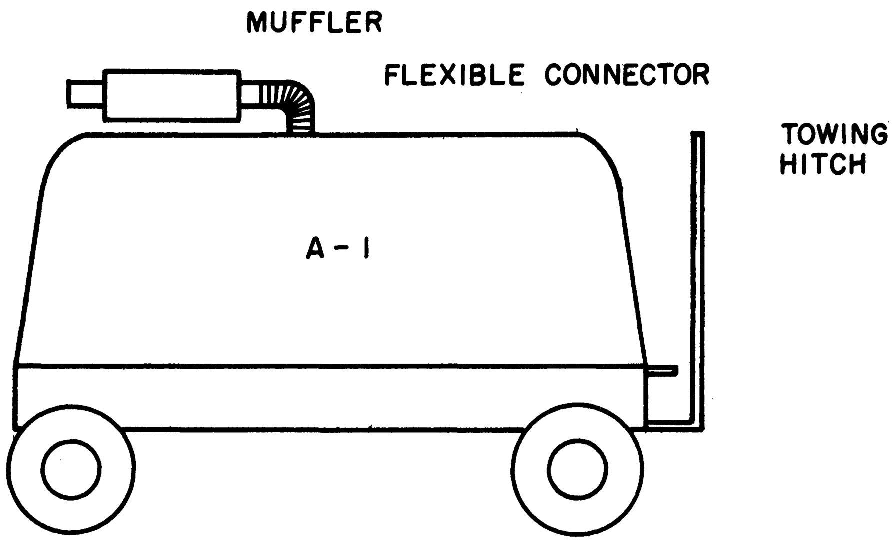



Fig. 3.6A. Schematic elevation of muffler installation on A-1.

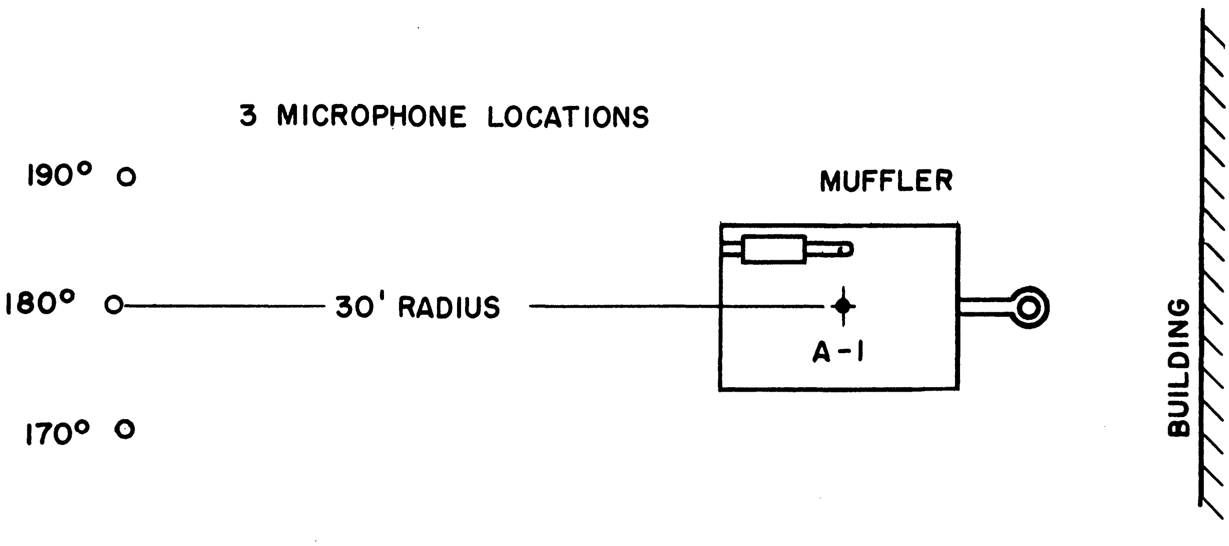
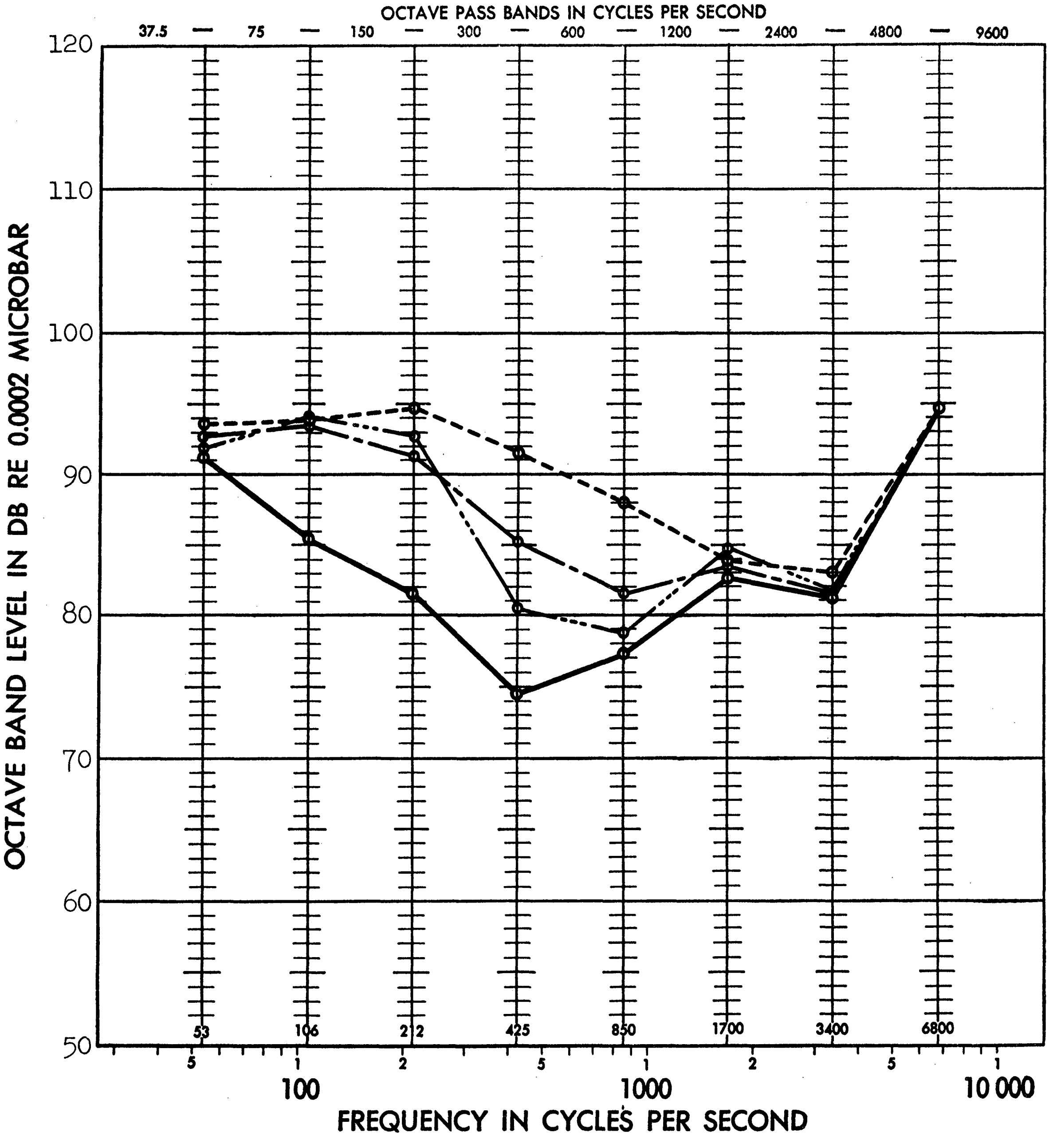



Fig. 3.6B. Plan view of A-1 muffler measurements.

Nonfree-field conditions
Microphone positions, see Fig. 3.6B
Average of 3 octave-band profiles




Type A-1 gasoline generator set equipped with muffler or extended exhaust line, see Fig. 3.6A
2410-2500 rpm, 195-205 amp, 28.6-28.9 volts dc
--------- Extended exhaust line without muffler
——---—— Riker HD-200 muffler (tractor muffler) no measurable back pressure
— --- — Walker 582 muffler (truck muffler) no measurable back pressure
———— Walker 639 muffler (passenger automobile muffler) back pressure less than 0.4 inch of mercury
Tested 29 August 1955

Fig. 3.7. Effect of mufflers on type A-1 generator noise.

Notice in Fig. 3.7 that none of the mufflers tested had any significant influence at high frequencies. This is entirely consistent with experience since ordinarily there is little high-frequency energy in the noise from internal-combustion-engine exhausts and, as mentioned above, the A-1's high-frequency noise is suspected to originate from other sources.

Cooling-Air-Blower Noise.—A qualitative discrete-frequency analysis of the high-frequency noise recorded locally was carried out by means of a Kay Vibralizer (see Appendix B). This analysis revealed discrete noise components at 2160 and 4320 cps, each of which corresponds to an octave band of high measured sound level. (Any higher harmonics, if present, fall beyond the frequency range of the Vibralizer.) To test the supposition that these tones radiate directly from the cooling-air-blower intake, an experiment with an externally mounted absorptive baffle was devised.

Figure 3.8 shows a schematic detail of this baffle. It consists of a 12-inch-diameter 16-gage sheet-metal disk centered over the cooling-air intake and spaced 3.75 inches away from the A-1 housing on four long screws. This baffle was lined with a 12-inch-diameter disk of 2-inch-thick fiberglass (Owens-Corning Fiberglas PF-335) constrained with a 12-inch-diameter disk of 1/4-inch-mesh hardware cloth. The final air space with the absorptive liner in place was about 1.75 inches wide. This baffle was designed to cover the air-intake opening, intercepting the radiating beam of high-frequency sounds without blocking the air flow. The absorbing material was selected on the basis of its 2000- and 4000-cps sound-absorption coefficient and the sheet-metal thickness on the basis of its 2000-cps sound-transmission loss. Local acoustic measurements were made with and without the baffle at each of six measuring positions according to the arrangement illustrated in Fig. 3.9. The noise reductions accomplished in the 1200-2400 and 2400-4800 cps octaves can be seen in Fig. 3.10. Directly in front of this baffle (position 4) reductions of 8 and 5.5 db, respectively, are found. However, at other measuring positions reductions are marginal, while the noise level observed at position 6 increased.

The disappointing behavior of this baffle was traced to sound being multiply-reflected through the intake cone of the A-1 and directed outward so obliquely as to miss the absorptive baffle. This interpretation was substantiated by additional tests and a new set of measurements were taken as illustrated in Fig. 3.11. A liner of one-inch-thick fiberglass was constructed to fit inside the intake cone. The results of the new tests are summarized by the bar graphs of Fig. 3.12. Here it can be seen that either the absorptive liner for the intake cone or the external absorptive baffle alone yields moderate attenuation, while in combination, they produce attenuations of 14.5 and 13.5 db, respectively, in the 1200-2400 cps and 2400-4800 cps octaves.

Internally Mounted Intake Silencer.—At the experimental stage reached above where, on the basis of local measurements, both the prominent low-frequency and high-frequency noises have been greatly reduced, one might logically carry out another free-field survey to verify the progress quantitatively, and to ascertain the character of the remaining noise-reduction problem. However, it was felt that although the externally mounted intake silencer served laboratory purposes very well, it possessed several objectionable features from the Air Force's viewpoint regarding possible equipment modification. Hence it was decided to attempt to devise an equally effective absorptive-type intake silencer capable of internal mounting within the A-1 cart, and which would be more representative of a practical installation.

The A-1 Generator Set provided a rather ideal piece of equipment on which to

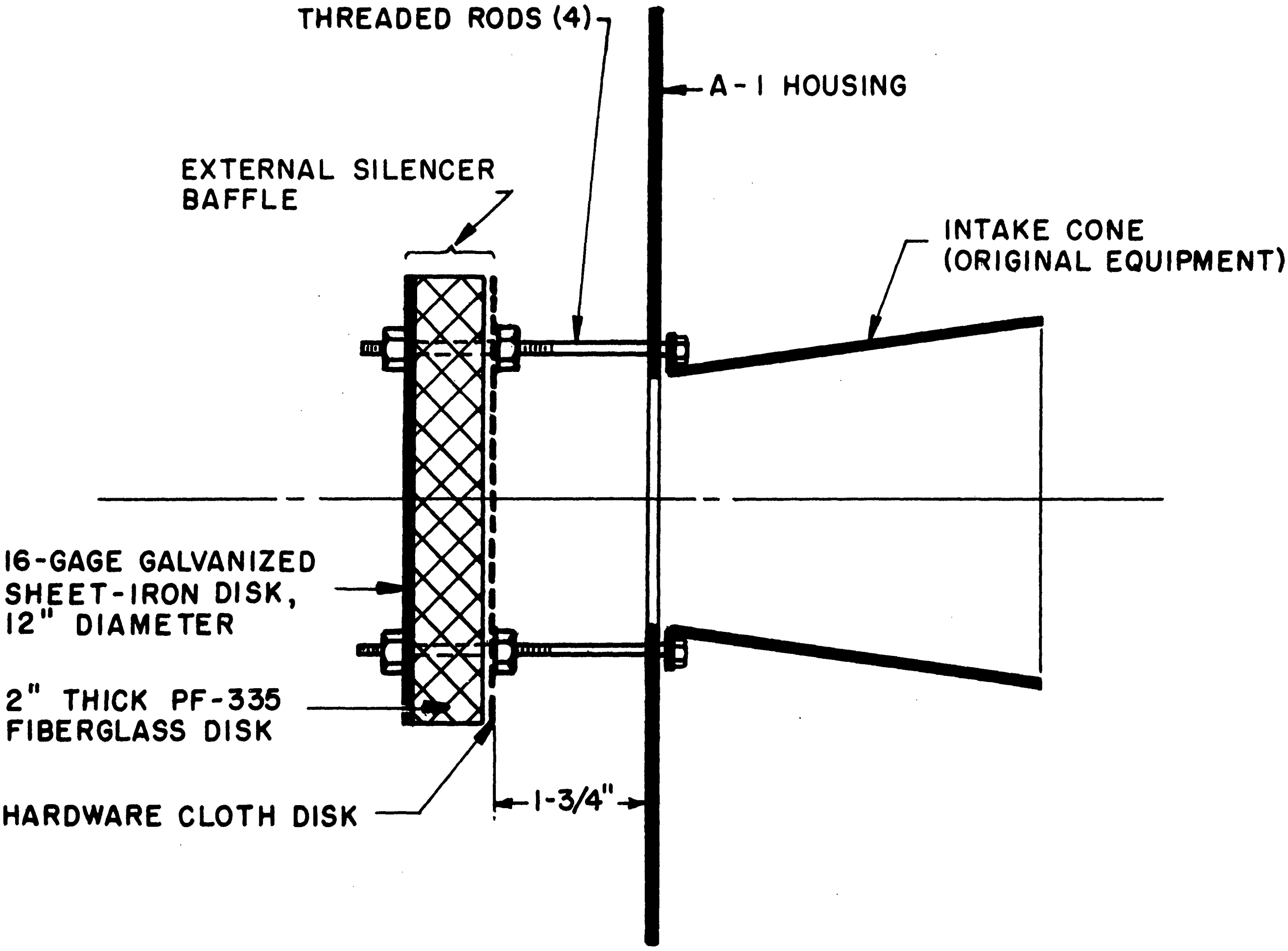
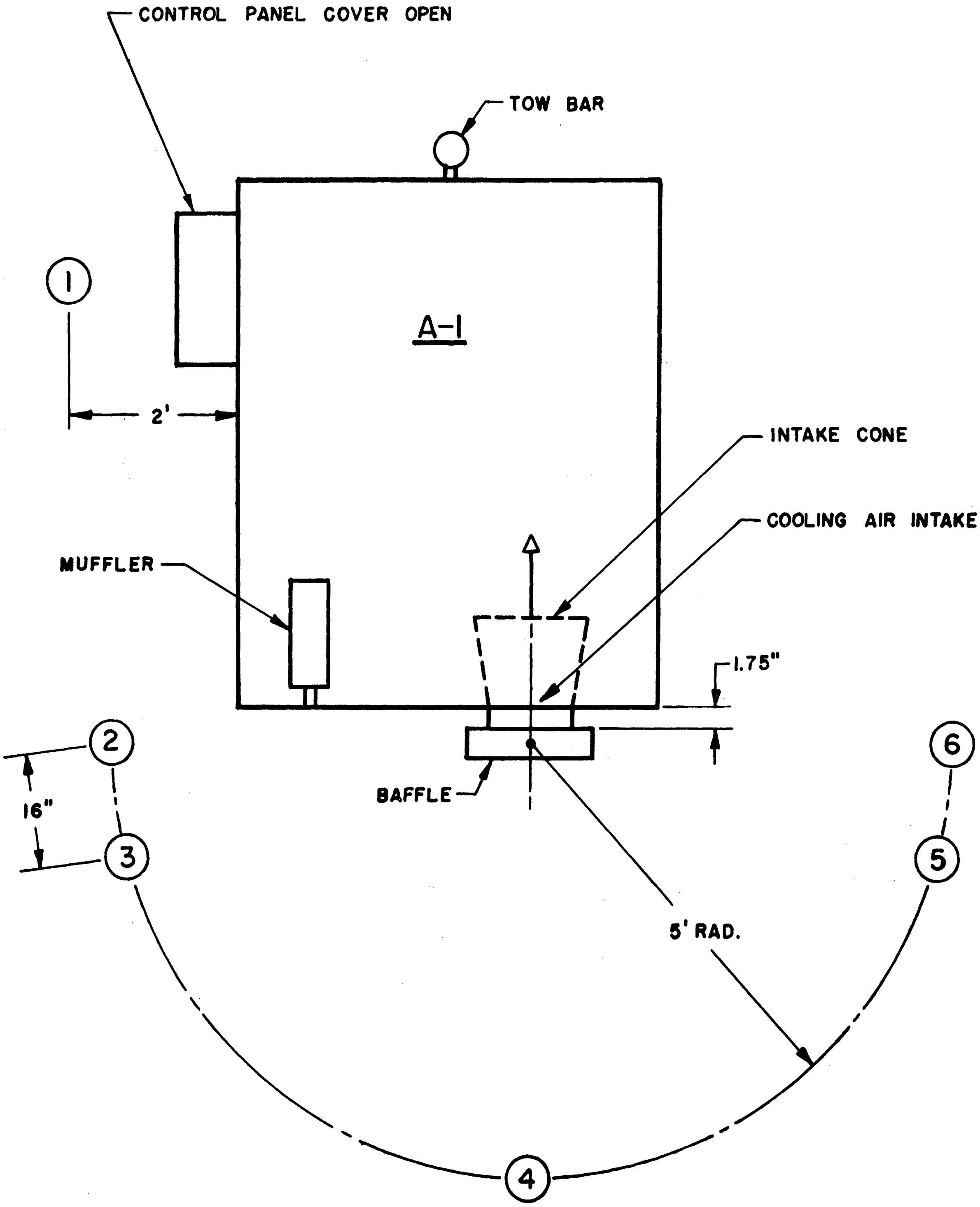



Fig. 3.8. Schematic section of external silencer baffle.




READINGS WERE TAKEN AT POSITIONS 1–6.

TOW BAR WAS LOCATED 7' FROM DOOR.

POSITION 1 WAS LOCATED 5'5" FROM GROUND.

POSITIONS 2–6 AND CENTER OF BAFFLE WERE LOCATED 3' 2.75" FROM GROUND.

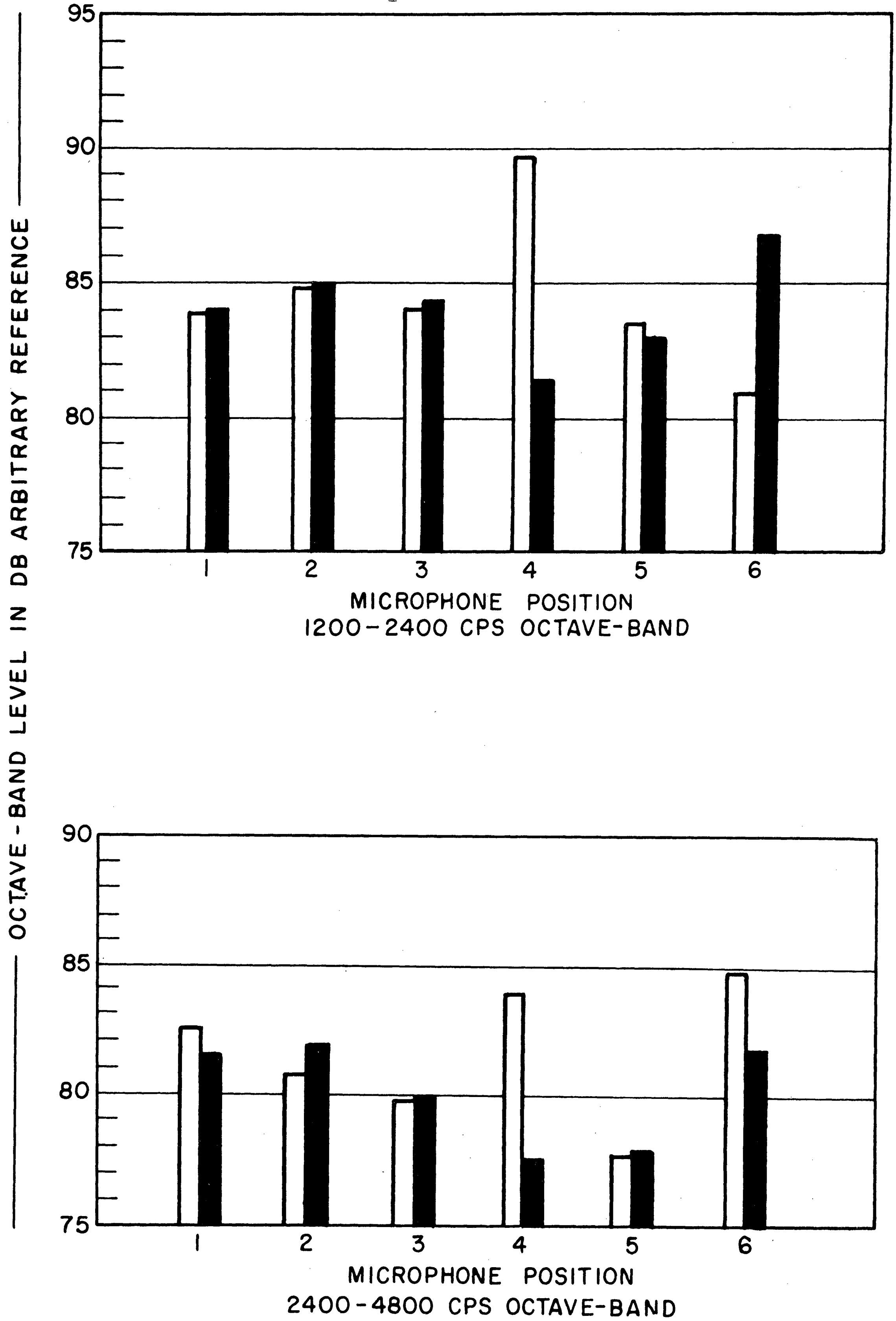
AIR INTAKE OPENING IS 7" DIA.

BAFFLE IS 12" DIA.

BAFFLE: 16 GA. GALVANIZED SHEET METAL DISC.
2 IN. THICK PF-335 FIBERGLAS DISC.
SCREEN.

Fig. 3.9. Measurement plan for external silencer baffle.

Nonfree-field conditions
Microphone position 4, see Fig. 3.9
Nonstandard measuring instrumentation. 1200-2400 and 2400-4800 cps octave bands, only internal differences have meaning




Type A-1 Gasoline Generator Set, equipped with Walker 639 muffler

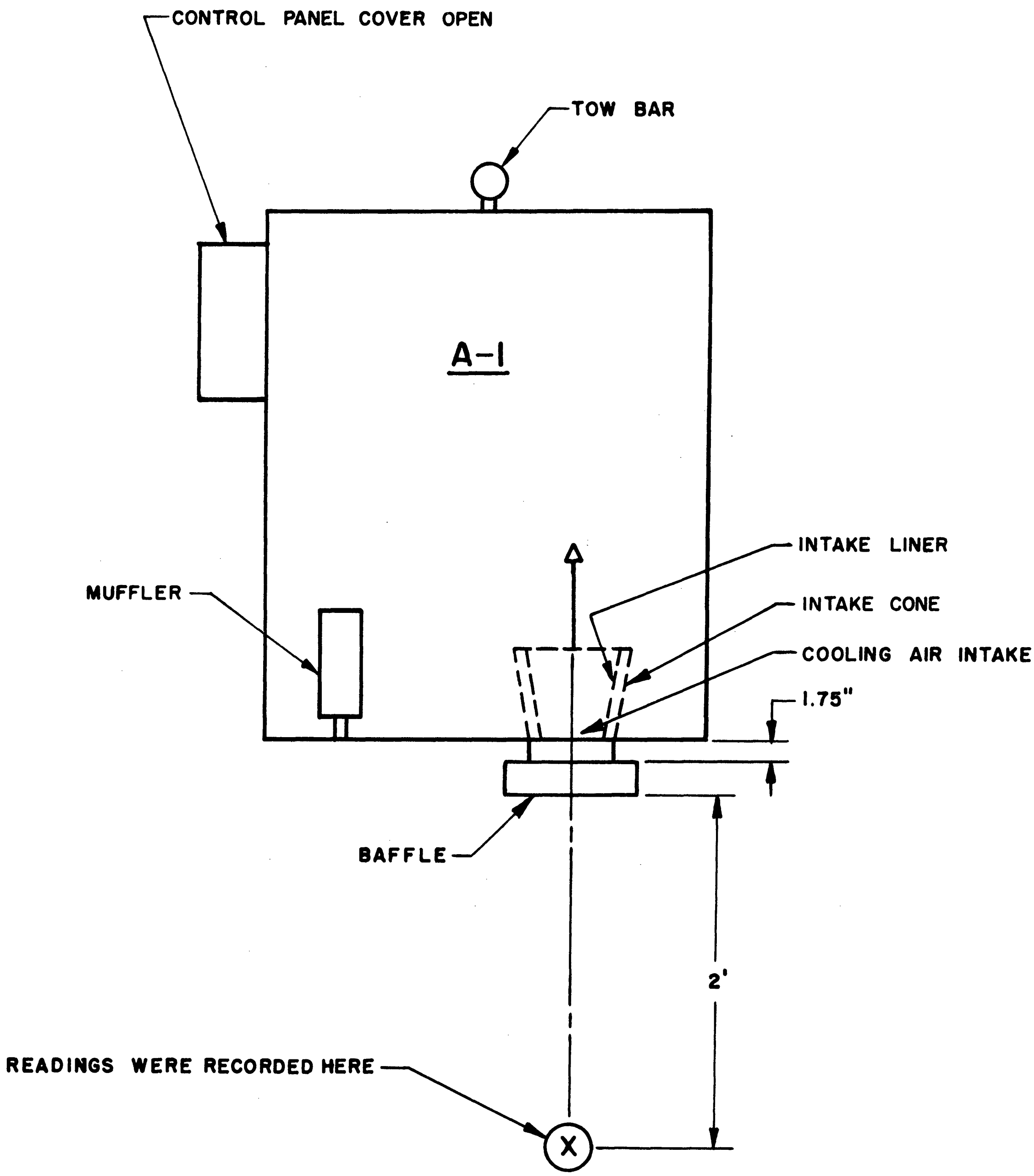
Baffle removed

Baffle installed

2440-2480 rpm, 197-200 amps, 28.2-285 volts dc

Tested 9 September 1955

Fig. 3.10. Effectiveness of external baffle at six microphone positions.




TOW BAR WAS LOCATED 7' FROM DOOR.

RECORDING INSTRUMENTS AND CENTER OF BAFFLE WERE LOCATED 3'2.75" ABOVE GROUND.

AIR INTAKE OPENING IS 7" DIA.

BAFFLE: 12" DIA.

16 GA. GALVANIZED SHEET METAL DISK.

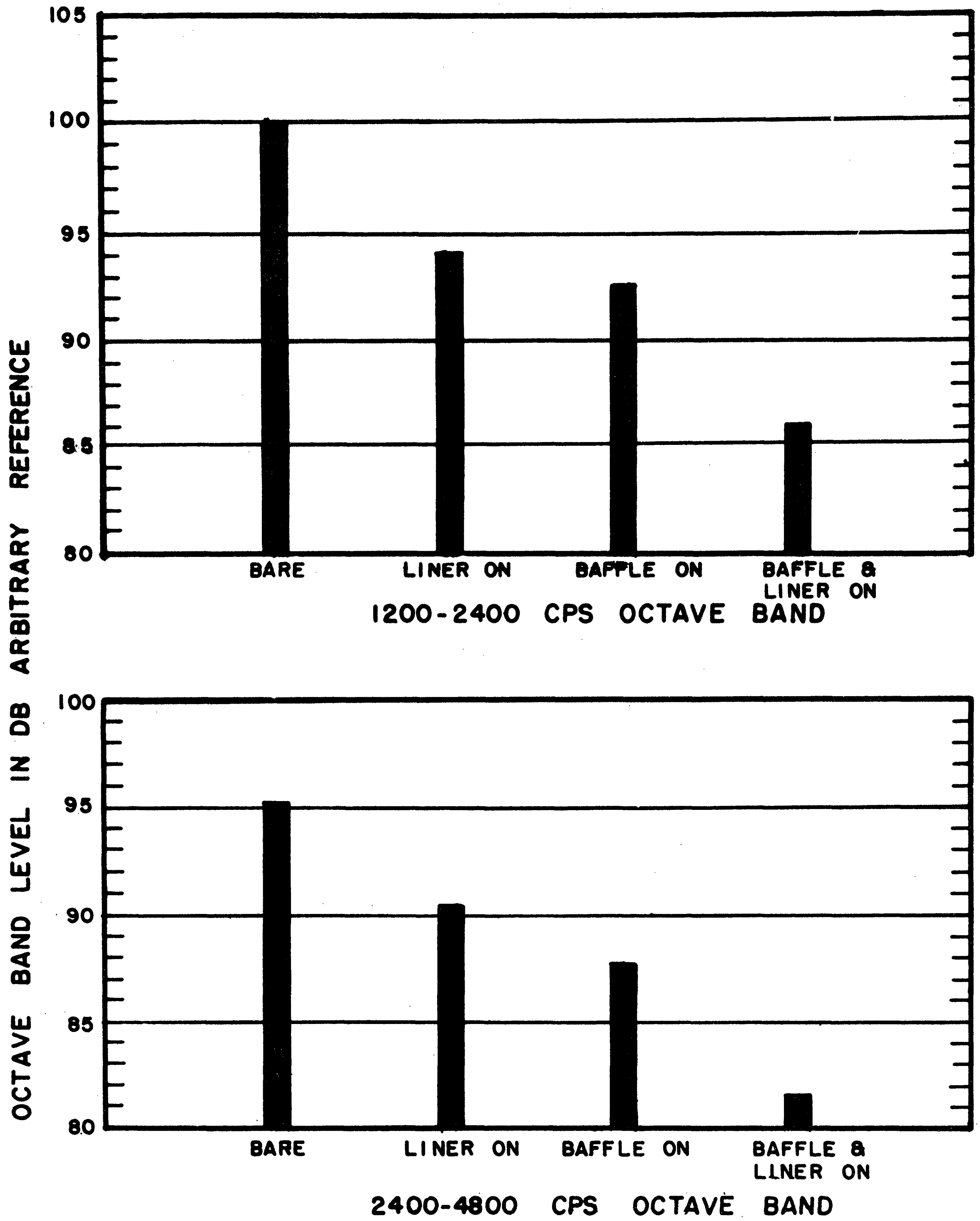
2" THICK PF—335 FIBERGLAS SHEET.

SCREEN.

LINER: 1" THICK FIBERGLAS

Fig. 3.11. Second measurement plan for external baffle.

Nonfree-field conditions
Microphone locations—see Fig. 3.11
Nonstandard measuring instrumentation. 1200-2400 and 2400-4800 cps octave bands, only internal differences have meaning




Type A-1 gasoline generator set, equipped with Walker 639 muffler
Approximately same operating conditions as for Fig. 3.10
Tested 14 September 1955

Fig. 3.12. Effectiveness of external baffle and cone liner.

attempt this development. The internally mounted intake cone could be removed from the cart, leaving a working space of about 8 inches between the blower and the end of the cart. This is not a particularly generous amount of space, but on the other hand it is not so cramped as to preclude a successful development.

Figure 3.13 shows a cross section of intake silencer No. 2. It consists of a cylindrical sheet-metal enclosure with its inlet-to-outlet path blocked by a flat absorptive baffle. The entire unit mounted directly onto the blower housing, leaving a 1/2-inch minimum clearance from the end of the cart. The same measuring arrangement shown in Fig. 3.9 was employed with the microphone located at position 4. Silencer No. 2's effectiveness, as illustrated in Fig. 3.14, was found to depend upon the use of an absorptive peripheral lining inside the cylindrical housing. However, this lining restricted the air flow, reducing the static pressure behind the blower from 7/16 to 1/4 inch of mercury which resulted in significantly higher engine temperatures.

Next the absorptive intake silencer was redesigned to make better use of the available space while retaining the absorptive peripheral lining. Figure 3.15 shows a cross section of the resulting intake silencer No. 3. Close-in measurements (position 4, Fig. 3.9) yielded attenuations of 13.6, 9.0, and 17.2 db, respectively, in the 1200-2400, 2400-4800, and 4800-9600 cps octaves. This silencer only affected the blower's static pressure slightly and was deemed satisfactory for the purposes of this program.

Final Free-Field Noise Survey.—A complete free-field noise survey was conducted on the A-1 Gasoline Generator Set with the Walker 639 muffler and intake silencer No. 3 installed. Operating parameters were the same as those of the "as-received" survey. (See Appendix C for detailed data.) Figures 3.16, 3.17, and 3.18 present the results. For comparison purposes, the "as-received" survey curves from Figs. 3.3 and 3.4 have been included again in Figs. 3.16 and 3.17. Figure 3.16, showing the polar distribution of the computed overall noise, shows that highly significant noise reductions have been achieved at all bearings. The average overall noise level at 30 feet has been reduced 8.9 db from 98.6 db in the "as-received" condition to 89.7 db. Some individual octaves at particular bearings have been reduced as much as 19 db, that is, to levels only about 10% as high as originally. Some directional variation is now evident in the overall sound-pressure levels and will be discussed below.

Figure 3.17, showing the average noise profiles, illustrates how the character of the spectrum has been changed by the addition of the exhaust muffler and the absorptive silencer on the cooling-air blower. The spectrum is now much flatter and on the basis of sound-pressure level, the 37.5-75 cps octave-band noise predominates. In Fig. 3.18 we see that the lower frequency octaves exhibit some directionality with a maximum in the neighborhood of 180 degrees. The higher frequencies show a reduced directionality and a general shift in the form of the directional pattern. The original maximum in the 180-degree direction has been replaced by a dip, and two smaller maxima appear at each side, roughly at 90 and 240 degrees. Figure 3.19, a polar plot of the 4800-9600 cps octave noise, has been included here to emphasize how drastic some of the changes have been.

Interpretation of Results Achieved.—The results reported above require some interpretation from the point of view of the treatments used. The original symmetry of the overall sound-pressure levels arose from the domination of the overall noise by comparatively low-frequency exhaust tones radiated from a vertical exhaust stack. In

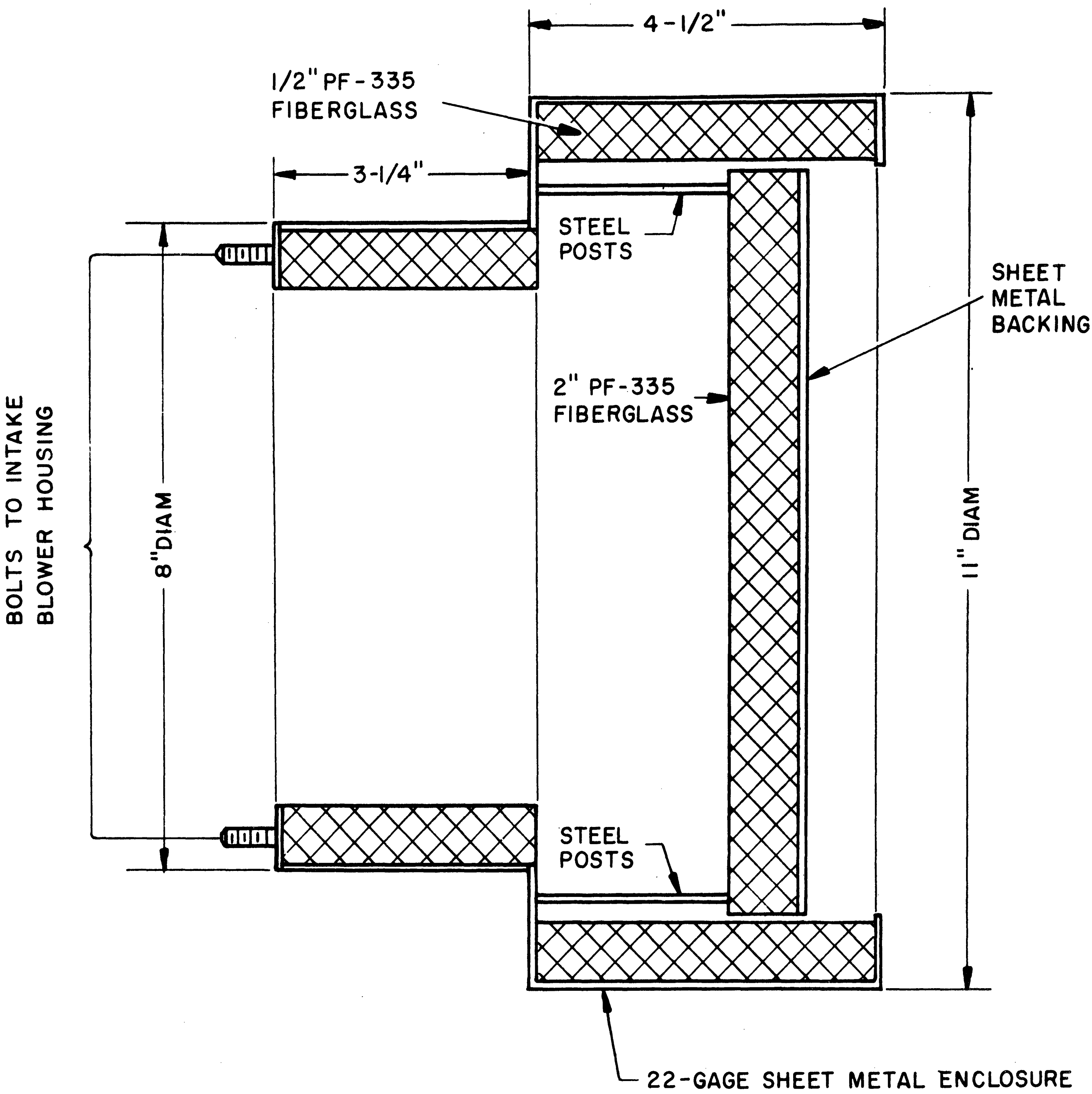
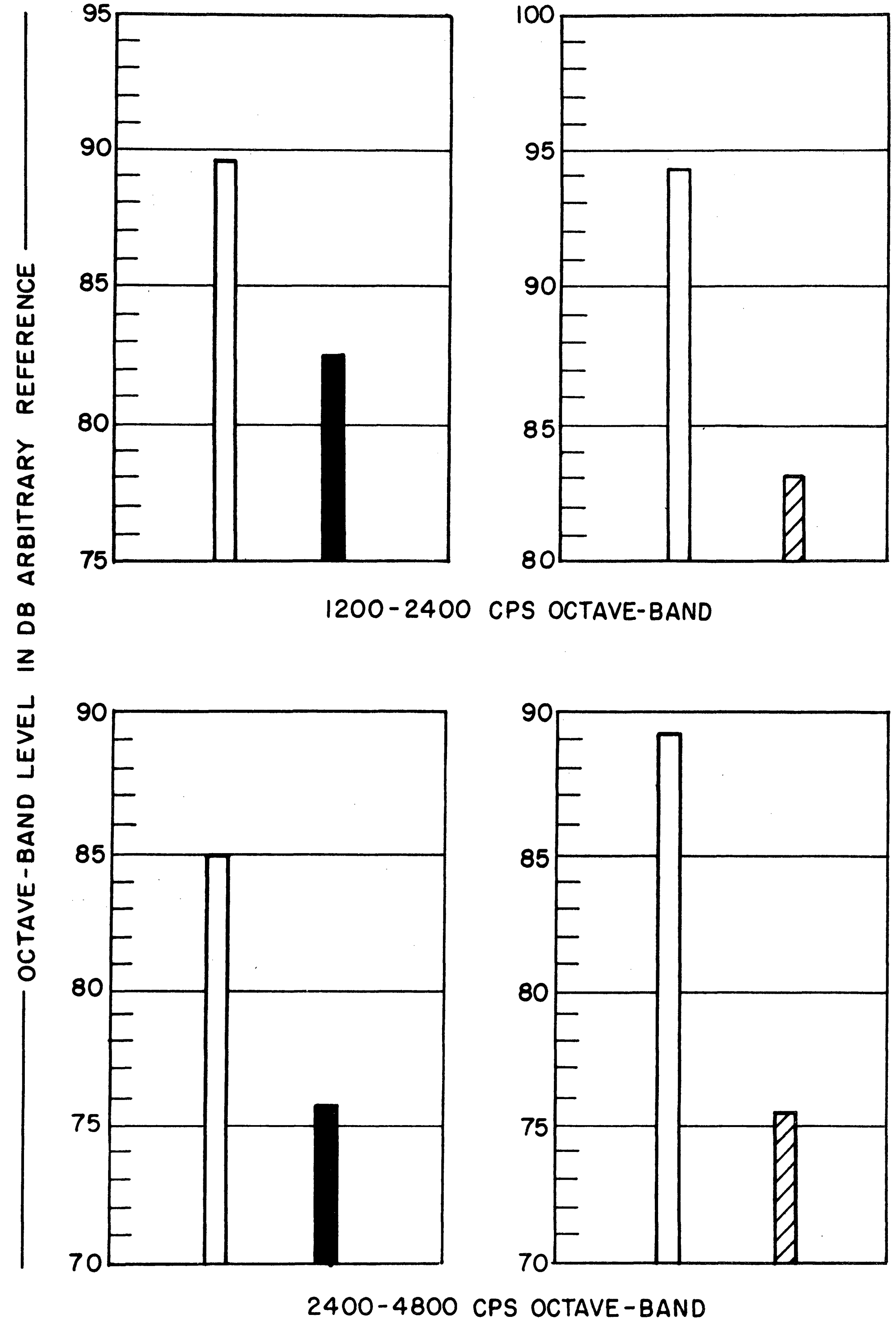



Fig. 3.13. Cross section of intake silencer No. 2.

Nonfree-field conditions
Microphone locations, see Fig. 3.9
Nonstandard measuring instrumentation. 1200-2400 and 2400-4800 cps octave bands, only internal differences have meaning




Type A-1 Gasoline Generator Set equipped with Walker 639 muffler 2410-2520 rpm, 197-207 amp, 27.9-28.4 volts dc

Silencer removed

Silencer installed without peripheral liner

Silencer installed with peripheral liner

Tested 26-27 September 1955

Fig. 3.14. Effectiveness of intake silencer No. 2.

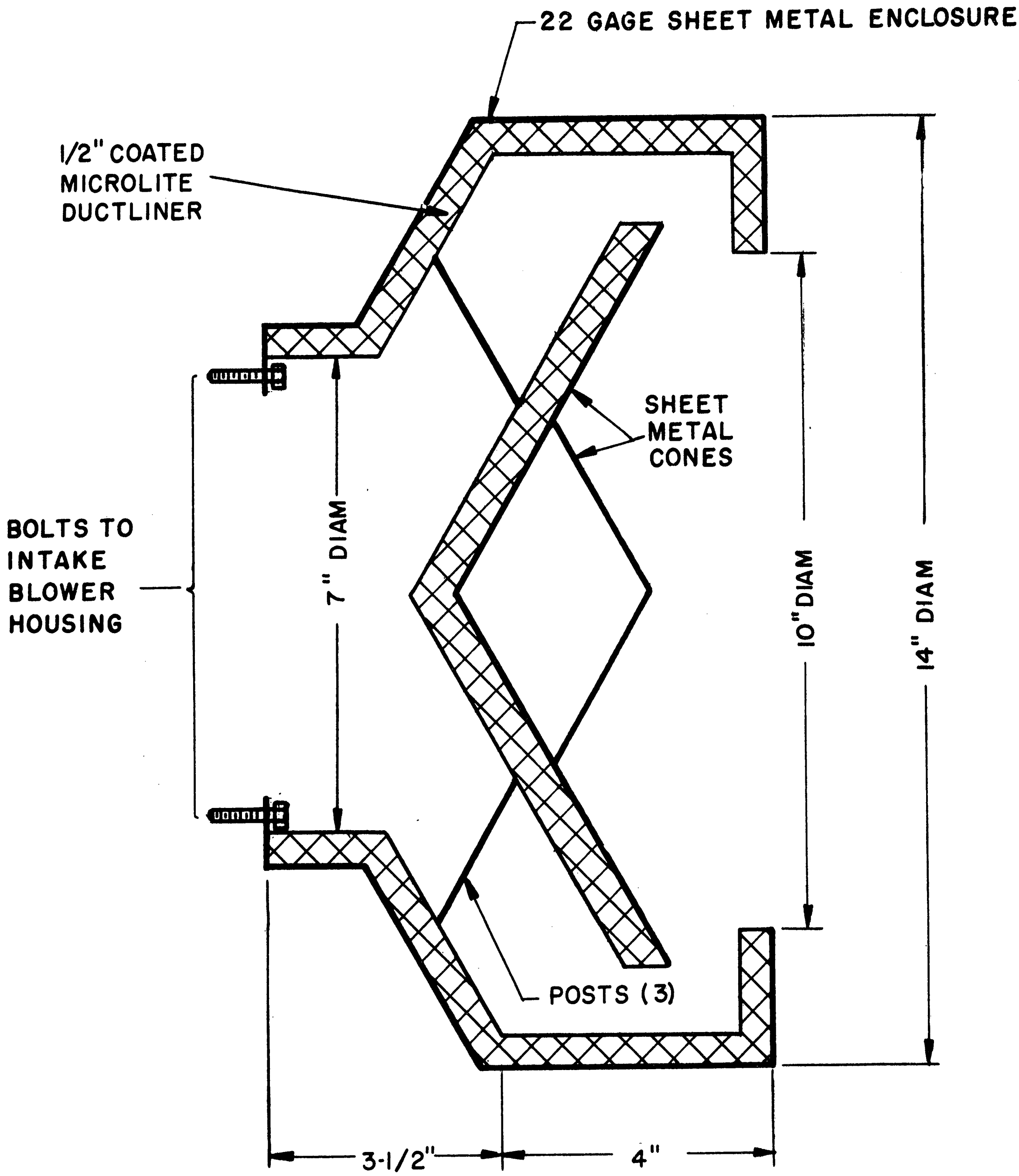
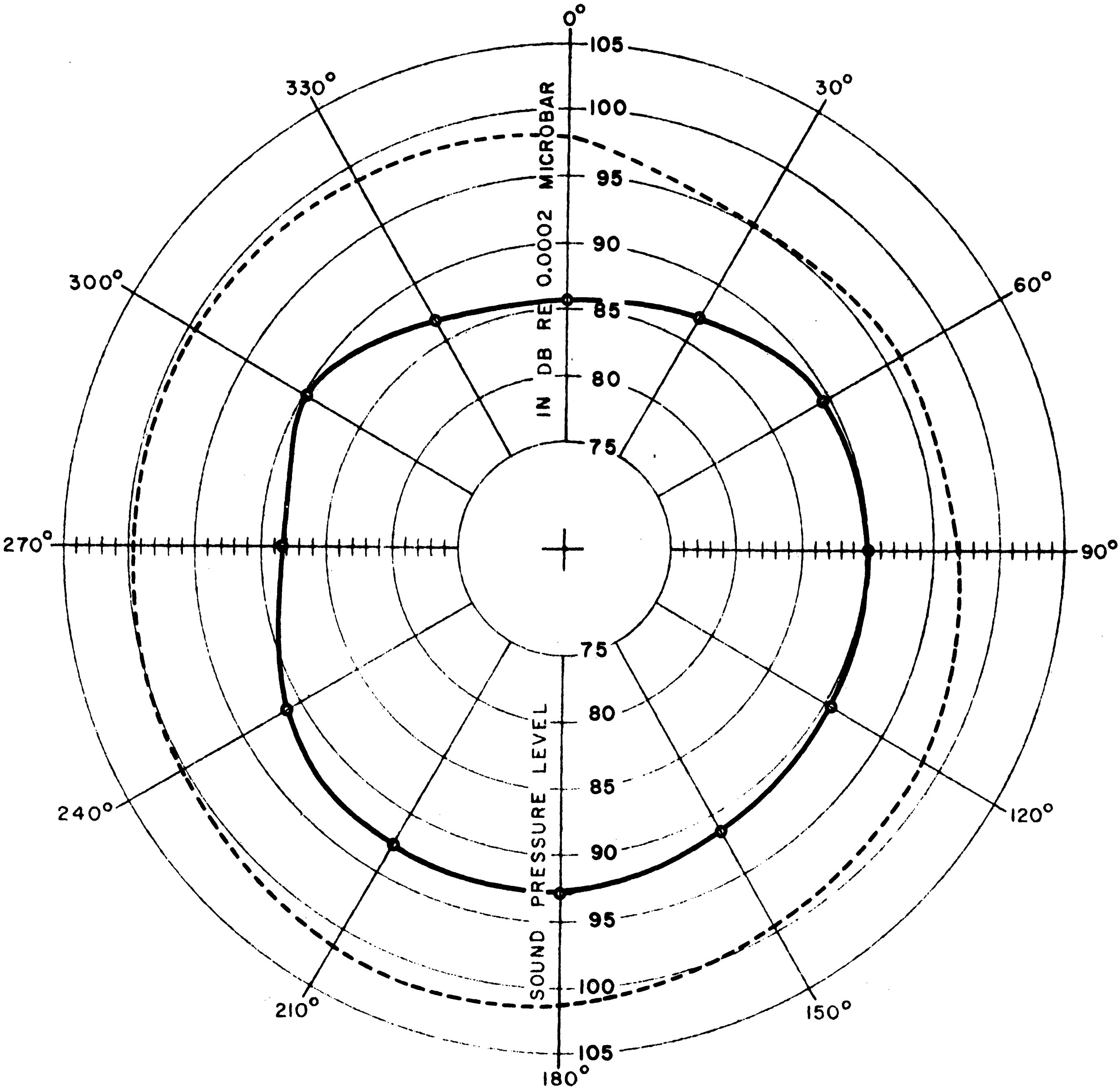



Fig. 3.15. Cross section of intake silencer No. 3.

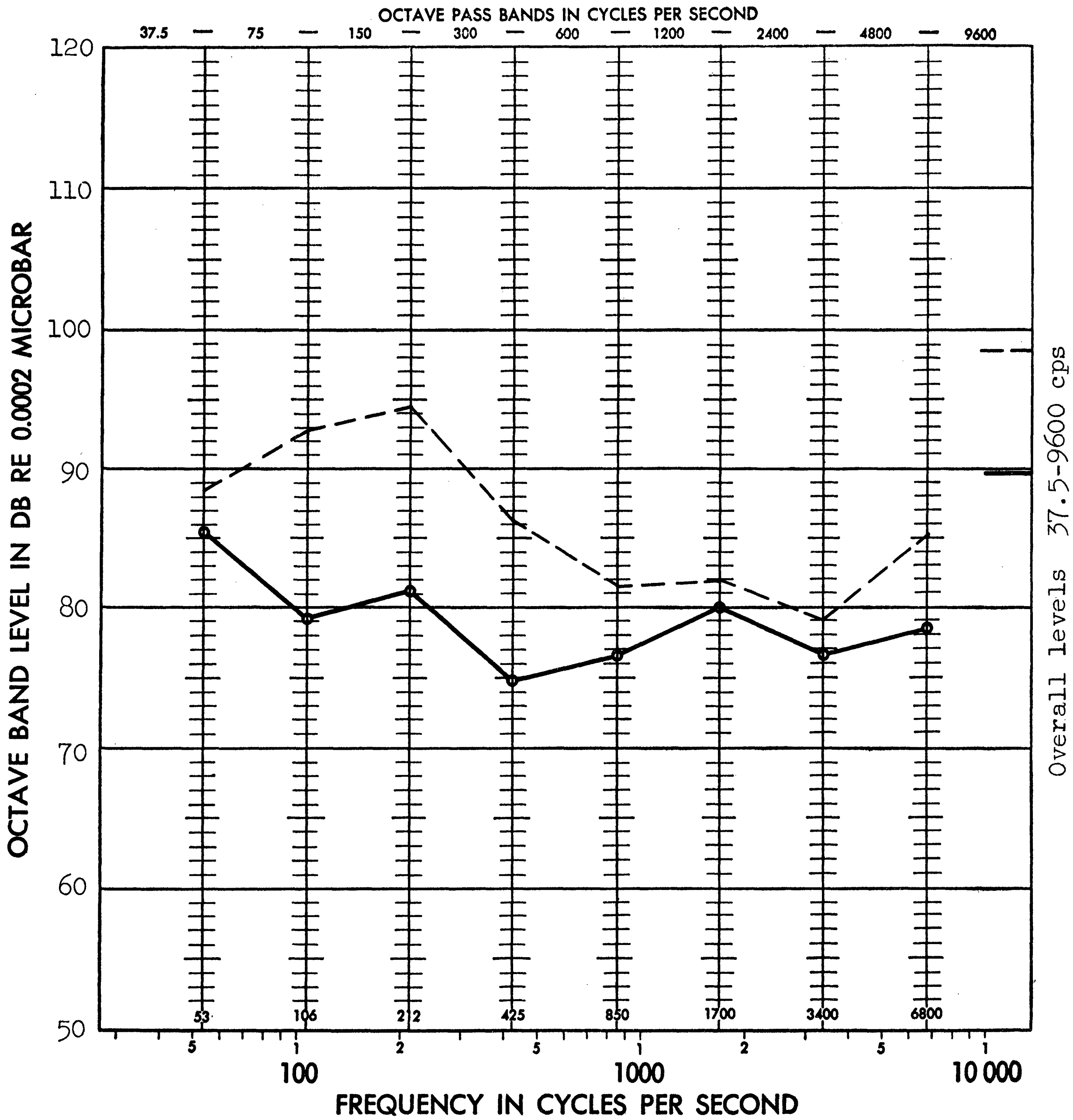
Free-field conditions
Microphone distance 30', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type A-1 gasoline generator set, equipped with Walker 639 muffler and intake silencer No. 3
2490-2570 rpm, 203-215 amp, 27.9-29.0 volts dc
Tested 1 November 1955
----- As-received condition, see Fig. 3.3

Fig. 3.16. Polar distribution of overall noise; A-1, treated.

Free-field conditions
Microphone distance 30', height 5'5"
Average of 12 octave-band sound pressure levels




Type A-1 gasoline generator set, with Walker 639 muffler and intake silencer No. 3 attached
2490-2570 rpm, 203-215 amp, 27.9-29.0 volts dc
Tested 1 November 1955

Fig. 3.17. Average octave-band noise profile; A-1, treated.

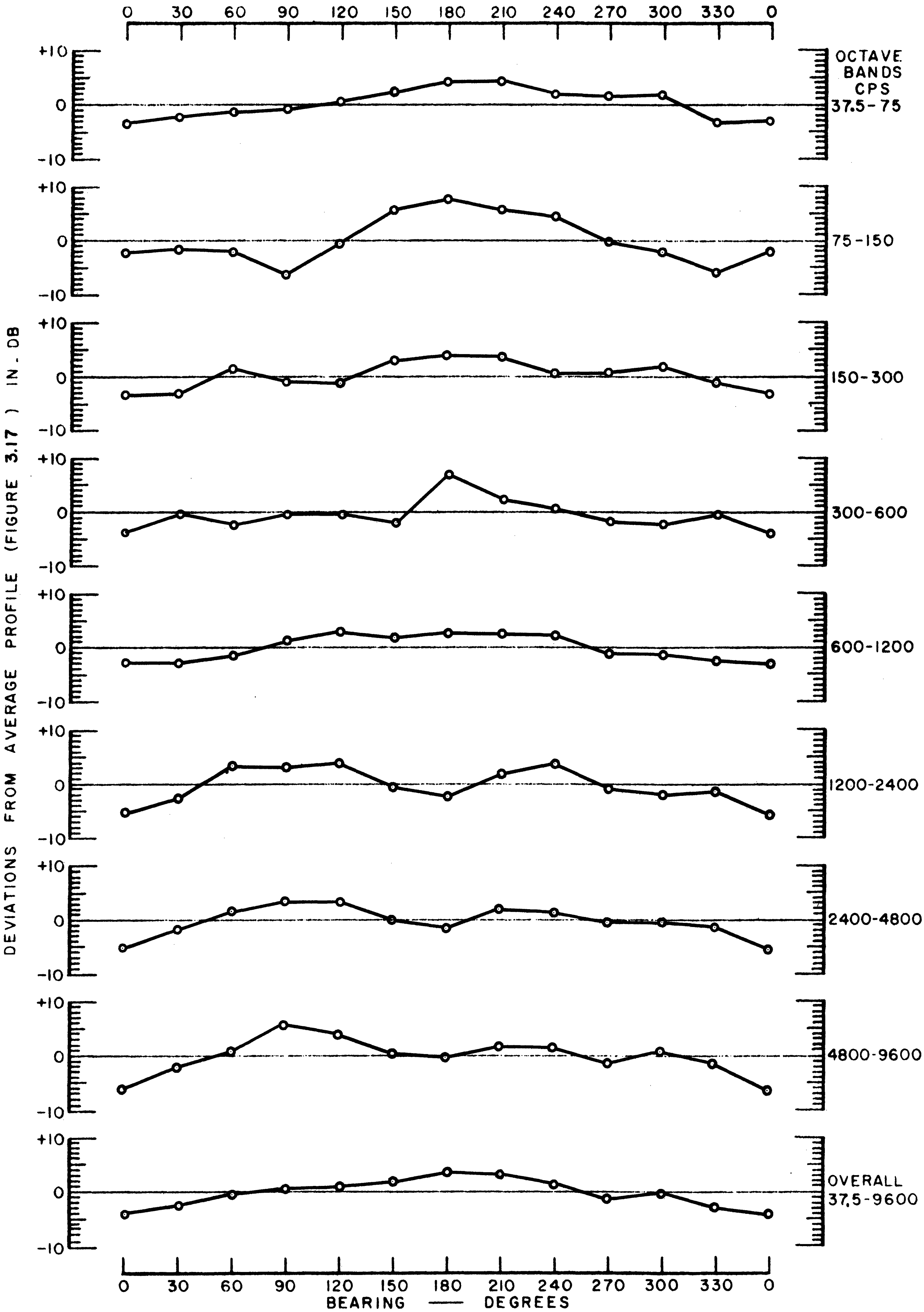
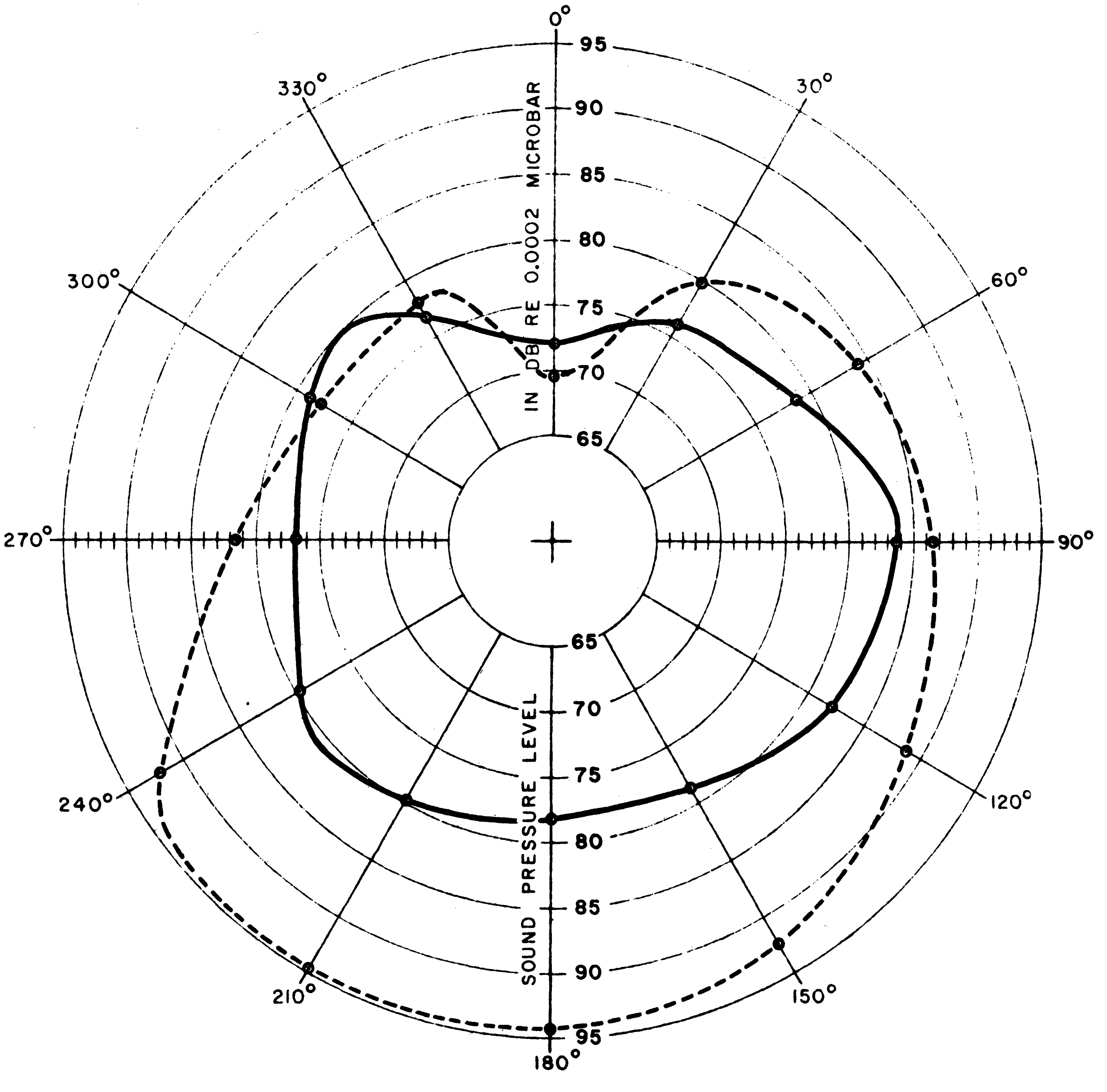



Fig. 3.18. Directional deviations from average profile; A-1, treated.

Free-field conditions
Microphone distance 30', height 5'5"
4800-9600 cps octave-band only




Type A-1 gasoline generator set, same operating parameters as for Figs. 3.4 and 3.17
----- As-received condition
——— Treated condition

Fig. 3.19. Polar distribution of 4800-9600 cps octave-band noise; A-1, as-received and treated.

this final survey, the overall sound-pressure level is dominated by an even lower frequency octave, i.e., 37.5-75 cps. Much of this remaining noise probably also arises from low-frequency exhaust tones which are insufficiently attenuated by the Walker muffler. The directionality is probably largely due to the experimental arrangement which now directs the exhaust in the 180-degree direction. If the muffler were mounted inside the cart and exhausted vertically, a nearly symmetrical pattern would be expected.

The domination of the overall levels by the low-frequency exhaust noise also obscures the large decrease in the blower noise in the neighborhood of the 180-degree bearing. For instance, Fig. 3.19 reports a decrease of over 16 db at the 180-degree bearing, while the low-frequency dominated overall noise plotted in Fig. 3.16 only shows slightly more than 8 db. At high frequencies, the average noise profile presented in Fig. 3.17 is also quite misleading. It suggests that only moderate reductions have been achieved, whereas Fig. 3.19 presents the situation more clearly. The original high-frequency noise was largely confined to the neighborhood of the 180-degree bearing and this noise has been very effectively reduced. The averages of the levels at all bearings include many moderate levels which have not changed appreciably and therefore cause the average high-frequency noise reduction reported in Fig. 3.17 to appear disproportionately small.

The remaining high-frequency directionality appears as two smaller lobes extending from the sides of the machine. It would require further testing to be certain, but since the interior cooling-air ducting is relatively thin and the sides of the engine compartment contain open louvers, it is plausible that the blower noise is radiating out through these louvers. It is also likely that gear whine, generator brush noise, etc., may now contribute significantly. Moreover, with the originally dominant sources treated, it is also more likely that mechanical vibrations communicated directly into the structure of the cart are causing it to radiate airborne noise, particularly in the low- and medium-frequency ranges.

The above studies have re-emphasized the necessity for installing effective exhaust mufflers as a first step in reducing the noise radiated by internal-combustion engines. The radiated noise originating from the cooling-air blower constitutes a prime example of a high-frequency noise "leak" from an enclosed machine. The application of a silencer of practical configuration illustrates an effective cure. This absorptive-type silencer compared to a resonant absorber has the distinct advantage because it is broadly tuned, and hence maintains its effectiveness over a wide range of frequencies and machine operating speeds.

The studies on the A-1 Gasoline Generator Set reported here are limited in the sense that they encompass only one noise-reduction cycle, i.e., delineation of the original problem, treatment of the predominant sources discovered, and verification of the effectiveness of the treatments. This systematic cyclic procedure may be continued as necessary to yield the desired end results. However, since higher priority was assigned to studies on other items of ground-support equipment, it was felt that studies on the A-1 Gasoline Generator Set should be terminated at this juncture. These studies have served their multifold purpose of demonstrating a general approach to a noise-reduction problem, of yielding specific recommendations for achieving quite large noise reductions quickly and economically, and of illustrating in detail the diagnosis and practical treatment of a particular noise source, i.e., the cooling-air blower.

Recommendations.—An exhaust muffler and an inlet silencer for the cooling-air blower are the necessary first modifications to the A-1 Gasoline Generator Set. Neither item used in the above tests was necessarily optimum for this application. It is believed, however, that with the development of silencer No. 3 as shown in Fig. 3.15, a point of diminishing returns has nearly been reached for the space limitation imposed by the present cart structure and by the machinery layout. Of course, the application of production fabrication methods could yield a more impressive looking package and perhaps reduce flow resistance even further without sacrificing the silencer's acoustical efficiency.

In the case of the exhaust muffler, one designed specifically for this particular engine would undoubtedly yield even better results leading to reduced backpressure and greater acoustic attenuation. Ultimately, an effective muffler which could be installed within the A-1 cart and also provide ejection pumping of cooling air could certainly be devised by a competent muffler manufacturer. This type of interior installation was not attempted during the course of these studies since the cart would have had to be altered permanently.

Beyond those stated above, further specific recommendations are difficult to formulate in view of the limited scope of this noise-reduction study. Probably a general acoustic sealing up of the cart to realize its full enclosure potential would be the next step. This would involve closing all openings or equipping them with acoustical low-pass filtering, preventing rattles, providing acoustical absorption within the cart, damping the panels of the cart to prevent resonant transmission, and isolating the cart structure as well as possible from the machinery-produced vibration.

## TYPE C-26 GASOLINE ENGINE GENERATOR SET

The Type C-26 Gasoline Generator Set consists of a six-cylinder opposed, four-cycle aircooled aircraft engine (Continental Model PE 150-2 rated 180 hp at 2400 rpm) geared to four electrical generator units. Three of the generators are 30-volt 500-ampere d-c aircraft generators (No. G-32-3), while the fourth is a 115-volt 15-kva, 380-800 cycle alternator (Eclipse-Pioneer Type 28E03-3-A). Cooling air for the engine is supplied by a blower which is an integral part of the gear box assembly connecting the engine to the generators. Cooling air for the generators is supplied by a separate electric-motor-driven axivane fan operating at 8000 rpm. The entire C-26 unit is housed in an integrally framed dolly or cart made from formed and riveted sheet aluminum, and the machinery assembly is attached to the cart at four points by means of rubber mounts. The overall size is about 7-1/2 feet long by 4 feet wide by 4-1/2 feet high.

The end of the C-26 cart adjacent to the towing hitch contains the fuel tank, batteries, power cable storage, generator cooling blower, generator control circuitry, and the engine instruments. The generators and the gear box are located approximately in the center of the cart. The opposite end of the cart contains the engine proper and a small engine-control panel. Figure 3.20 shows a schematic elevation of the C-26's interior arrangement, while Fig. 3.21 is a schematic plan view which identifies the approximate location of various important features and defines the azimuthal directions at which all free-field sound-pressure measurements were made.

Engine cooling air is taken in through four louvered panels, two on the rear and two on the side of the unit. This air sweeps through the engine and generator com-

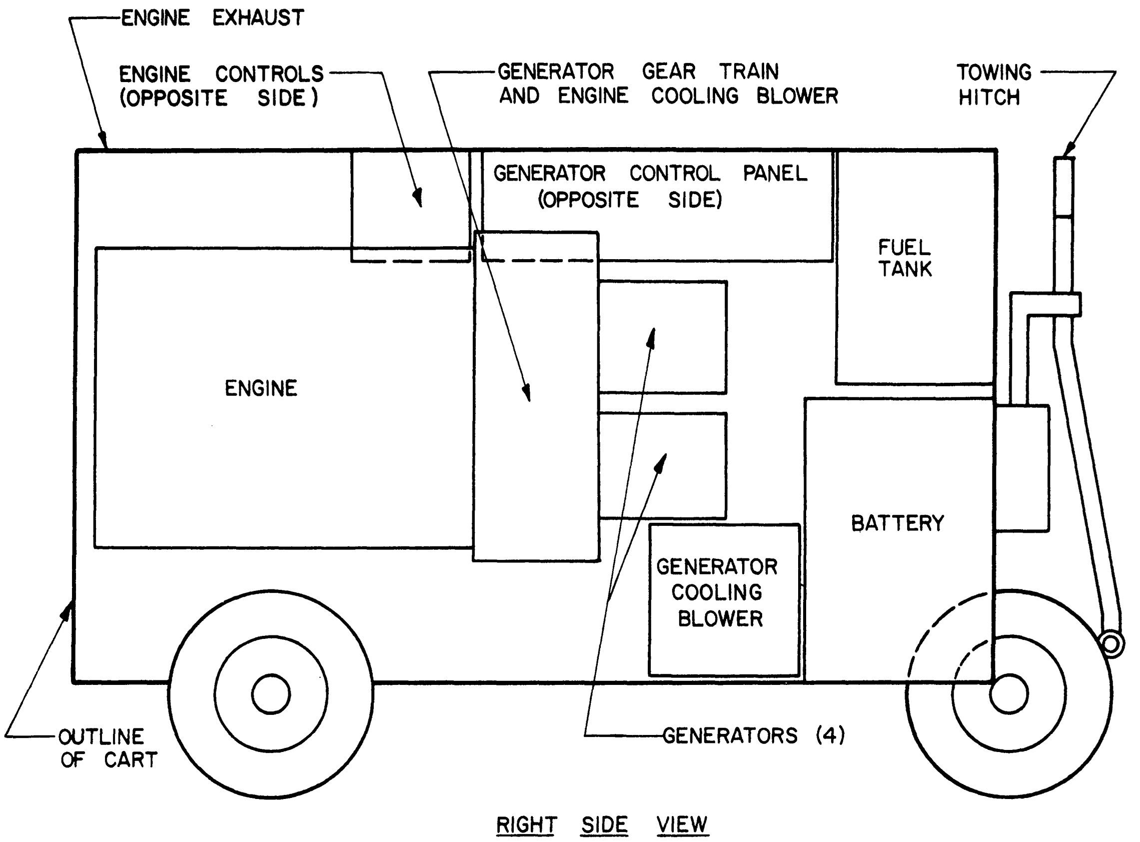



Fig. 3.20. Schematic elevation of C-26 interior arrangement.

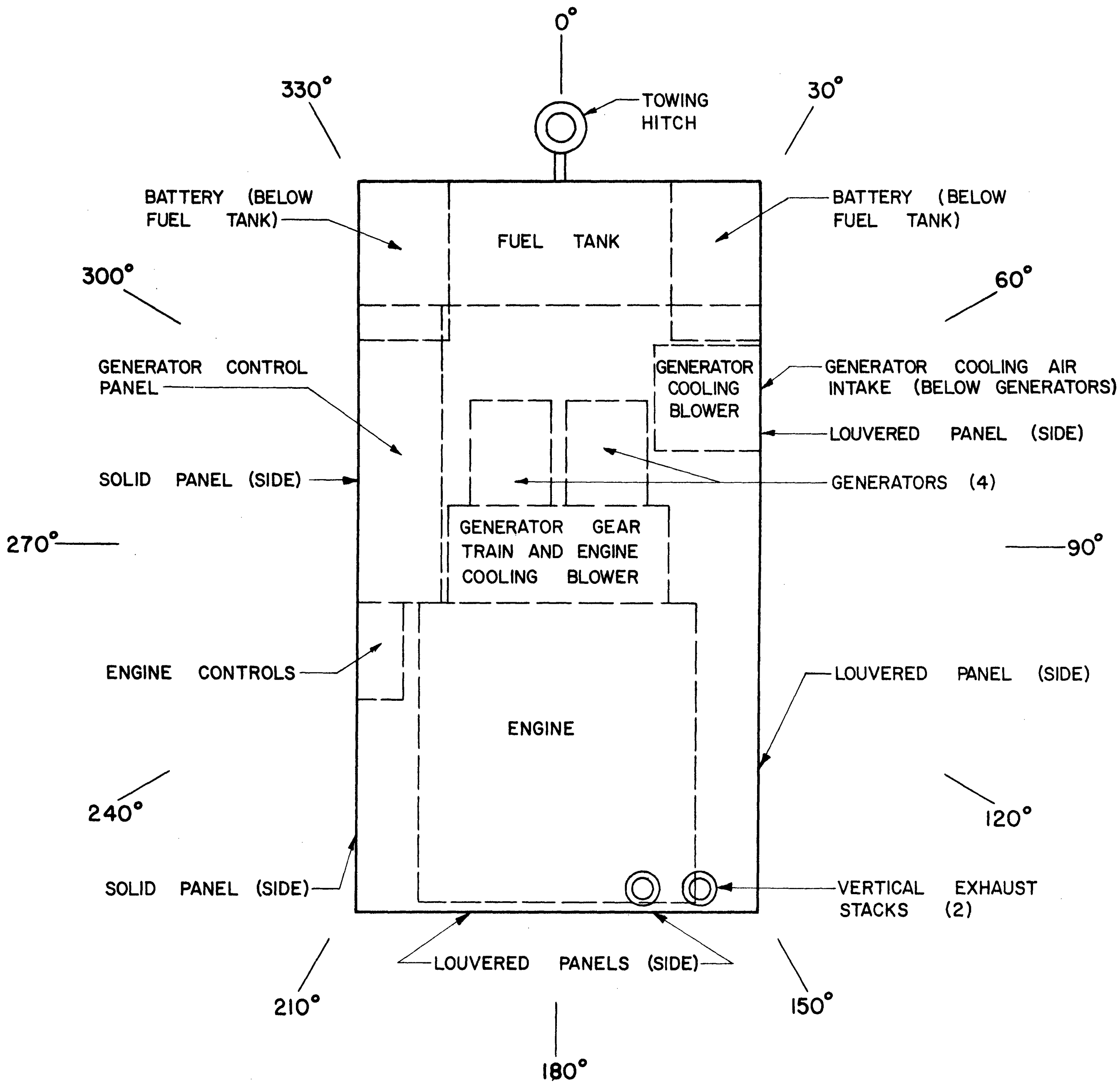



Fig. 3.21. Schematic plan view of C-26 generator set.

partments to the engine cooling blower mounted on the inboard end of the engine integral with the gear housing. This blower forces the air over the engine, which is tightly cowled to assure proper distribution of the cooling air to the cylinders. Finally the hot air is exhausted out through a louvered panel in the bottom of the cart directly beneath the engine. A small amount of the engine cooling air is also directed through ducting concentric with the exhaust lines as they traverse the engine compartment. This flow of air around the exhaust lines is aided by augmentor stacks around the final portions of the two exhaust lines. Cooling air for the generators is supplied by a separate blower system. This air is taken in through a grill in the side of the cart and ducted to the individual generators. After flowing through the generator housings, the hot air is exhausted into the generator compartment where it mixes with the engine cooling air. During normal operation, all panels may be closed except for two small panels covering the generator and engine controls. The generator control panel is rather tightly installed and communicates only with a separate control compartment in the cart; hence it could hardly constitute an acoustical leak. The engine control panel opens into the engine compartment, and although the resulting apertures are small in size, they do communicate directly with the machinery spaces, thus constituting potential acoustical leaks.

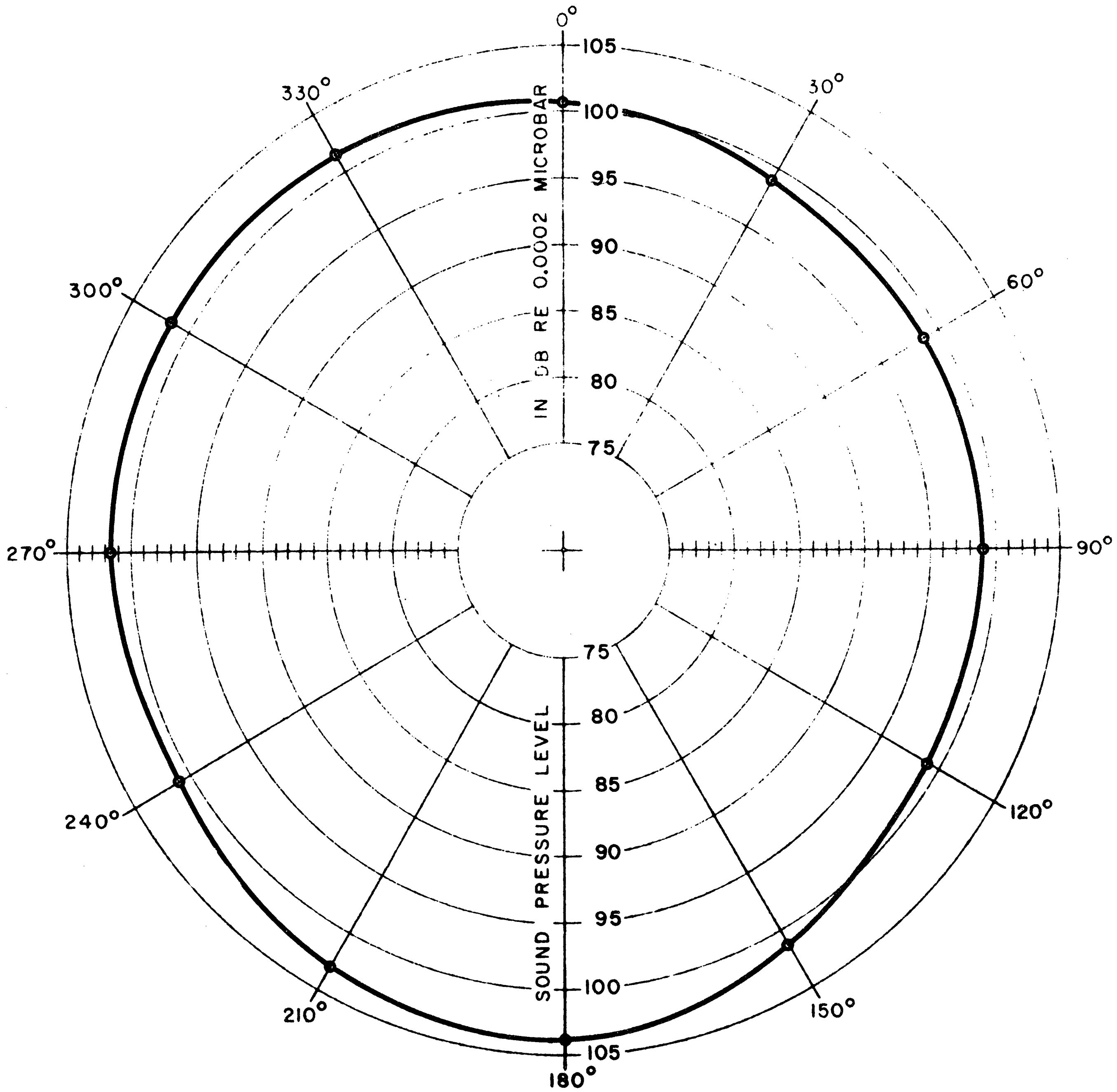
As-Received Free-Field Noise Survey.—As in the case of the A-1, the step preliminary to evaluating the problem of reducing the noise radiated by the C-26 Generator Set was to conduct a free-field octave-band noise survey with the machine in its "as-received" condition. These measurements were carried out at the free-field measuring site (see Appendix B) with the microphone located at a radius of 40 feet from the center of the C-26 and at a height of 5 ft 5 in. above the ground. As before, the measurements were taken at 30-degree intervals around the machine.

The normal maximum steady-state rated load for the d-c generators is about 1200 amperes. However, a series-parallel connection of the electrical loading banks even with water-cooling could only dissipate something of the order of 950 amperes (the exact limiting value depended on the ambient temperature and the rate of water flow), and so a current slightly in excess of 900 amperes was selected as a suitable load for noise tests on the C-26. No attempt was made to load the a-c generator. Cooling water for the loading coils had to be trucked to the free-field test site. During measurements, the trailer with the cooling water and the loading coils were always placed at some distance away from the C-26 on the side away from the microphone so that they would interfere as little as possible with attaining acoustically valid free-field measurements.

To maintain as close regulation of the C-26 during surveys as possible, the engine regulator was readjusted for best operation. This first free-field survey was taken at a load current of 910 amperes ± 10% with the engine operating at 1890 rpm ± 2%. (See Appendix C for additional engine data.) In this case also, two complete sets of octave-band data were obtained at each microphone position as a check on the short time stability of the experiment. Since good agreement was obtained, the two sets of data have been averaged together for presentation here.

The results of this noise survey are presented in Figs. 3.22, 3.23, and 3.24, showing, respectively, the polar distribution of the computed overall noise, the average octave-band profile, and the directional deviations from the average octave-band profile. Figure 3.22 shows that, under the above conditions, the overall noise from the C-26 has an almost symmetrical distribution at a level of about 100 db at a distance of forty feet from the unit, and Fig. 3.23 indicates that the overall noise

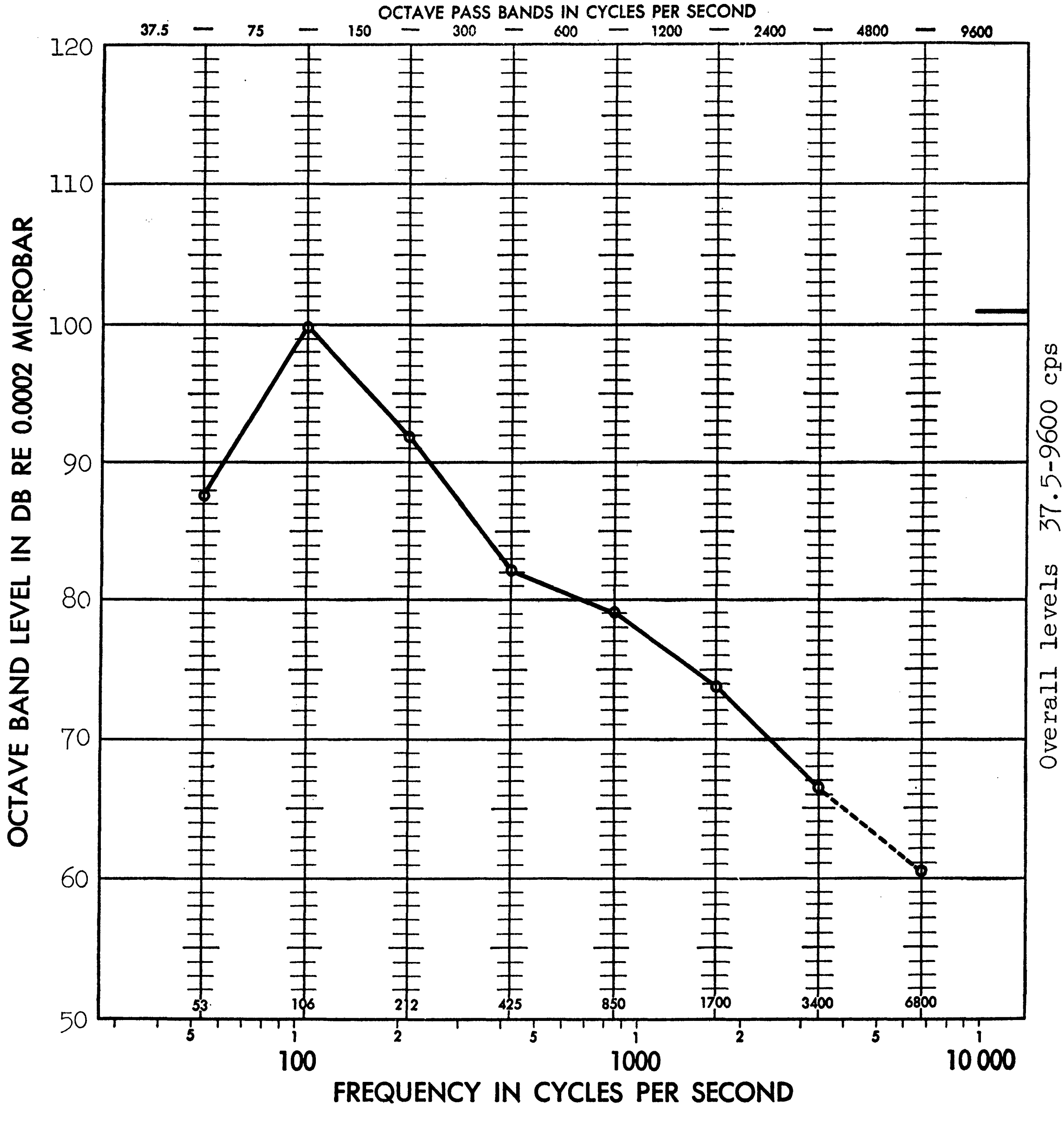
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type C-26 gasoline generator set, as-received condition
1880-1895 rpm, 800-955 amp, 29.0-30.0 volts dc
Tested 4-5 August 1955

Fig. 3.22. Polar distribution of overall noise; C-26, as-received.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave-band sound pressure levels




Type C-26 gasoline generator set, as-received condition
1880-1895 rpm, 800-955 amp, 29.0-30.0 volts dc
Tested 4-5 August 1955
Dotted line indicates that actual level probably lies lower than plotted value

Fig. 3.23. Average octave-band noise profile; C-26, as-received.

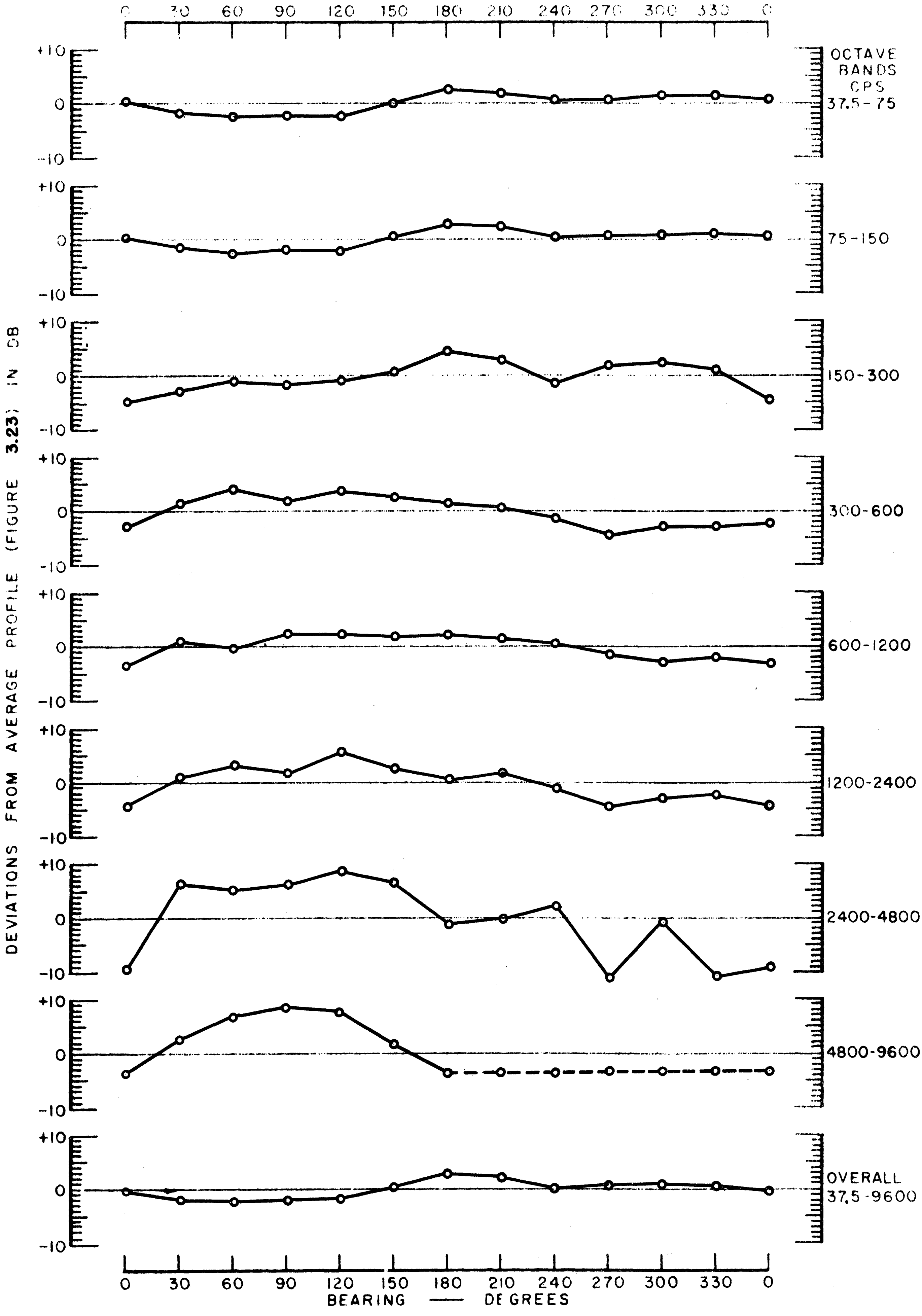



Fig. 3.24. Directional deviations from average profile; C-26, as-received.

is completely dominated by the 75-150 cps band. Figure 3.24 emphasizes that the lack of prominent directionality in the overall noise levels displayed in Fig. 3.22 is the direct result of the domination of the overall noise by the 75-150 cps octave-band noise. The lower 37.5-75 cps octave band has approximately the same minimal directionality characteristics, but all higher octaves exhibit distinctly different patterns, the high-frequency directionality being quite pronounced. The existence of a dotted line in the 4800-9600 cps graph of Fig. 3.24 means that for these bearings the actual levels were still lower but could not be measured due to interfering background noise.

Analysis of Initial Noise-Reduction Problem.—Using all the information available, the initial noise-reduction problem may be interpreted as follows. Figure 3.23 is the most revealing for it shows a complete domination of the general radiated noise by the noise contained in the 75-150 cps octave band. Since we are again dealing with an unmuffled internal-combustion engine, this low-frequency noise undoubtedly consists almost exclusively of engine or exhaust tones. In fact, the shape of the octave-band plot in Fig. 3.23 is typical of the noise radiated by a normal unmuffled internal-combustion engine free of prominent noise contributions from associated equipment. (For contrast, consider the case of the A-1 Generator Set, Fig. 3.4, where the high-frequency portion of the profile was strongly influenced by blower noise.) Since the engine operated at 1890 rpm, one would expect a series of exhaust tones spaced at an interval of about 15.8 cps (1/2 engine rps) and with the sixth harmonic at about 95 cps among others being particularly loud. This supposition is completely consistent with the experimental findings. Moreover, the lack of directionality exhibited by the overall noise or, what amounts to the same thing, by the 75-150 cps octave band, would be expected because of the vertical orientation of the exhaust stacks.

In view of the overwhelming predominance of the 75-150 cps octave-band noise, there is little practical value in proceeding much farther with the analysis until muffling has been accomplished. However, the directionality displayed in Fig. 3.24 by the noise in other octaves tempts a few observations. The higher frequency bands all show a minimum in the neighborhood of zero degrees bearing. This may be attributed to the acoustic shielding effected by the gasoline tank which occupies this end of the cart. The generator cooling-air blower faces the 90-degree direction and probably accounts for some of the higher frequency noise measured in this general direction. Finally, the side of the cart represented by bearings increasing from perhaps 240 degrees through 360 or 0 degrees is occupied by the generator control circuitry and is closed in the acoustical sense, which is probably responsible for the decreased noise levels observed in this area above 300 cps.

Muffler Tests.—As explained above, the essential first noise-reduction step is to muffle the exhaust. In view of the considerable success with the Walker 639 muffler when installed on the A-1 Generator Set in controlling exhaust tones in this same frequency range, it was decided to try two of these mufflers on the C-26's dual exhaust system also. Of course it was realized that the Walker 639's certainly were not acoustically optimum for this application, but they did constitute an immediately available laboratory expedient. In addition, to simulate a practical alteration a bit more closely with this laboratory experiment, the mufflers were installed within the C-26 cart rather than just coupled on externally. Figure 3.25 shows schematically the internal alteration necessary to permit mounting the two Walker 639 mufflers at the rear of the C-26's engine compartment.

This internal mounting of the mufflers required the removal of the augmentor

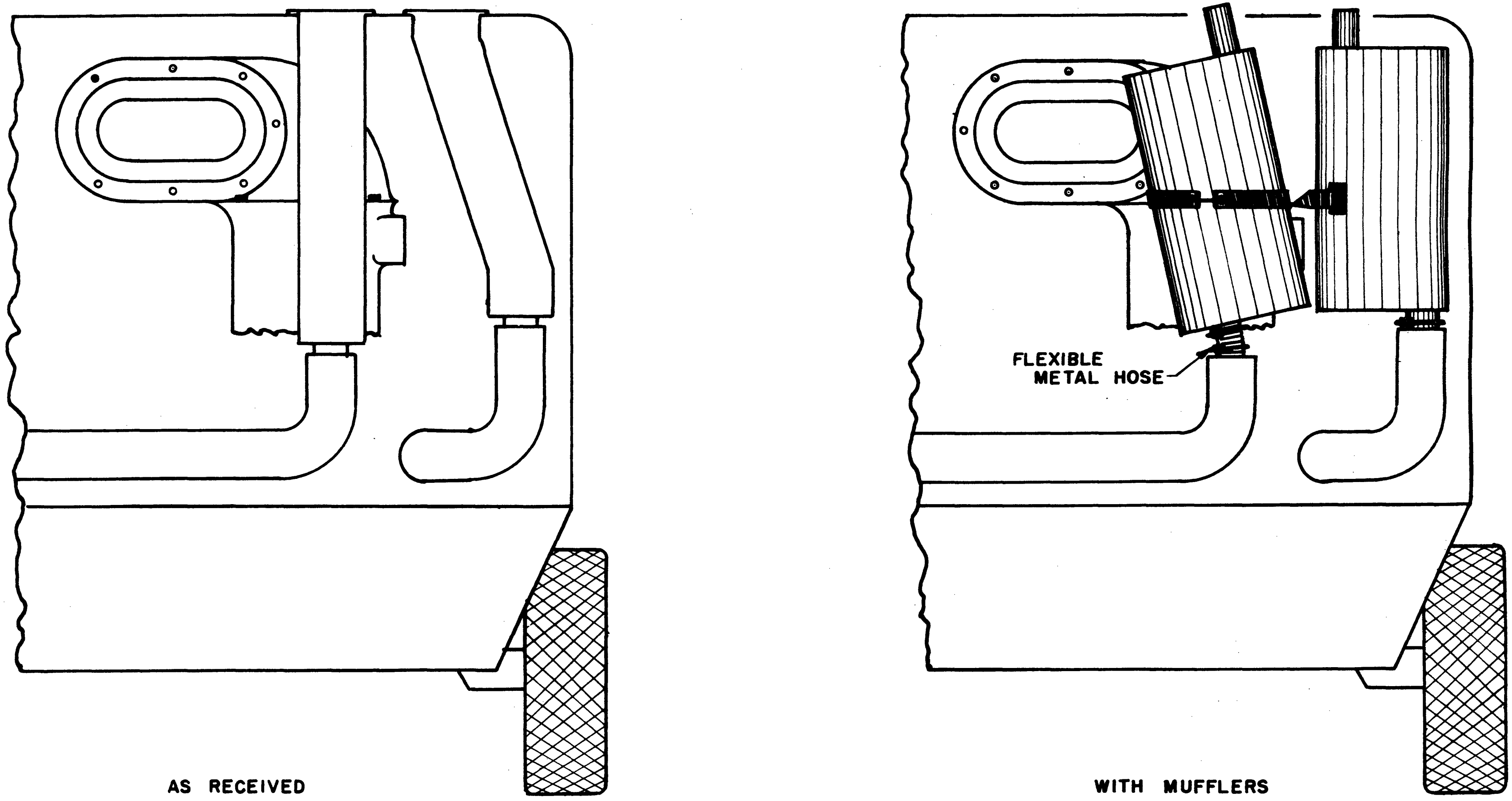



Fig. 3.25. Schematic view of C-26 muffler installation.

stacks, but the resulting loss of the augmentor action did not appear to result in any prohibitive increase of engine or compartment temperatures, at least not from the standpoint of laboratory testing. However, if optimum mufflers ultimately are designed for interior installation, there appears to be no fundamental reason why the augmentor action could not be re-incorporated.

With the muffler installation completed, another free-field noise survey was conducted, the results of which are presented in Figs. 3.26, 3.27, and 3.28. The values obtained in the original "as-received" survey have been included as dashed lines in Figs. 3.26 and 3.27 to facilitate comparison. The C-26 was operated as closely as possible to the same speed and load conditions used in the first tests. Measurements of the static pressure in the exhaust manifold indicated backpressures contributed by the mufflers amounting to from 5 to 7 inches of water. This amount of backpressure may be unacceptably high for a final production modification, but in these tests it merely emphasizes the need for mufflers designed specifically to match this model of engine.

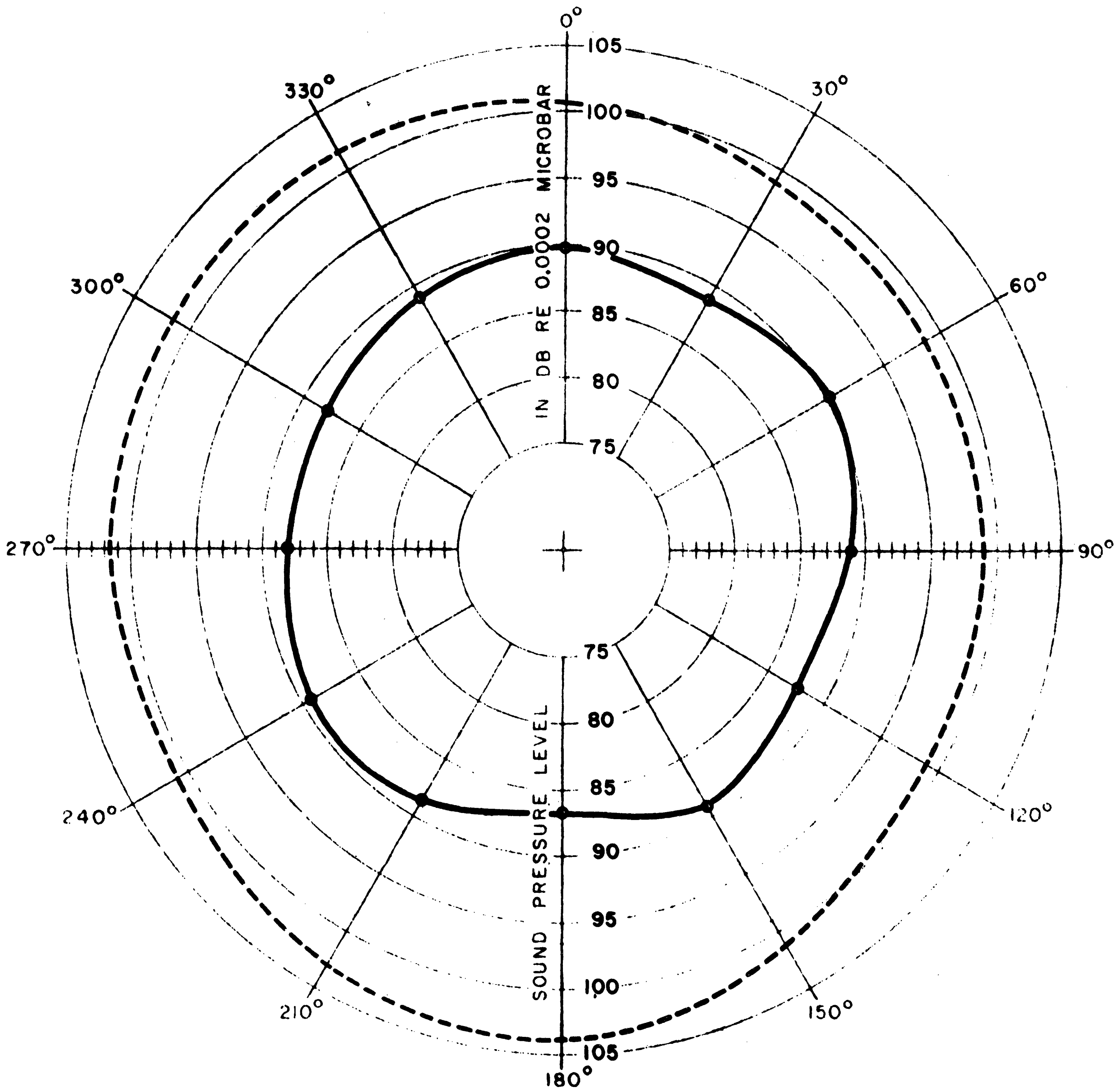
It is immediately apparent from Fig. 3.26 that large reductions in overall noise have been accomplished at all angles. These reductions range from 8.5 db at 60 degrees bearing to 17.3 db at 180 degrees bearing and average 12.2 db. In other words, the noise now on the average is only 25% of that radiated originally. Figure 3.27 shows that all this reduction was accomplished in the five lowest frequency octave bands. The largest average octave-band reduction, 14.0 db, occurred in the 150-300 cps octave closely followed by a 12.4-db reduction in the 75-150 cps octave. No significant change in the average noise level was detected above 1200 cps, a result to be expected since exhaust noise usually contributes very little to the high-frequency portion of the spectrum.

The octave-band directional deviations from the average profile, shown in Fig. 3.28, present a somewhat confusing picture. The patterns for most octaves seem to vary a bit more irregularly from one microphone position to another than they did before, and also seem to suggest a tendency toward noise maxima in the directions of the "closed" control-panel and fuel-tank areas. These directionality features are uninterpretable in detail from the limited data now available. About the only logical conclusion which can be drawn at this stage is that the originally predominant exhaust noise has been reduced to the point where the residual radiated noise in many octaves consists of approximately equivalent contributions from several sources, and perhaps reaches the measuring location by several devious paths out through the C-26's enclosure.

Nevertheless, Fig. 3.27 indicates that in spite of the large noise reductions accomplished, the residual noise still occurs predominantly in the 75-150 cps octave band. This suggests that even more effective mufflers are required to accomplish further noise reductions. Indeed, some of the related studies presented in Section IV of this report indicate that more effective muffling can be achieved on the C-26. However, since more effective mufflers were not readily available, it was decided at this point to investigate several other aspects of the C-26's noise.

Load Variation Tests.—While carrying out the tests reported above, it was suspected that the amount and spectral distribution of the radiated noise might depend strongly on the imposed load conditions. Therefore, it was decided to return to the "as-received" condition and determine in detail how the noise depended on the applied load for at least one orientation of the machine. Figure 3.29 shows the free-field

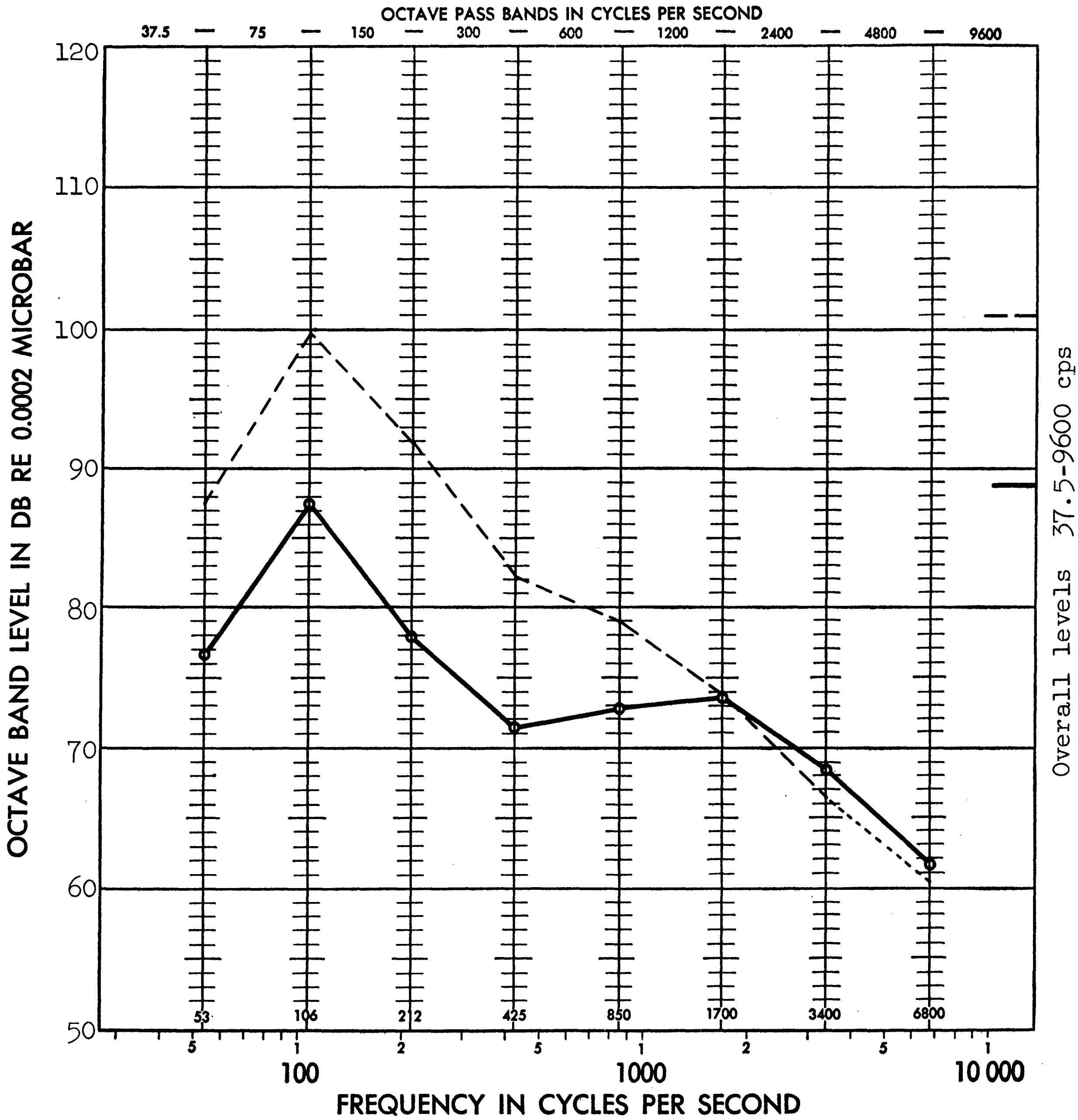
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type C-26 gasoline generator set, equipped with two Walker 639 mufflers, see Fig. 3.25
1840-1960 rpm, 910-1000 amp, 29.5-30.0 volts dc
Tested 28 October 1955
-----As-received condition, see Fig. 3.22

Fig. 3.26. Polar distribution of overall noise; C-26, muffled.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave-band sound pressure levels




Type C-26 gasoline generator set, equipped with two Walker 639 mufflers, see Fig. 3.25
1840-1960 rpm, 910-1000 amp, 29.5-30.0 volts dc
Tested 28 October 1955

Fig. 3.27. Average Octave-band noise profile; C-26, muffled.

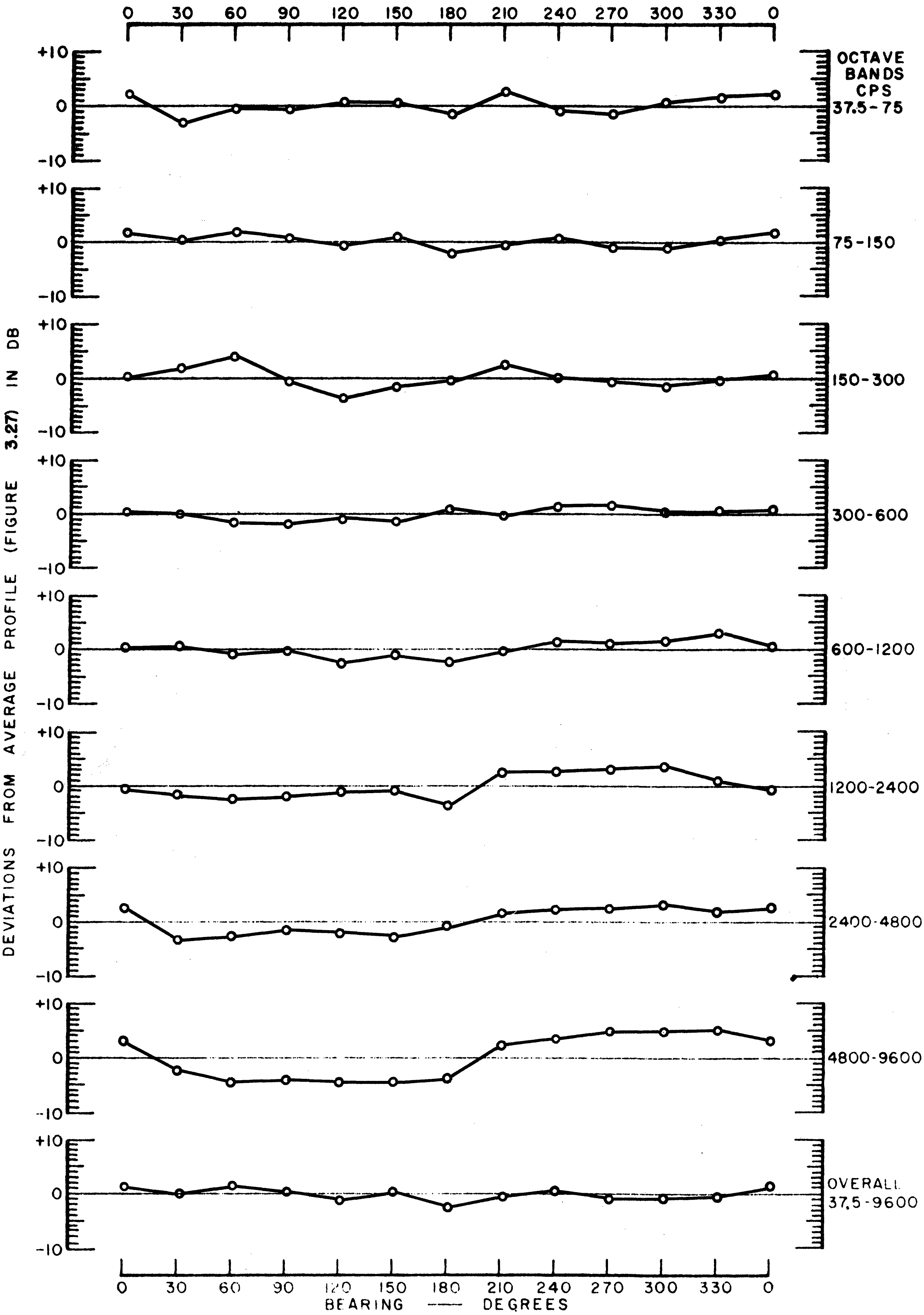
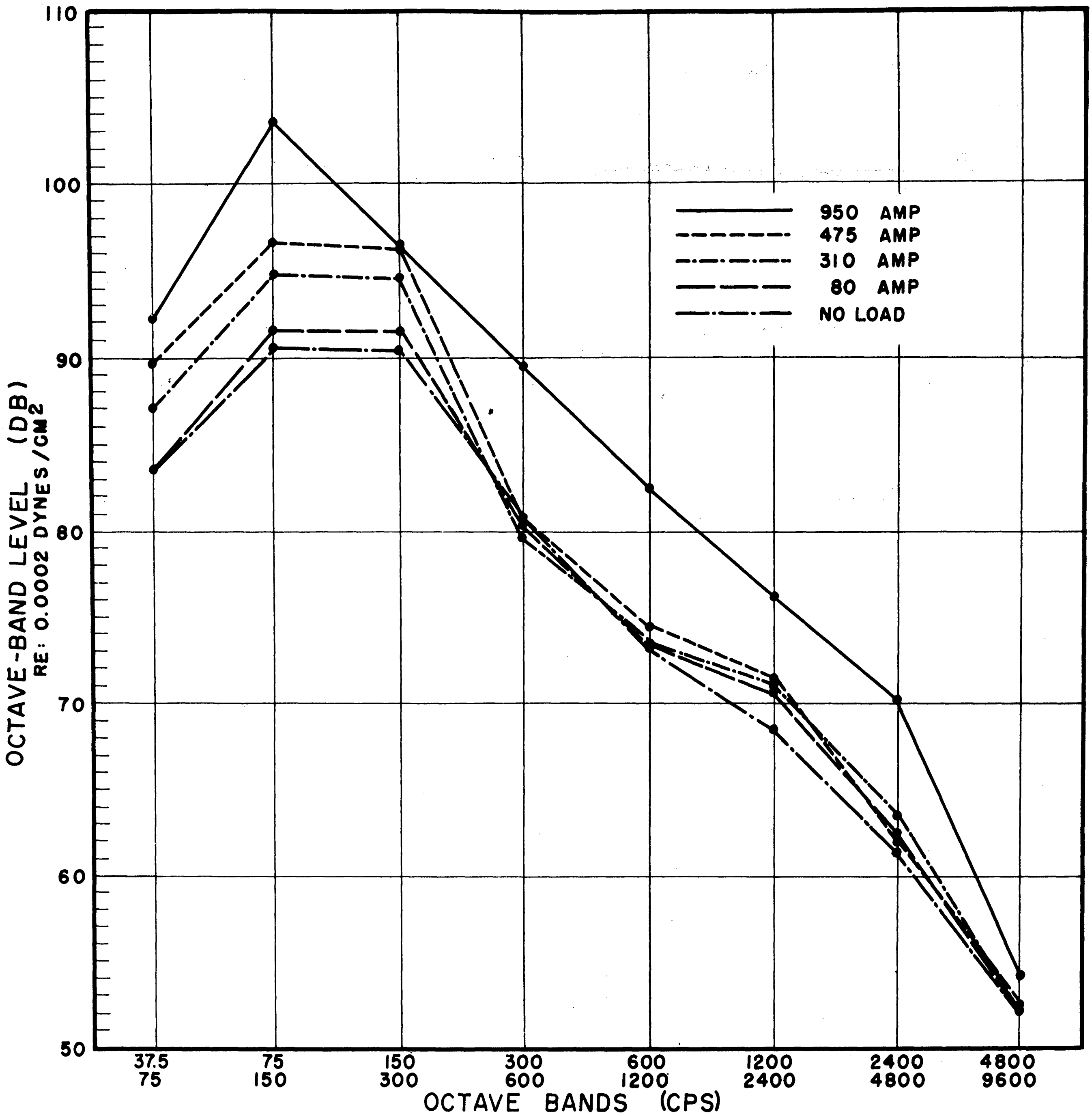
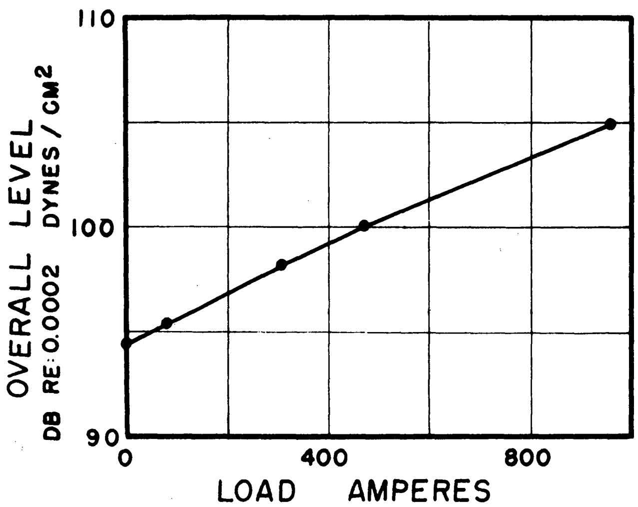



Fig. 3.28. Directional deviations from average profile; C-26, muffled.




AS - RECEIVED CONDITION
MICROPHONE DISTANCE = 40'
180° BEARING

Fig. 3.29. Variation of C-26 noise level with load.

noise levels measured at the 180-degree bearing for load currents of 950, 475, 310, 80, and 0 amperes. The engine operating speed was maintained fairly constant throughout these tests. The 80-ampere load is the current drawn from the generators when only the generator cooling-air blower is operated, the no-load condition being obtained by switching off the cooling blower.

Two octave bands, 75-150 and 150-300 cps, which are dominated by exhaust tones, show a steady increase in level with load and, because they are the highest level octaves, largely determine the overall noise level which can be seen to increase almost linearly with load current. Most of the high-frequency bands seem relatively unaffected by load up to at least a load current of 475 amperes. The two distinctly different load-noise characteristics exhibited by the several octave bands are displayed more clearly in Fig. 3.30. The three lowest frequency octave bands show a relatively steady increase of noise level with increasing load while the higher frequency octave bands exhibit relatively constant noise levels until high loads are reached. The dotted line in a portion of the 4800-9600 cps graph denotes uncertainty in the measurements due to interfering ambient noise. Probably the actual levels for this octave band fall somewhat below the levels indicated.

Data accumulated during the load-dependence tests permit a check of the reproducibility of noise measurements made on the C-26 after a prolonged interval of time. Figure 3.31 shows the correspondence of data accumulated on 4 August 1955 at the 180-degree bearing during the original "as-received" noise survey with similar data from the load-dependence tests conducted on 10 November 1955. The general agreement between these two sets of data is very good, in fact, better than is usually expected in machinery-noise measurements of this type. It is probably significant that the two octave bands, 300-600 cps and 2400-4800 cps, showing the greatest divergences in Fig. 3.31 are demonstrated in Fig. 3.30 to be strongly dependent on load current near the higher load condition of 950 amperes.

Because the increase in overall noise level of the unmuffled C-26 with load can be traced to an increase in exhaust noise, a similar load-dependence test was performed with the two Walker 639 mufflers reinstalled. In addition, the generator cooling-air blower was resiliently mounted. (A discussion of this phase of the noise-reduction study is presented in the following section.) The results of this load-dependence test are presented in the same manner as before in Figs. 3.32 and 3.33. The data for the C-26 without mufflers appearing originally in Fig. 3.30 have been replotted as dashed lines in Fig. 3.33 to facilitate comparison.

Figure 3.32 shows that at small loads, for the 180-degree orientation at least, the 75-150 cps octave-band noise just barely succeeds in dominating the overall levels, and that some of the higher frequency bands, particularly the 600-1200 cps octave band, contribute significantly to the overall noise. However, as the load is increased, the lower frequency octaves, particularly the 75-150 cps octave band, rise in prominence and reach complete domination at the 980-ampere load. This demonstrates the marginal acoustic adequacy of the Walker 639 mufflers under no-load operation and the pronounced inadequacy at higher loads. The overall noise level rises with load also, but not as smoothly as before, which is probably indicative of an altered interplay among various contributory factors. The exact nature of this interplay has not been revealed clearly by any of these measurements.

Figure 3.33 is consistent with the results of the complete free-field survey presented in Fig. 3.27 in that the mufflers have significantly reduced the noise in the

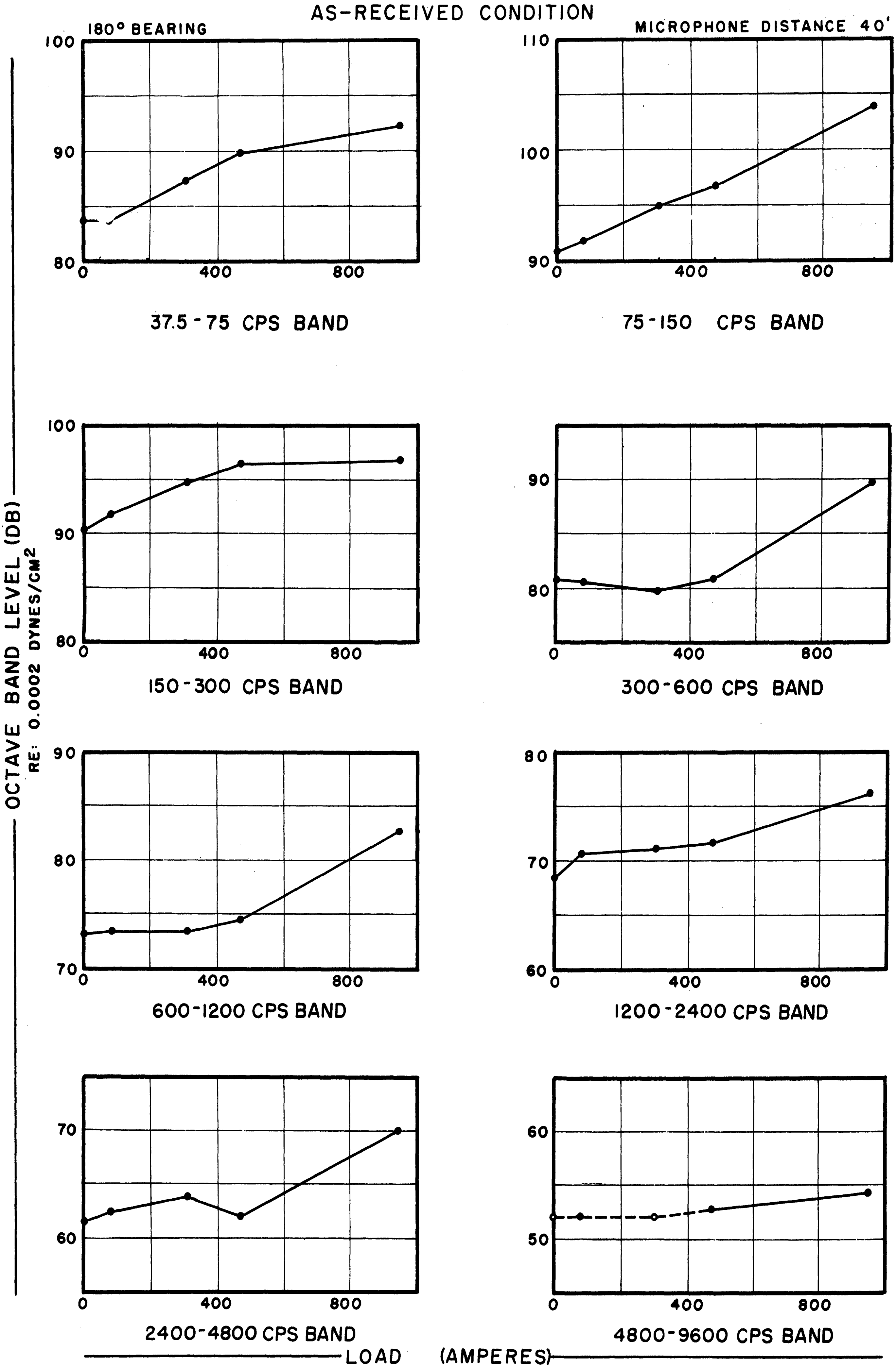
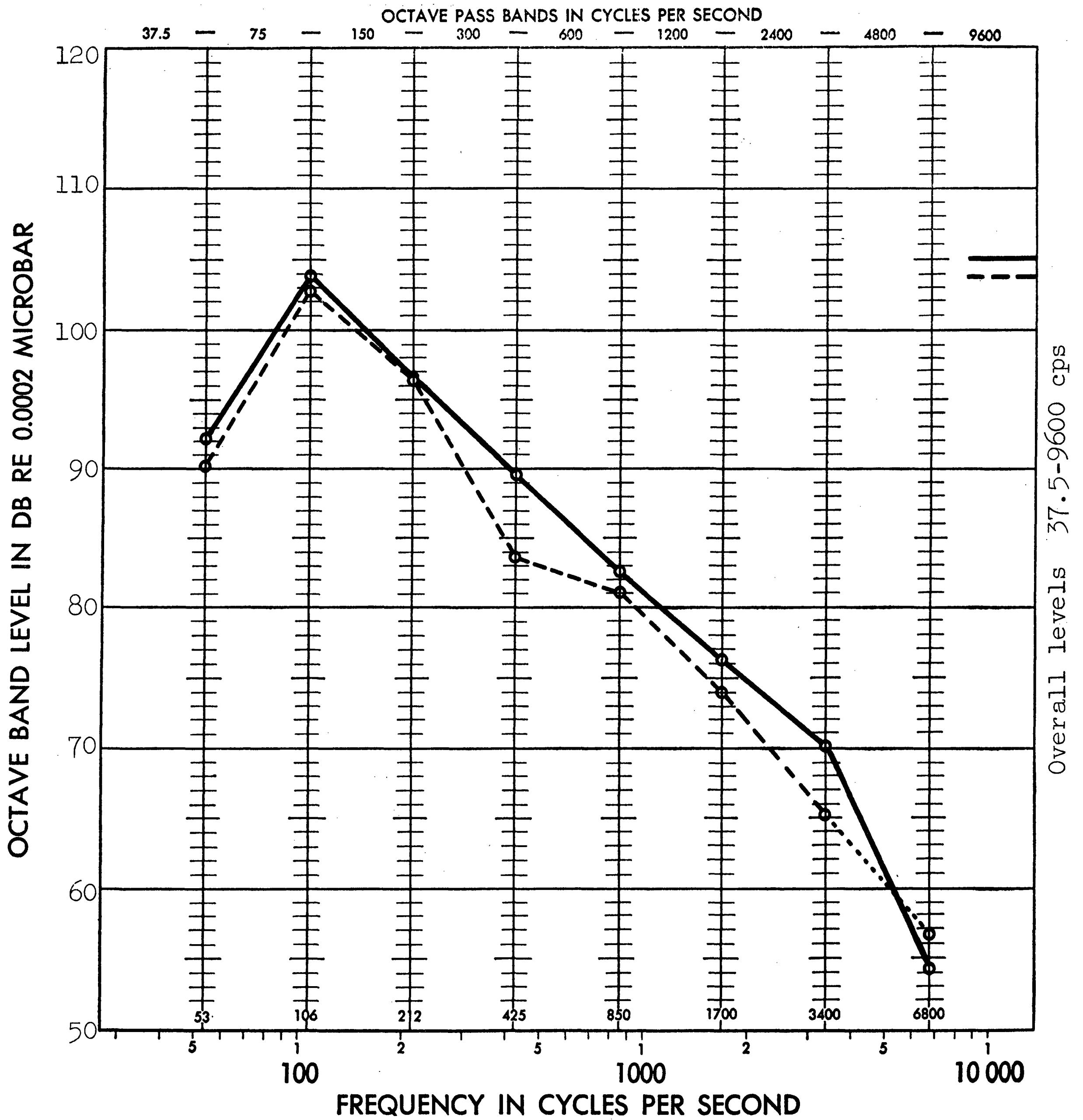



Fig. 3.30. Variation of C-26 octave-band noise levels with load condition.




Type C-26 gasoline generator set, as-received condition
---------- Data taken 4 August 1955, 930 amp load
———————— Data taken 10 November 1955, 950 amp load
Dotted line indicates actual level probably below plotted level

Fig. 3.31. Reproducibility of C-26 noise level profiles.

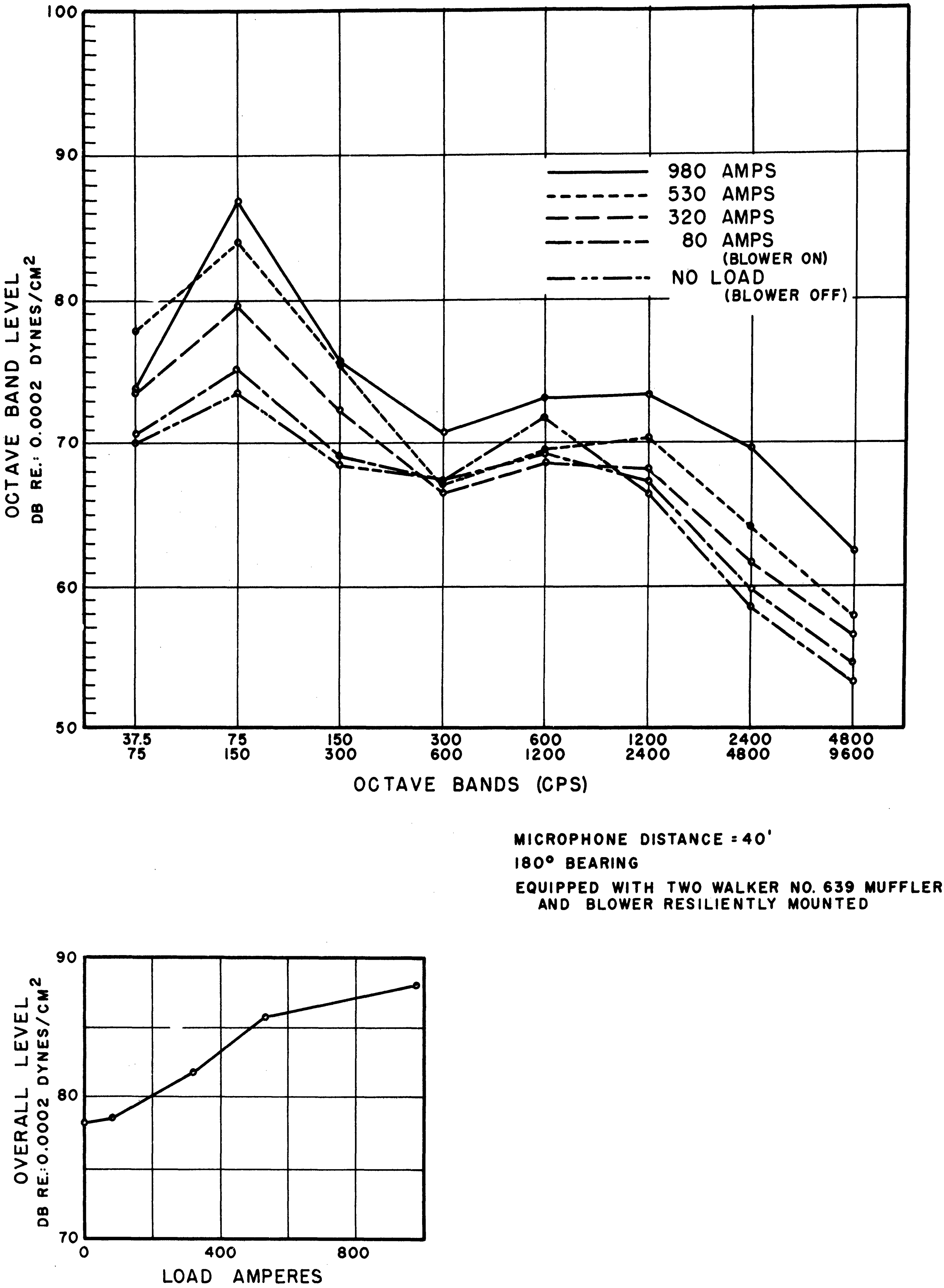



Fig. 3.32. Variation of C-26 noise level with load.

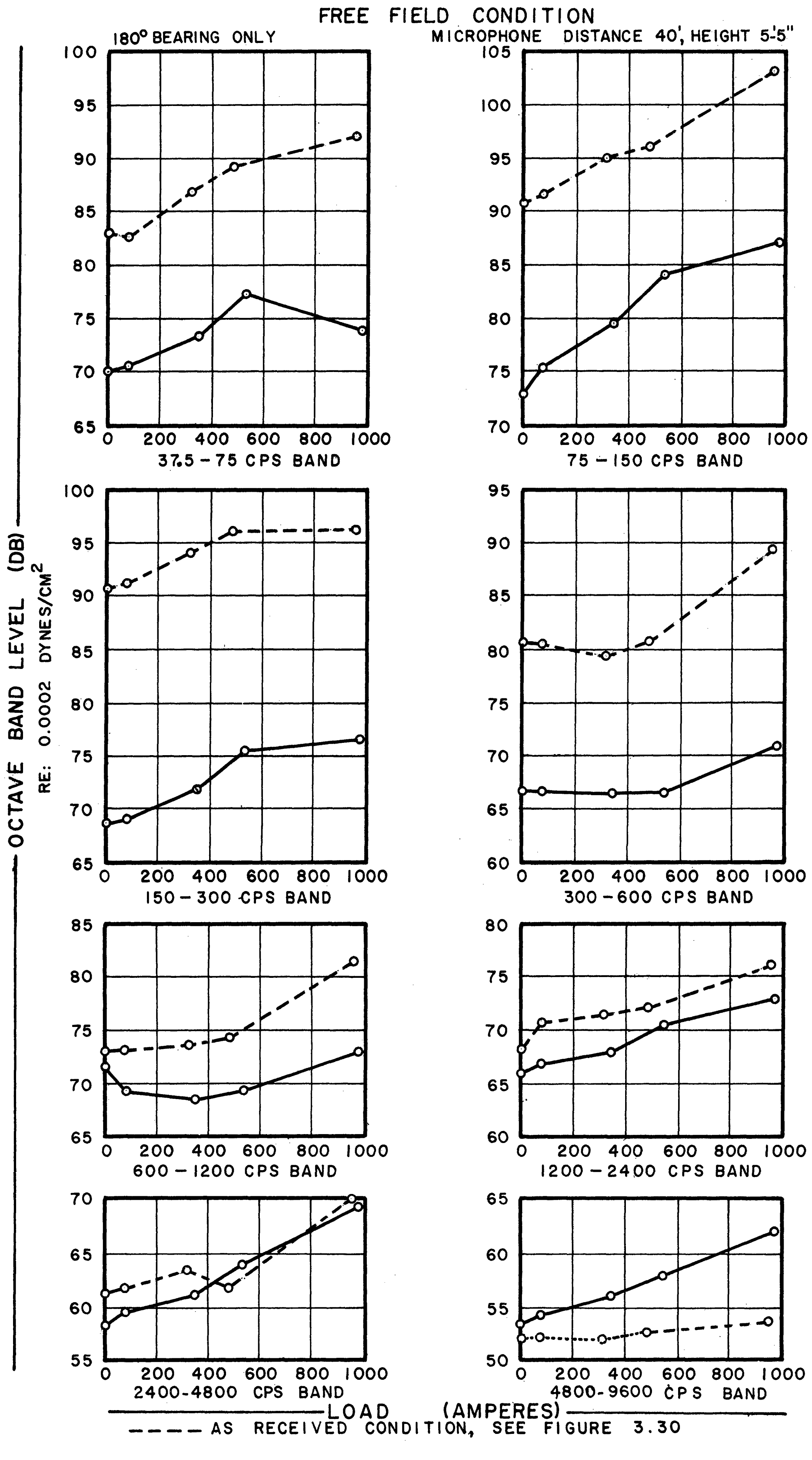



Fig. 3.33. Variation of C-26 octave-band noise levels with load condition. (2 Walker mufflers and resilient mounting)

five lowest octave bands. This is seen to be true at all loads tested. The highest octave bands present a confusing picture but this does not interfere appreciably with the main course of the research because these high-frequency octave bands are of comparatively low sound-pressure level and hence have little effect as yet on the noise-reduction program for the C-26. In fact, the very lowness of the high-frequency noise levels probably contributes to the confusion since these low measured values are more susceptible to influence by interfering ambient noises. However, considering as a whole the measurements made on the C-26, there does seem to be some evidence of gradually increasing generation of high-frequency noise, perhaps due to normal wear of the machinery.

Resilient Mounting of Generator Cooling-Air Blower.—During some of the exploratory tests in connection with the load-dependence experiments discussed above, it was noticed aurally and verified by tactile estimation of the vibration of the various panels of the C-26 cart that an abnormally large increase in noise and vibration accompanied the application of the 80-ampere load. Thus it was suspected that the generator cooling-air blower could in some way contribute appreciable noise by itself. Probably the excitation was due to unbalance of the blower motor and thus consisted of a series of frequency harmonics related to the rotational speed of the motor. Since the normal operating speed was about 8000 rpm, the fundamental should be at about 133 cps, which falls within the 75-150 cps octave band.

It was found that the inordinately large amount of noise generated by the operation of the cooling-air blower could be demonstrated easily in the laboratory by operating the blower momentarily from the C-26's batteries without starting the engine. This procedure was eminently satisfactory for quick qualitative listening tests or demonstrations, but the limited battery capacity would not permit the extended operation necessary for quantitative acoustic surveys.

It was planned to conduct a series of special measurements with the microphone located a few inches away from each of the panels of the C-26 to gain some insight into the relative noise contributions from or through these panels for a range of operating parameters. To provide a clearer picture unhampered by the C-26's own engine noise, the A-1 Generator Set was to be used to supply electrical power to the C-26's cooling-air blower. The A-1 could be located remotely outside of the laboratory building, thus minimizing acoustical interference from it. About the time that these experiments were nicely underway, the splined drive shaft in the A-1 fractured, and a replacement part could not be secured for several months. Since no other source of d-c power sufficient to operate the cooling-air blower was immediately available, it was necessary to operate the C-26's engine to supply the electrical power for the blower. Hence the close-in evaluation of the noise generated by the cooling-air blower and the effectiveness of the subsequent resilient mounting of this blower had to be carried out in the presence of obscuring noise from the engine.

The generator cooling-air blower was resiliently mounted on four Lord 102P-8 mounts by an arrangement shown schematically in Fig. 3.34. These particular mounts were utilized because they were the most suitable ones immediately available from laboratory stock. For maximum effectiveness they were located as closely as possible to the horizontal plane containing the center of gravity as the space confines would allow. It was practically impossible to insert a flexible duct connector between the blower and the C-26 cart at the rear of the blower at the position where the blower was originally mounted. Instead, a flexible cloth duct connector was installed between the intake bell of the blower and the outside of the C-26 cart, thus

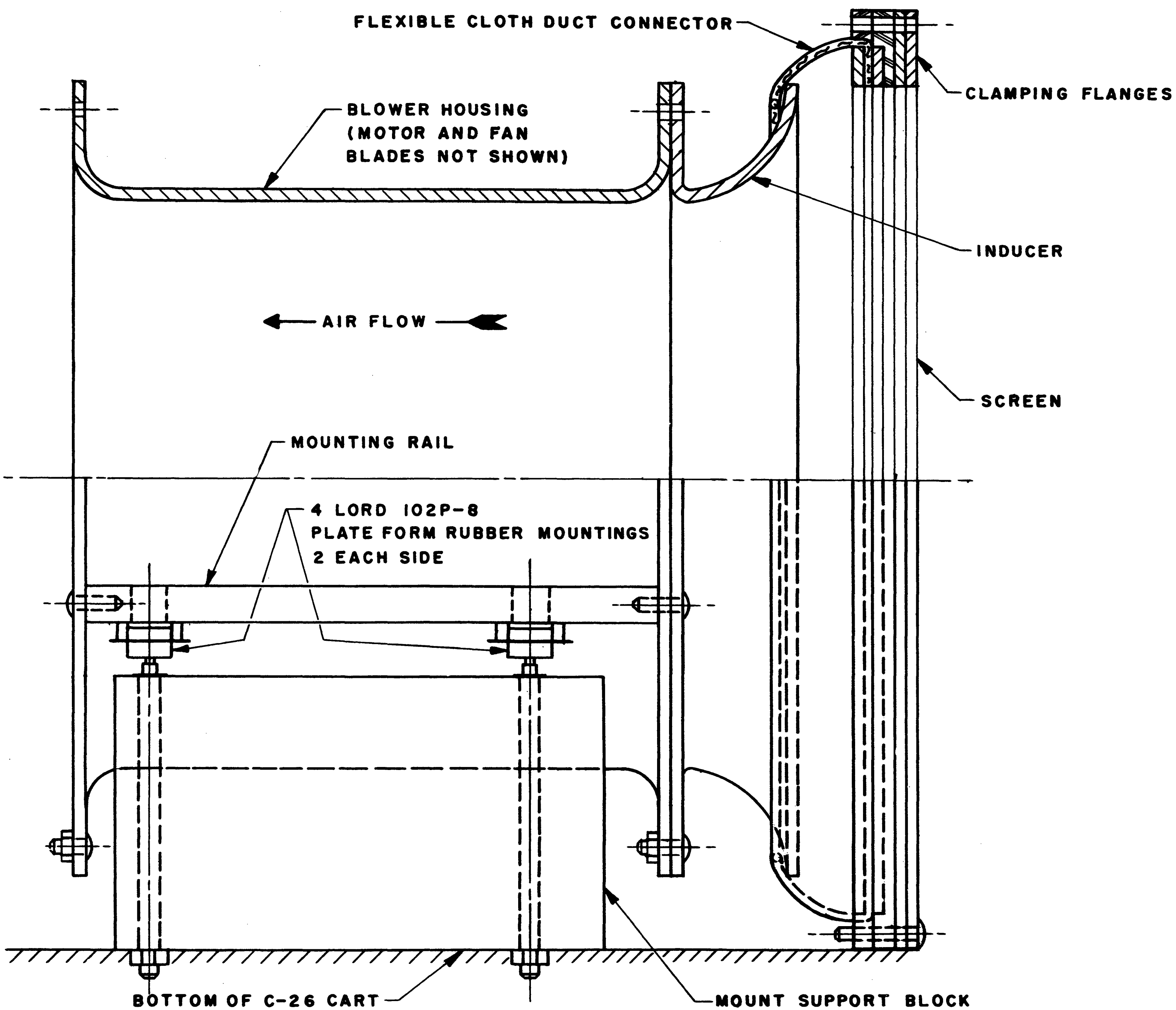


Fig. 3.34. Half section, schematic of resiliently mounted C-26 cooling air blower.

converting the entire blower compartment into a plenum chamber. Every precaution was taken to assure that the resilient mounts could not be accidentally acoustically shorted out.

Brief listening tests with battery operation verified the effectiveness of the resilient mounts but it was still necessary to prove that the resilient mounting brought about a measurable decrease in noise during actual operation of the C-26. Close-in measurements were taken at several positions for the C-26 equipped with the Walker mufflers and the increase in noise investigated which occurred when switching from no load to blower load. Figure 3.35 shows the results for the two conditions of the blower rigidly mounted and the blower resiliently mounted. Position No. 7 is near the engine compartment's solid panel on the opposite side of the C-26 from the blower. Position No. 16 is directly in front of the blower intake and position No. 19 is adjacent to the top of the fuel tank. These measurements indicate that the vibration generated by the cooling-air blower was widely distributed throughout the entire C-26 cart and radiated as airborne noise from all parts of the cart.

Following these close-in measurements, a limited free-field survey was conducted to see if the improvement just described could be detected and corroborated in the far-field radiated noise. To reduce the amount of labor, measurements were conducted at only four bearings, namely 0, 90, 180, and 270 degrees for the conditions of no load, blower load, and 990-ampere load with the cooling-air blower first rigidly and then resiliently mounted. Since the noise profiles were very similar at each bearing, the data for all four bearings have been averaged together for presentation here. Figure 3.36 shows the results with the blower rigidly mounted. The inordinately large increase in the 75-150 and 150-300 cps octave-band noise due to blower operation is evident. Figure 3.37 presents the data for the same three operating conditions with the cooling-air blower resiliently mounted. The reduction in the noise levels for blower operation particularly in the 75-150 and 150-300 cps octave bands is evident. Small reductions in the noise levels for the 990-ampere load also appear. These small reductions may be due partly to diminished contributions from the blower and partly to the larger random scatter of data, since only four sets of data instead of twelve have been averaged together.

By plotting in juxtaposition both the increases in noise level due to the rigidly and to the resiliently mounted blower, as illustrated in Fig. 3.38A, the net effect of the resilient mounting can be demonstrated more clearly. The maximum reduction in radiated noise occurring in the 75-150 and 150-300 cps octave bands is entirely consistent with the findings of the near field measurements presented in Fig. 3.35. On the basis of the computed overall levels, the resilient mounting of the blower has reduced the radiated noise for blower operation alone from 83.4 db to 78.6 db.

In Fig. 3.38B, the increases of the octave-band sound-pressure levels due to applying a 990-ampere load are shown. Consistent with previous results, the large increases occur in the 75-150 and 150-300 cps octave bands. The magnitude of the high-frequency increases is somewhat less certain than at lower frequencies since, as Figs. 3.36 and 3.37 have already demonstrated, the high-frequency levels are comparatively low, and they are affected more by ambient noise. Thus the high-frequency increases displayed in Fig. 3.38B are the differences between two relatively small and less precise values, and therefore are themselves somewhat questionable.

Special Muffler Studies.—Section IV of this report discusses in detail certain studies conducted with a laboratory-built experimental muffler using the C-26 as a

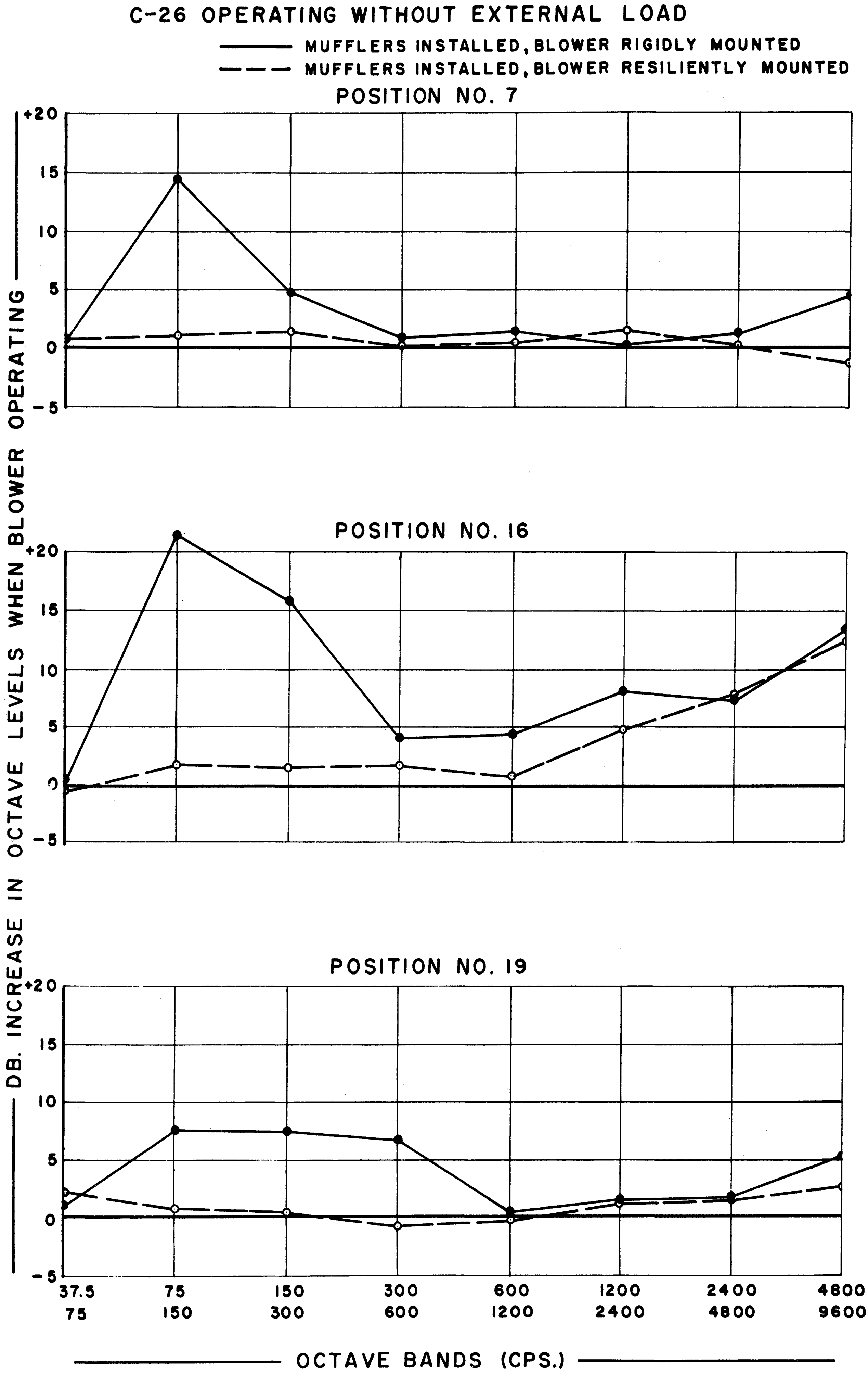
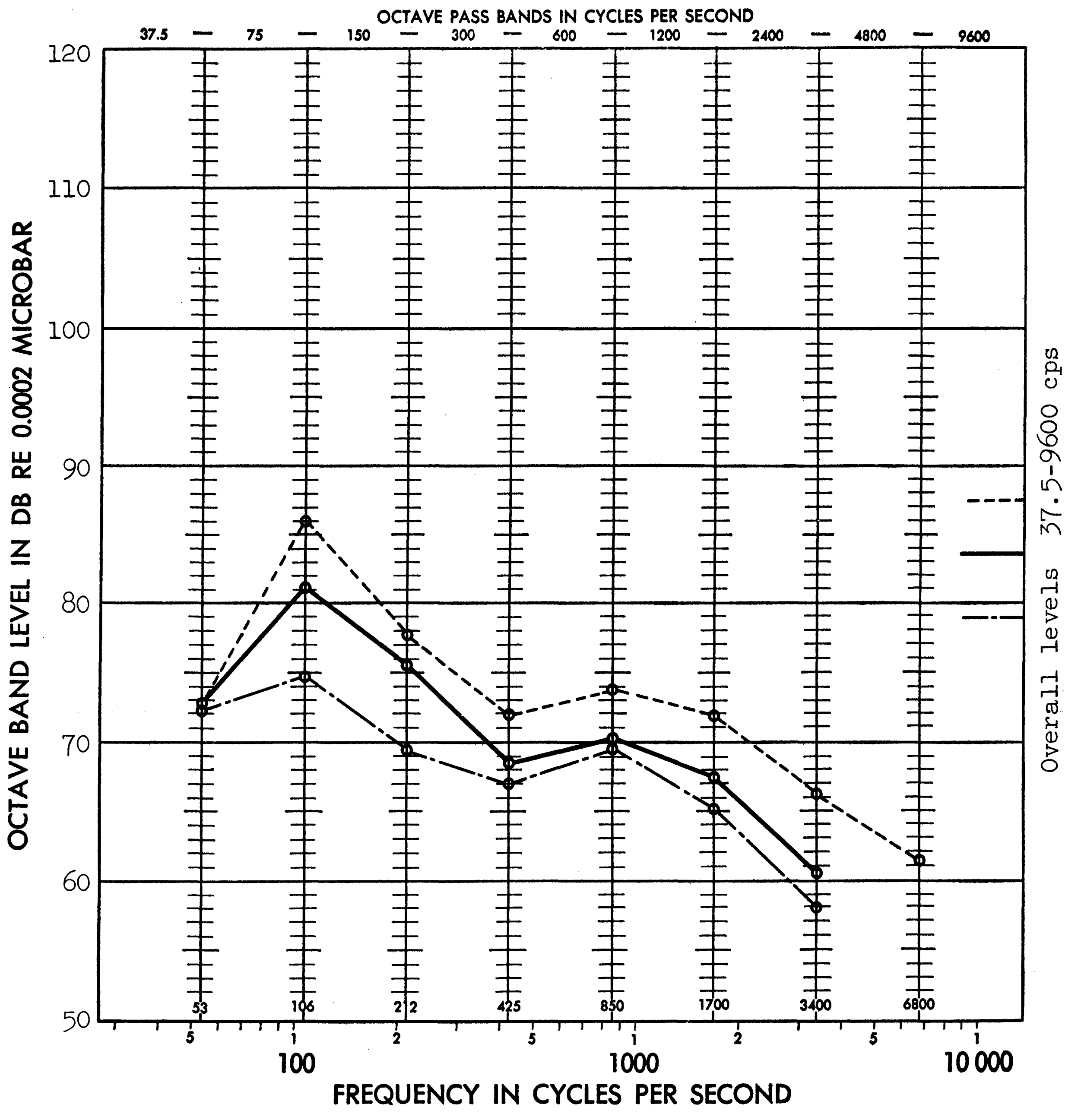



Fig. 3.35. Effect of operating blower with and without resilient mounts—nonfree-field measurements.

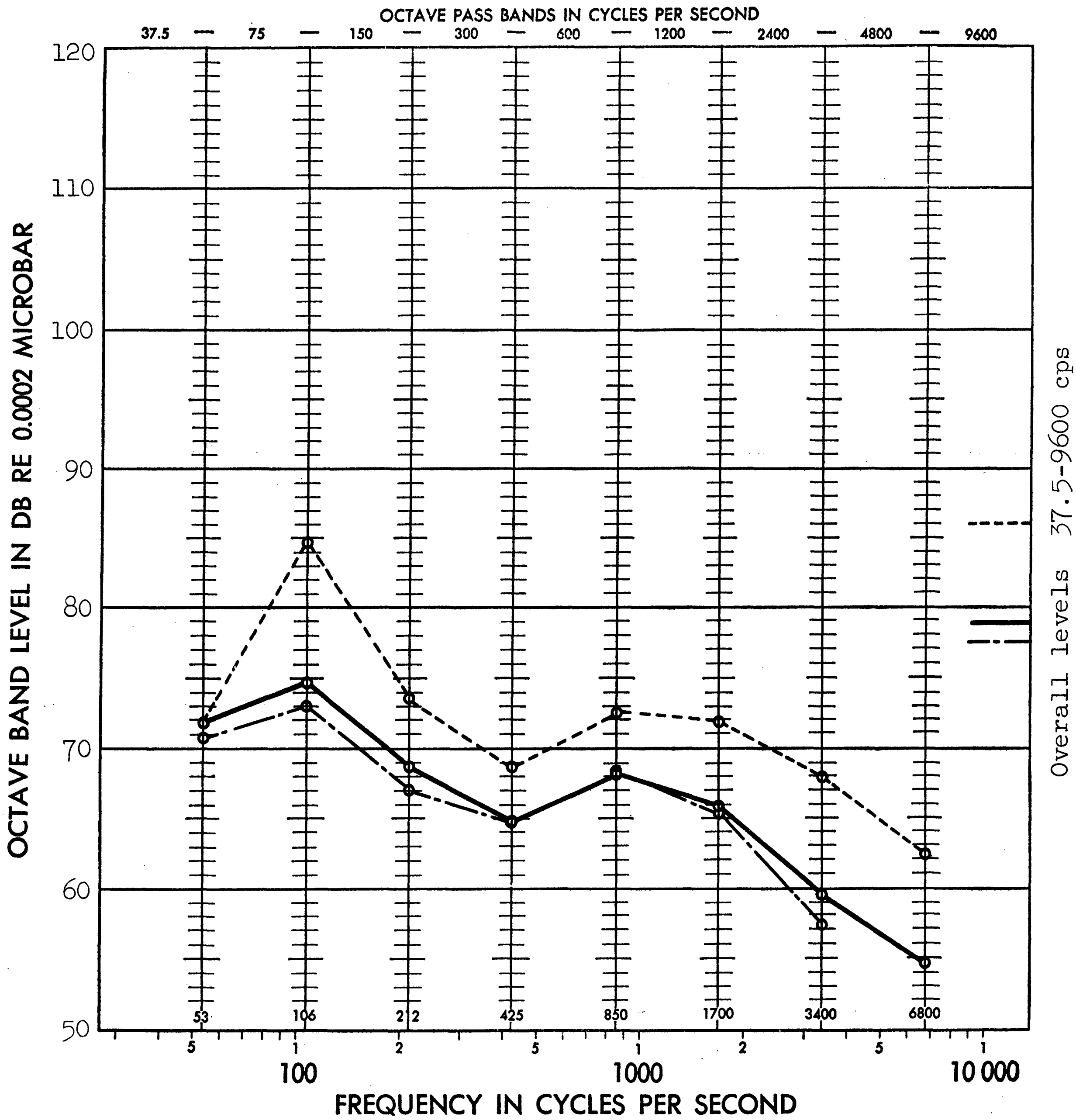
Free-field conditions
Microphone distance 40', height 5'5"
Average of 4 octave-band sound pressure levels (0°, 90°, 180°, 270°)




Type C-26 gasoline generator set, equipped with two Walker 639 mufflers, generator cooling blower rigidly mounted
1740-2000 rpm, 28.0-29.7 volts dc
—— - —— No load
———— 95 amp load (cooling blower only)
--------- 970-1010 amp load
Tested 30 December 1955

Fig. 3.36. Average octave-band noise profiles; C-26, blower rigidly mounted.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 4 octave-band sound pressure levels (0°, 90°, 180°, 270°)




Type C-26 gasoline generator set, equipped with two Walker 639 mufflers, generator cooling blower resiliently mounted, see Fig. 3.34
1720-2010 rpm, 28.0-30.0 volts dc
—— - —— No load
———————— 80-85 amp load (cooling blower only)
--------- 980-990 amp load
Tested 21-22 February 1956

Fig. 3.37. Average octave-band noise profiles; C-26, blower resiliently mounted.

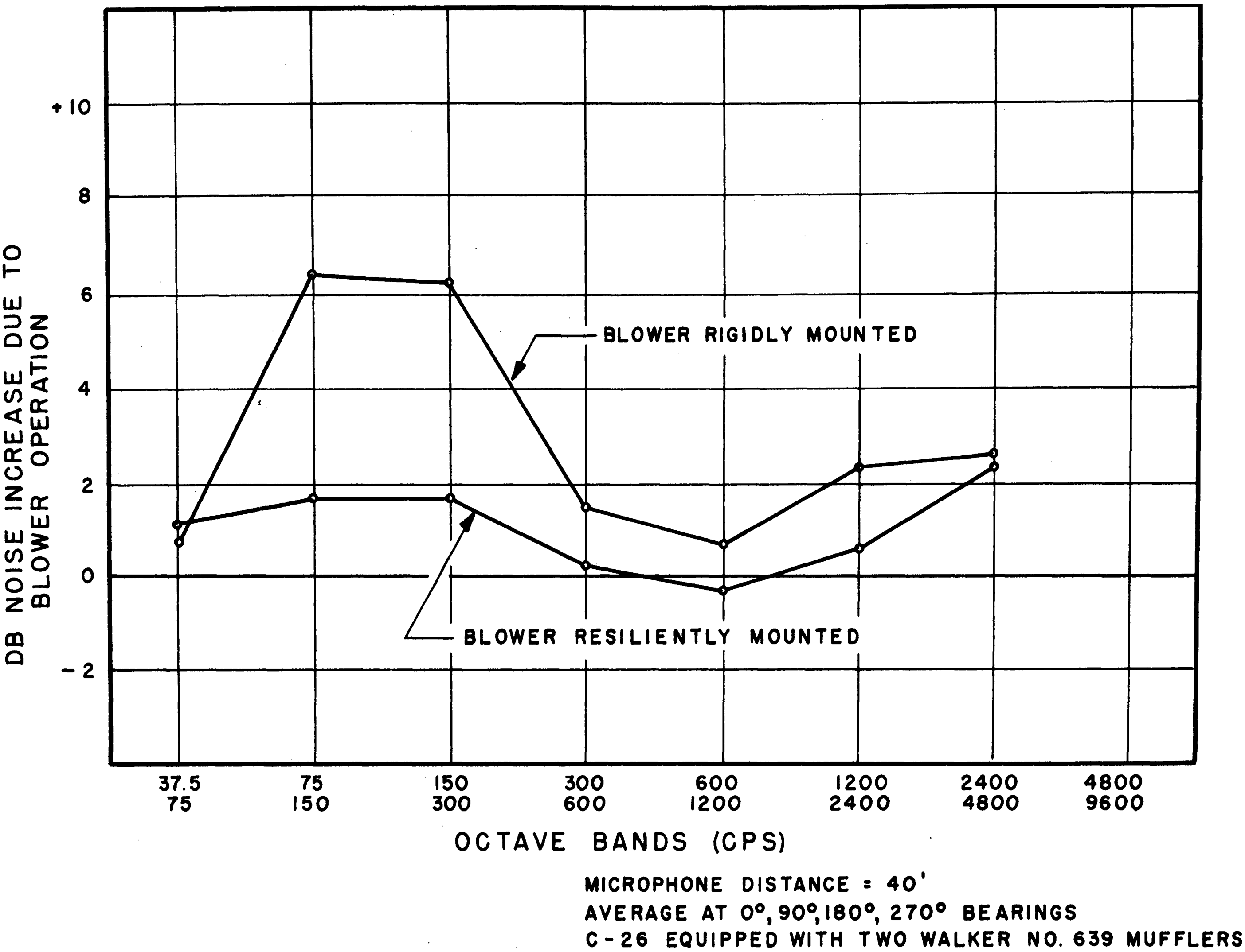



Fig. 3.38A. Effect of resiliently mounted blower.

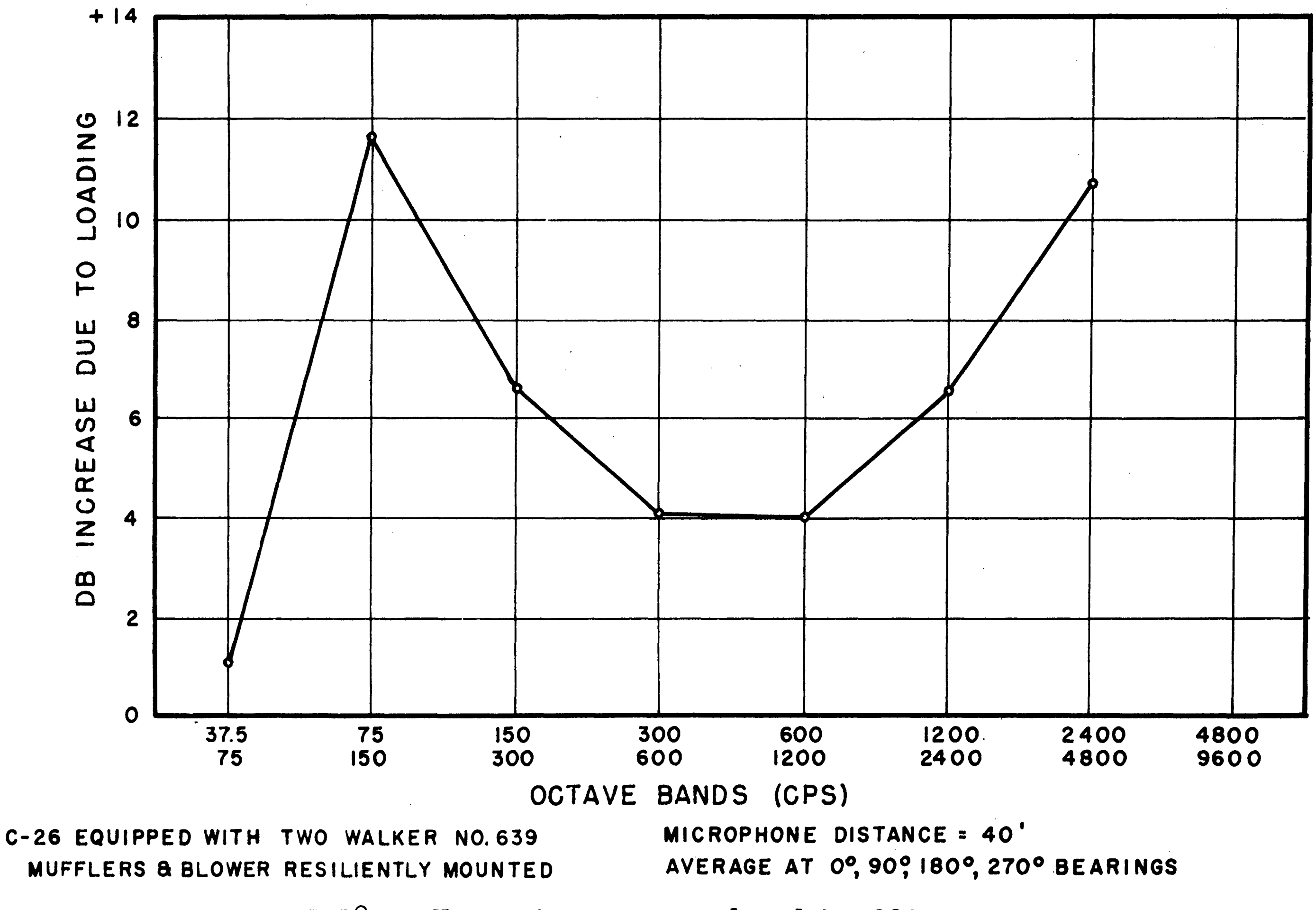



Fig. 3.38B. Noise increase produced by 990 ampere load.

test vehicle. There is no point in duplicating that discussion, but for the sake of a complete presentation of the C-26 noise problem certain pertinent aspects of those special muffler studies will be summarized here.

Measurements taken during the special muffler studies were all close-in positions in an arbitrary acoustic environment immediately outside the laboratory building; hence only comparisons possess validity. Discrete frequency analyses were carried out on the noise in the suspect range from about 75 to 2500 cps. These analyses confirmed the presence of particularly prominent discrete tones, identifiable as harmonics of the engine firing frequency, in the 75-150 and 150-300 cps octave bands. Furthermore, the intensities of several of these harmonics, e.g., at roughly 100, 150, 200, and 250 cps, increased markedly with engine loading, thereby accounting for the correspondingly large increases observed in the free-field octave-band measurements.

Additional discrete frequency analyses with the Walker 639 mufflers installed verify that these mufflers essentially eliminate the higher frequency exhaust tones and considerably attenuate the lower frequency tones. However, with increasing load, the Walker mufflers do not maintain sufficient control over the low-frequency exhaust tones. It was found possible to adjust the parameters of the experimental muffler to attenuate the lower frequency exhaust tones very effectively, much more so than with the Walker mufflers. However, to obtain a well-sounding muffler, it is necessary to adjust carefully the proportion of attenuation in the various parts of the noise spectrum. Ultimately it was found possible to adjust the parameters of the experimental muffler so that it sounded better than did the Walker 639 muffler, and in addition, the experimental muffler provided enhanced low-frequency attenuation.

Absorptive Lining of Cart.—A perusal of the C-26 studies up to this point indicates that the various treatments employed have had comparatively little effect on the high-frequency radiated noise. This higher frequency noise is just beginning to show prominence since the low-frequency noise has been greatly reduced and is particularly important now that ways have been demonstrated to secure even more low-frequency attenuation. The high-frequency noise seems to originate, at least partially, in the generators or generator gear train. The application of palliative absorptive treatments within the C-26 housing to attenuate these higher frequencies, along with a general acoustic tightening of the housing, would seem to be the next logical step. Absorptive panels were fabricated from fiberglass and used to line as much of the engine and generator compartments as possible. To simulate more closely an absorptive treatment which might be acceptable from an environmental standpoint, fiberglass sheets 1 or 2-1/2 inches thick (Owens-Corning Fiberglas PF-335, 1 in. thick, 0.75 lb/cu ft and PF-334, 2-1/2 in. thick, 0.5 lb/cu ft, respectively) and of the desired size and contour were enclosed in a single layer of 0.0015-inch-thick aluminum foil. This protects the fiberglass from fumes, moisture, and oil, while retaining reasonably high absorption.

Although reverberation room evaluations of the diffuse absorption coefficients of these composite treatments were not actually carried out, it is possible to synthesize the probable values with a fair degree of accuracy. This can be accomplished using known values of the diffuse absorption coefficients for the plain fiberglass blankets placed in direct contact with a hard surface and the known effects of adding impermeable surface septa. In the present case, the absorption coefficient data for 2-1/2-in. PF-334 and 1-in. PF-335 on No. 4 mounting were available from technical data sheets,[1] and the effect of an impermeable surface septum was estimated from other tests involving surface facings (see Fig. 3.5 of Reference 2). The results of this synthesization are shown in Fig. 3.39, and are believed to represent the actual values fairly well.

Although the diffuse absorption coefficient as evaluated by the reverberation room method is not a perfect representation of the behavior of the composite treatment as installed within the C-26 cart, still it is probably as correct as any materials-measurement which might be selected. The important ramification to be derived from Fig. 3.39 is the large amount of absorption available in the 250-, 500-, and 1000-cps ranges when a protective facing is employed. The loss of high-frequency absorption due to the facing is not serious since the remaining absorption is still adequate, and because machinery in general does not generate much noise in these higher frequency ranges.

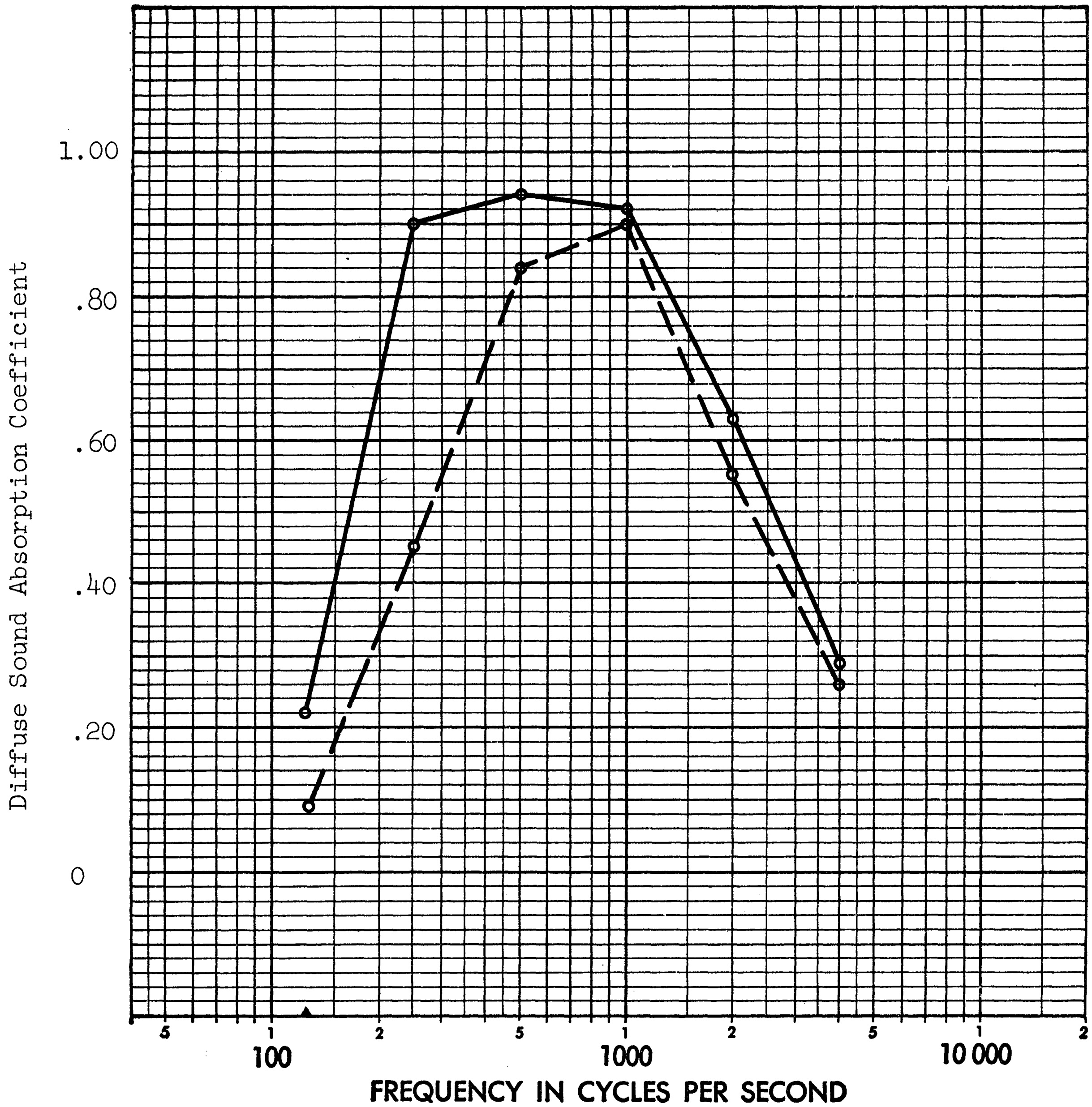
Returning to the experimental installation in the C-26, added physical protection for the composite treatments was afforded by covering their exposed surfaces with 18/14-mesh aluminum screening. These absorptive panels were attached to the C-26 housing by means of adhesive cement (Abesto Plastic Cement, Abesto Manufacturing Corporation), wire, and sheet-metal clips. An 18-gauge sheet-metal baffle was installed behind each louvered opening in the C-26 housing, but spaced out far enough to permit air flow. The interior surfaces of these sheet-metal baffles were lined with the same absorptive treatment described above.

All these absorption treatments were installed in such a way that they did not interfere with the accessibility of the machinery. Moreover, they were all installed inside the present C-26 housing except for the one baffle covering the louvering at the end of the engine compartment. It had to be installed externally to avoid permanent interior modifications, but it only projects 3-1/2 in. beyond the cart and does not interfere with opening the engine hood. (See Appendix C for details of absorptive panel size and placement.) Where the treatment is attached flat against panels of the C-26 housing, it provides considerable vibration damping capacity also.

It had been observed earlier that openings directly into the engine compartment existed around the engine control panel. During C-26 operation, sound seemed to beam directly at the operator's station from these openings. Consequently an absorptive panel and sheet-metal septum were installed inside the housing to block any direct sound paths in this area. No treatment of the exhaust opening for the engine cooling air in the bottom of the cart was attempted. This is a very difficult location to install any treatment, and furthermore it was felt that at this stage this opening could constitute only a second-order leak with respect to the far-field noise. High-frequency noise escaping by this route must undergo at least one reflection from the ground before reaching any measuring location.

<u>Final Free-Field Noise Survey</u>.—No individual evaluation of the effectiveness of the absorptive treatment described above was attempted. Instead a complete free-field survey was conducted using all the successful modifications simultaneously, that is, two Walker 639 mufflers were installed, the generator cooling-air blower was resiliently mounted, and the absorptive treatment was in place. The C-26's operating parameters closely approximated those used in the earlier free-field surveys (see Appendix C).

Figures 3.40, 3.41, and 3.42 present the results of this free-field survey in the usual form, i.e., the polar distribution of the computed overall noise, the average octave-band noise profile, and the directional deviations from the average profile,



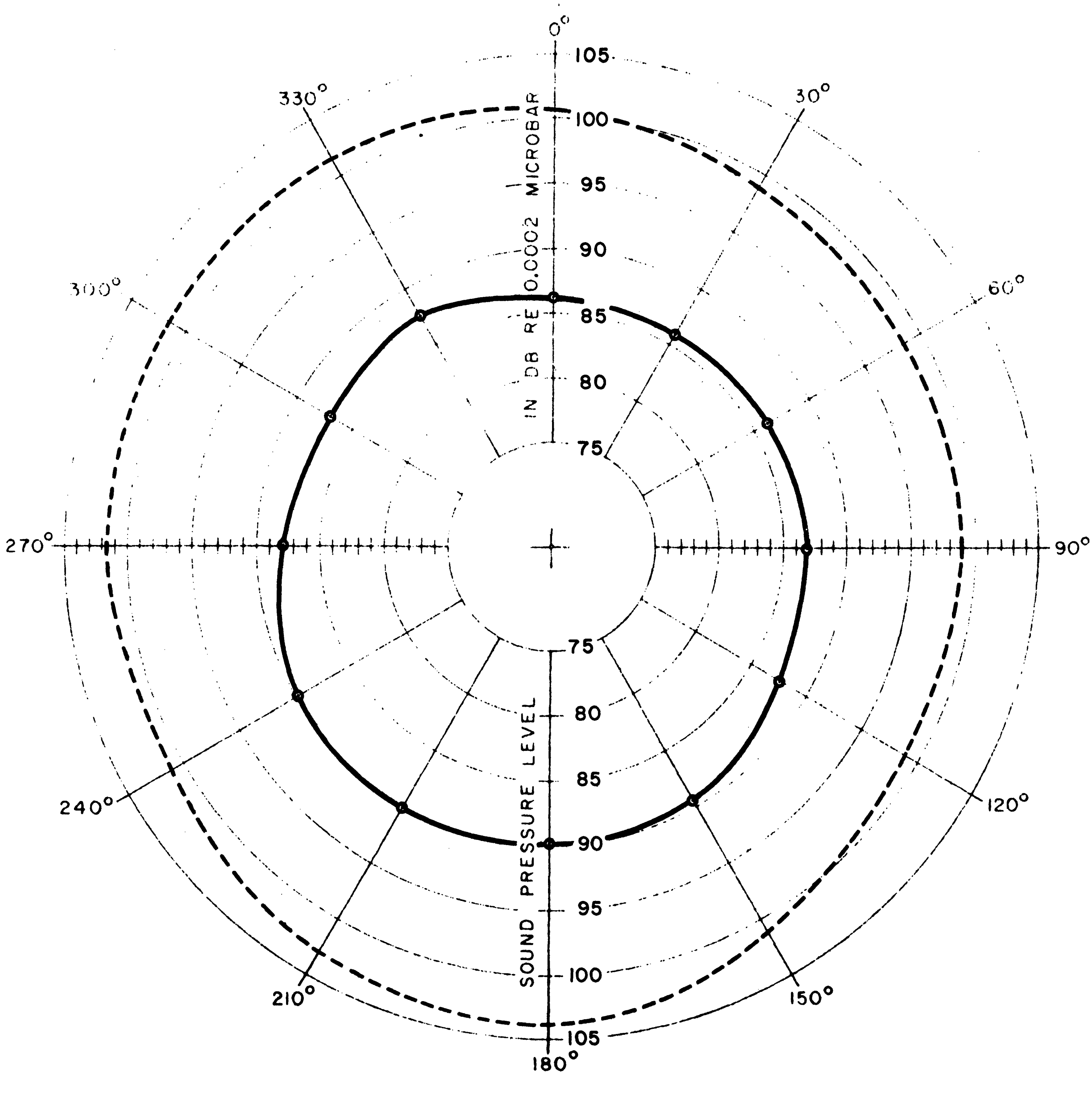

——————— 2-1/2" PF-334 Fiberglas 0.5 pound per cubic foot density with aluminum foil surface facing, on No. 4 mounting (directly against hard wall surface)

— — — — 1" PF-335 Fiberglas 0.75 pound per cubic foot density with aluminum foil surface facing, on No. 4 mounting

These curves are predicted, not experimental

Fig. 3.39. Synthesized diffuse sound absorption coefficients for composite treatments.

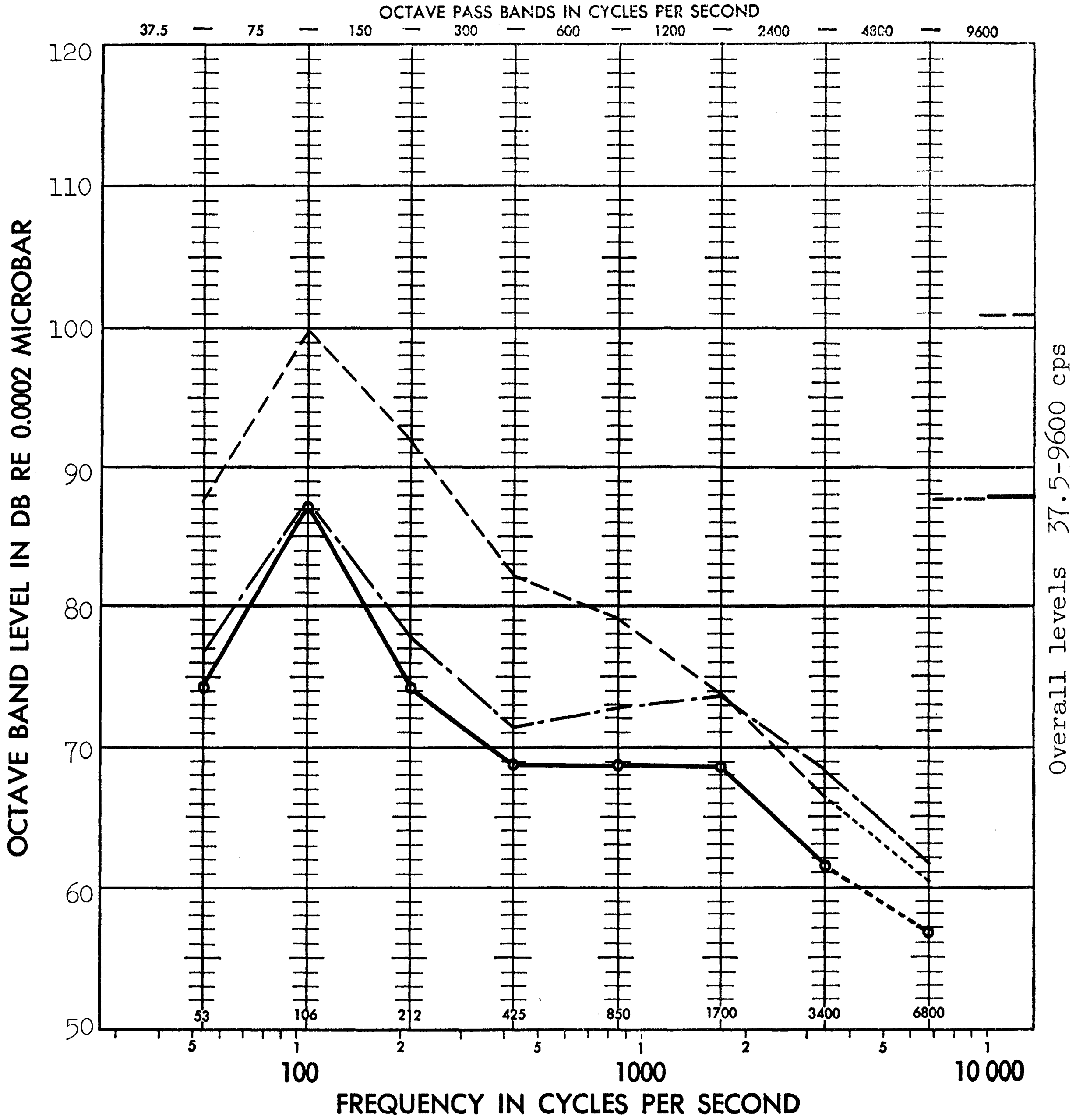
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type C-26 gasoline generator set, equipped with two Walker 639 mufflers, generator cooling blower resiliently mounted and absorptive lining installed in cart
2000-2030 rpm, 930-930 amp, 29.5-30.2 volts dc
Tested 27 June 1956
-----As-received condition, see Fig. 3.22

**Fig. 3.40.** Polar distribution of overall noise; C-26 treated.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave-band sound pressure levels




Type C-26 gasoline generator set, with two Walker 639 mufflers installed, blower resiliently mounted and absorptive lining in cart
2000-2030 rpm, 930-930 amp, 29.5-30.2 volts dc
Tested 27 June 1956
Dotted line indicates that actual level probably lies lower than plotted value

Fig. 3.41. Average octave-band noise profile; C-26, treated.

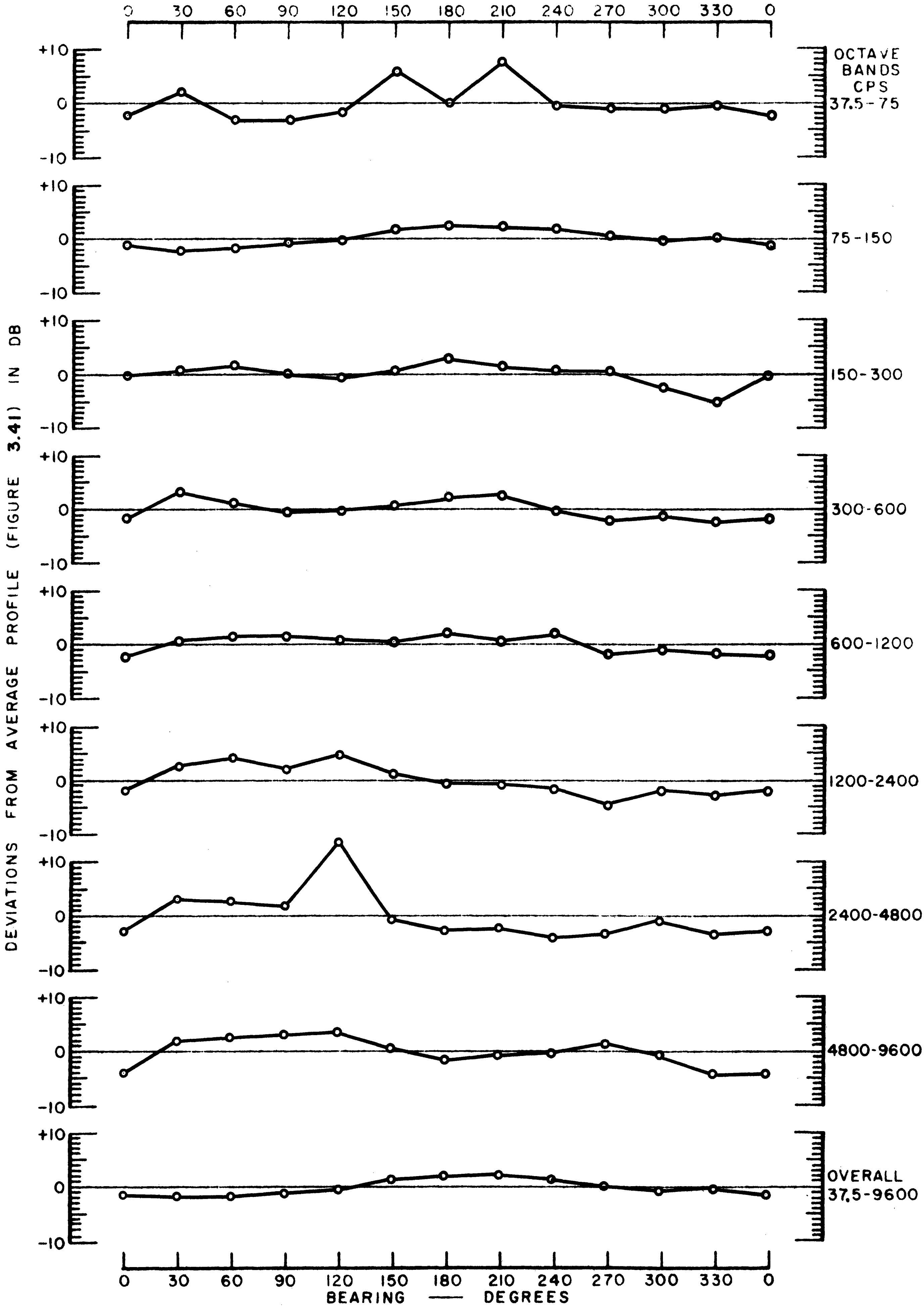



Fig. 3.42. Directional deviations from average profile; C-26, treated.

respectively. The corresponding curves for the "as-received" survey and the survey with two Walker mufflers have been included for comparison. Figure 3.40 shows that in general the reduction in overall noise has been achieved equally in all directions. The actual amount of the final reduction ranges from 11.5 db at 120-degrees bearing to 14.5 db at zero-degrees bearing, and averages about 13.1 db. Thus on an overall basis the radiated noise has been reduced to about 22% of the amount originally present.

Figure 3.41, showing the average noise profiles for the several treatment conditions, illustrates how the character of the acoustic spectrum has been altered. The installation of the mufflers greatly reduced the lower frequency noise, but had only a slight effect on the high-frequency noise. The resilient mounting of the generator cooling-air blower and the addition of the absorptive treatment have brought further reductions, the absorptive treatment appearing particularly successful above about 600 cps. The cumulative effect of the several treatments has been a suppression of the radiated far-field noise in all portions of the spectrum. Figure 3.42 shows that, except for the 37.5-75 cps octave band and the 2400-4800 cps octave band, no particularly outstanding directional characteristics were observed.

Interpretation of Results Achieved.—As in the case of the A-1 Generator Set, the original symmetry of the C-26's overall sound-pressure levels arose from the domination of these noise levels by comparatively low-frequency exhaust tones radiated from vertical exhaust stacks. In the final treated configuration reported above, the overall levels, considerably reduced, are still dominated by the same low-frequency exhaust tones radiated from the vertical muffler outlets; hence symmetry is still to be expected. This is indeed the case as shown in Fig. 3.40; however, slightly more noise is observed from the 180-degree bearing than from the zero-degree bearing. This may be the result of acoustic shielding by the fuel tank and greater radiation from the engine compartment; however, definite conclusions about the origins of the residual directionality cannot be drawn from these data alone.

Figure 3.41 indicated that the installation of the Walker mufflers dropped the low-frequency noise so that the average noise profile was appreciably flatter even though the exhaust noise still dominated. Further treatment produced some additional improvement in the lower frequency ranges but had its major effect at higher frequencies. Thus the final profile, although much lower at all frequencies, retains a frequency distribution resembling the "as-received" condition. The implication is that the use of very effective mufflers along with absorptive lining of the housing and resilient mounting of the cooling-air blower could accomplish almost as much additional noise reduction as the installation of the Walker mufflers did at the first trial. If this were actually carried out, then the overall noise on the basis of sound pressure would have been reduced a whole order of magnitude to less than 10% of that originally radiated.

The directionality of the residual noise as reported in Fig. 3.42 is probably due to the increasing importance of multiple secondary noise "sources" since they are now less completely submerged by the exhaust noise. However, these last free-field measurements alone are not sufficient to permit detailed identification. No particularly satisfying hypothesis about the origin of the 37.5-75 cps octave-band directionality comes to mind. However, the peak in the 2400-4800 cps octave-band directionality curve at about 120-degrees bearing is probably caused by one or more tones radiating from the cooling-air blower's intake. Under the final test conditions, high-frequency tones attributed to this source could be detected easily by ear in this general direction.

The use of an absorptive-type intake silencer, similar in acoustic design to the silencer utilized on the A-1 (see Fig. 3.15) should effectively control this noise.

The above studies have again emphasized that installing highly effective exhaust mufflers is an essential first step in reducing the noise radiated by internal combustion engines. The vibratory excitation of large potentially efficient radiating surfaces by small machinery items has been demonstrated, and the resilient isolation of the generator cooling-air blower illustrates how isolation can be used to control such difficulties. The closing of various acoustical leaks in a machinery enclosure and the necessary addition of absorption within the enclosure to render it acoustically effective has been demonstrated also.

The acoustical studies on the C-26 Generator Set reported here have been limited in several respects. The first cycle of experimental noise reduction identified the exhaust noise as the only predominant source and treated this source as effectively as possible with readily available mufflers. The second cycle of experimental noise reduction established that in spite of a large reduction in noise level, the same source still predominated the C-26 noise spectrum. Normally, at this stage, one would concentrate on improving the muffling until exhaust noise was no longer the limiting factor. This approach was not taken since it was felt that it could easily use up the remaining project time and funds and still only provide background information on a very narrow and specialized phase of noise-reduction research. Rather, the remaining effort was concentrated on investigating other noise sources even though they were still partially obscured by the residual exhaust noise. The additional noise reduction thus achieved is only slightly reflected upon examination of the overall noise. The reductions are apparent in the average noise profiles, but even here they are not as dramatically evident as would have been the case if the exhaust noise had been eliminated.

The final phases of the research on the C-26, which although it included resilient mounting of the cooling-air blower, acoustically blocking louvered openings, and incorporation of absorption within the cart, represent only part of the investigation necessary to develop the full enclosure effectiveness of the C-26's housing. In general, the control of a noise source by an acoustical enclosure requires the simultaneous application of resilient isolation of the entire enclosure, vibration damping of the panels, absorption within the housing, and the securing of a high transmission loss for airborne sound. Further major investigations in this direction would have required modifications much too involved to be successfully completed within the scope of this project. For example, the tests with the generator cooling-air blower demonstrated that vibration reaching the exterior panels of the cart readily found its way into far-field airborne noise. Also the necessity in the original C-26 design for resiliently mounting much of the generator control circuitry testifies to the level of cart vibration. Even though the engine and generators are mounted on rubber within the cart, it is likely that, because of the mechanical impedences involved, considerable forced vibration of the cart still occurs resulting in radiated noise. A complete study of enclosure effectiveness would have involved such experiments as operating the machinery completely removed from the cart, constructing special new housing more completely isolated from the machinery vibration, etc. All these experimental studies are obviously difficult, expensive, and time-consuming. Therefore the present studies were limited to those experiments and modifications which were most readily accomplished in the laboratory and which might most easily be adapted to existing service equipment.

Recommendations.—Obviously, the first recommendation is to install a pair of very effective mufflers. It appears that the exhaust noise is so predominant that almost any muffler acceptable from the standpoint of backpressure, etc., will bring some noise relief. However, to realize the greatest reduction, an engineering development will need to be undertaken to provide acceptable mufflers capable of producing sufficient acoustic attenuation. It should be emphasized that the development required is really of an engineering nature because the suitability of mufflers to control the C-26's noise has been established and the effectiveness of the various acoustic configurations or components used in muffler design has been well demonstrated. It remains to assemble these into a suitable package for C-26 application.

Presupposing reasonably effective muffling, the generator cooling-air blower requires attention. This would be an appropriate place to apply some noise-reduction-at-the-source technique by supplying a more carefully balanced blower. Even with better balancing, resilient mounting will probably still be required. Center-of-gravity mounting using soft equal-stiffness mounts and attending to all the other features of good resilient-mounting practice will help to yield optimum results. The use of absorptive linings in the machinery compartments and absorptive baffling of the various louvered openings is also recommended if mufflers at least as effective acoustically as the Walker 639's are used. If poorer mufflers are employed, absorptive linings will probably not be worthwhile.

Presuming that very effective mufflers are employed and that the several other treatments suggested above have been used, the question arises of how even greater quieting might be obtained. The answer to this is not at all clear since, as described above, the present research had already pushed considerably beyond the scope to be expected with the Walker mufflers installed.

## TYPE MA-1 GAS-TURBINE-DRIVEN AIR COMPRESSOR

The Type MA-1 Gas-Turbine-Driven Air Compressor consists of a small, compact gas turbine (manufactured by the Continental Aviation and Engineering Corporation) mounted in a three-wheeled trailer. The centrifugal compressor has an air-handling capacity about 50% greater than can be utilized for combustion in the gas turbine proper. This excess air is bled from the turbine housing and constitutes the useful output of the unit. When the unit is operating at the rated speed of 35,000 rpm, the useful air delivery is in excess of 125 lb/min at 50 psi absolute. The entire unit is housed in an aluminum trailer about 6 feet long by 4 feet high by 4 feet wide. The trailer itself consists of a structural chassis with the aluminum body directly attached. The gas turbine mounts to the trailer chassis at three points by means of two rubber resilient mounts and one metal-to-metal mount.

The end of the trailer adjacent to the towing hitch contains the exhaust end of the turbine, while the instrument and control panel and the compressor air intake are located at the opposite end. Fuel is contained in two tanks extending the full length of the trailer on both sides of the turbine. The various accessories are clustered around the turbine and the output air is taken from the side of the trailer by means of a flexible duct about 30 feet long by 3-1/2 inches inside diameter. The operation of the MA-1 is largely automatic, all critical control and metering functions being taken over by special devices. Figure 3.43 shows a schematic view of the machinery arrangement within the trailer, while Fig. 3.44 is a schematic plan view identifying

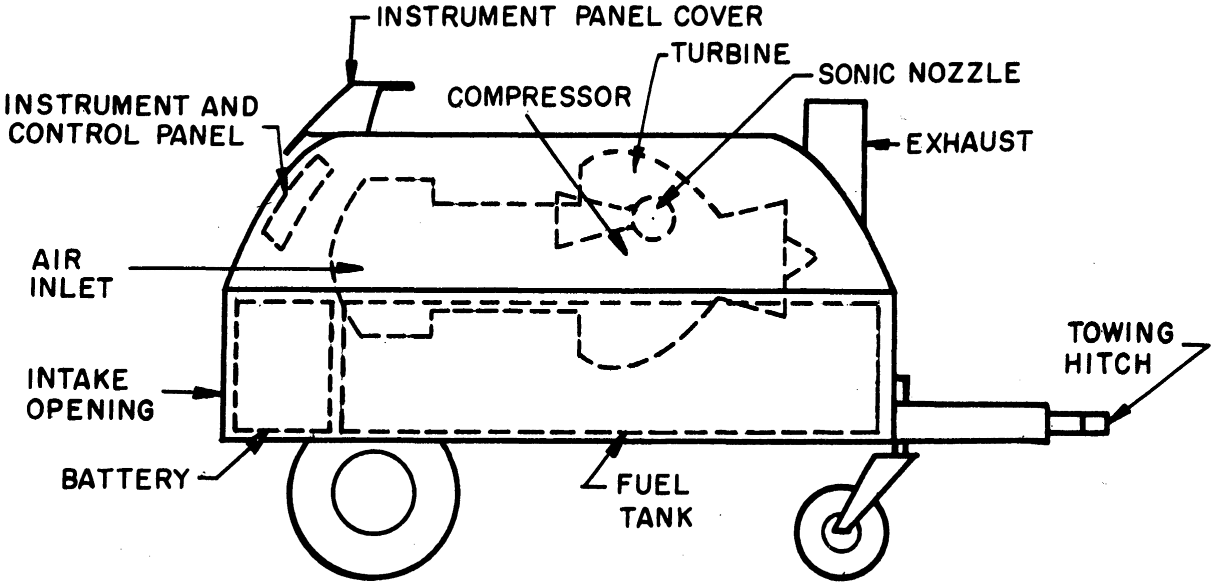



Fig. 3.43. Schematic elevation of MA-1 interior arrangement.

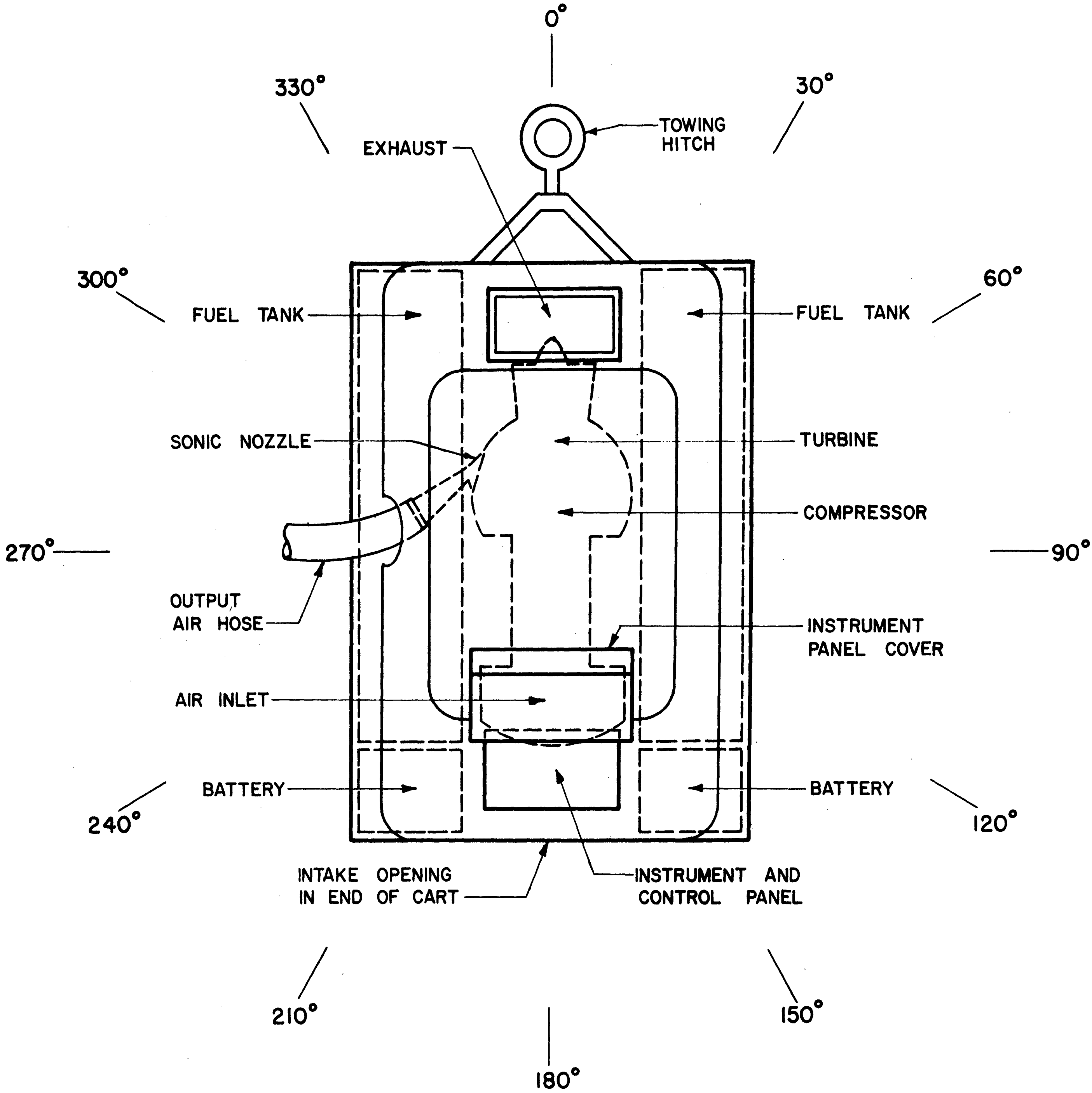



Fig. 3.44. Schematic plan view of MA-1 gas turbine driven air compressor.

various important features of the MA-1 and defining the azimuthal directions used for free-field measurements.

Compared with the reciprocating engine powered units examined, the MA-1 Turbine-Driven Air Compressor seems much more completely enclosed by its housing since there are no louvered ventilating panels and the only major openings are those utilized for intake air and exhaust. Under normal operation, the unit is controlled from a small panel at the 180-degree end of the trailer. This panel fits into the housing fairly closely, although some intake air can flow in around this panel. The main intake air comes through an opening located below the instrument panel. The turbine exhaust is deflected from horizontal to vertical flow immediately after leaving the turbine's tail cone by a short vertical exhaust stack located at the towing-hitch or zero-degree end of the trailer. A hinged cover over the exhaust stack provides a slight amount of horizontal deflection again in the zero-degree direction. The output air is bled from the turbine housing through a sonic nozzle and is delivered to the flexible air delivery duct, or else by-passed out through the exhaust stack if the delivery duct is closed off.

Air-Load Silencer.—Since the useful output of the MA-1 consists of a fairly large volume of compressed air, operation of this unit at full load condition for acoustical measurement purposes presents some difficulties not experienced with the other units tested. Their output was in the form of electrical energy and could be conveniently and silently dissipated in resistor banks. In the case of the MA-1, some equivalent means must be devised to dissipate its output silently enough to avoid interference with the necessary acoustic measurements. Since the sonic nozzle of the MA-1 is self-regulating, no particular flow control of the output air needed to be accomplished to assure proper turbine operation.

Preliminary observations, made with the flexible delivery duct extended, and exhausting unobstructed into the open air, confirmed that the escaping compressed air itself generated considerable noise. With the air duct at maximum extension and exhausting in the opposite direction from the microphone, the detection of the noise of the escaping air was marginal in the presence of the MA-1's direct noise at the free-field microphone position. At other positions, however, the added noise contribution from the freely escaping air could be distinguished easily by ear.

With this situation, conceivably the "as-received" free-field noise survey could be conducted in a reasonably successful manner by always orienting the air-delivery duct away from the measuring microphone. However, with only marginal freedom from interference in the "as-received" condition, it would be practically impossible to measure and evaluate any actual noise reduction or possible noise reduction achieved subsequently. Therefore, it was deemed imperative to silence the noise generated by the jet of air issuing from the delivery tube as a preliminary step to any actual survey measurements.

Figure 3.45 shows the features of a 4 ft x 4 ft x 2.5 ft absorptive-type air-load silencer constructed of 3/4-in. plywood and lined with 2-in.-thick PF-335 Fiberglas retained by hardware cloth. This type and thickness of fiberglass was used to provide a large amount of high-frequency absorption with reasonably large amounts of absorption still available at low frequencies. The air flow had to reverse twice before escaping from the silencer, thus providing an effective geometry for sound absorption,and the final exit aperture of the silencer was about 15 x 28 inches, thus allowing greatly reduced flow velocity.

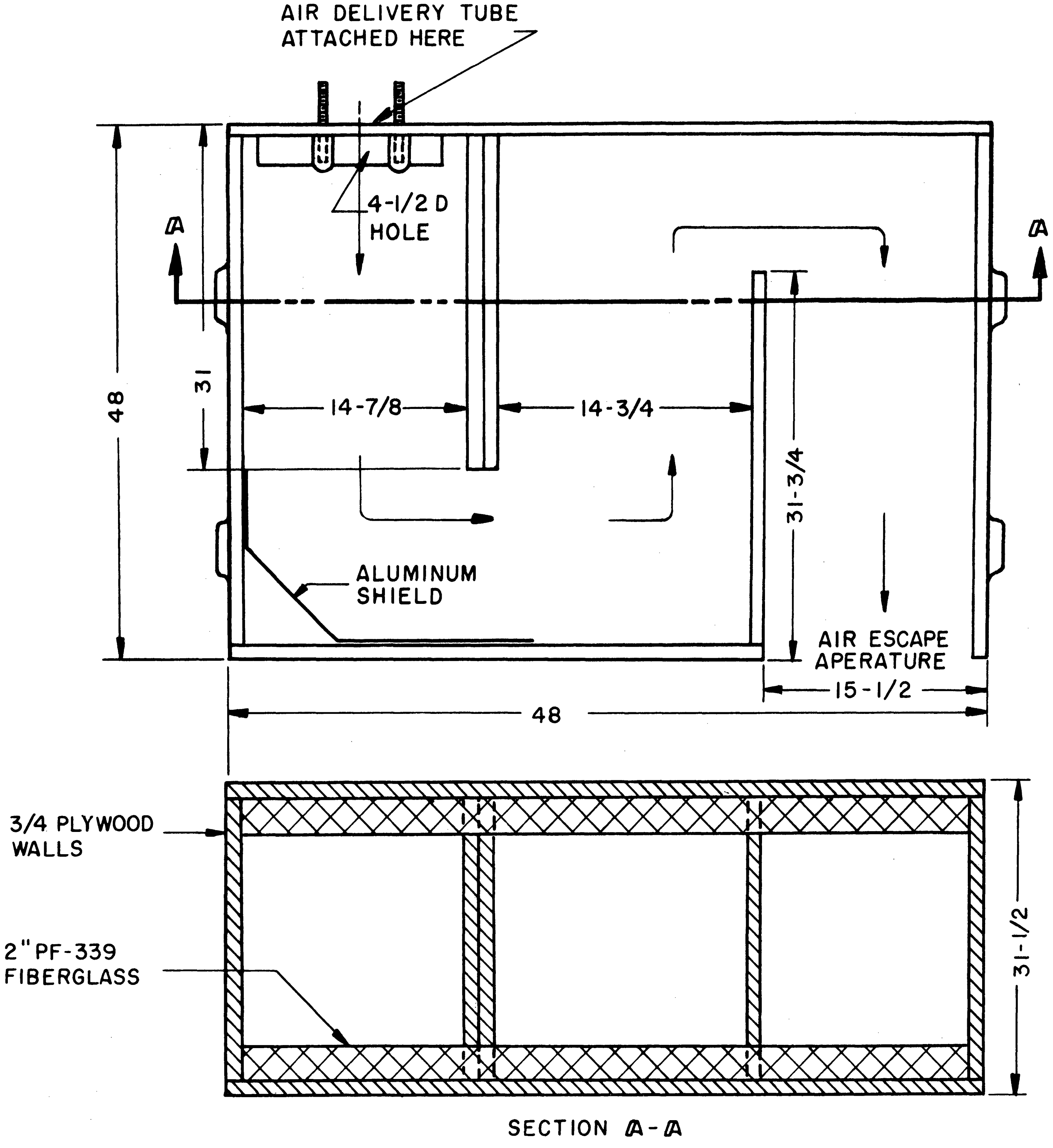



Fig. 3.45. Schematic cross section of air-load silencer for MA-1 gas turbine.

The air-load silencer described above was completed at a time when the octave-band filter set had been returned to the factory for repairs. Consequently, preliminary evaluation of the silencer's effectiveness was carried out on a broad-band basis with the MA-1 operated at its rated speed of 35,000 rpm. A substitution method was employed in which the air was first allowed free escape from the delivery tube, and then, without altering the test geometry, exhausted through the silencer. This was done both with and without the fiberglass absorptive lining in place. Figure 3.46 shows the test arrangement and the measuring locations situated at the free-field test site. Figure 3.47 presents the acoustic results in bar-graph form, but since the reference levels differ, absolute levels cannot be compared between the two parts of this figure.

The results of these wide-band evaluations are entirely consistent with what one would expect. Microphone position 1 represents the normal free-field survey measuring location and, as Fig. 3.47 shows, only marginal effects are observed. This verifies the earlier remarks about the "as-received" relative importance of the MA-1 noise, and that noise generated by free air escape from the delivery tube. Microphone position 2 corresponds roughly to the operator's position, and one would expect the direct noise from the MA-1 to dominate here. At microphone positions 3 and 4, considerable reduction, presumably due to shielding, was observed for the unlined silencer and further significant reductions were obtained when the absorptive lining was installed. On the basis of these broad-band measurements, it appeared that the silencer provided at least a 12-db reduction of interference from the noise generated by the free escape of the load air. If this is true, then on the basis of the marginal interference evidenced by the free air escape, this silencer should enable noise reductions of the order of 12 db on the MA-1 itself to be evaluated before serious interference from the delivered air noise becomes significant again.

<u>As-Received Free-Field Noise Survey</u>.—Following the completion of these initial silencer studies and the arrival of the repaired octave-band filter set, an "as-received" free-field survey of the MA-1 noise was undertaken. The air-load silencer was installed and located as far as possible opposite from the microphone with the silencer exhaust opening pointed away from the measuring area. The MA-1 gas turbine was operated at its full rated speed of 35,000 rpm, and the full amount of air which the unit is capable of pumping at this speed was delivered to the silencer. Measurements were taken at the usual 30-degree intervals with the microphone located a distance of 40 ft away and 5 ft 5 in. above the ground.

Figures 3.48, 3.49, and 3.50 present the overall directionality, the average octave-band noise profile, and the directional deviations from the average profile, respectively. The overall directionality shows a tendency toward a four-lobed pattern and somewhat more variation from one bearing to the next than evidenced by similar curves for the A-1 and C-26 reciprocating-engined machines studied earlier. Figure 3.49 indicates how extremely different in shape the octave-band noise profile is compared to the profile generated by an unmuffled reciprocating engine. The preponderance of high-frequency noise generated by the MA-1 gas turbine is particularly striking. Figure 3.50 verifies that considerable directionality is exhibited in all octaves, and that there is a distinct tendency for all octaves to possess roughly the same directionality pattern.

<u>Analysis of Initial Noise-Reduction Problem</u>.—It is difficult at this point in the studies to assemble a clear and satisfactory picture of the noise problem posed by this gas turbine. Octave-band directionality patterns and octave-band profiles of

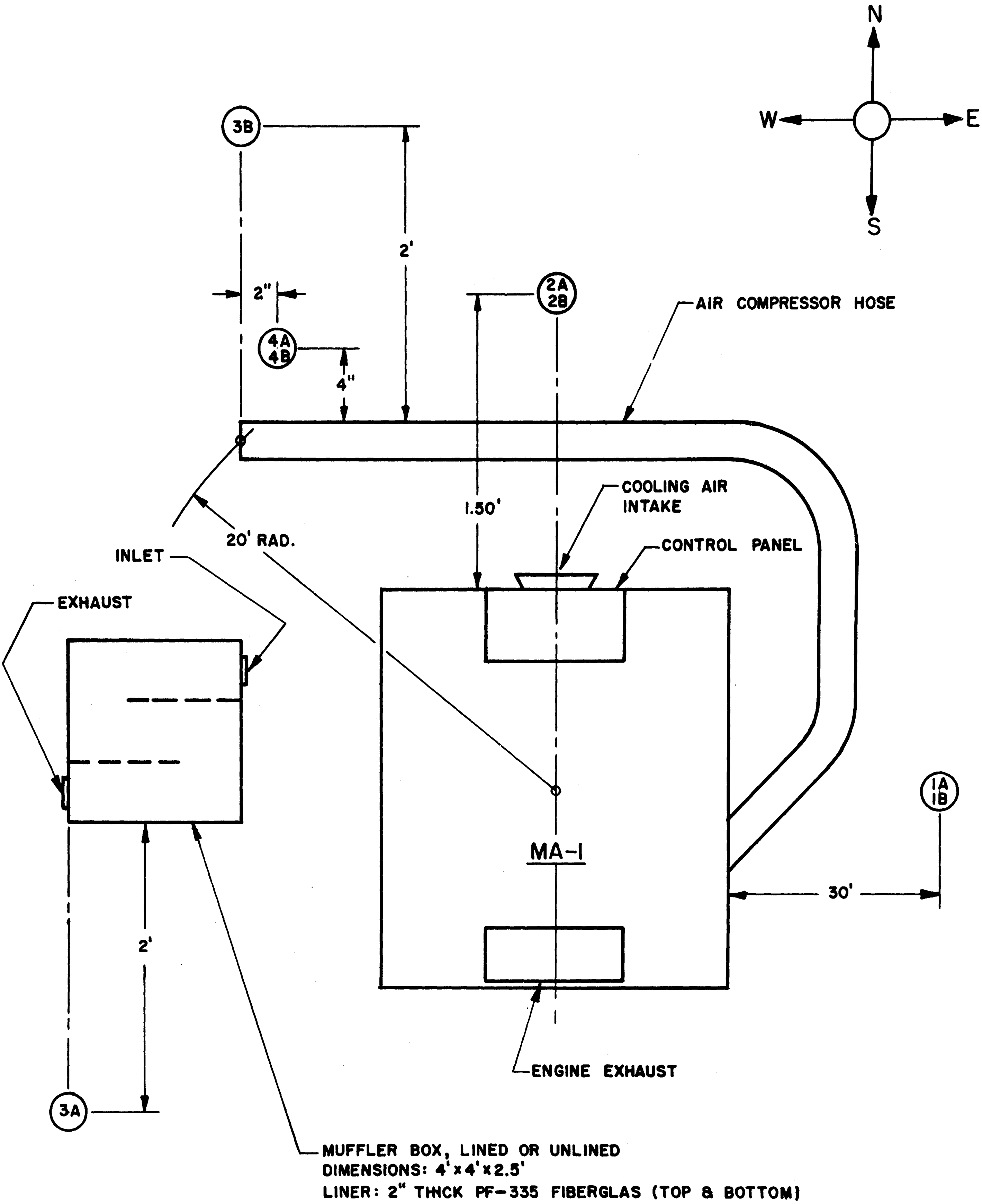
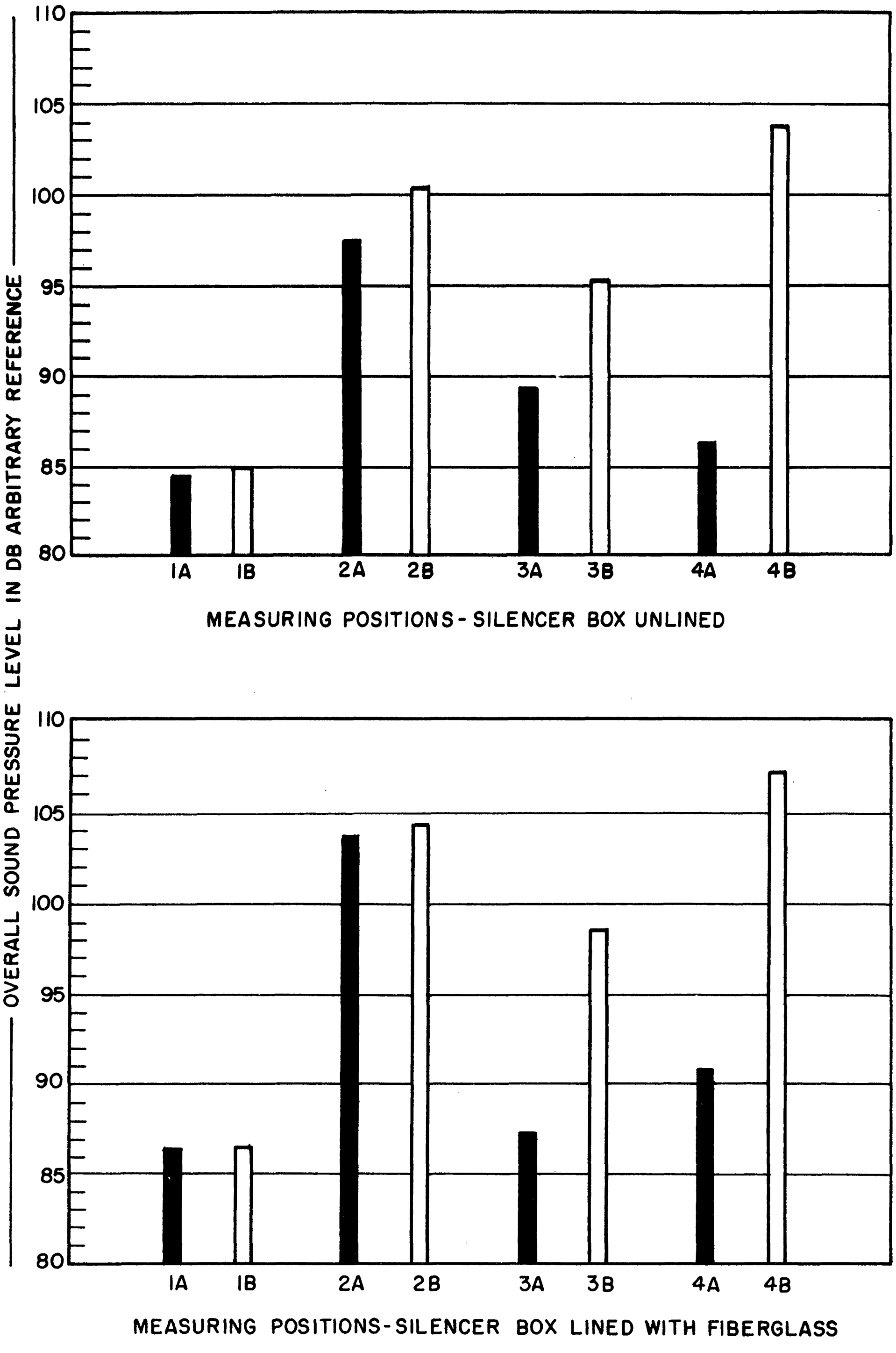



Fig. 3.46. Test plan for evaluating air-load silencer.

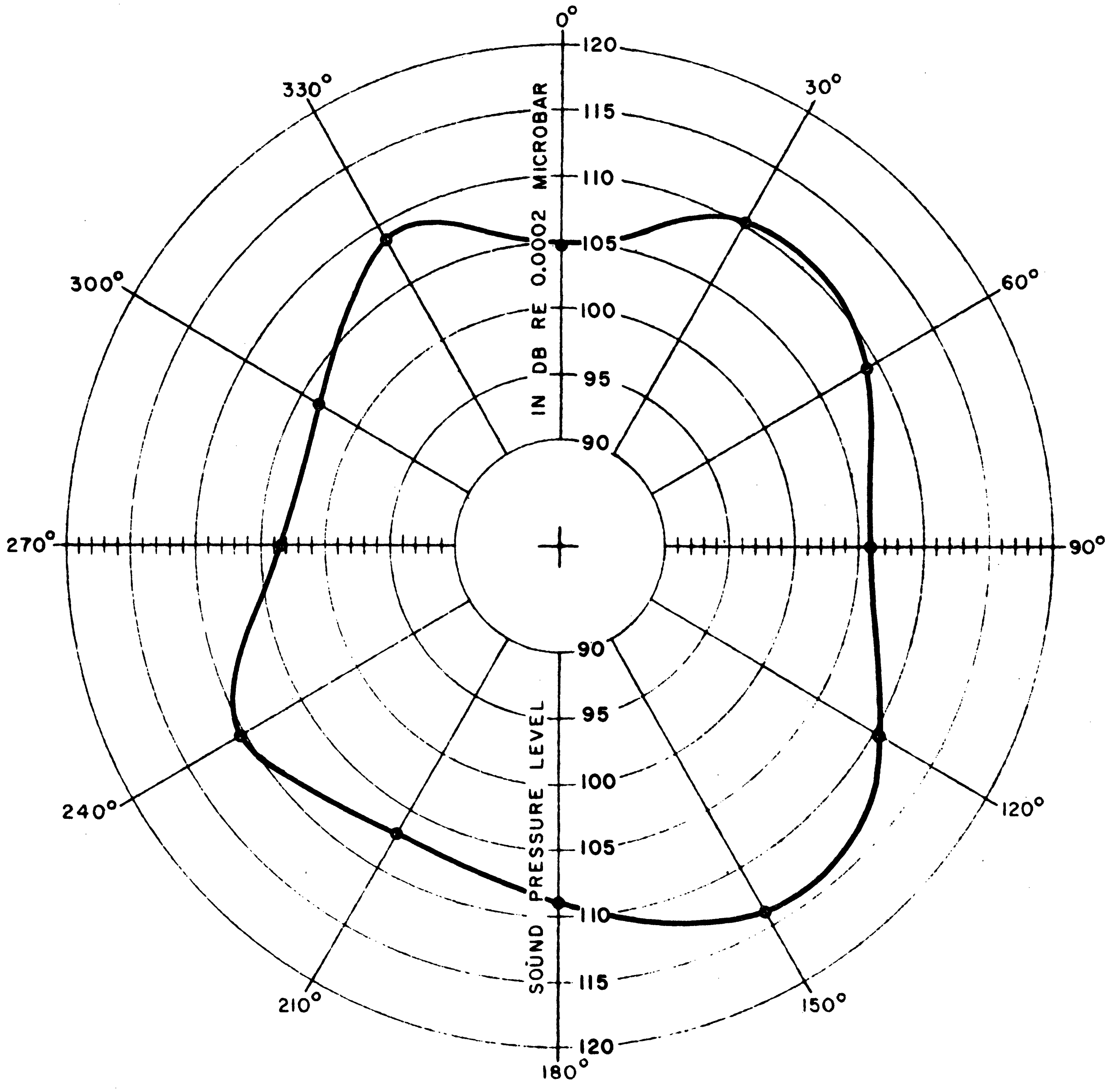
Free-field conditions
Microphone locations, see Fig. 3.46
Nonstandard measuring instrumentation, broad band measurements covering the approximate frequency range from 37.5-9600 cps




Type MA-1 Gas Turbine Driven Compressor with output air directed through air-load silencer or allowed free escape, see Fig. 3.45
34,800-35,200 rpm, 1010-1090°F exhaust temperature
Silencer attached (A)
Silencer removed (B)
Tested 14 September 1955 and 21 September 1955

Fig. 3.47. Air-load silencer evaluation by means of broad band measurements.

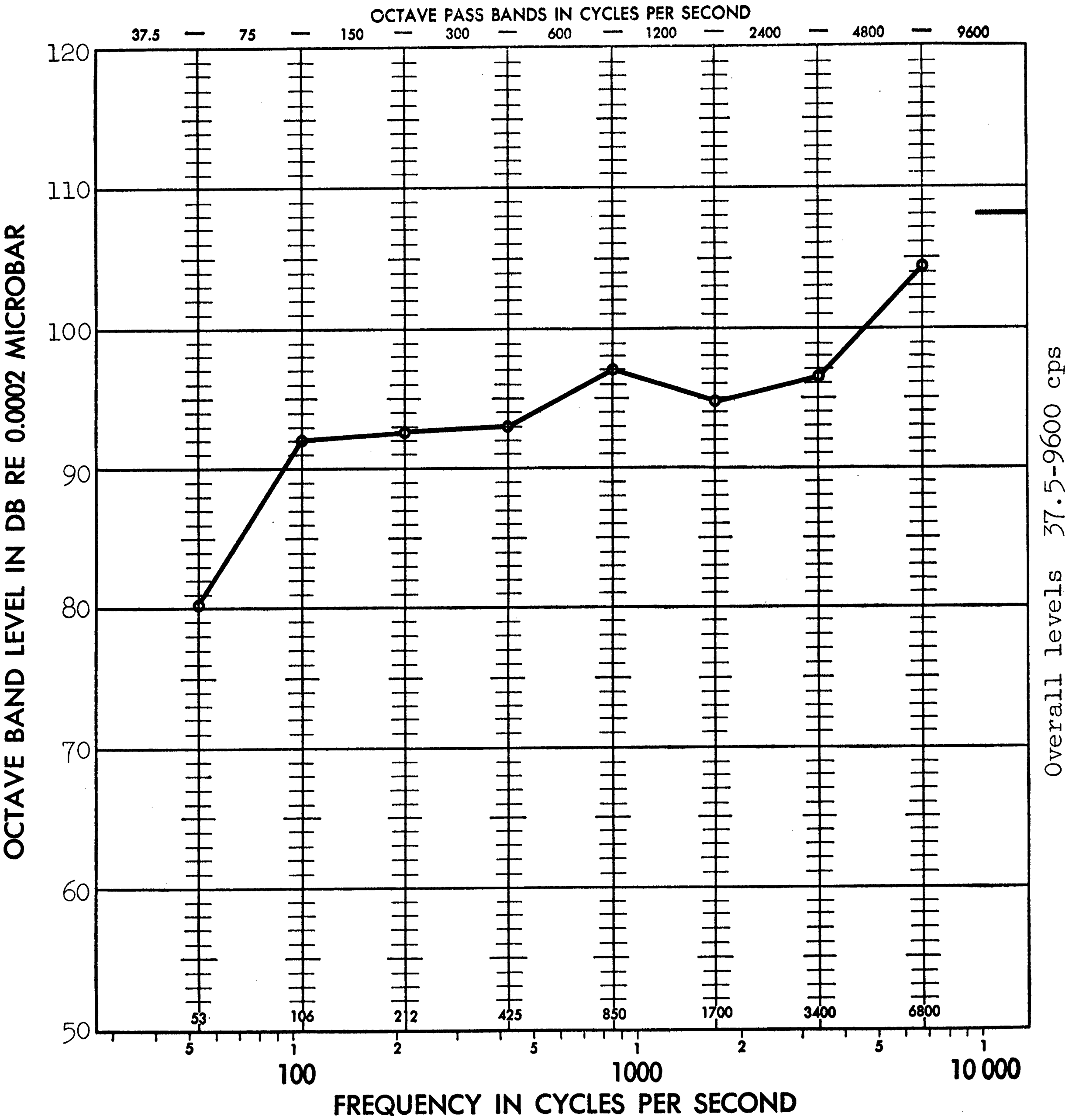
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type MA-1 gas turbine driven air compressor, as-received condition with output air directed through the air-load silencer, see Fig. 3.45
34,800-35,100 rpm, 1040-1080°F exhaust temperature
Tested 26 October 1955

Fig. 3.48. Polar distribution of overall noise; MA-1 gas turbine, as-received, with air-load silencer attached.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave-band sound pressure levels




Type MA-1 gas turbine driven air compressor, as-received condition with output air directed through the air-load silencer, see Fig. 3.45
34,800-35,100 rpm, 1040-1080°F exhaust temperature
Tested 26 October 1955

Fig. 3.49. Average octave-band noise profile; MA-1 gas turbine; as-received, with air-load silencer attached.

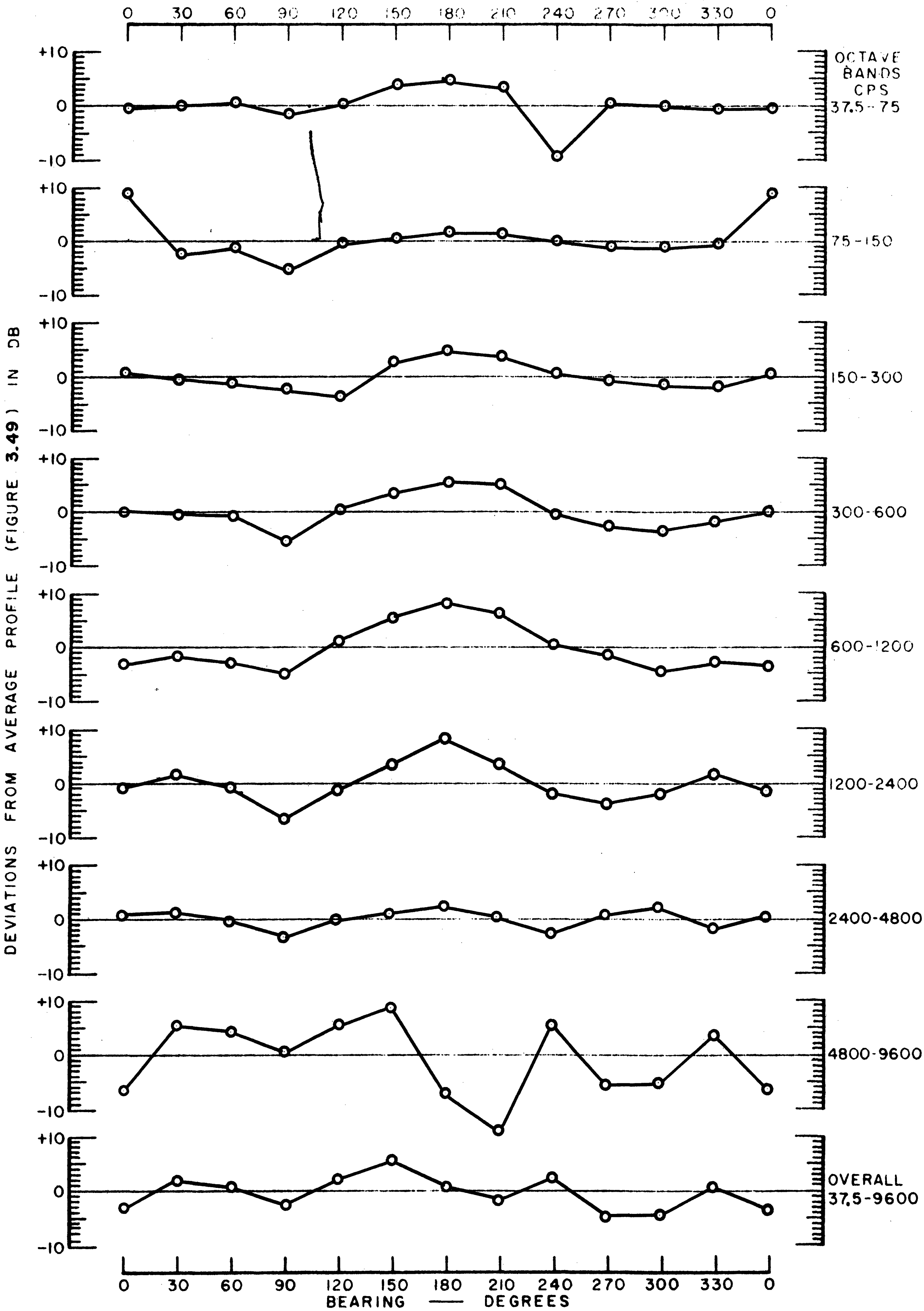



Fig. 3.50. Directional deviations from average profile; MA-1 gas turbine, as-received, with air-load silencer attached.

noise sources are not sufficient for this purpose. It is necessary to know the detailed characteristics of the noise source to interpret octave-band data intelligently. In the case of the reciprocating-engined machinery, aural evaluations coupled with considerable experience acquired over a period of years provided the additional clues necessary to interpret the octave-band data collected in the "as-received" surveys. In the present case, detailed background information about jet noise was lacking, and the evidence from the literature proved to be somewhat confusing. Between the time when these studies were conducted and the preparation of this report, the general description of jet noise in the literature has been clarified to a large extent, and additional experience has been gained in this laboratory on jet-type noise sources. Consequently, by hindsight, some of the then confusing details encountered in the collection of data are much better understood and would now result in a somewhat altered approach. However, dealing with just the data at hand, the most striking feature is the change in shape of the average profile from those typical of reciprocating engines. The gas turbine appears to generate comparatively less low-frequency noise and much more high-frequency noise. Listening tests suggest that, aside from a discrete tone or two probably generated by the compressor, the rest of the noise spectrum is largely white in character. (This statement, of course, was subject to verification by means of precise discrete-frequency analyses.)

The observed directionality characteristics seem to be of negligible help at this time in clarifying the overall noise picture. One fact, not immediately evident from the data as presented in Figs. 3.48, 3.49, and 3.50, is that the highest octave-band level at each bearing which predominates the computed overall level for that particular bearing varies from one octave to another at different bearings. For example, the 75-150 cps octave band predominates at zero-degrees bearing, the 600-1200 cps octave band predominates at 180- and 210-degrees bearings, and the 4800-9600 cps octave band predominates in most other directions. This is cited only to emphasize that the average octave-band profile shown in Fig. 3.49 is the average, not only of a range of numerically different levels at the several bearings, but also of profiles of characteristically different shapes. Further detailed interpretation of why this situation occurs is not possible from these limited data.

At this point, one begins to suspect that the noise-reduction problem presented by the MA-1 Gas Turbine is fundamentally different from those of the A-1 and C-26 machines, much more so than is apparent from the difference in spectral characteristics. The large amount of comparatively high-frequency "white" noise suggests the characteristic noise generated by the turbulent exhaust stream of jet engines. If this is true, then much of the noise may originate in the exhaust stream external to the MA-1 housing, making palliative, containment-type, noise-reduction treatments, on which all of the experimental work proposed under this contract was based, of very limited use here. It may be necessary either to find a way to accomplish noise reduction at the source or to bring the sources within the boundaries of an "enclosure" by means of duct work external to the present housing, where they can be treated by palliative techniques.

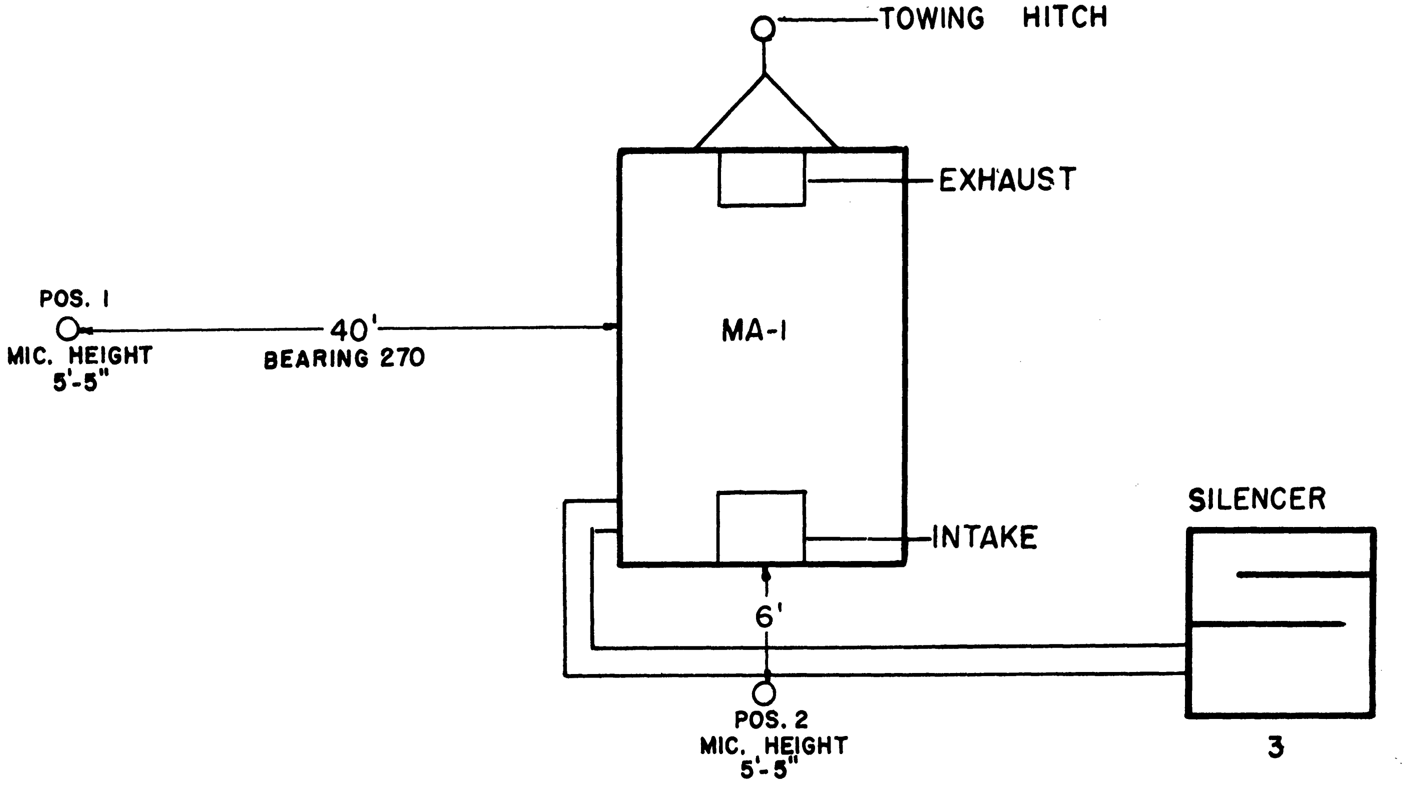
The discrete tone (or tones detected aurally and presumed to be due to the centrifugal compressor) does not seem to present any fundamental difficulty. Since the frequencies are high, it should be possible, in principle at least, to provide a very effective absorptive-type intake silencer.

Another factor, not demonstrated explicitly by this first survey alone, is the large variability of the noise generated by this gas turbine. Certain preliminary measurements and the behavior during this first full survey gave indications that considerable variability might be encountered, although the interpretation at that time was not at all clear. Later tests have firmly established the existence of the suspected variability but its cause and interconnection with various operating parameters is not known.

In conducting the various tests on the MA-1 Gas Turbine, the unit was always brought up to rated speed, and the tail-pipe temperature was monitored to be certain that safe limits were not exceeded. If the temperature was satisfactory and the rated speed was reached, then measurements were taken after a short period of time was allowed for the unit to stabilize. As in all such acoustic tests, the experimenters maintain a close aural alert for any telltale changes or fluctuations in the characteristic sound. (Aural monitoring is an exceedingly sensitive qualitative test in the case of reciprocating engines and many other types of machinery which produce sounds of predominantly pitch character associated with operating speed.) No changes in noise deemed significant were noticed. However, now that more is known about the noise characteristics of gas turbines, it is realized that aural monitoring is probably not very sensitive for such sources. As long as the rpm remains the same, the few discrete tones remain at the same frequency so the ear would detect nothing unless these tones fluctuated wildly in amplitude. The white noise is very difficult to judge since for this type of noise the ear is not very sensitive either to displacement of the profile along the frequency or amplitude axes, unless such displacement takes place abruptly and with considerable magnitude. Slow gradual changes in the white noise probably would go completely unnoticed. Similarly, when a time factor enters, such as judging whether the same sound occurs after stopping and restarting the turbine, the ear does not permit a very sensitive comparison.

In any event, since the data obtained in the initial free-field survey were not considered completely satisfactory, a number of additional tests were undertaken. First, because the effectiveness of the air-load silencer had been based only on preliminary overall measurements, it was decided to reverify its behavior on an octave-band basis. Figure 3.51 shows the geometry of the experiment which has been altered slightly from the previous tests. Again a substitution procedure was used in which the load air was first permitted to escape freely and then was directed through the silencer. Thus only a few minutes elapsed between comparable sets of octave-band measurements, although it was necessary to slow down the gas turbine to about 28,000 rpm. This was done so that the load air would be cut off by the motorized valve, facilitating the safe handling of the nozzle. Microphone position 1 corresponds to the normal free-field measuring location at 270-degrees bearing. The results with and without the silencer, presented in Fig. 3.52, show that the effect of the air-load silencer is negligible. The largest divergences occurring in the 300-600 cps and 4800-9600 cps octave bands are considered within the range of variation which seems to be normal for this machine. The computed overalls (106.9 db withoutout silencer and 107.2 db with silencer) corroborate the data obtained for the overall levels measured in the original evaluation.

Since the above measurement corresponded to the 270-degrees free-field test geometry, it is legitimate to compare this profile to the corresponding profile from the original free-field survey since both profiles were obtained with the silencer attached. These two profiles, presented in Fig. 3.53, are very discouraging regarding reproducibility. In the 300-600 cps octave band, a divergence of 13.2 db is noted with lesser but still seriously large divergences of 5.5 and 4.0 db in the 600-1200 and 1200-2400 cps octave bands, respectively. However, the full import of these di-



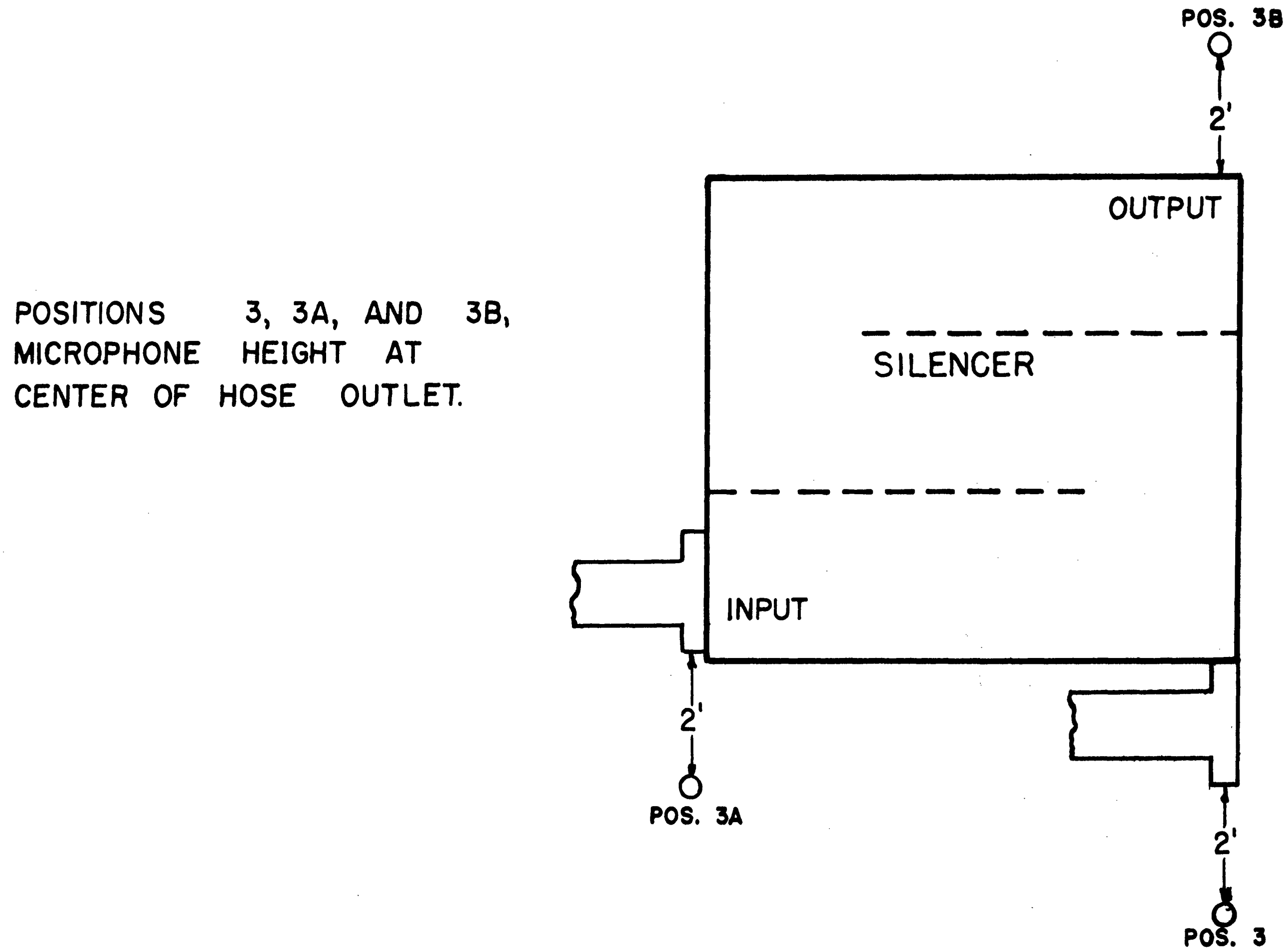
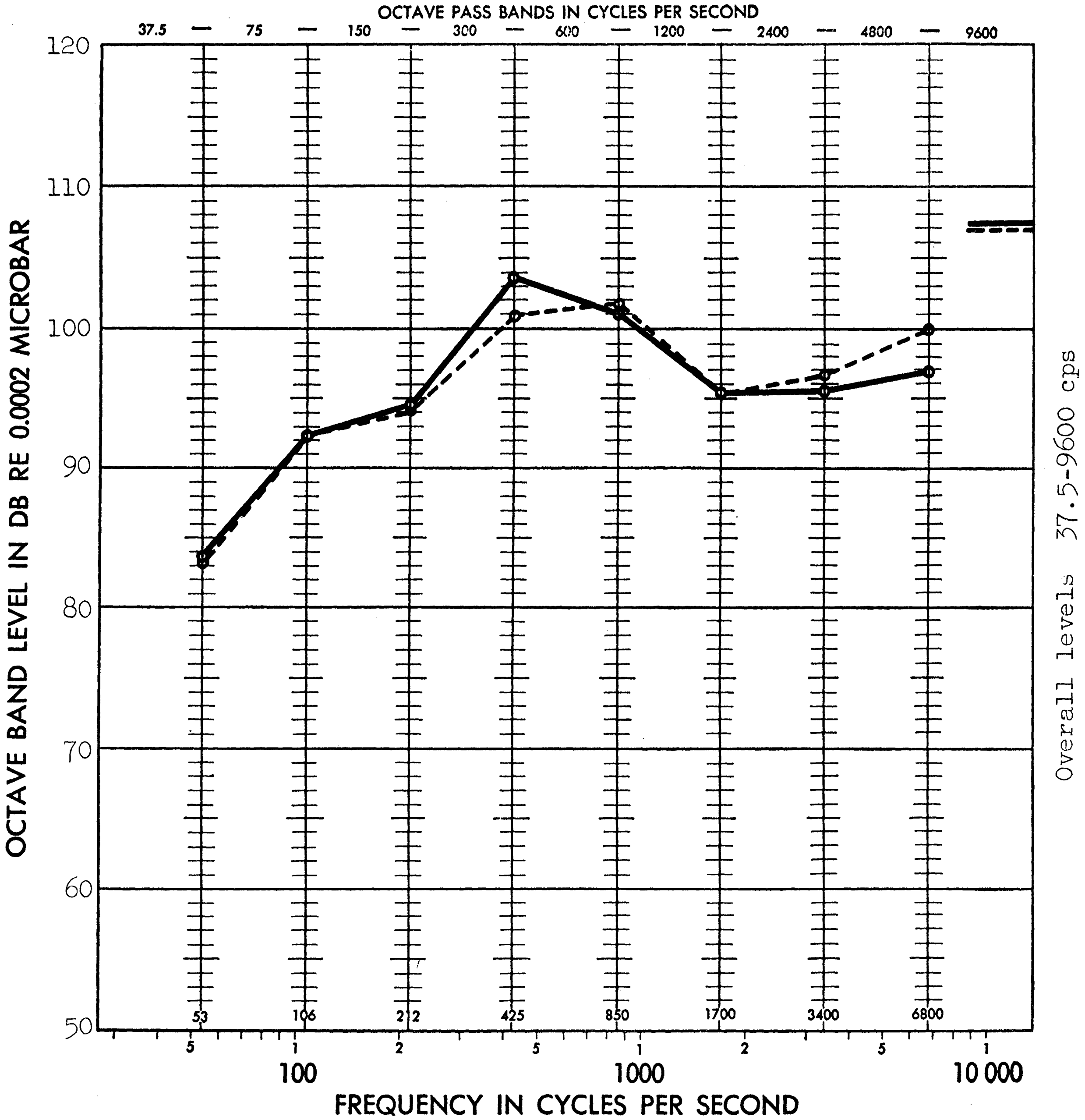




Fig. 3.51. Test plan for octave-band evaluation of air-load silencer.

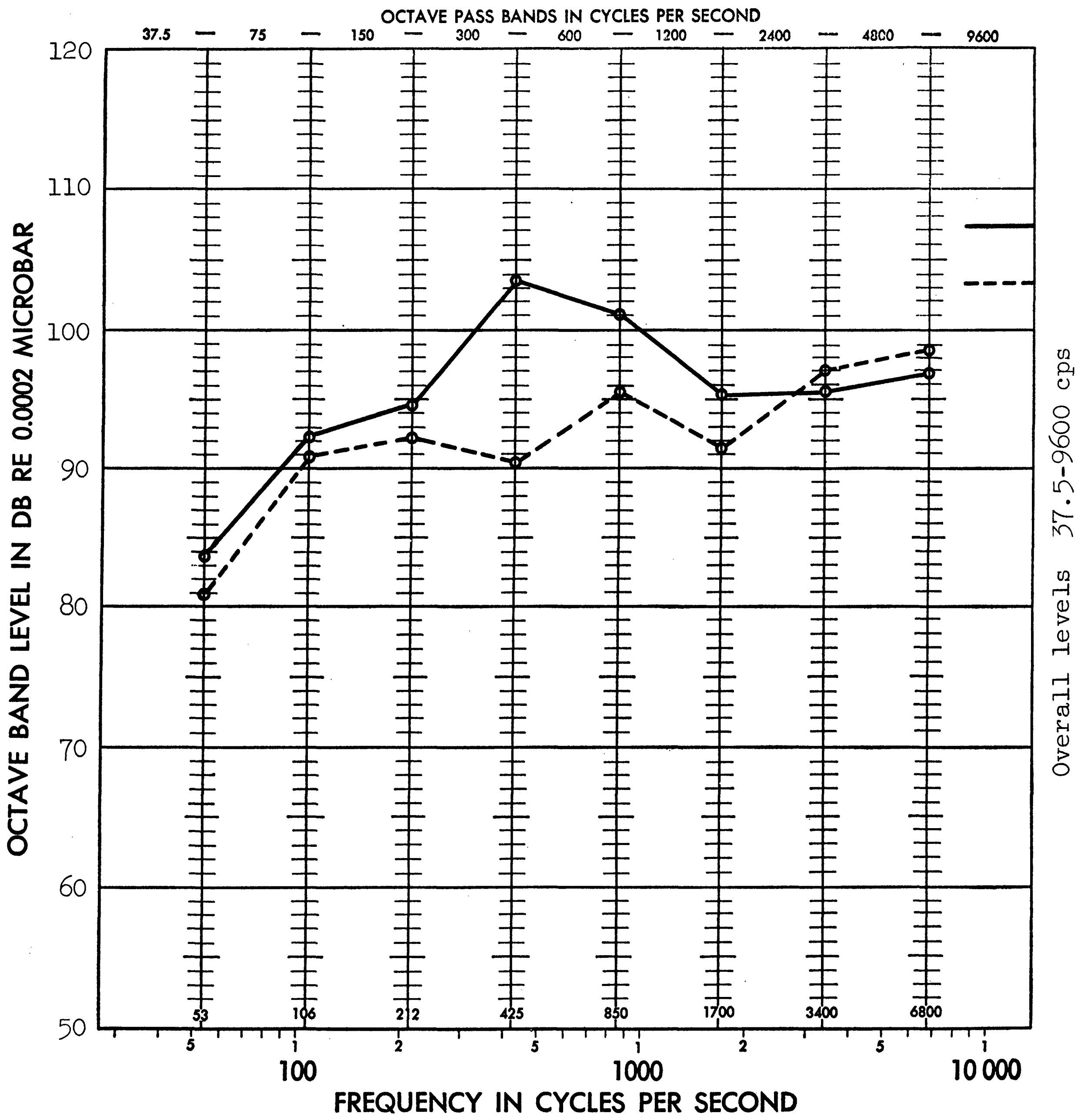
Free-field conditions
Microphone distance 40', height 5'5", 270° bearing only (microphone position 1) see Fig. 3.51




Type MA-1 gas turbine driven air compressor
34,300-34,400 rpm, 1010-1020°F exhaust temperature
--------- Free escape of load air (silencer off)
———— Air-load silencer installed
Tested 17 December 1955

Fig. 3.52. Far field effect of air-load silencer.

Free-field conditions
Microphone distance 40', height 5'5"
270° bearing only




Type MA-1 gas turbine driven air compressor, equipped with air-load silencer

--------- Tested 29 October 1955, 35,000 rpm, 1080°F exhaust temperature

———— Tested 17 December 1955, 34,300 rpm, 1000°F exhaust temperature

Fig. 3.53. Comparison of noise profiles; MA-1 gas turbine with air-load silencer.

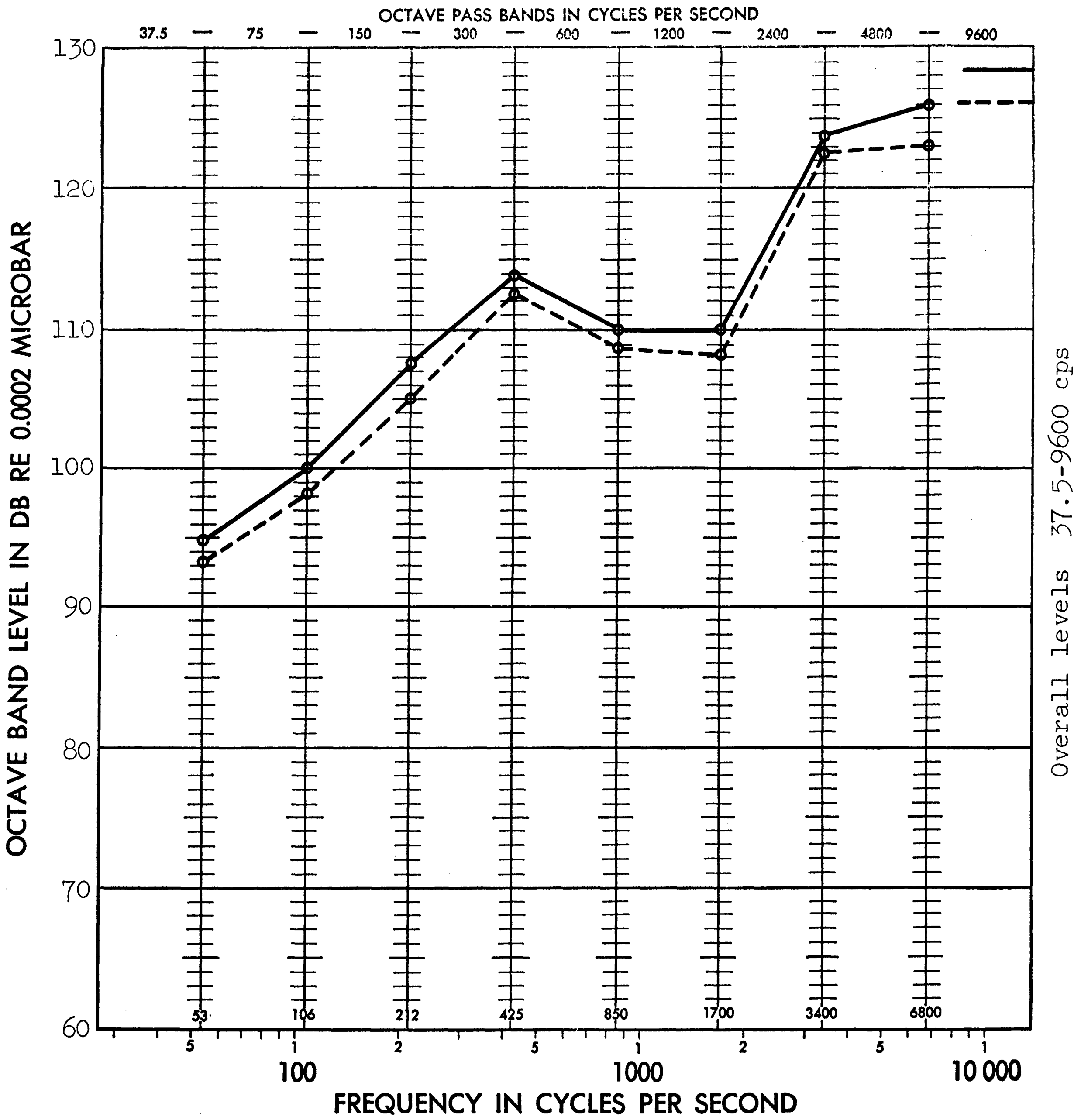
vergences was not realized at the time of the measurements due to a considerable time lapse before final data reduction permitted this detailed comparison.

Continuing with the octave-band comparison of the effectiveness of the air-load silencer, Fig. 3.54 shows the behavior obtained at microphone position 2 near the operator's station. Although the levels measured utilizing the silencer consistently lie a db or two above the corresponding levels for no silencer, this difference is probably not significant in view of the variability already noted. The computed overall levels show only a small effect, thus essentially agreeing with the earlier results reported in Fig. 3.47. However, a comparison of this profile shape secured at the operator's position with the average far-field profile in Fig. 3.49 indicates that the high frequencies (2400-4800 and 4800-9600 cps octave bands) are even more prominant at the operator's position. The significance of this will be discussed later after some discrete frequency analyses have been carried out.

Figures 3.55 and 3.56 show the effects of the silencer measured near the escaping load air. To first approximation the observed differences may be attributed to the transmission loss of the silencer walls and the absorption characteristics of the silencer, respectively. The reductions of 13.1 db and 14.2 db in computed overall noise levels compare favorably with values of 11.3 db and 16.4 db, shown previously in Fig. 3.47B. Actually the curves of Figs. 3.55 and 3.56, if they uniquely represent the effect of the silencer, demonstrate very satisfactory behavior. Notice, however, that Fig. 3.55 indicates an apparent amplification due to the silencer in the 75-150 cps octave band, and Fig. 3.56 shows similar behavior for the 75-150 cps and 150-300 cps octave bands. It is felt that these "amplifications" are probably not very significant in view of the source variability. However, at the time of these measurements and based on a preliminary data analysis only, these "amplifications" were thought to represent serious anomalies. Thus it was considered essential to determine the cause of the "amplifications" if they were real, and to do something to eliminate them. For instance, if they were caused by measurement technique, e.g., pressure doubling due to microphone proximity to the silencer box, then it was necessary to demonstrate how these "false amplifications" had been caused and to devise a measurement scheme yielding "correct" measurements.

To this end, a series of exploratory measurements were planned, concentrating attention exclusively on the suspected 75-150 cps octave band to limit the labor involved in investigating a number of measurement geometries. Also, it was realized that if the air-load silencer could not be used successfully, then some other method of operation would have to be adopted to proceed with noise-reduction work. It was learned from design engineers at the Continental Aviation and Engineering Corporation that the motorized valve on the MA-1 could be rewired so that all output air would be by-passed into the turbine exhaust, presumably without affecting the normal behavior of the MA-1. It was reasoned that by-passing the output air might be an effective way of disposing of this load air, since one would only be adding a little more characteristically jet noise (white) to the existing jet exhaust noise and at the same source location with respect to the MA-1. Furthermore, this by-passed condition was a common occurrence in practice any time the aircraft to be started was not actually accepting the output air. To evaluate the effect of by-passing the load air, the motorized valve was rewired so that operation could be switched easily from delivery through the flexible duct to the by-pass operation. Before changing from one mode of operation to the other, it was necessary to decrease the MA-1 turbine speed to below 28,000 rpm to shut off momentarily all load air during the switchover.

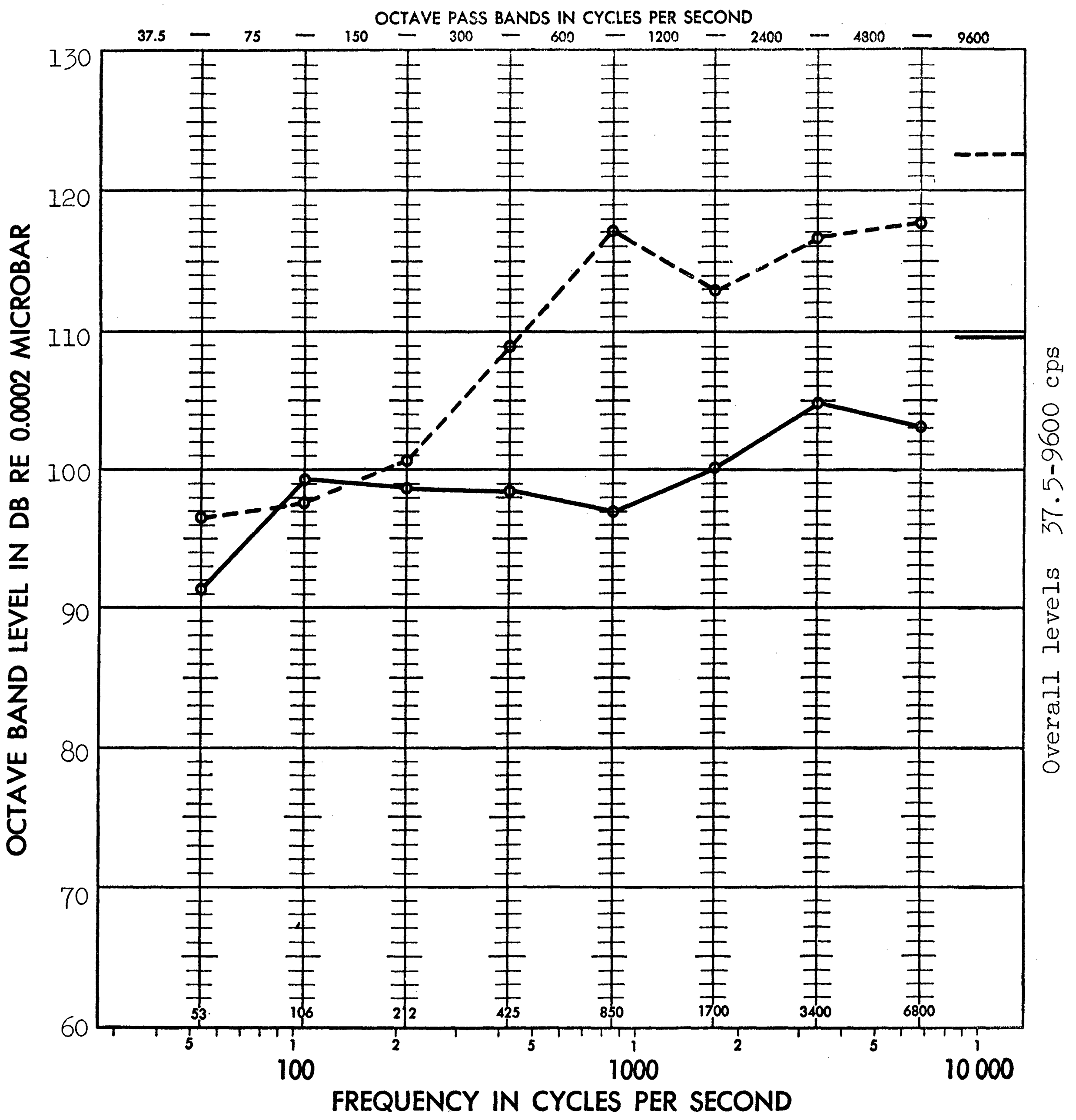
Free-field conditions
Microphone at operator's position, see Fig. 3.51, height 5'5"




Type MA-1 gas turbine driven air compressor
34,300-34,400 rpm, 1010-1020°F exhaust temperature
----- Free escape of load air (silencer off)
——— Air-load silencer installed
Tested 17 December 1955

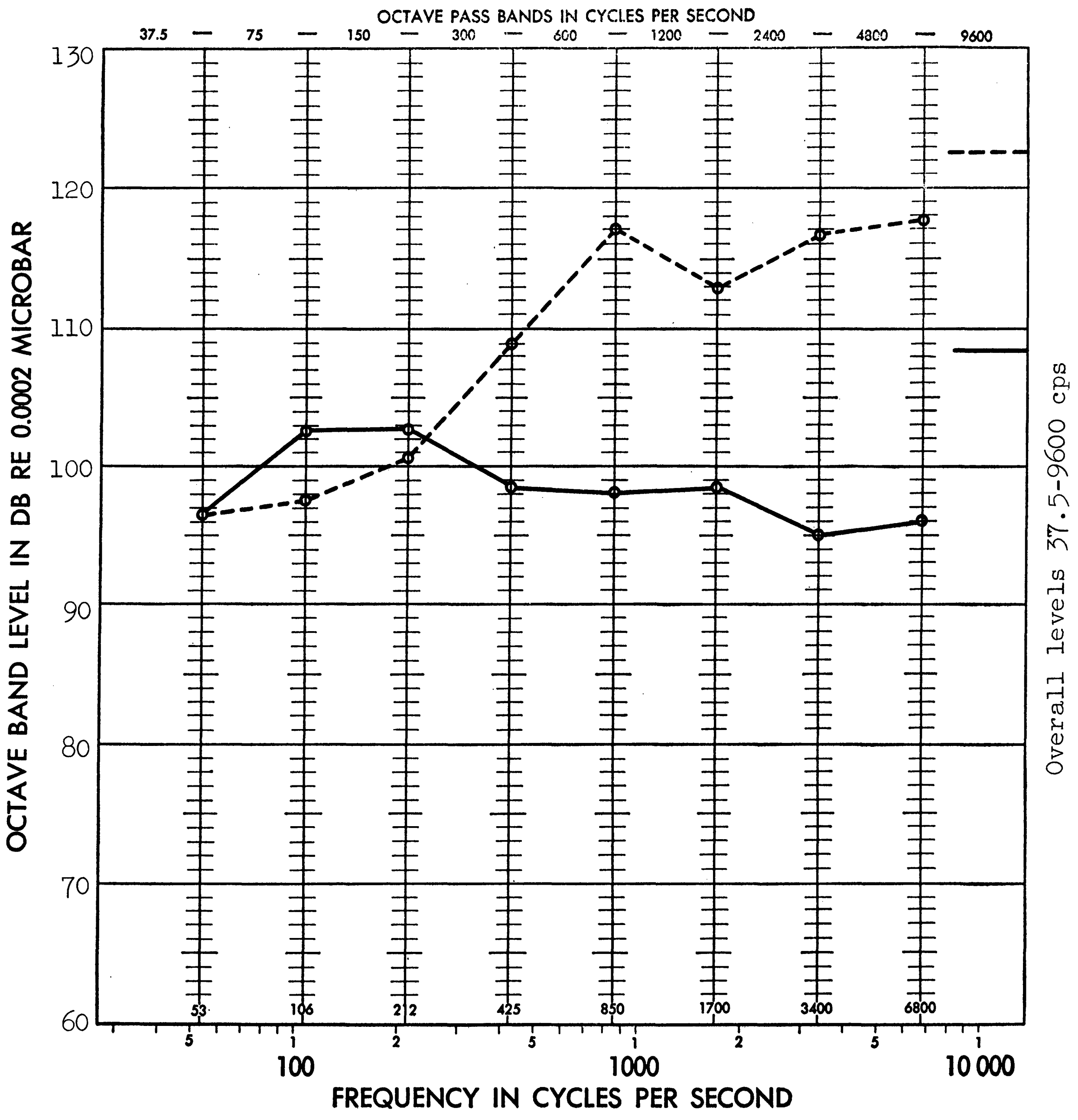
Fig. 3.54. Effect of air-load silencer at operator's position; MA-1 gas turbine.

Free-field conditions
Microphone locations, see Fig. 3.51




Type MA-1 gas turbine driven air compressor
33,800-34,200 rpm, 990-1010°F exhaust temperature
----- Free escape of load air, microphone location 3
——— Air-load silencer installed, microphone location 3A

Fig. 3.55. Local effect of air-load silencer ("Transmission Loss"); MA-1 gas turbine.




Type MA-1 gas turbine driven air compressor
33,800-34,700 rpm, 990-1030°F exhaust temperature
----- Free escape of load air, microphone location 3
_____ Air-load silencer installed, microphone location 3B

Fig. 3.56. Local effect of air-load silencer ("Absorptive Loss"); MA-1 gas turbine.

With this modification completed, it was decided to conduct exploratory experiments in the 75-150 cps octave band as mentioned above. The scheduled tests called for an intercomparison of three modes of operation: free escape of load air from the delivery duct, by-passed operation, and delivered air with the silencer attached as before.

The test geometry as shown in Fig. 3.57 was employed. For each operating condition, the microphone was moved successively to the three measuring positions, and the 75-150 cps octave-band levels were measured before proceeding with another test condition. The data are tabulated chronologically in Table 3.1 below.

TABLE 3.1. FIRST FREE-FIELD EVALUATION OF EFFECTS OF BY-PASSING LOAD AIR

Date: 30 Jan. 56 Test geometry: See Fig. 3.57
Turbine rpm: 34,800-35,200
Exhaust temperature: 940-1070°F
Turbine air pressure: 44.5-47.0 psi
Ambient air temperature: 30°F
Sound-pressure levels in db re 0.0002 dynes/sq cm
75-150 cps octave band only

| Test No. | Time P.M. | Load Air | Remarks | Microphone Location, db A | B | C |
|---|---|---|---|---|---|---|
| 1 | 2:53<br>3:07 | Delivered | First run after source stabilized | 89.1 | 106.6 | 97.1 |
| 2 | 3:07<br>3:15 | Delivered | Repeat of first run | 89.1 | 107.6 | 97.5 |
| 3 | 3:15<br>3:18 | Delivered | Stopped turbine and started again | 89.0 | 106.6 | 97.2 |
| 4 | 3:18<br>3:30 | By-passed | Slowed down to switch to by-pass operation | 88.7 | 107.3 | 94.8 |
| 5 | 3:30<br>3:41 | By-passed | Stopped turbine and started again | 101.2 | 115.3 | 106.1 |
| 6 | 3:41<br>3:45 | By-passed | Repeat previous run | 100.8 | 115.2 | 105.6 |
| 7 | 3:45<br>3:48 | Delivered | Slowed down to switch to delivered air | 99.8 | 114.1 | 112.6 |
| | | Cooled microphone, checked instrument calibration, wait, etc. All subsequent tests made between 4:30 and 5:30 P.M. | | | | |
| 8 | -- | Delivered | Start up, stabilize, and test | -- | -- | 97.6 |
| 9 | -- | By-passed | Slowed down to switch to by-pass operation | 88.7 approx. | 107.3 approx. | 94.8 |

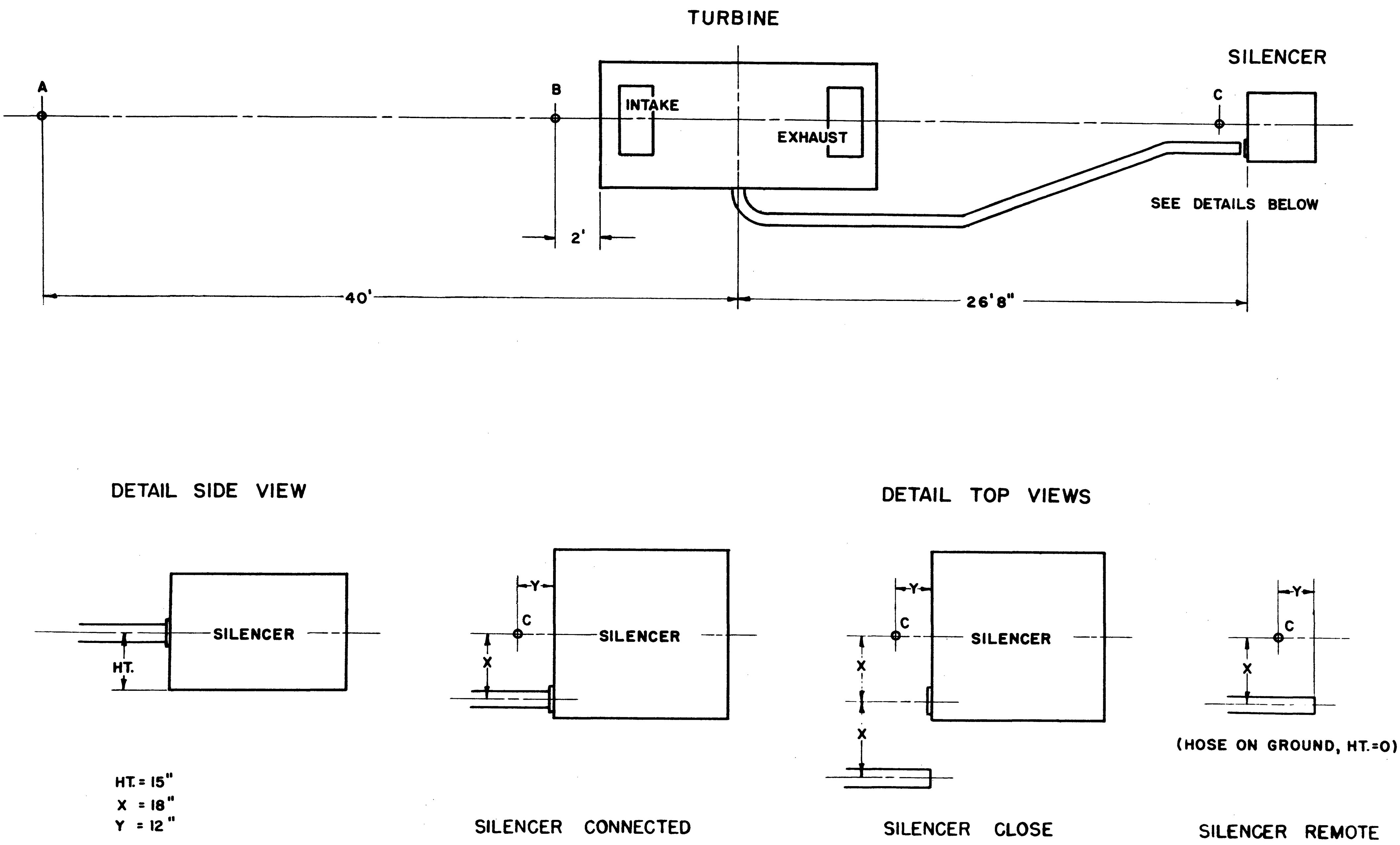



Fig. 3.57. Test plan for 75-150 cps octave-band tests.

Tests No. 1 and No. 2 show good repeatability with the MA-1's operation maintained as constant as possible. Between tests No. 2 and No. 3, the MA-1 was stopped completely and then brought back up to speed. Again the excellent reproducibility was obtained at all three measuring positions. Next the turbine was slowed down, switched to by-passed operation and brought up to speed and measured with the results shown as test No. 4. No significant differences in noise level were noted for microphone positions 1 and 2. Microphone position 3 indicates a drop, as expected, due to the elimination of the local noise source. These values would indicate that the free-air escape was contributing about 93.4 db at microphone position 3. The MA-1 was stopped and started again in by-passed operation with the results listed as test No. 5. Something very drastic happened between tests No. 4 and No. 5 because the 75-150 cps octave-band levels were now higher at all three microphone positions by amounts ranging from 8.0 to 12.5 db. An immediate recheck, test No. 6, confirmed the higher levels. Next the MA-1 was slowed, switched over to delivered air, and tested again. The higher levels were again confirmed for all three microphone positions for this changed mode of operation.

All instrumentation was carefully checked and a quick field calibration conducted, including calibration of the microphone itself, and all were found to be functioning normally. All subsequent noise measurements given at the bottom of Table 3.1 indicated that the MA-1 had reverted to its previous lower noise levels. This set of tests was conducted on a rather cold winter day, and as a final precaution the microphone was exposed until it had certainly reached ambient temperature (indicated in Table 3.1 by the expression "cooled microphone"). However, no detectable difference was evidenced in the microphone's sensitivity.

The sudden increase in level of the MA-1 Gas Turbine's noise followed by a reversion to the originally observed noise levels presents a serious situation acoustically because this behavior could not be correlated with a corresponding change in any known parameter. Apparently the noise levels generated by the MA-1, in the 75-150 cps octave band, can vary at least as much as 12.5 db and perhaps more. It is clearly impossible to collect acoustical data useful for noise-reduction purposes under such conditions.

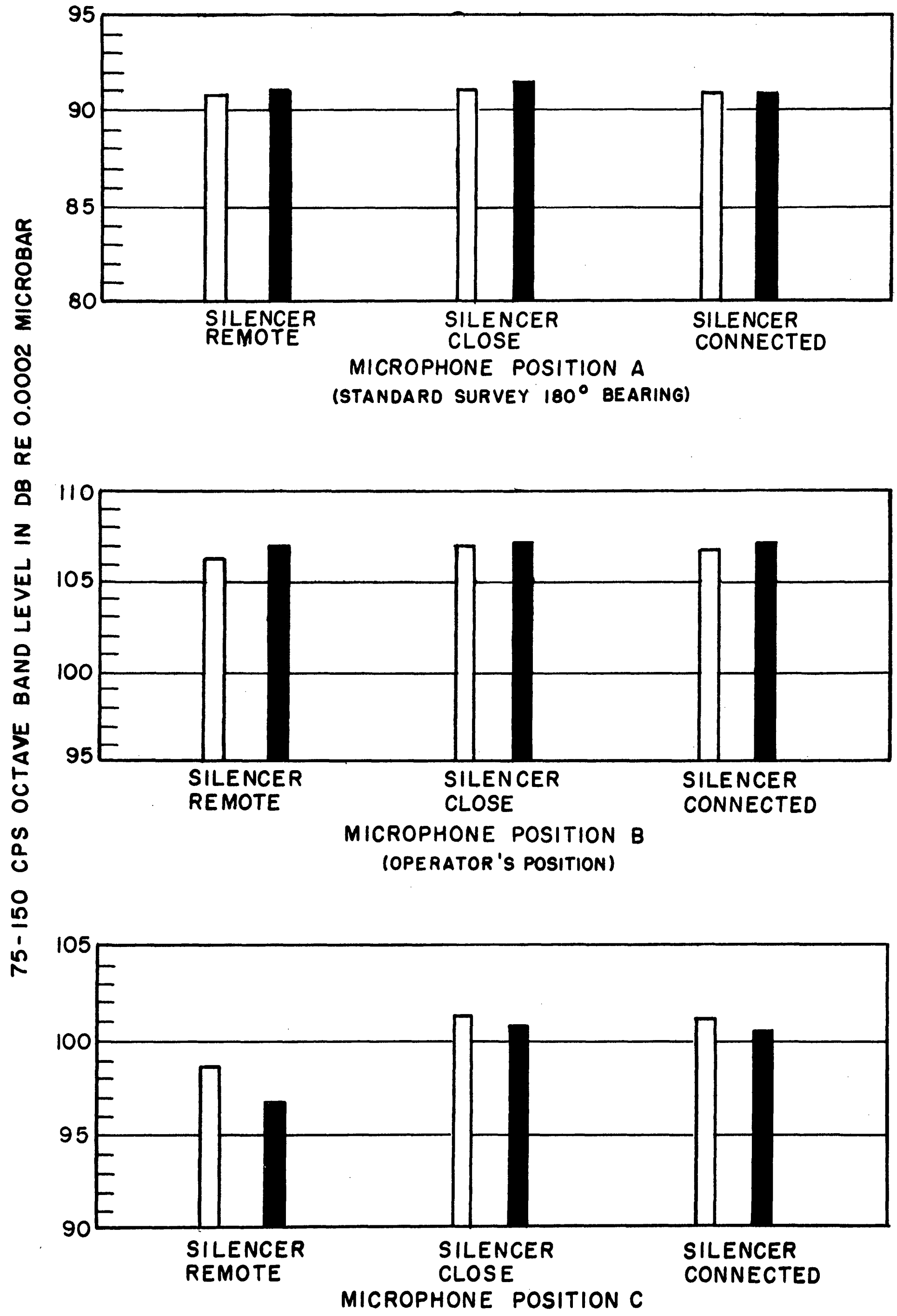
Another series of tests was conducted in which attempts were made to induce this large change in radiated noise. The test geometry was the same as shown in Fig. 3.57 with the microphone located at position 3, and the MA-1 was operated in both the free escape of the load air and in the by-pass condition. (See Appendix C for detailed list of individual tests.) Every conceivable sequence of starting the MA-1 and running it up to speed was tried, all to no avail. During by-passed operation the measured levels ranged from 95.6 to 96.4 db, while for free escape the levels ranged from 97.2 to 98.3 db. Thus only small variations in level were observed and, based on these limited data alone, the MA-1 appears to be a stable and reproducible noise source.

The large jump in noise level reported above has never been detected again in any subsequent test, but on the other hand no satisfactory explanation can be offered for the original occurrence. It seems that there must have been either a real change in the noise output of the MA-1 or some effect of the acoustical environment such as an unusual temperature gradient reflecting and concentrating the sound energy, or else some obscure interaction between the MA-1 and its environment. On the day that that test was performed, there was a rapid change in ambient temperature, which lends credence to the possibility of the acoustical environment contributing to the anomaly. Nevertheless, it is impossible to discard completely the conclusion that the acoustic output of the MA-1 may actually have changed by the observed amount.

When no explanation of the anomalous behavior seemed forthcoming, attention was returned to the problem of explaining the low-frequency "amplification" by the air-load silencer as illustrated in Figs. 3.55 and 3.56. Comparative tests in the 75-150 cps octave band were conducted at three microphone positions according to the test geometry shown in Fig. 3.57. The three silencer configurations illustrated by the bottom of this figure were examined. "Silencer Remote" means that the silencer box was removed completely from the measuring area and the microphone was located as shown with respect to the end of the delivery duct. "Silencer Close" means that the silencer was placed at the normal position with respect to the microphone, but the air was still allowed free escape from the end of the delivery duct. Finally, the third configuration, "Silencer Connected," is self-explanatory. Measurements were taken for each configuration with the air delivered and with the air by-passed. The results of these tests are displayed as a bar graph in Fig. 3.58. It is evident that just placing the unconnected silencer box near the microphone caused an increase in the observed noise levels indicating pressure doubling due to the proximity of a reflecting surface. Air flow through the silencer than increased the local noise level slightly. As far as normal free-field measurements are concerned (microphone position A, 75-150 cps octave band only), the use of either the silencer or the by-passed operation seems satisfactory. This also holds true for measurements at the operator's position (microphone position B). A further octave-band check on by-passed operation at microphone position A suggests that perhaps slight increases in the high-frequency radiated noise do occur as a result of by-passed operation.

As-Received Free-Field Noise Survey—By-Passed Operation.—When it was realized that operation of the MA-1 with the output air by-passed into the exhaust would probably be a more convenient operational procedure for many of the laboratory noise-reduction studies, a complete free-field survey was conducted using this mode of operation. As usual, the speed of the MA-1 was maintained at 35,000 rpm and the microphone located forty feet away and 5 ft 5 in. above the ground. Figures 3.59, 3.60, and 3.61 present the results of this new survey. The average computed overall sound-pressure level of 108.4 db compares very well with the previous value of 108.0 db when the air-load silencer was used. However, the polar plots of the overall noise levels, Figs. 3.59 and 3.48, demonstrate poor agreement. The average profile presented in Fig. 3.60 shows agreement in general shape with the profile obtained previously with the silencer (dashed line), but the actual levels of the individual octave bands show considerable variation. These differences may indeed be due to the two different modes of operation, particularly since preliminary evidence had indicated probable increases in the high-frequency levels due to by-passed operation. On the other hand, in view of some of the large random variations already encountered for supposedly identical tests, the variations observed above may really have little significance. The directional deviations from the averages, shown in Fig. 3.61, now exhibit a definite and consistent pattern in all octaves. There is a clear maximum at the zero-degree bearing, minima near the 90- and 270-degree bearings, and a tendency toward a weaker maximum at the 180-degree bearing, with all patterns becoming more prominent in the high-frequency octaves. Again the more consistent behavior observed for this survey compared to the earlier free-field survey using the air-load silencer cannot be explained.

Special Close-Up Tests.—In view of the limited project time and funds remaining, it was decided to proceed with testing which might suggest some useful noise-reduction




Type MA-1 Gas Turbine Driven Air Compressor operated with output air either delivered through flexible tube or by-passed into exhaust.
34,500-35,000 rpm, 920-1050°F exhaust temperature

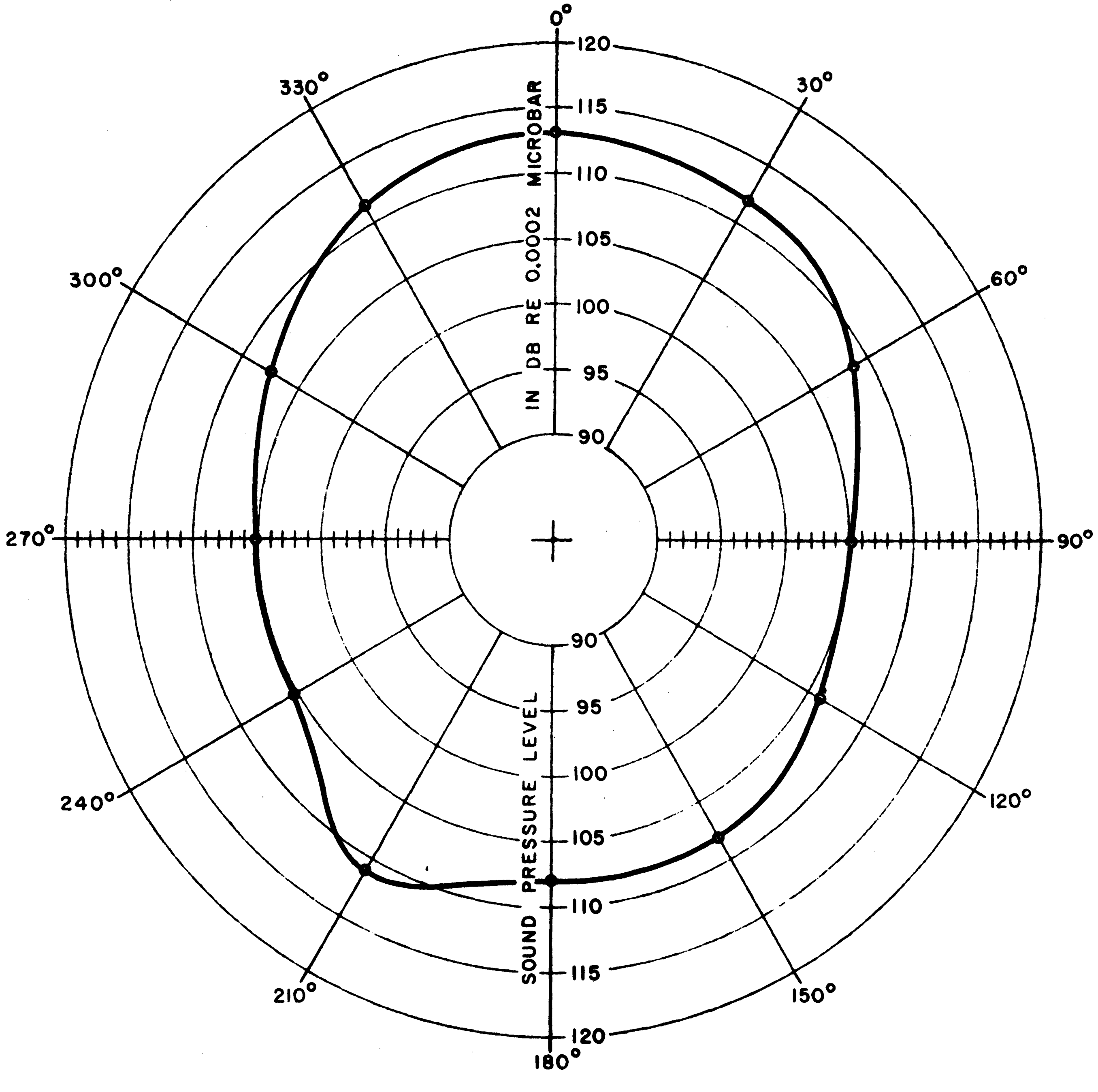
☐ Output air delivered through tube

■ Output air by-passed into exhaust

Tested 1 February 1956

Fig. 3.58. Investigations of silencer vs. by-passed operation of MA-1 gas turbine.

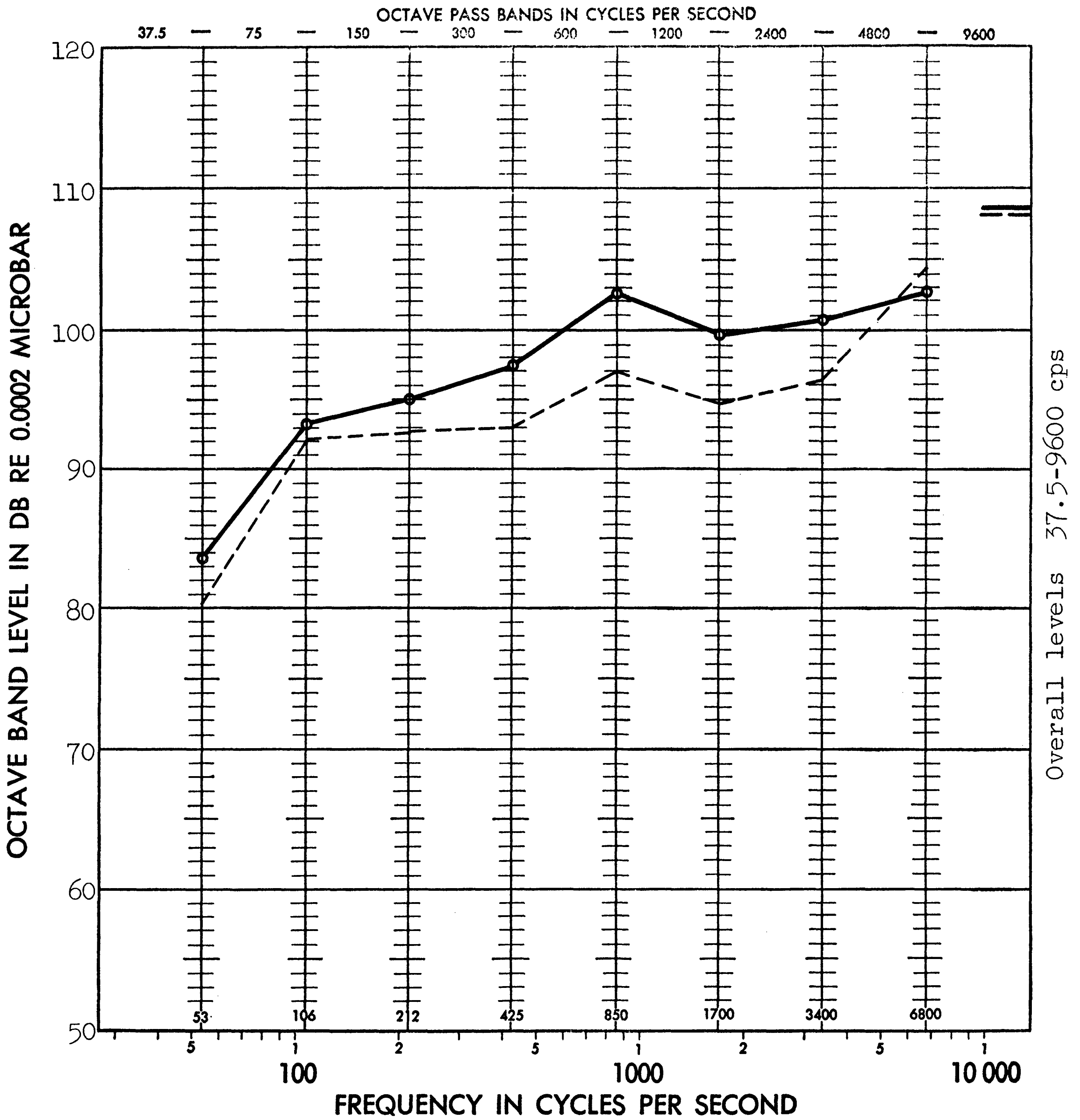
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Type MA-1 gas turbine driven air compressor, as-received condition with output air by-passed into exhaust
33,700-34,200, 900-905°F exhaust temperature
Tested 18 January 1956

Fig. 3.59. Polar distribution of overall noise; MA-1 gas turbine, as-received, air by-passed.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave band sound pressure levels




Type MA-1 gas turbine driven air compressor, as-received condition, operated with output air by-passed
33,700-34,200 rpm, 900-905°F exhaust temperature
Tested 18 January 1956

Fig. 3.60. Average octave-band noise profile; MA-1 gas turbine, as-received, air by-passed.

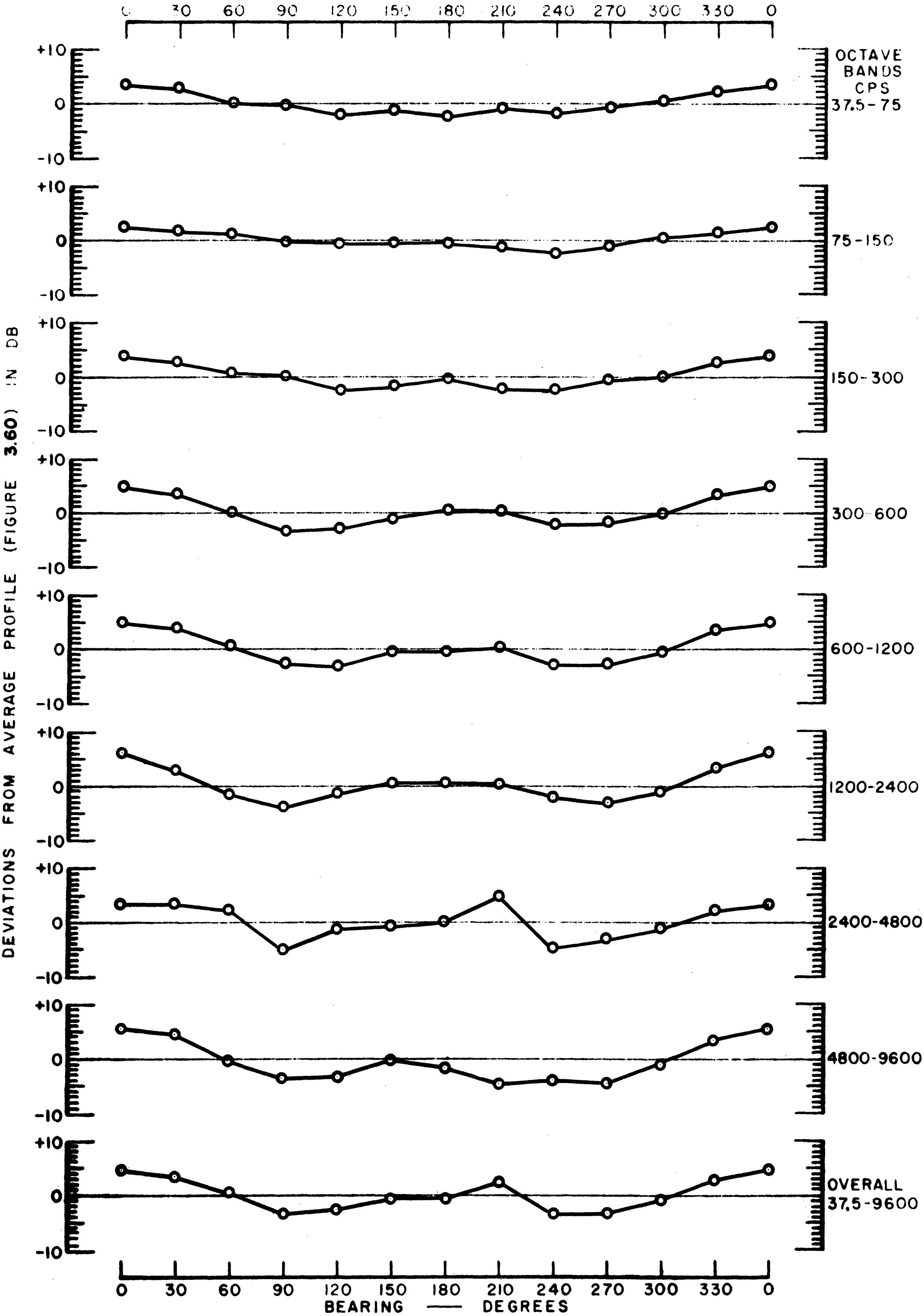



Fig. 3.61. Directional deviations from average profile; MA-1 gas turbine, as-received, air by-passed.

techniques. This, of course, represents a deviation from the rigorous and systemitized noise-reduction approach outlined in the original research proposal. However, this deviation is justified because the labor, time, and expense involved in completely characterizing the MA-1 noisewise, in view of the difficulties uncovered, would be completely prohibitive under the present contract limitations.

An experiment was conducted to determine the relative importance of panel-transmitted noise compared to intake and exhaust noise. The approach involved comparative noise measurements taken very close to the MA-1's housing in its original untreated condition, and again after a treatment designed to produce a large change in the transmitted noise was applied. For this purpose, an external cover was fabricated from 16-gauge galvanized sheet steel to conform to the top of the MA-1's housing between the control panel and the exhaust port. A 1/2-inch-thick layer of fine fiberglass (Microlite duct liner, Owens-Corning Fiberglas Corporation) was compressed between the MA-1's aluminum top panel and the steel cover by the weight of the steel cover alone. This sandwich construction is known to provide exceptionally high damping for both panels involved so that the possibility of resonant transmission of sound is virtually eliminated for the area covered by this composite treatment. The damped steel cover by itself should be capable of providing direct transmission losses on the basis of weight law ranging from about 20 db at 75 cps to 63 db at 10,000 cps. The physical arrangement of this test is illustrated by Fig. 3.62. A small sheet-metal baffle covered with fiberglass was used to block the possible direct-reflection path of intake noise to the measuring positions.

Comparative octave-band noise measurements were taken at the three microphone positions, indicated in Fig. 3.62, both with and without the composite panel treatment. These tests were performed under non-free-field conditions and with the MA-1 in normal operation with the air-load silencer attached. Because of the high noise levels encountered, a protective acoustic attenuator had to be attached to the microphone. This attenuator alters the "microphone response" but this is of no consequence here since only differences between measured levels are required. Furthermore, the non-free field measuring location renders absolute calibration useless anyhow.

No significant difference in the measured noise could be detected as a result of adding the sheet-steel panel. This result is interpreted to mean that in the "as-received" condition, the intake and exhaust noise definitely predominate the MA-1 noise spectrum, and that noise transmitted directly through the MA-1's housing need not be considered until both the intake and exhaust noises have been reduced substantially.

Discrete frequency analyses were made of the noise at the operator's position and at other locations close to the intake end of the MA-1. These were carried out using a tape recorder and a ± 1% proportional bandwidth analyzer (see Appendix B). The analyses revealed a single prominent tone at about 4700 cps corresponding to eight times rotational speed. This tone undoubtedly originates from the eight-bladed centrifugal compressor rotor. (Actually the rotor has sixteen blades, but since alternate blades are of different length, it presents a basic acoustic periodicity of eight.) Probably higher harmonics are also radiated but at the normal operating speed these lie above the measuring range of the microphone. The existence of higher tones from this or other sources is easily established by aural tests during the start and run-up sequences. Tones are heard at moderate speeds which, as the turbine's speed increases, also increase in frequency and fade out as they exceed the observer's high-frequency hearing limit.

Semi-quantitative measurements at the locations mentioned above indicate that

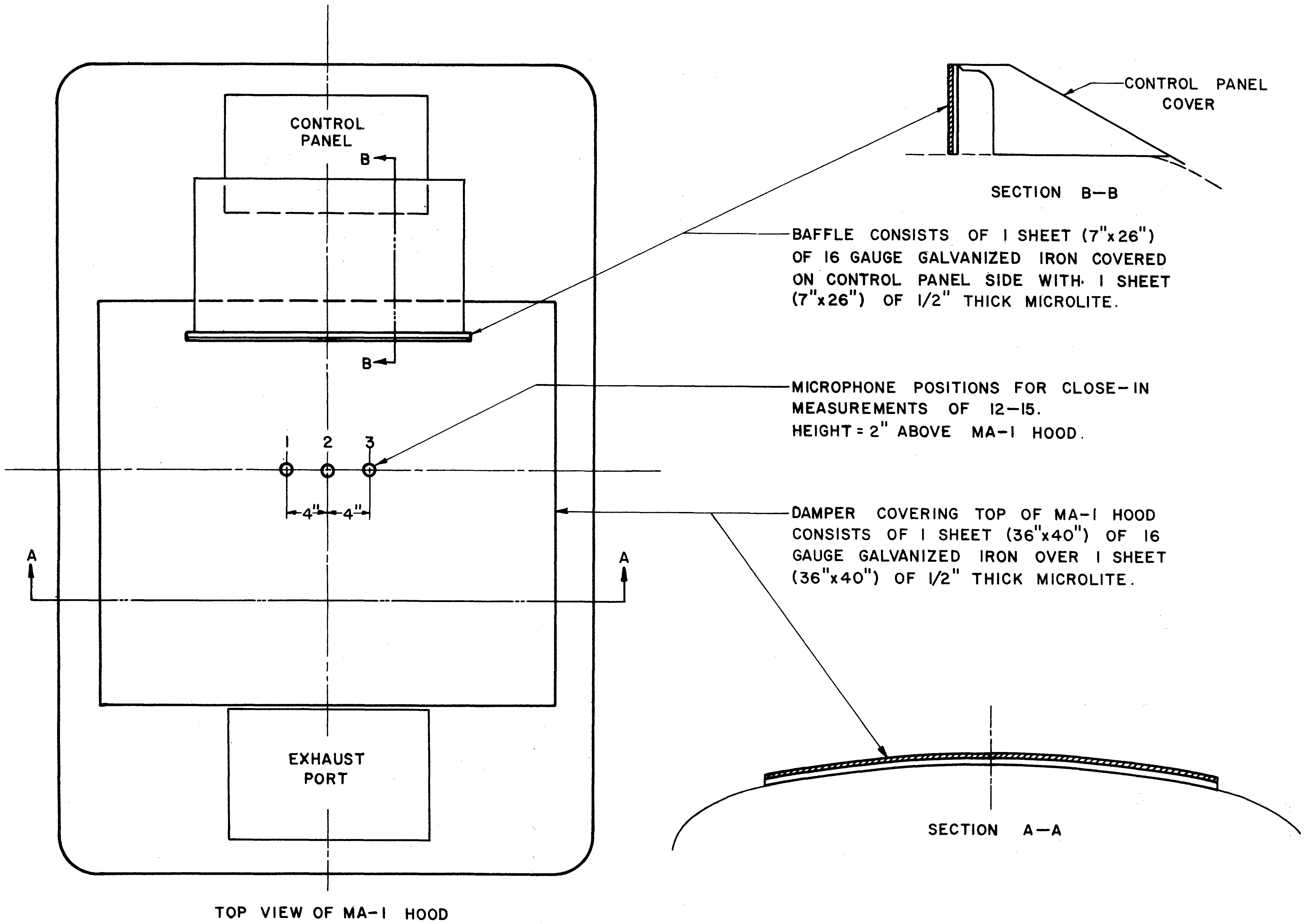



Fig. 3.62. Arrangement of cart transmission tests.

this compressor tone is of the order of 20 db more intense than any other noise within the frequency range of the measuring equipment. The frequency location of this tone at the boundary between the 2400-4800 and 4800-9600 cps octave bands also explains the high noise levels observed for both of these octave bands at the microphone positions used for these discrete-frequency analyses. This constitutes an excellent example of how a single prominent tone can dominate two adjacent octave bands and render the interpretation of the octave-band data impossible until the detailed character of the noise is known. The results of these discrete frequency analyses corroborate and explain the high levels measured previously at the operator's position under free-field conditions as presented in Fig. 3.54.

This same high-frequency tone could be detected by discrete-frequency analysis at other non-free-field measuring positions around the MA-1 trailer, even adjacent to the exhaust port, but the tone was not as prominent here as compared to the other essentially white noise. These facts suggest that the high-frequency compressor tone may be responsible for most of the excess 2400-4800 and 4800-9600 cps octave-band noise observed at the operator's position under free-field conditions. However, the corresponding far-field high-frequency noise probably contains this tone in lesser proportion, and the rest of the noise is contributed by the white noise originating from the turbulent exhaust stream.

Some preliminary measurements were taken at intervals along a 20-foot vertical path adjacent to the cone of exhaust gases by elevating the microphone along a guyed steel pole. These measurements were not very informative, but seem to be consistent with the interpretation that much of the noise is generated by turbulent exhaust flow, and as such the "sources" are distributed along the path of the exhaust gas flow external to the MA-1's trailer. The ordinary means of palliative noise reduction are thus not directly applicable until these sources can be confined within an enclosure of some type.

<u>Exhaust-Muffler</u> <u>Tests</u>.—In keeping with the altered approach explained above, experimental configurations of exhaust silencers for the MA-1 were fabricated and jected to limited testing. Neither configuration was intended to simulate anything t would be used in service; rather, both mufflers were designed mainly to investie the applicability of acoustic principles and to facilitate laboratory testing.

The first exhaust-muffler configuration tested, designated Muffler No. 1, is shown in Fig. 3.63. This muffler consists of a short flared stack of rectangular section terminated in a perforated metal cover and was based on the general principles described by J. Tyler and G. Towle.[3] The perforated sheet-metal cover consisted of 1/16-in.-diameter holes at 1/8-in. staggered spacing, resulting in about 22.5% open area. To install this muffler, the top cover of the MA-1's housing had to be removed. The muffler was then mounted vertically directly on the existing short rectangular exhaust duct.

Muffler No. 1, as originally constructed, somewhat restricted the flow of exhaust gases. No attempt was made experimentally to adjust the perforated area to regain the original flow condition; instead, the MA-1's speed was restricted to 30,000 rpm for the purposes of this test. Also for convenience, the output air was by-passed into the exhaust and hence through the muffler.

Comparative octave-band measurements were made under non-free-field conditions at both the operator's position and 40 feet away from the exhaust end of the trailer

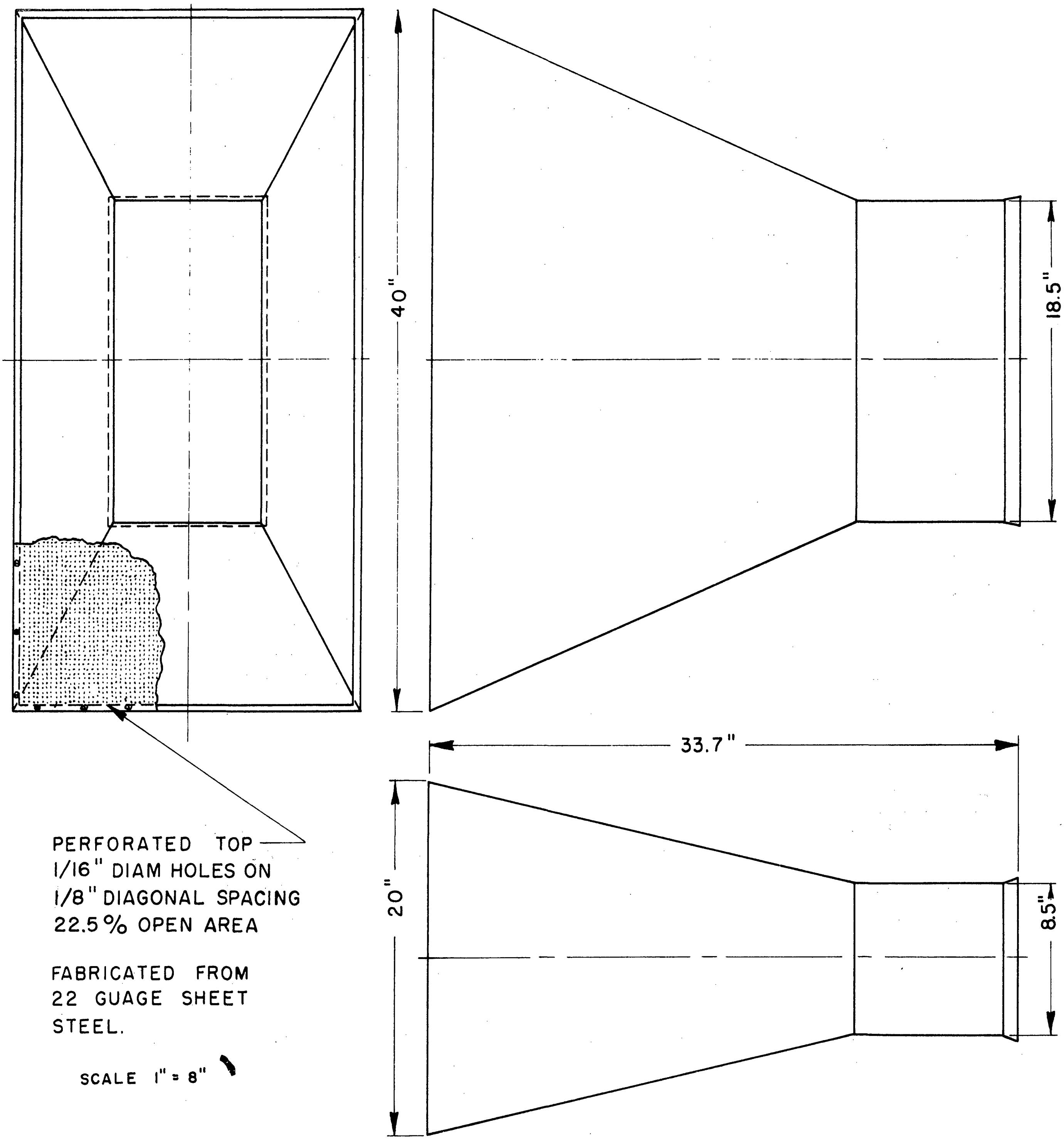



Fig. 3.63. MA-1 gas turbine exhaust muffler No. 1.

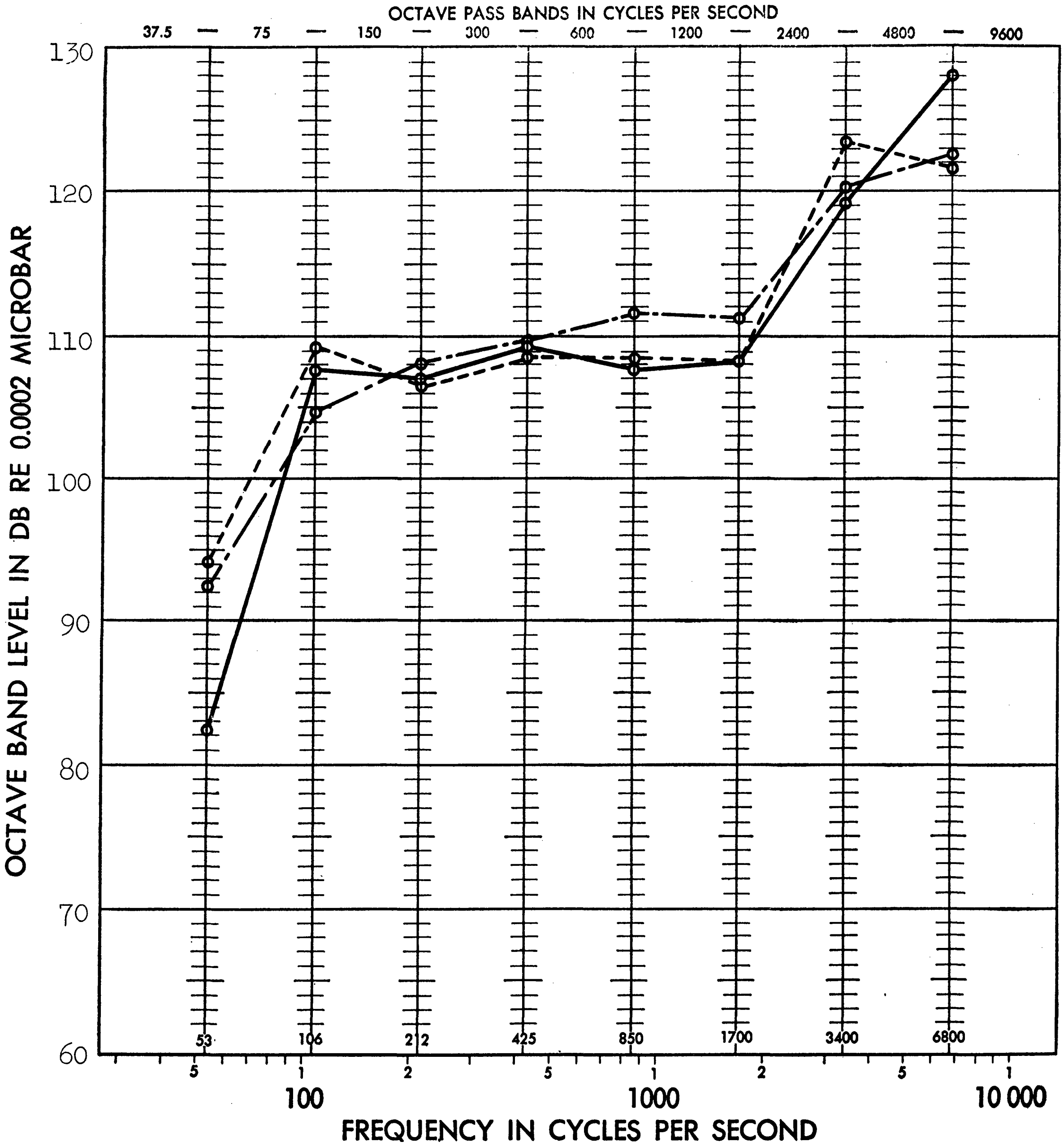
(zero-degree bearing) for three test configurations: (1) complete exhaust muffler installed, (2) flared muffler stack installed but perforated top cover removed, and (3) experimental muffler completely removed. Listening tests seemed to indicate some change in character of the noise due to the silencer. The results of the octave-band measurements are displayed in Figs. 3.64 and 3.65. No consistent or significant changes could be detected at the operator's position. At 40 feet from the exhaust, the high frequencies appeared attenuated while the low frequencies were amplified slightly. The low-frequency amplification may have been the result of forced vibration of the muffler's sheet-metal housing. The vibration amplitude was large enough to be observed easily by eye. Above 600 cps, reductions ranging from about 6 to 10 db were observed. However, in view of the non-free-field measuring conditions, about all that can safely be concluded is that there appears to be some reduction of the high-frequency noise. No free-field tests were conducted and something of much greater effectiveness is sought because of the high noise levels generated by the MA-1 in original condition.

A second muffler, Muffler No. 2, was constructed as shown in Fig. 3.66. The principal design consideration here was to obtain a long section of ducting arranged for maximum experimental flexibility. The sides of this muffler diverge gradually at an angle of about 8 to 10 degrees so that it will act much like the diffuser section of a wind tunnel, thus minimizing flow separation and additional turbulence. It was fabricated in five sections so that various treatments could be assembled, bends and tuned absorbers inserted, etc. In addition, turning vanes were inserted into the bends to guide the air flow. The final exit area is about twice the area of the rectangular exhaust duct of the MA-1. It is emphasized again that this muffler was designed to facilitate experimental investigation and was not expected to resemble a device suitable for service application.

In the first acoustic tests with Muffler No. 2, the simplest configuration was employed as shown in Fig. 3.67. Neither an absorptive lining nor a coupled resonant absorber nor any other acoustic control device was inserted, so that for these preliminary tests Muffler No. 2 merely constituted a rectangular pipe which changed the location of the exhaust aperture with respect to the MA-1's trailer. Comparative octave-band measurements were taken with and without the muffler at two microphone locations near the operator's position, and at two locations near the exhaust aperture as illustrated in Fig. 3.67. For the latter measurements, the distances were measured from the center of the particular exhaust aperture in question; hence the microphone was at different distances from the MA-1's trailer depending upon whether or not the muffler was installed. The MA-1 was operated at 35,000 rpm and the output air was by-passed. As in the case of the previous muffler test, a non-free-field location was used for these preliminary tests.

Listening tests again confirmed a change in the character of the noise and Fig. 3.68 presents the results obtained at the two microphone locations near the operator's position. Moderate reductions appear in the low and intermediate octave bands; however, since the installation of the muffler moves the effective location of the jet exhaust noise farther away, the most probable cause of the noise reduction observed is simply that resulting from the increased distance between the exhaust and the microphone. Figure 3.69 presents the results of measurements taken near the exhaust aperature. In this case most of the reduction is observed to occur above 600 cps, increasing toward higher frequencies. These moderate reductions are probably real, but again, because of the non-free-field measuring conditions, no particular credence should be attached to the observed magnitudes of reductions.

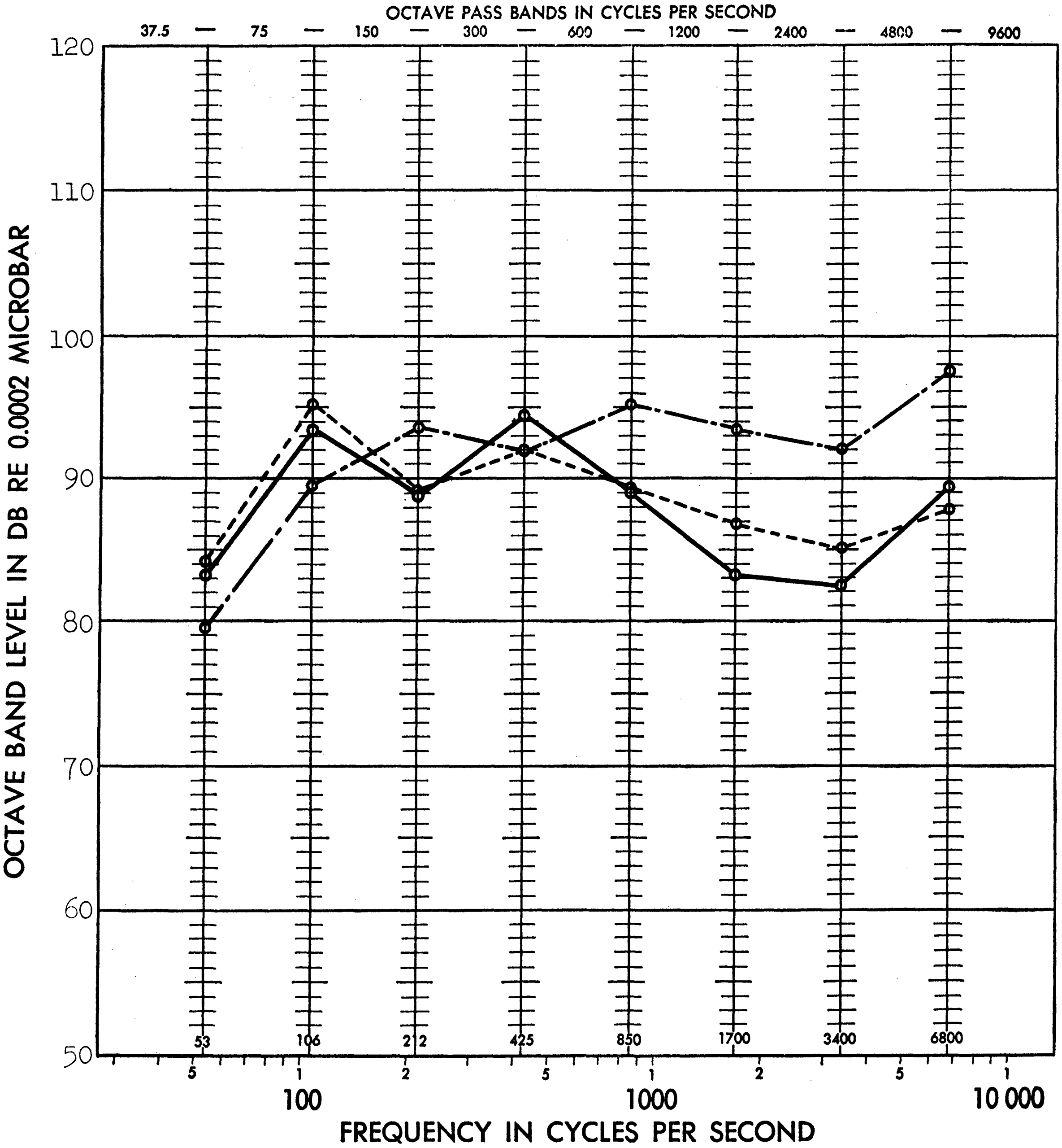
Nonfree-field conditions
Microphone located near operator's position distance 2' out from end of cart, height 5'5"




Type MA-1 gas turbine driven air compressor, output air by-passed
30,000 rpm
———— With muffler No. 1 attached, perforated top installed
--------- With muffler No. 1 attached, perforated top removed
—— - —— With muffler No. 1 removed completely
No overall levels included because of nonfree-field conditions

Fig. 3.64. Effect of muffler No. 1 at operator's position; MA-1 gas turbine.

Nonfree-field conditions
Microphone located 40' distant from exhaust end of cart, height 5'5"




Type MA-1 gas turbine driven air compressor, output air by-passed
30,000 rpm
———— With muffler No. 1 attached, perforated top installed
--------- With muffler No. 1 attached, perforated top removed
—— - —— With muffler No. 1 removed completely

Fig. 3.65. Effect of muffler No. 1 40' from exhaust; MA-1 gas turbine.

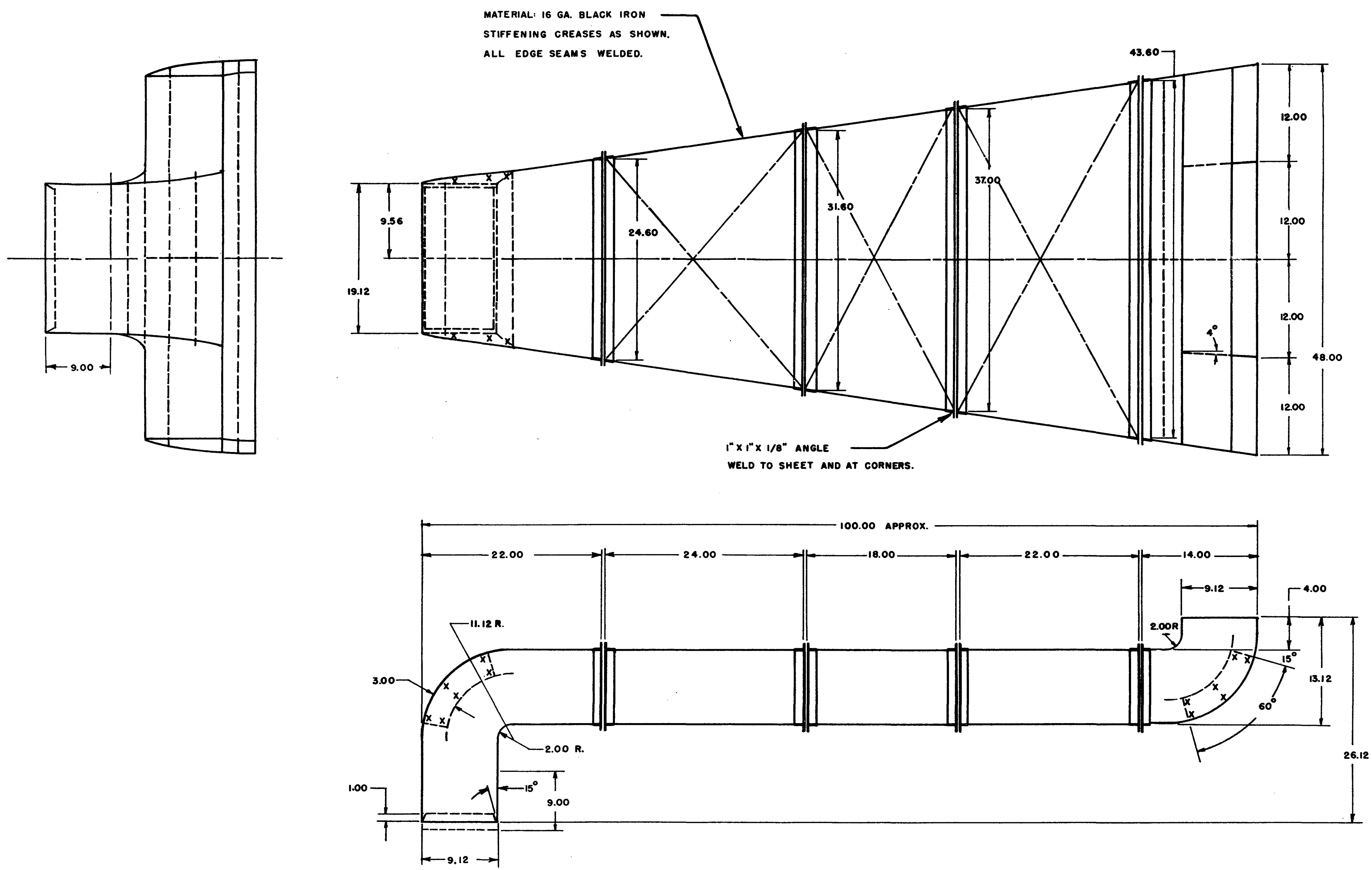
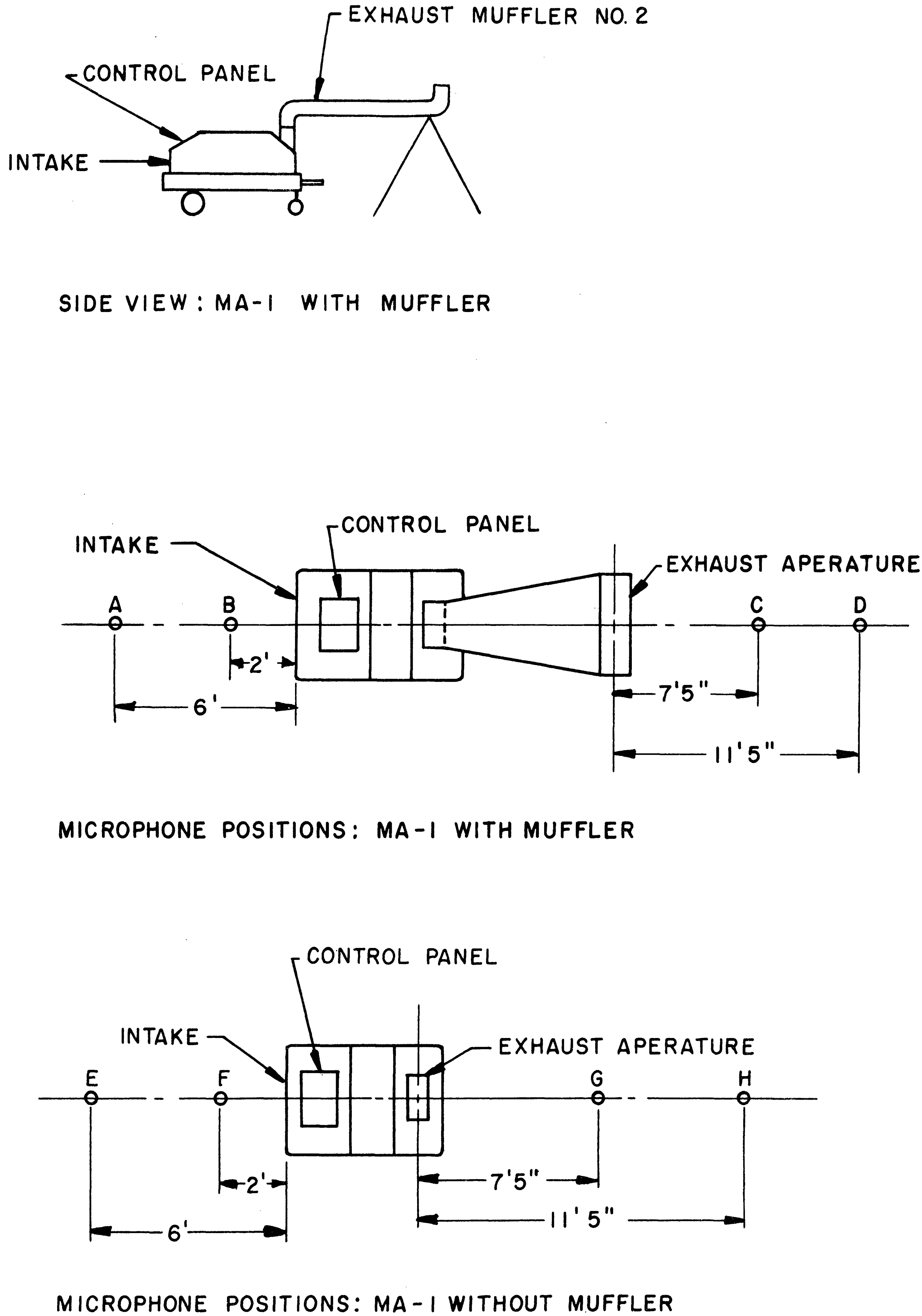


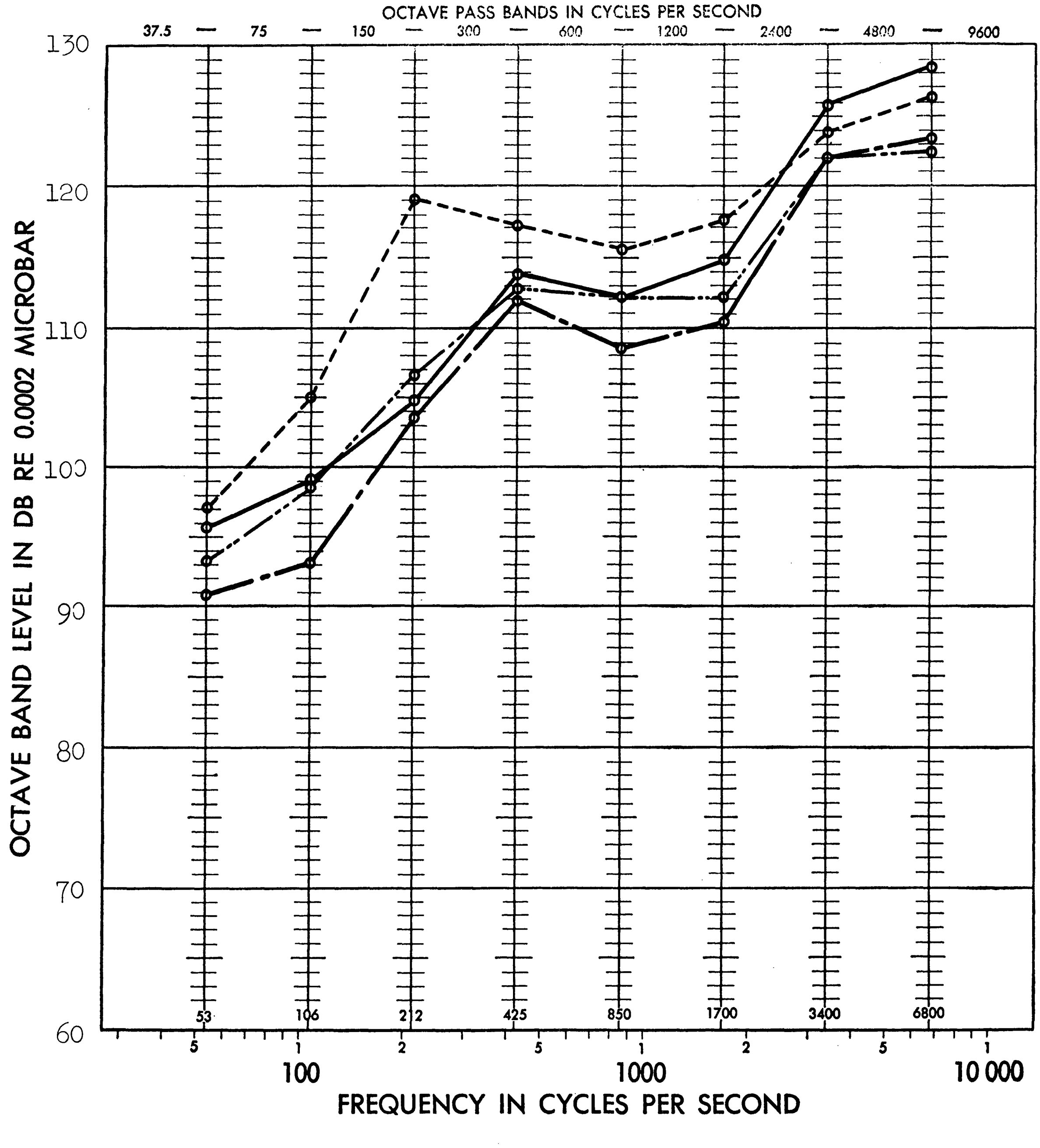

Fig. 3.66. MA-1 gas turbine exhaust muffler No. 2.




O - MICROPHONE LOCATION (5'5" ABOVE GROUND FOR ALL POSITIONS)

Fig. 3.67. Test arrangement for muffler No. 2.

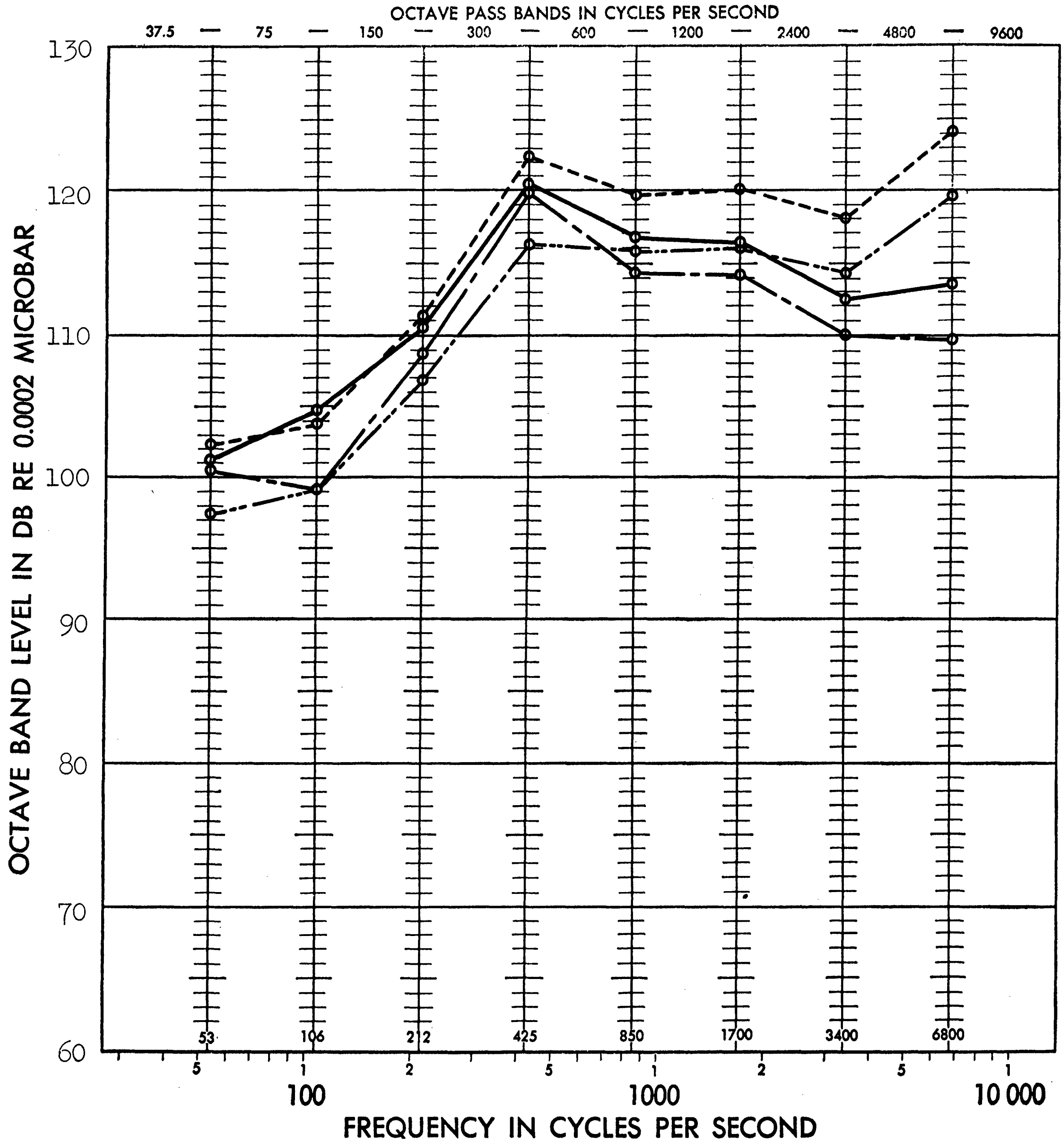
Nonfree-field conditions
Microphone located near operator's position, see Fig. 3.67




Type MA-1 gas turbine driven air compressor, output air by-passed
34,800-35,000 rpm, 930-940°F exhaust temperature
———————— With muffler No. 2 attached, microphone distance 2'
----------- With muffler No. 2 removed, microphone distance 2'
——— - ——— With muffler No. 2 attached, microphone distance 6'
——— -- ——— With muffler No. 2 removed, microphone distance 6'

Fig. 3.68. Effect of muffler No. 2 at operator's position; MA-1 gas turbine.

Nonfree-field conditions
Microphone located near exhaust aperture, see Fig. 3.67




Type MA-1 gas turbine driven air compressor, output air by-passed
34,900-35,000 rpm, 930-950°F exhaust temperature
———— With muffler No. 2 attached, microphone distance 7'5"
-------- With muffler No. 2 removed, microphone distance 7'5"
—— - —— With muffler No. 2 attached, microphone distance 11'5"
—— -- —— With muffler No. 2 removed, microphone distance 11'5"

Fig. 3.69. Effect of muffler No. 2 near exhaust; MA-1 gas turbine.

It was originally planned, using Muffler No. 2, to conduct various additional tests, such as inserting various absorptive linings both in the straight sections and in the bends. Resonant chamber absorbers and other acoustic geometries were also to be considered. Thus the preliminary tests reported above, which do not indicate anything very spectacular by themselves, were to be followed by a whole series of tests which were expected to yield considerable information, e.g., the applicability of absorptive ducts in controlling jet exhaust noise. However, these additional tests were not completed due to the expiration of the contract period and lack of funds.

Present Status of Problem and Recommendations.—All noise-reduction research included in the scope of this project was proposed and undertaken on the assumption that palliative noise-reduction procedures alone would suffice to make large improvements in each of the pieces of ground-support equipment to be studied. The reasons for this were set forth in the original research proposal (see Appendix A). At that time, based on brief listening tests under non-free-field conditions and on certain panoramic analyses performed by others, it was believed that the MA-1 Gas-Turbine-Driven Air Compressor's noise was completely dominated by high-frequency compressor scream. Thus palliative techniques would have sufficed for at least the first cycle of noise-reduction research.

The tests reported above do indeed confirm the presence of at least one high-frequency compressor tone in the measuring range and that this tone does appear to predominate at the operator's position. However, these same tests also establish the existence of large amounts of jet exhaust noise. This "white" noise contributes significantly to all portions of the average noise profile or spectrum above about 75 or 100 cps. Therefore, it is necessary to reduce the jet exhaust noise if much significant noise reduction is to be accomplished. However, the jet exhaust noise originates in the turbulent flow of the decelerating exhaust gases external to the MA-1 trailer, placing these sources beyond the reach of normal palliative noise-reduction techniques. The problem of noise control on the MA-1 thus necessarily includes noise-reduction-at-the-source techniques which are orders of magnitude more complex and expensive than the proposed palliative program.

The actual experimental work on the MA-1 was beset by three broad areas of difficulties. The first difficulty was providing suitable disposition (from the acoustical viewpoint) of the compressed-air output. The second difficulty arose from the apparent inherent variability of the MA-1 as a noise generator. The third difficulty arose from the disparity between the assumed initial noise-reduction problem and the real noise-reduction problem as it was gradually revealed during the measurement program. The original assumptions dictated certain measurement approaches, and the resulting interpretational difficulties, even allowing for experimental variability, required additional measurements. Thus the true picture of the problem developed through a series of essentially negative experimental results. A few additional, scattered, preliminary experiments were conducted near the end of the program to gain some insight into the magnitude of the noise-reduction-at-the-source problem.

The fact that a major portion of the MA-1's noise is generated by the jet exhaust stream may partially explain some of the variability observed. The total noise generated by such an exhaust stream is generally considered to depend on the eighth power of the jet velocity and the first power of the cross-sectional area of the nozzle. Since the only parameters of the MA-1's operation readily monitored were the rpm and the exhaust temperature, it is conceivable that small changes in exhaust velocity occur unnoticed from one run to another. Because the noise generated is such a rapidly changing function of velocity, these unnoticed velocity changes might be responsible for considerable acoustic variation. It is unlikely, however, that the one very large acoustic anomaly occurred in this way. If it were assumed that a similarly large jump occurred in the total noise, then the exhaust velocity would have had to change by something of the order of 20%, an improbably large amount.

As mentioned earlier, a pure tone in the neighborhood of 4700 cps was found to be rather prominent and has been identified as originating from the compressor. This sound is propagated out of the air inlet and finds its way out of the MA-1's housing either by direct or indirect paths after reflection within the housing. Thus, in principle, it should be easy to reduce this noise by proper sound absorption treatments. This can be accomplished by proper positioning of the absorptive treatment so that the sound must reflect at least once from sound-absorbing material before escaping from the housing. Preferably, all possible escape paths should involve multiple reflections from the absorbing treatment, and sound-absorbing materials having high absorption coefficients in the proper frequency range should be used. Such an absorptive treatment was not actually tried for several reasons. The rather cluttered interior of the MA-1's trailer surrounding the air intake makes it extremely difficult if not impossible to obtain an effective absorptive installation without limiting the intake air flow or without permanently altering the trailer interior. Furthermore, to demonstrate appreciable noise reduction, it is necessary to diminish the jet exhaust noise concurrently, and the means for controlling the jet exhaust noise had not yet been worked out; only a few preliminary experiments had been conducted.

In connection with some other acoustic research carried out recently by this laboratory, acoustic measurements have demonstrated the applicability of short, absorptively lined ducts in controlling jet noises caused by hot gases issuing from nozzles at near-sonic velocities. Preliminary evaluations have shown reductions of 15 db or more with reasonable geometries and sizes of ducting. Thus it appears that carefully designed silencing-type nozzles in conjunction with absorptively lined ducting might provide an acceptable solution to this jet exhaust noise problem.

Within the limitations of the measurement program described in this section, the MA-1 noise problem seems to be about as follows. The far-field noise consists of "white" jet exhaust noise extending from comparatively low frequencies upwards and including important contributions at the high frequencies from one or more compressor tones. At the operator's position, the compressor noise is more prominent, but the jet exhaust noise is not negligible. Under the present circumstances, noise transmitted through the housing walls appears to be completely negligible.

In terms of silencing, an effective jet exhaust muffler is required along with absorptive reduction of the compressor scream. The nature of the troubled absorptive treatment is known but some modification of the present MA-1 trailer will be required for effective installation. The detailed nature of the required jet exhaust muffler is not known, although several approaches appear promising and might ultimately be developed into configurations acceptable for service installation.

## CONSOLIDATED MA-1 MULTIPURPOSE UNIT

The Consolidated MA-1 Multipurpose Unit is a compact four-wheeled vehicle capable of a number of distinct functions. It has an enclosed cab and can be driven as a normal automotive vehicle. It is capable of towing a rather considerable load. Moreover, its engine, which was a standard water-cooled automotive type, may be coupled to a cluster of three electrical generators (two d-c generators and one a-c generator) located at the rear of the cab. The a-c generator operates a 3500-psi air compressor located at the side of the cab. Figure 3.70, copied from T.O. 19-45-403, shows a side view of this unit together with a legend identifying the important features. Figure 3.71 is a plan view, taken from the same source, on which has been superimposed the identification of the azimuthal directions used during the free-field survey.

Although the driver's cab itself was reasonably weathertight, from an acoustical viewpoint this entire unit was quite open. Various engine and generator compartment panels are louvered, and during normal operation of the generators and air compressor, several compartment panels have to be open to expose the operating controls and instruments. Also from the acoustical viewpoint, the principal feature differentiating this reciprocating-engined unit from the others tested and reported here is that this engine was already equipped with a moderately effective muffler.

Free-Field Noise Survey.—To provide comparison data, the Consolidated MA-1 Multipurpose Unit was tested under free-field measuring conditions in a manner compatible with the other surveys described in this report. The microphone was located 40 feet away, 5 ft 5 in. above the ground. The water-cooled electrical loading coils were used to dissipate a d-c load of about 880-890 amperes at 30 volts. The a-c generator was used to operate the high-pressure air compressor, and by carefully adjusting the output air valve, a steady level of operation of 3000-3400 psi was maintained throughout the free-field survey. Incidentally, this air compressor gave considerable mechanical trouble and had to be serviced several times during the survey to keep it operating properly.

Figures 3.72 and 3.73 are photographs of this unit located at the free-field measuring site. Figure 3.72 is taken from a bearing of about 315 degrees, and shows the electrical loading coils just beyond the radiator. Figure 3.73 is taken from the opposite bearing of about 135 degrees. The sighting arrangement used to establish the relative bearing is clearly seen attached to the roof of the cab. Likewise, the open condition of the body panels during operation is evident. The microphone is located outside the picture area to the left. Several other photographs taken during this same free-field survey illustrate the nature of the free-field test site and will be found in Appendix B.

Figures 3.74, 3.75, and 3.76 present the results of this free-field survey in the usual manner. The overall directionality pattern of the Consolidated MA-1 Multipurpose Unit as displayed in Fig. 3.74 shows a slight tendency to bulge in the fore and aft directions corresponding to the engine and generator ends of the unit, respectively. Probably the most noticeable feature is that in no direction did the sound-pressure level exceed 89 db. This is drastically different from the as-received overall noise values of about 101 db and 104 db obtained for the A-1 and the C-26, respectively.

Figure 3.75 presents the average octave-band noise profile. Again the difference from the similar information for the A-1 and C-26 Generator Sets (Figs. 3.4 and 3.23) is most striking. The profile for the Consolidated MA-1 is not only much lower in magnitude but also is quite flat by comparison. The noise which is generated appears rather evenly distributed among the several octave bands. The average overall noise level is only 86.4 db, compared to 100.9 db for the C-26. Thus from a comparison of

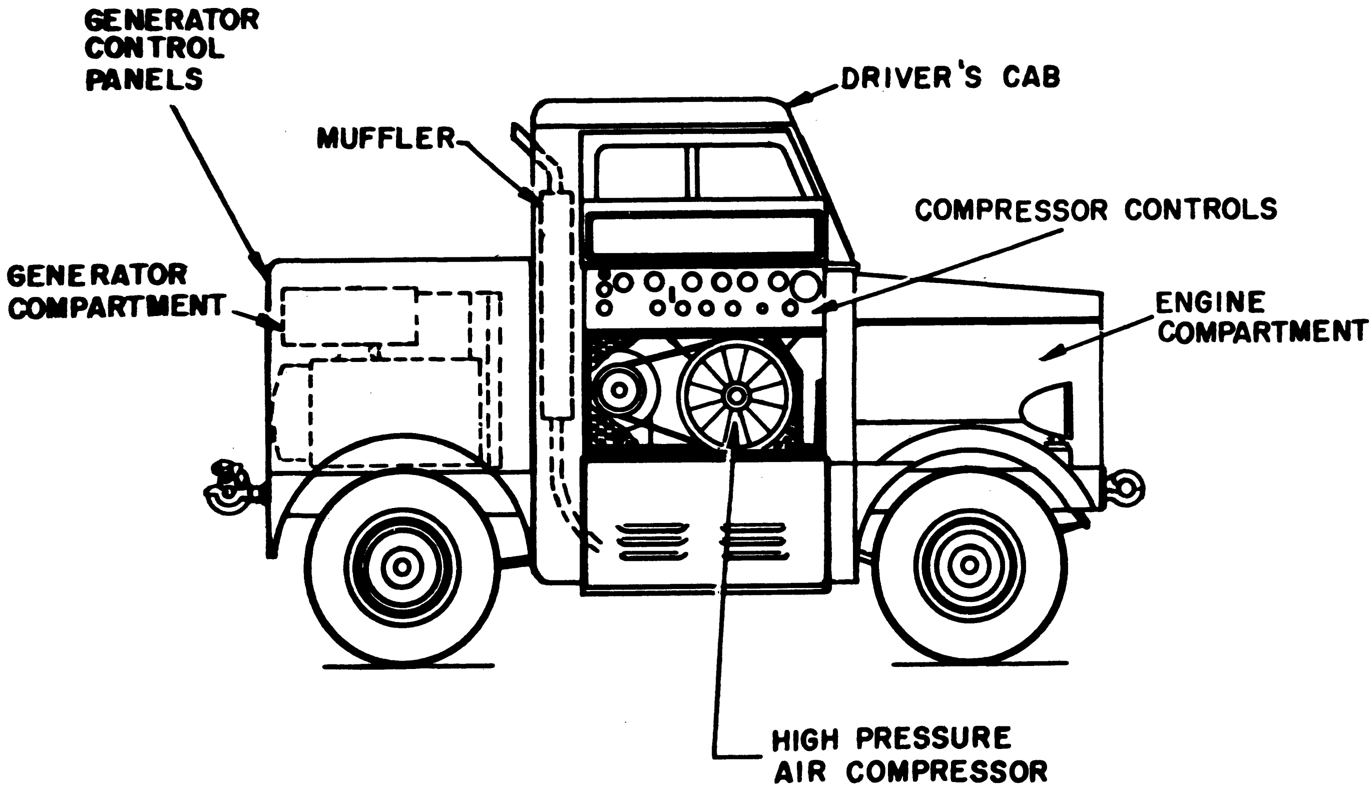



Fig. 3.70. Side view of Consolidated MA-1 multipurpose unit.

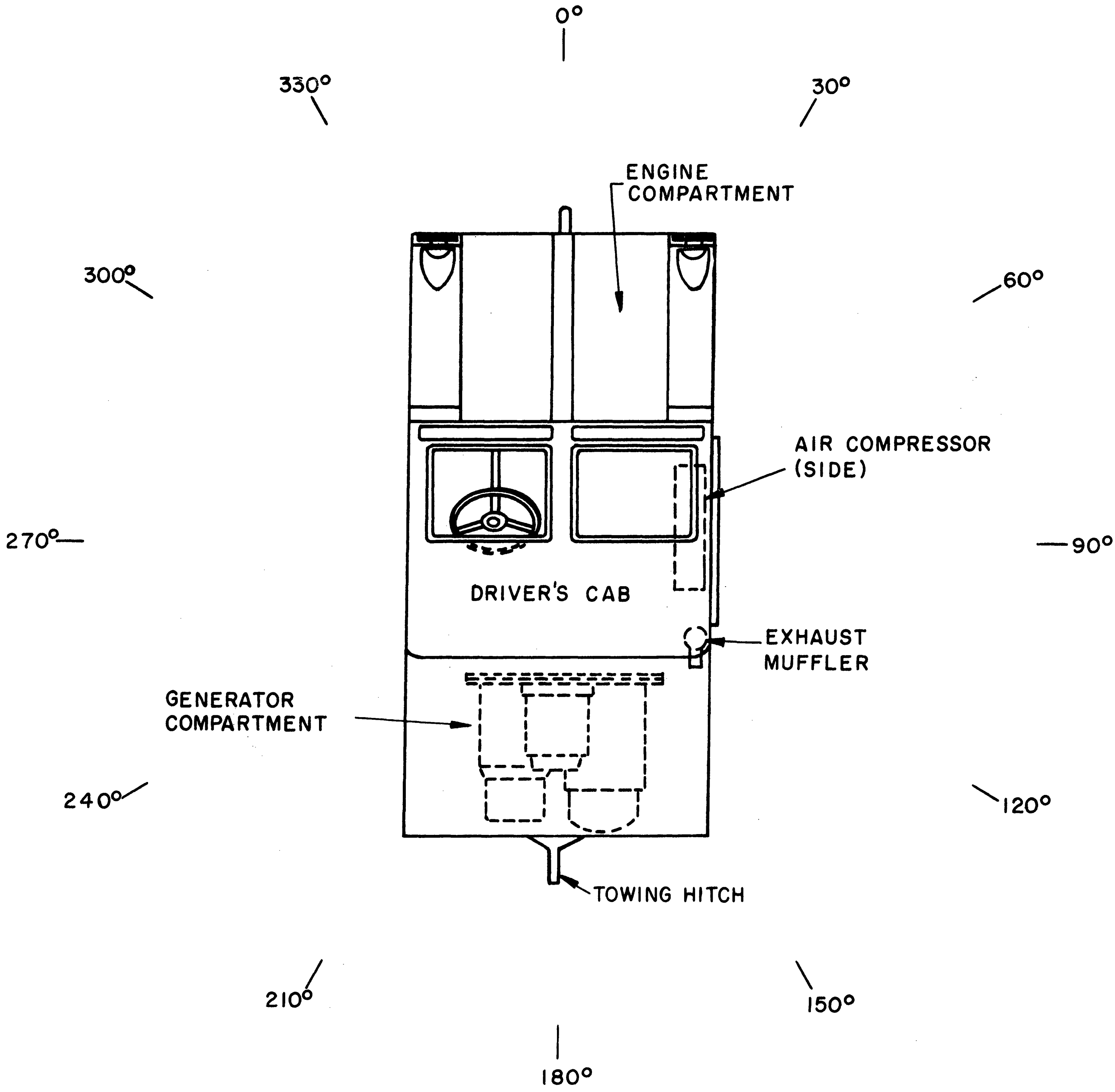
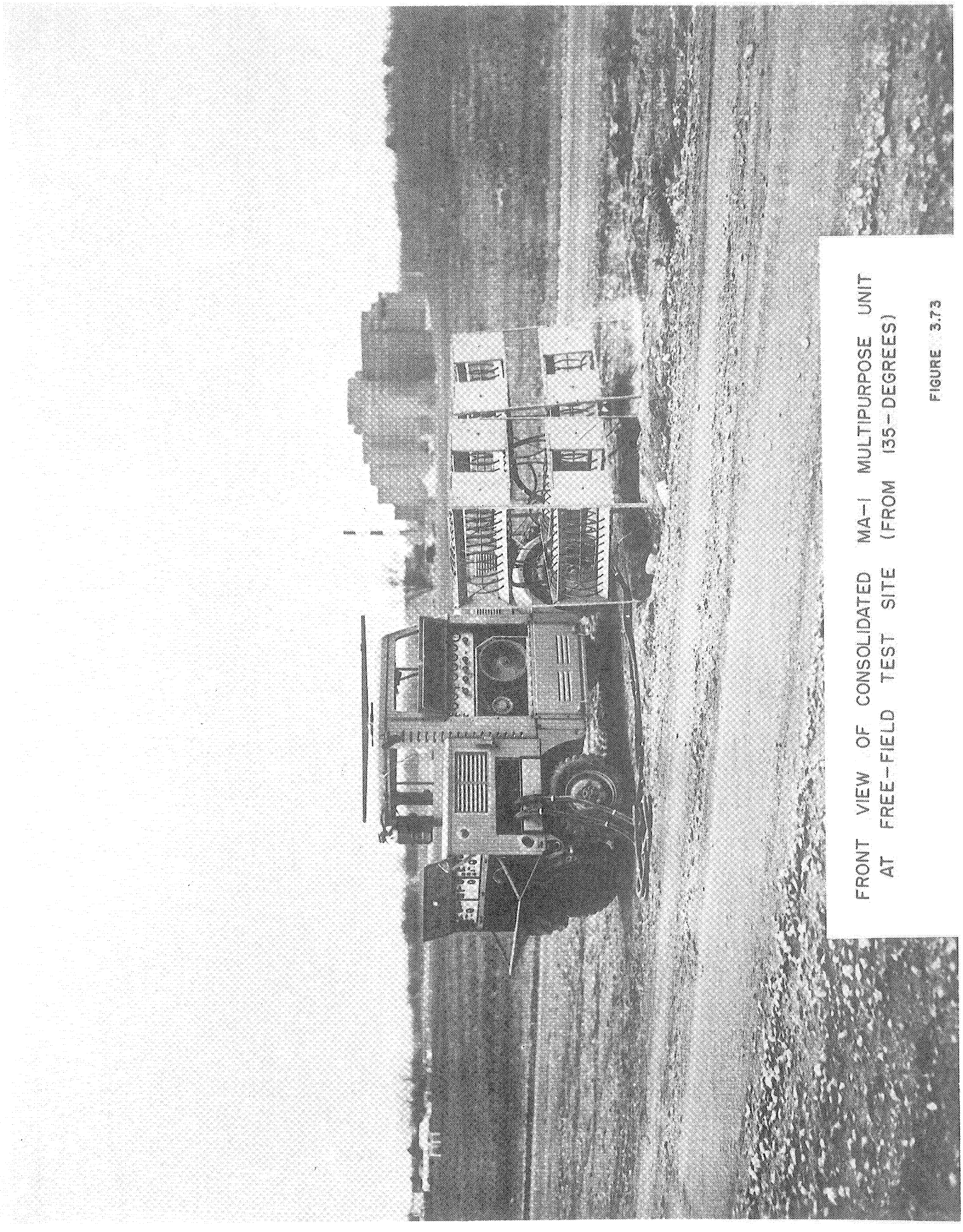



Fig. 3.71. Plan view of Consolidated MA-1 multipurpose unit.


FRONT VIEW OF CONSOLIDATED MA-I MULTIPURPOSE UNIT AT FREE-FIELD TEST SITE (FROM 315-DEGREES)

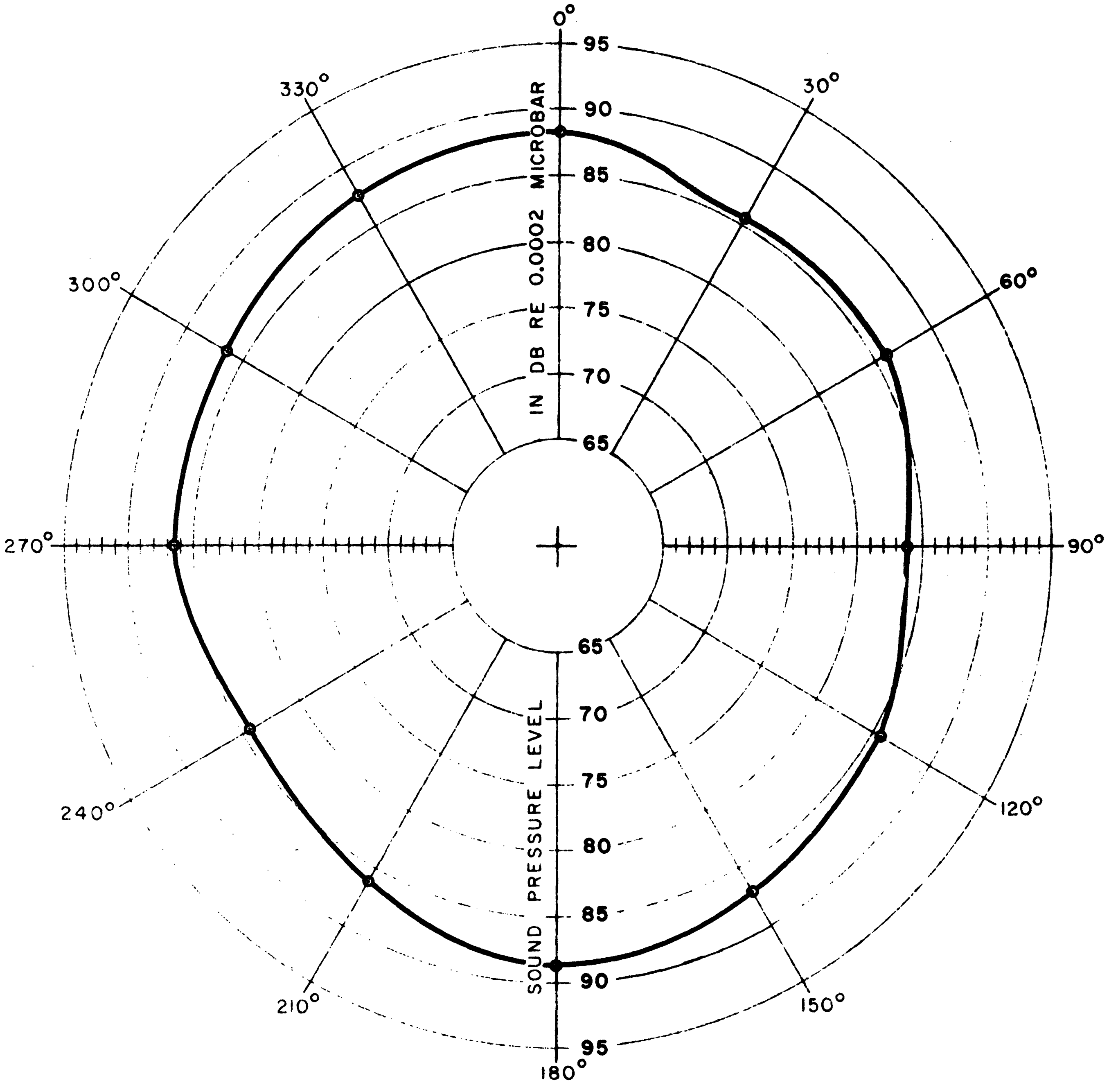
FIGURE 3.72



FRONT VIEW OF CONSOLIDATED MA-1 MULTIPURPOSE UNIT AT FREE-FIELD TEST SITE (FROM 135-DEGREES)

FIGURE 3.73

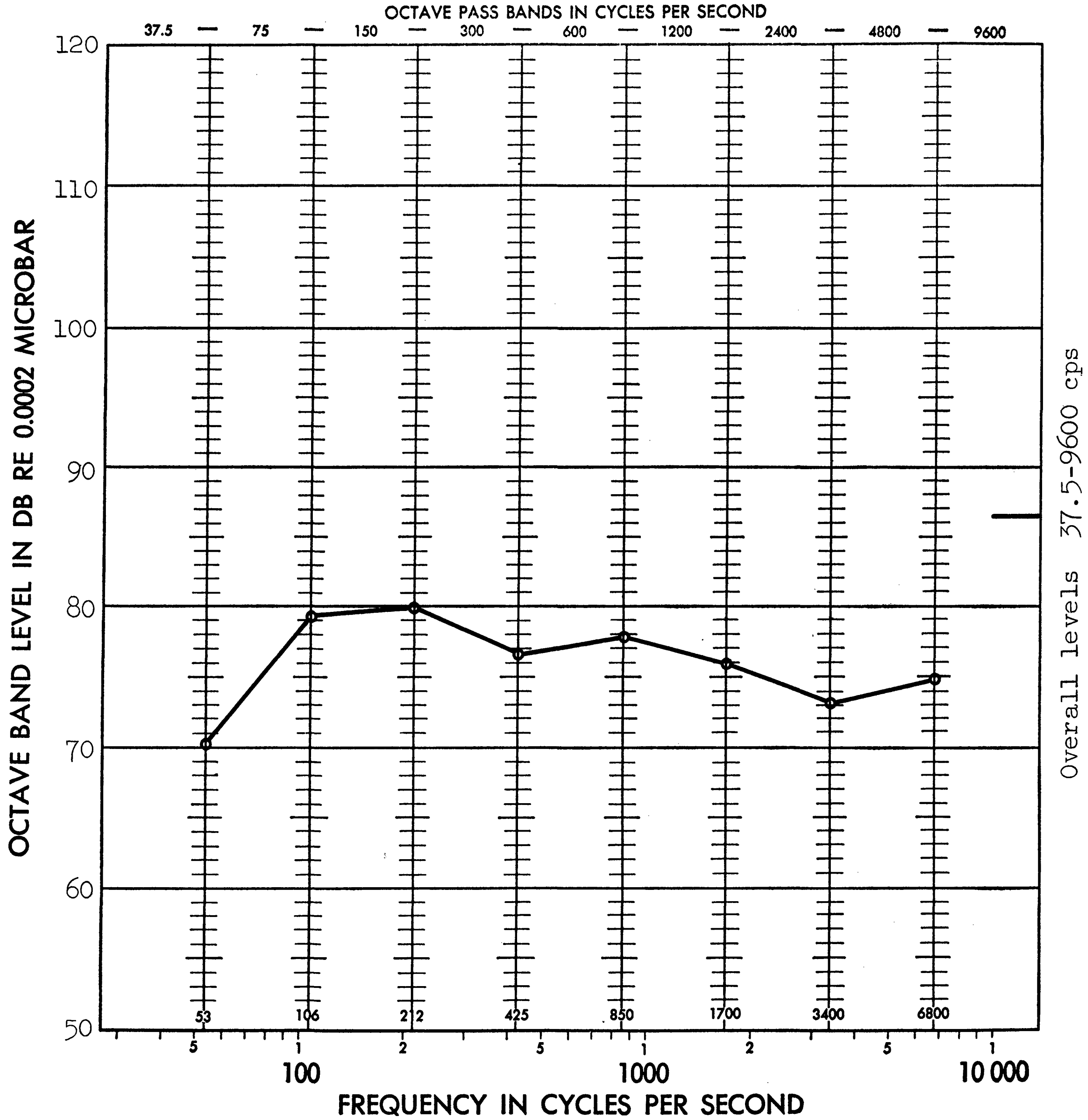
Free-field conditions
Microphone distance 40', height 5'5"
Computed overall sound pressure levels, 37.5-9600 cps




Consolidated MA-1 multipurpose unit, as-received condition
880-890 amps, 30.0-30.1 volts dc, 402-406 cps range of a-c power,
2950-3400 psi service line air pressure
Tested 18 November 1955

Fig. 3.74. Polar distribution of overall noise, Consolidated MA-1, as-received.

Free-field conditions
Microphone distance 40', height 5'5"
Average of 12 octave-band sound pressure levels




Consolidated MA-1 multipurpose unit
880-890 amp, 30.0-30.1 volts dc, 402-406 cps frequency range of a-c power, 2950-3400 psi service line air pressure
Tested 18 November 1955

Fig. 3.75. Average octave-band noise profile; Consolidated MA-1, as-received.

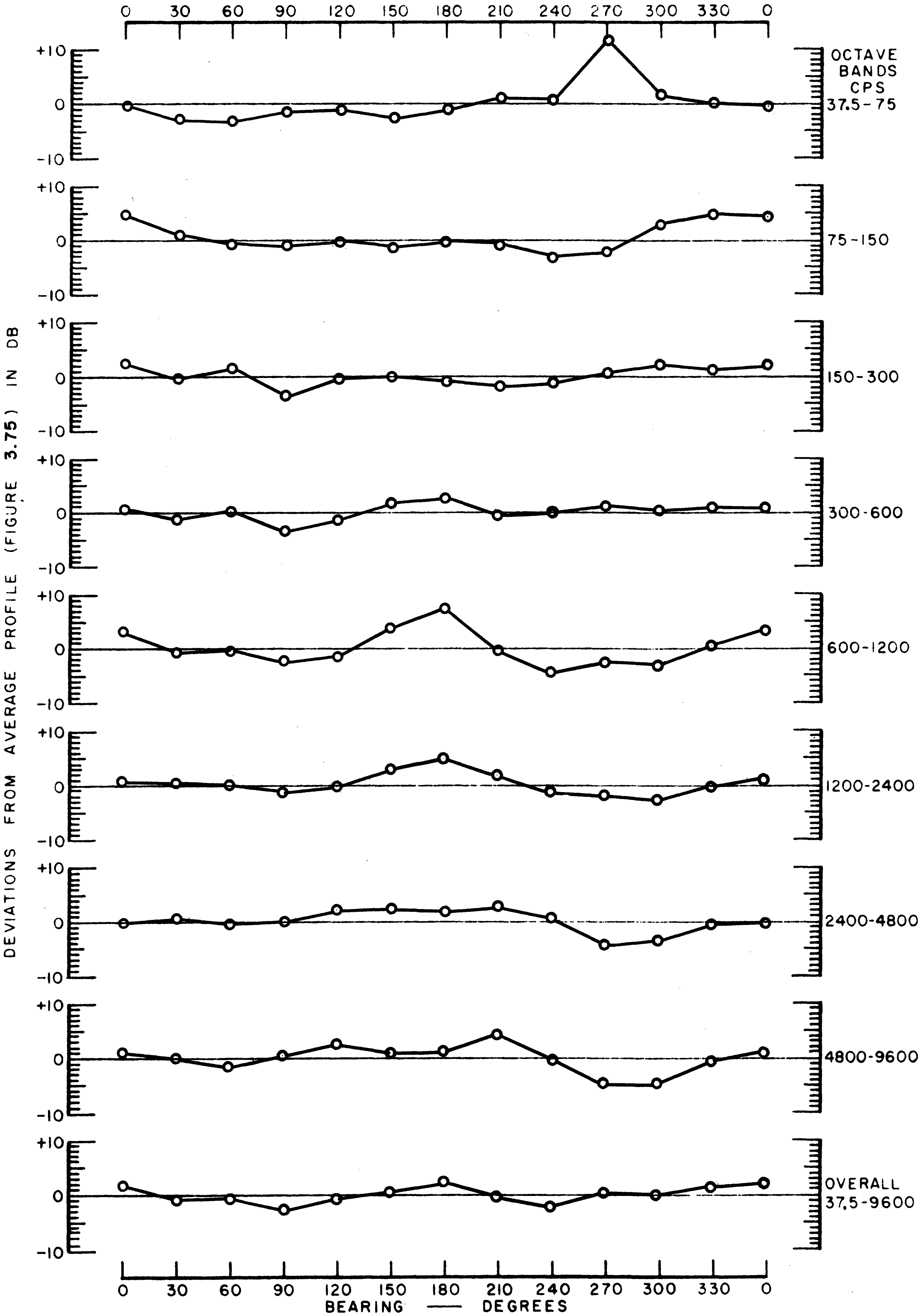



Fig. 3.76. Directional deviations from average profile; Consolidated MA-1, as-received.

sound pressures, this unit only makes around 20% as much noise as does the C-26 Generator Set when both noise units are measured in their "as-received" condition.

Figure 3.76, which presents the directional deviations from the average profile, shows several interesting features. The prominent rise in the 37.5-75 cps octave band at 270-degrees bearing cannot be readily explained. Since such sharp directionality at low frequencies is unexpected, the most likely but embarrassing interpretation is that the sound-level meter was misread and thus an error of 10 db was recorded in the original data.

The 75-150 cps octave band, in particular, shows marked level increases at bearings around the engine compartment. This is probably evidence of exhaust tones emanating from the engine compartment. In the opposite direction, increases in radiated noise are particularly noticeable in the 600-1200 cps octave band, and probably are the result of generator noise and/or generator-gear noise emanating from the generator compartment. Nothing of any interpretable consequence occurs at bearings around 90 degrees corresponding to the most likely direction for air-compressor noise. However, several of the high-frequency octaves show a pronounced dip in the 270-degree direction, which may be the result of acoustic shielding provided by the cab.

A close examination of the data reveals that the noise in the 75-150 cps and 150-300 cps octave bands predominates slightly at all bearings except the three toward the rear of the unit. At bearings of 150, 180, and 210 degrees, the noise in the 600-1200 cps octave band shows relative prominence.

Analysis of Initial Noise Problem.—Two features differentiate the noise radiated by the Consolidated MA-1 Multipurpose Unit from the noise radiated by the A-1 and C-26 Generator Sets. The noise levels of the Multipurpose Unit are appreciably lower than of the other units driven by reciprocating internal-combustion engines. In addition, the average profile is almost flat, not exhibiting the prominent low-frequency noise found in the surveys of the A-1 and the C-26. Both of these differences can be attributed to the installation of a reasonably effective muffler as part of the original equipment on the Multipurpose Unit. This muffler is mounted vertically in the corner of the cab behind the air compressor, and exhausts slightly to the rear and side near the top of the cab. Its position has been indicated in Fig. 3.70. This muffler has reduced the low-frequency exhaust tones associated with reciprocating engines until the resulting noise spectrum is almost flat. If the overall noise level of this unit, as judged on the basis of octave-band sound-pressure levels, is to be reduced still further, it will be necessary to attenuate all portions of the noise spectrum before much noticeable effect will be found.

A close study of the free-field survey data indicated that toward the front of this unit, the lower frequency octave bands, 75-150 cps and 150-300 cps had a slight tendency to predominate. Also, at the rear of the unit, the predominant noise shifts to the 600-1200 cps octave band. One may guess this to mean that the noise radiated from the front of the Multipurpose Unit probably originates principally from the engine, while that at the rear of the unit originates principally from the generators and the generator gear train. However, even if the above assumptions are correct, the data do not permit evaluating the significance of two possible sound paths. The sound may become airborne within the machinery compartments and propagate out through the louvers and open panels. On the other hand, the machinery may vibrate the body panels and surfaces, causing them to radiate. Thus, the excitation source, the energy transfer medium, and the resulting directionality of the airborne sound have yet to be correlated.

Some additional data were collected by tape-recording the noise at selected locations and analyzing these recordings with the Kay Vibralizer. Analyses from recordings made near the generators at the rear of the Multipurpose Unit revealed a series of prominent tones extending in frequency from about 1750 cps upward to about 4500 cps. Similar analyses of recordings made near the engine radiator grill showed strong tones in the range from about 200 to 450 cps, thus corroborating the predominance of noise in the lower frequency octave bands octave bands originally found in this direction. Plans for a more detailed inspection and investigation of this Consolidated MA-1 Multipurpose Unit with a view to formulating specific noise-reduction recommendations were interrupted by sudden notice of termination of the load period.

Recommendations.—Because definitive acoustic data are not available, detailed and specific noise-reduction recommendations cannot be formulated. However, a few suggestions, based on the limited data available, can be made, and a certain amount of guessing, based on previous experience, may be allowed.

Some further reduction of the low-frequency noise probably could be achieved by installing an even more effective muffler. Mufflers currently in use seldom reduce exhaust noise to insignificant levels. Beyond this, the noise at all frequencies could probably be reduced by developing the enclosure effectiveness more fully. If the noise is radiated into the air directly from the machine elements themselves, then it would be necessary to close up acoustically the panels and louvers and to add sound absorption within the housing. If the noise is radiated from the housing and cab panels after transmission to these areas as structureborne vibration, then it will be necessary to isolate the housing from the machinery by means of resilient mounts, and to damp the various panels to prevent resonant excitation. In all probability, both the airborne and structureborne paths contribute somewhat, and hence both types of treatment outlined above will be required to some extent.

The presence of a closed driver's cab on the Consolidated MA-1 Multipurpose Unit presents an opportunity to discuss briefly another aspect of noise control by means of enclosures. In the cases of the several machines discussed so far, the general idea behind palliative noise-reduction procedures has been to place the noise source within a "soundproof" enclosure. For this purpose, exhaust mufflers and intake silencers have been used to permit essential fluid flow in and out of the enclosure, and the enclosure itself has been resiliently mounted, its panels covered with vibration damping treatments, its walls designed to have considerable transmission loss, and sound absorption has been placed inside the enclosure to control reverberant buildup of sound. All these treatments were necessary simultaneously to a greater or lesser extent to enclose a noise source effectively, and thus isolate it from the surrounding space. However, these same techniques may be used to construct a quiet chamber so that the noise levels observed inside are much attenuated from those existing in the surrounding space. Thus by applying these various principles and treatments to the cab of the MA-1 Multipurpose Unit, it could be transformed into a portable haven against the high noise levels generated by aircraft during various field operations.

Whether "soundproofing" the cab of this or any other piece of ground-support equipment would have practical application in routine airport operations is not known. The discussion was presented here merely to emphasize that the techniques of palliative noise reduction are equally applicable to keeping noise in or out as the case may be. Furthermore, in both cases, a complete and balanced treatment is required. Only very limited effectiveness can be achieved by, for example, treating the air-borne sound paths and neglecting to consider the structure-borne vibration paths.

TYPE B10B GENERATOR SET AND MDX CART

The B10B Generator Set is a trailer mounted combination of a 4-cylinder air-cooled internal-combustion engine (Continental PE-90) and both a-c and d-c generators. In physical arrangement it is generally similar to the C-26 Generator Set but is is smaller. It can supply 30 kw of power at 208 volts, 400 cps. (Additional description and specifications of this unit were not immediately available when this report was prepared.)

The MDX cart referred to in some of the tests below was a new model generator set basically similar to the B10B but with a number of innovations. It was being fabricated by the Wolverine Diesel Power Company and was only partially completed at the time of these tests.

Measurement Site.—Although originally scheduled for free-field measurements, factory schedules for the B10B and MDX finally allowed time only for some non-free field measurements conducted in the delivery drive area of the Wolverine Diesel Power Company. The approximate layout of this measurement site is presented in Fig. 3.77, as drawn from memory, along with the two orientations of the B10B unit, labeled I and II, which were used during this test program. These testing locations and orientations were selected by ear to minimize the effects of constructive interference. Figure 3.78 identifies the eight microphone positions with respect to both the B10B and MDX units at which measurements were taken. The approximate positions and proportions of the various panel and control areas are shown, the exhaust ports being represented by the two cross-hatched circles in the engine cover panel.

By making octave-band measurements with the B10B in each of the two orientations shown in Fig. 3.77 with the microphone located at position 1, it is possible to demonstrate some of the effects of the non-free-field measuring site. Figure 3.79 presents the octave-band profiles obtained with the B10B operating at an a-c load of 116 amperes. Since the only change between the two sets of measurements was the machine's orientation with respect to its surroundings, the discrepancies which range from 1.4 to 10 db must be assigned to the influence of the acoustic environment. Moreover, there is no assurance that the maximum effects have been observed at these two orientations.

Under conditions such as those demonstrated above, where the measured sound levels are so profoundly affected by the acoustic environment, neither the absolute levels measured nor the difference in levels between different octave bands have any unique interpretable significance. Thus neither the height of a profile nor the shape of profile has any real meaning here. The only valid comparisons which can be drawn are the differences in level of the same octave band for two different operating conditions of the same machine in the identical orientation and configuration. Even in this case, one must avoid attaching too much significance to such differences since unexpected and uncontrolled factors can often influence the behavior of the sound field.

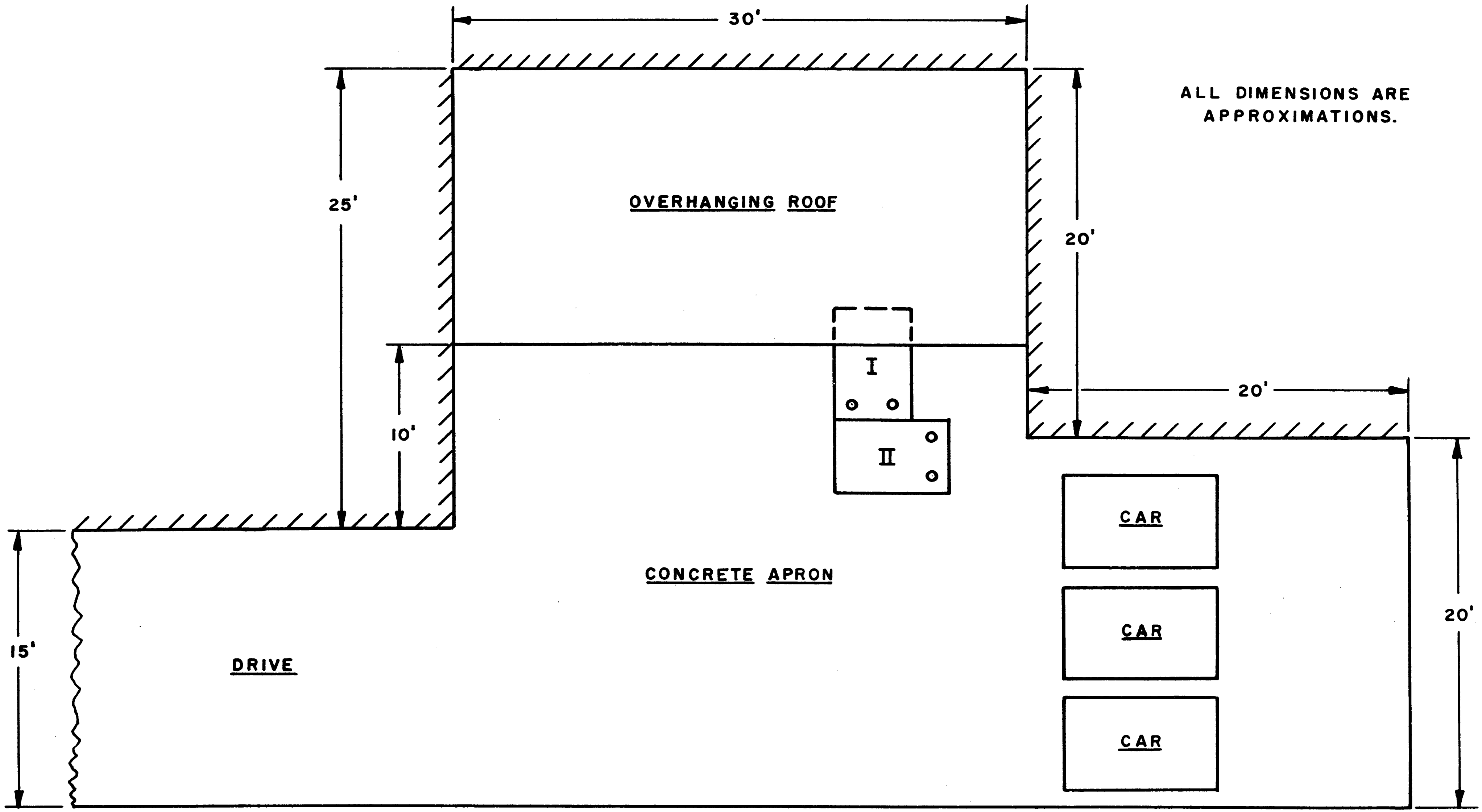



Fig. 3.77. Wolverine diesel B10B at plant-noise-survey measuring site.

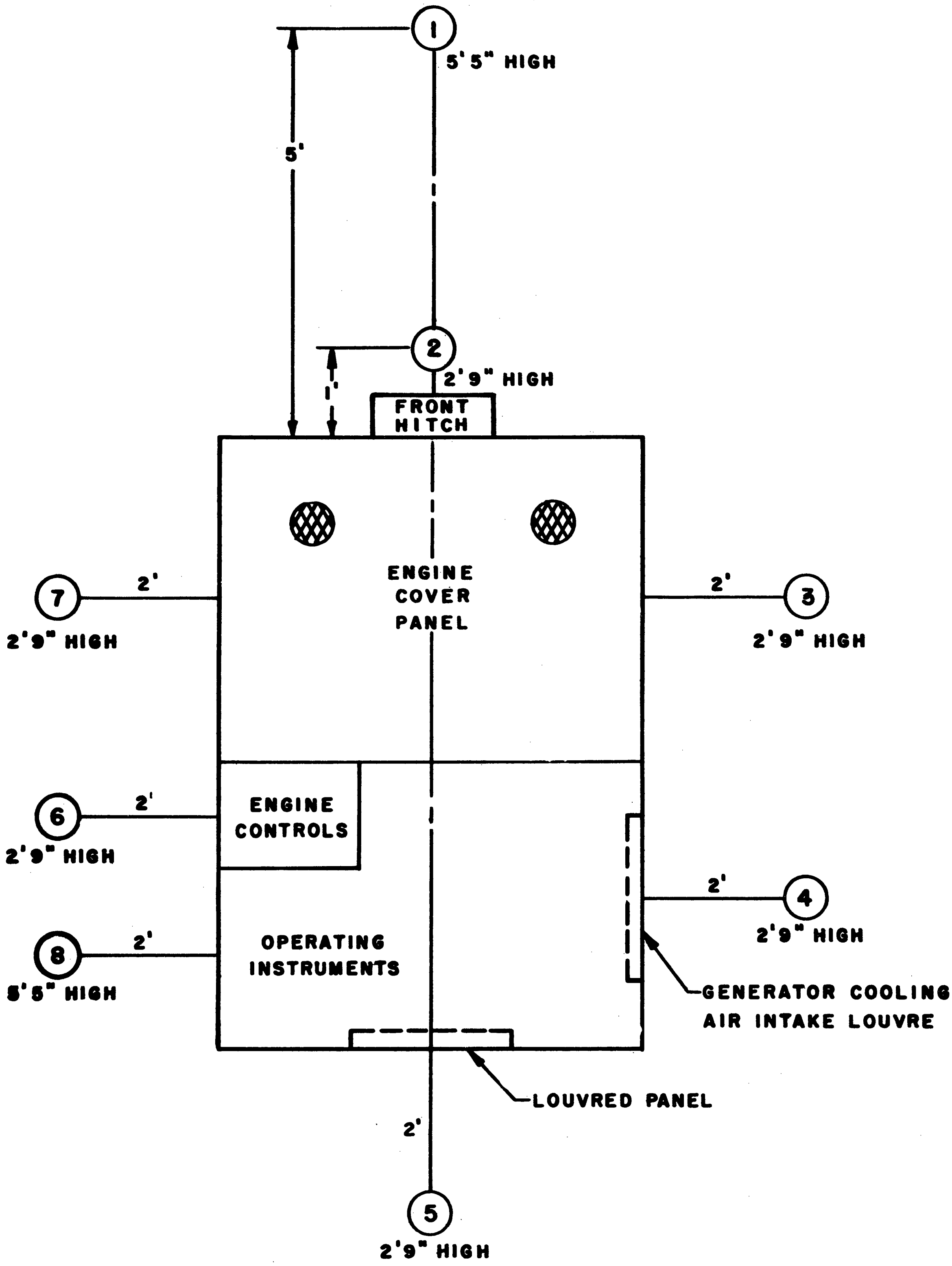
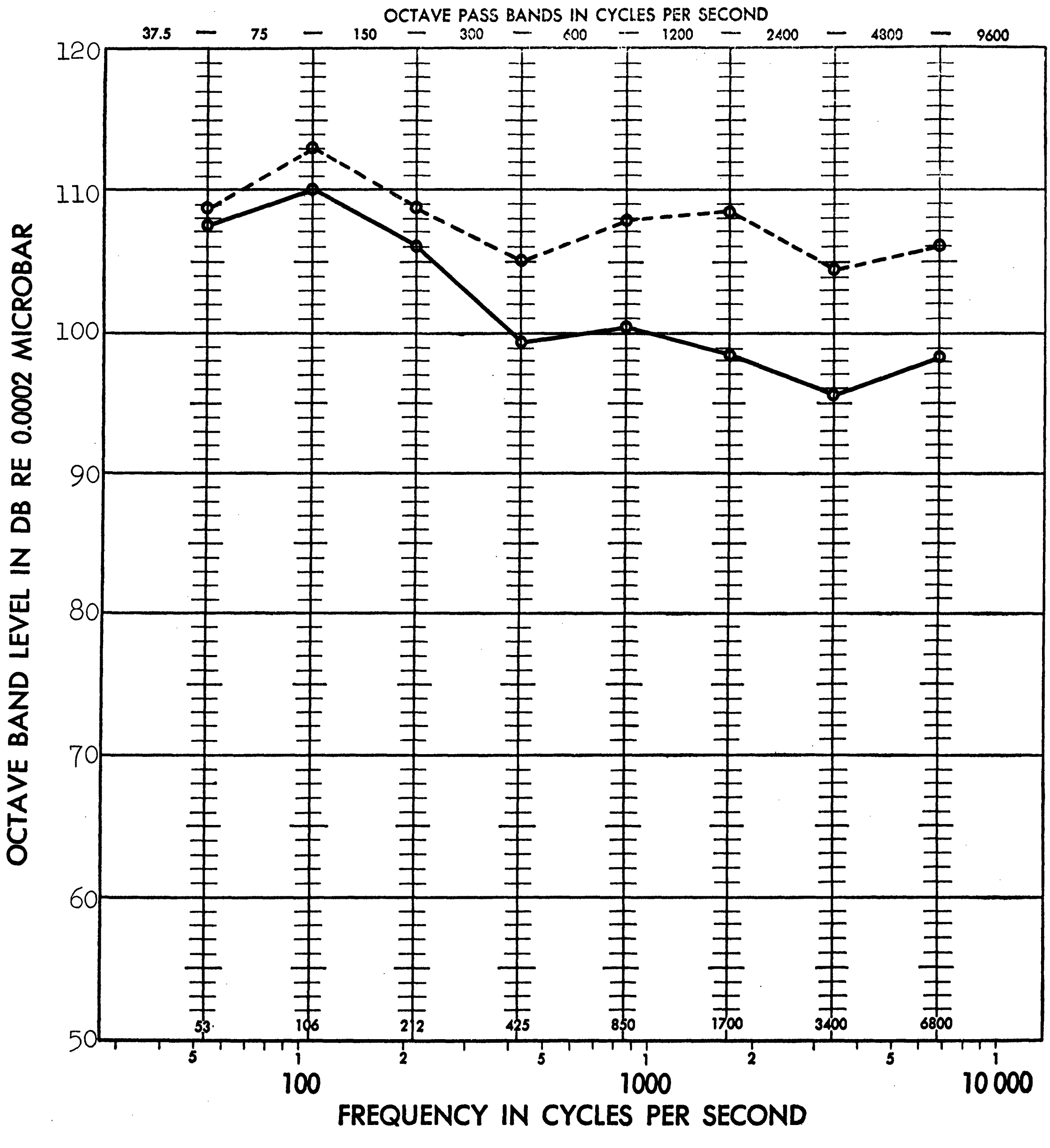



Fig. 3.78. Wolverine diesel B10B at plant-noise-survey microphone positions.

Nonfree-field conditions
Microphone position 1 (distance 5', height 5'5"), see Fig. 3.78




B10B generator set, 116 amp, 208 volt a-c load
--------- Orientation I, see Fig. 3.77
——————— Orientation II, see Fig. 3.77
Tested 22 November 1955

Fig. 3.79. Comparison of noise profiles for two orientations of B10B generator set.

Effect of Load.—Figure 3.80 shows the change in acoustic output which results when a 116-ampere load is applied to the B10B as measured at seven microphone positions. Comparison data for the no-load condition were not collected for microphone position 1. The reader is cautioned again to consider only differences and not the heights or shapes of the individual plots, even though they do resemble typical results for this type of machinery. The differences observed in Fig. 3.80 consistently show that the B10B generates increased noise when loaded. Furthermore, these differences seem to show that the increase in noise occurs approximately equally in all octave bands and at all microphone locations. Figure 3.81 presents similar load and no-load data for microphone position 1 with the B10B in orientation I.

Exhaust Aspirator Tests.—It was desired to evaluate the acoustic effectiveness of a newly designed exhaust aspirator. Figure 3.82 shows schematic cross sections of both the three-stage aspirator which is standard equipment on the B10B and of the new four-stage aspirator. This four-stage unit incorporates an interference tube assembly and swirl vanes, and is intended for possible use in the new MDX cart. Notice that the engine compartment cover forms the final stage of each aspirator. Because of the non-free-field testing condition, the two aspirators were to be compared when attached to the B10B. However, the four-stage interference-tube aspirator would not fit within the B10B housing, so it was necessary to remove the B10B's engine cover to permit installation, thus reducing this four-stage aspirator to a three-stage aspirator.

Because of the engine cover removal, a comparison of the noise generated by the normal B10B with and without the engine cover panel (thus effectively equipped with a three-stage and then a two-stage aspirator) was carried out as a preliminary test. The differences between the curves in Fig. 3.83 show the effect of removing the engine cover panel on the noise measured at two microphone positions for both load and no-load conditions. Changes certainly did occur but interpretation of the results obtained from these limited data is difficult. One is tempted to suggest, for those cases where removing the cover resulted in lower noise levels, that the cover was vibrating and radiating excess noise. However, remembering the non-free-field acoustic environment, the changes observed may be largely the results of changing the acoustic interactions at those particular microphone positions. A change such as removing a cover panel is too drastic to allow the assumption of a negligible change in the detailed acoustic environment.

Figure 3.84 shows the effect of the changing load at two microphone positions with the two aspirators installed in the B10B minus its engine cover panel. The increases in noise are similar to those observed for the normal B10B. Figure 3.85 presents a comparison of the behavior of the two different aspirators (engine cover panel removed) at the same microphone positions and under the same load conditions. These last comparisons are acoustically less valid since the physical geometry of part of the source configuration has been changed, and it is impossible to state with certainty from these data alone whether one aspirator is really better than the other.

Figure 3.86 shows the effect of a load change on the noise output of the MDX unit which had the new aspirators installed. Since the engine cover was now in place, the new aspirators were in their normal, complete four-stage configuration. The rest of the MDX cart was somewhat open because a number of the cover panels had not yet been installed. For these measurements, the MDX cart was placed in orientation II and microphone position 1 was 5 ft from the end of the cart just as indicated in Fig. 3.78. However, because of the increased height of the MDX cart relative to the B10B, the

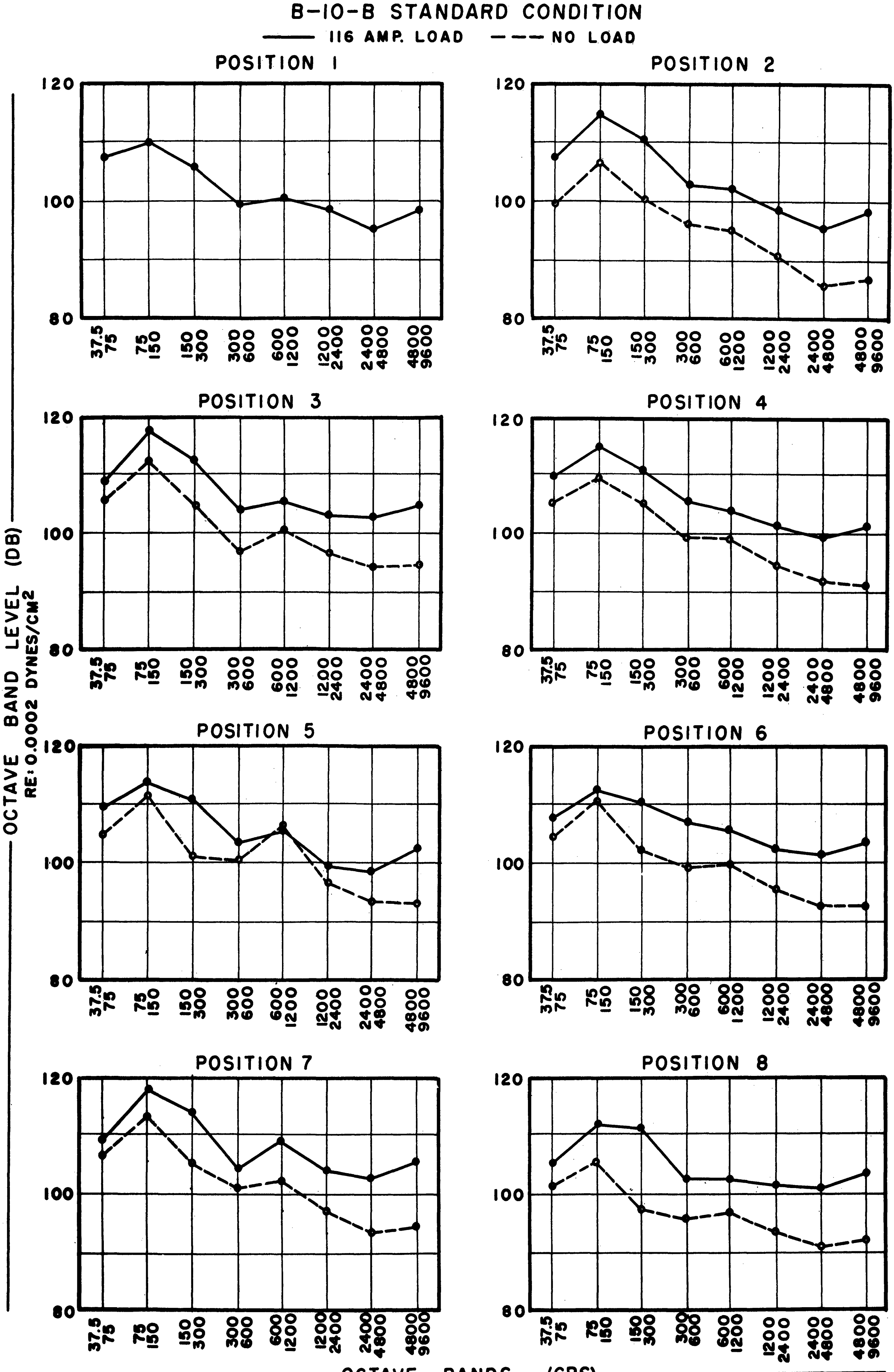
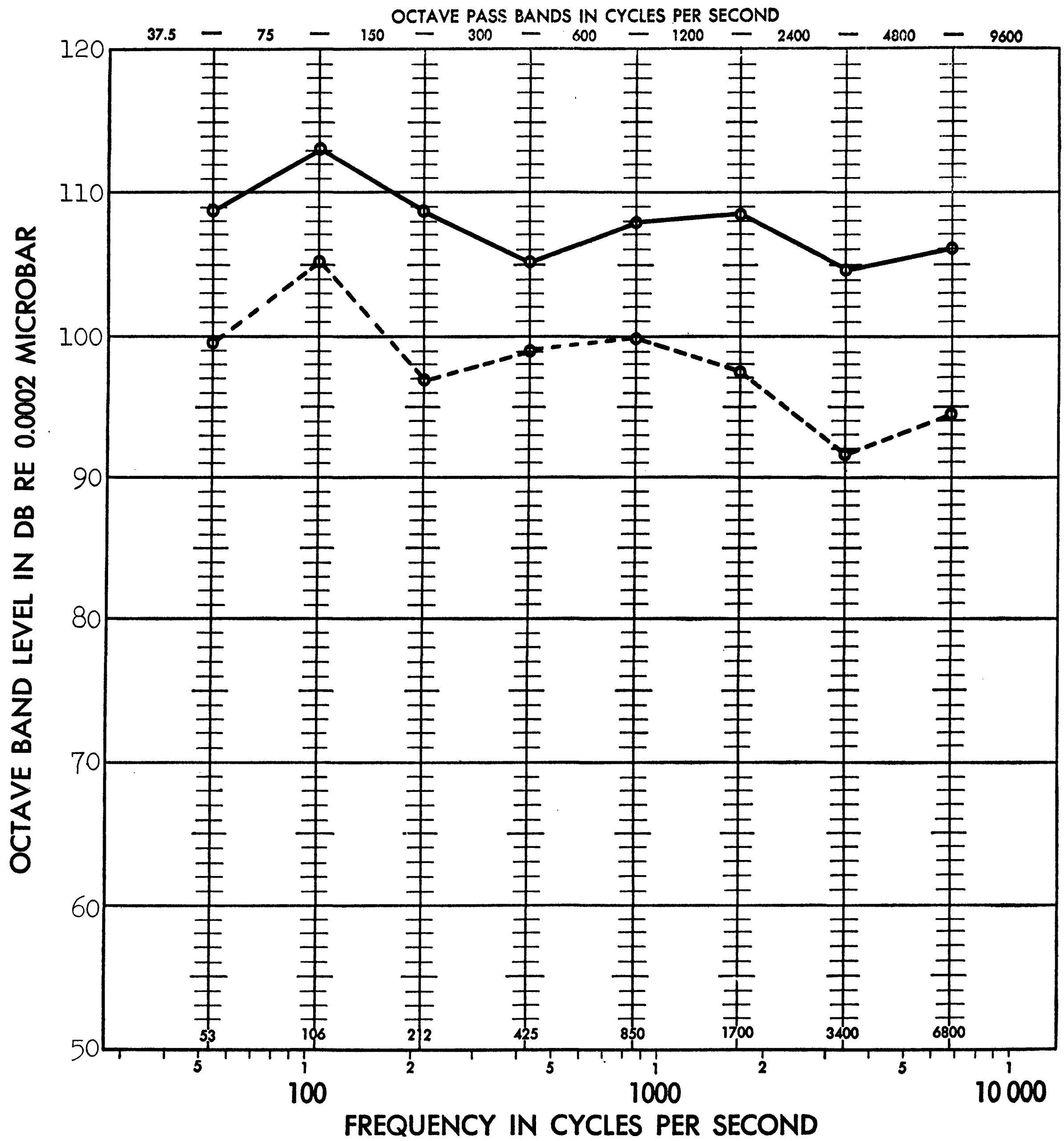



Fig. 3.80. Effect of load on noise generation.

Nonfree-field conditions
Microphone position 1, orientation I, height 5'5", see Figs. 3.79 and 3.80




B10B generator set equipped with 3-stage aspirators and with housing closed

--------- No load

————— 116 amp 208 volt a-c load

Tested 22 November 1955

Fig. 3.81. Effect of load on noise generation, orientation I; B10B generator set.

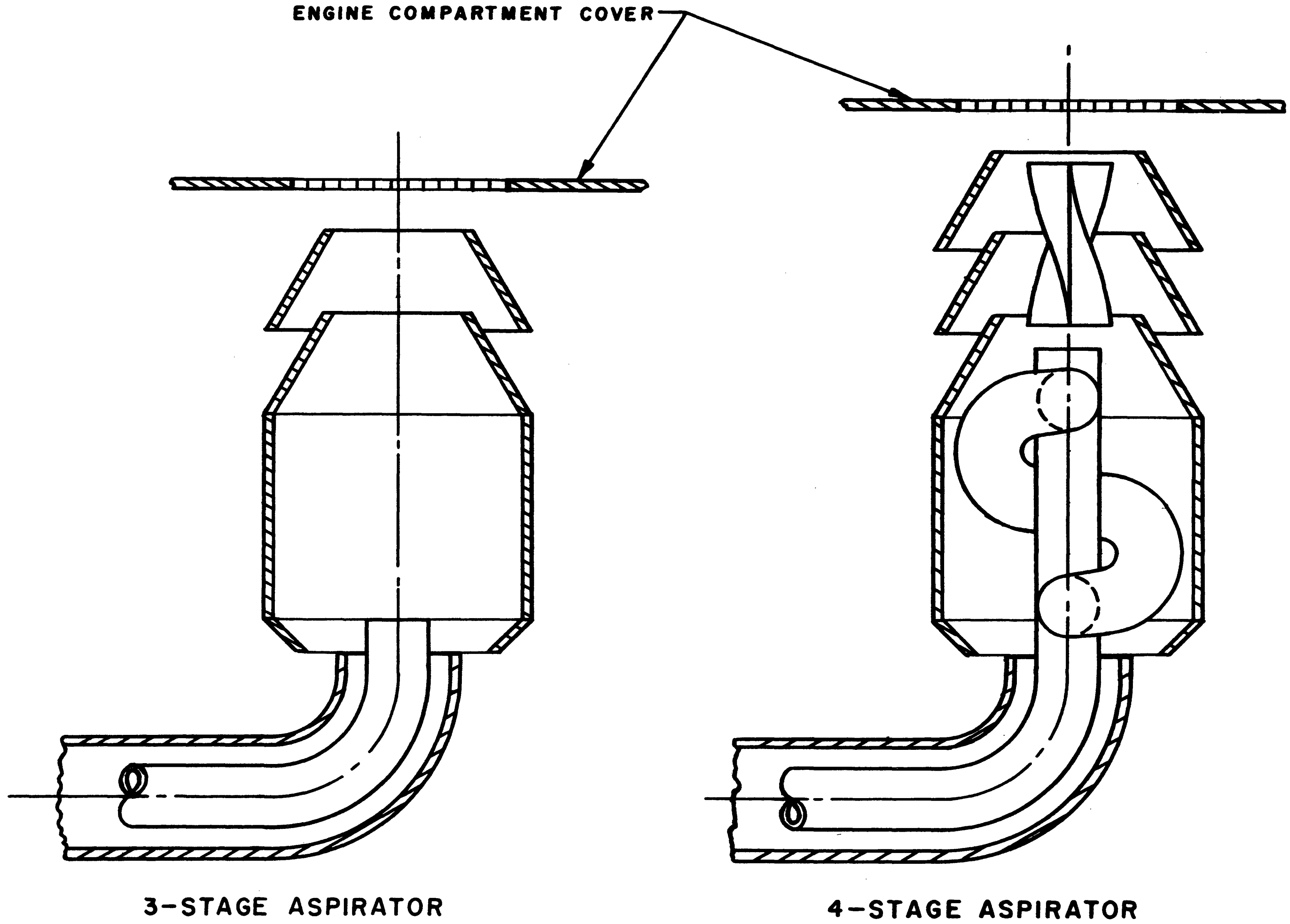
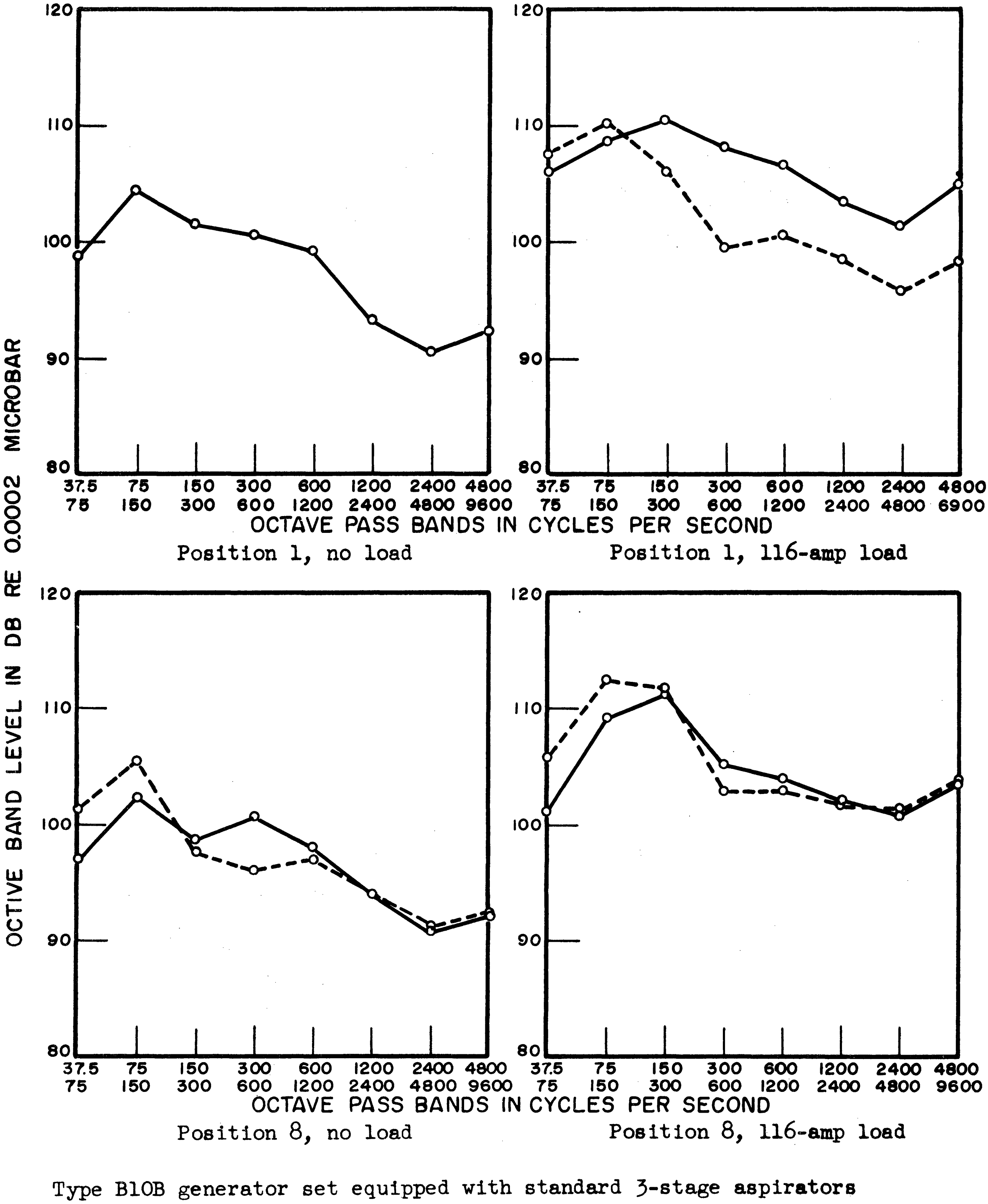



Fig. 3.82. Schematic cross sections of aspirators; B10B, MDX.

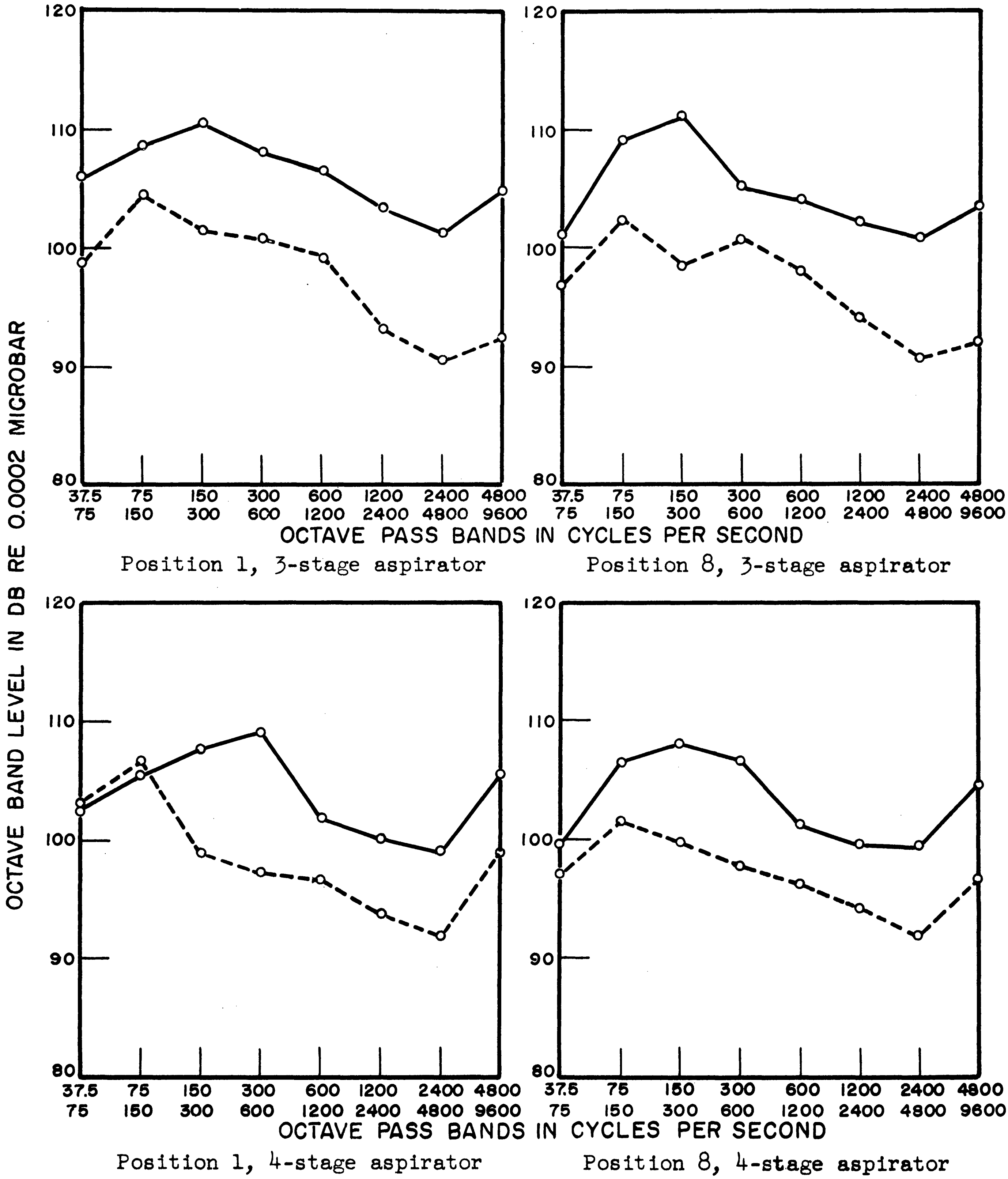
Nonfree-field conditions
Orientation II, microphone positions 1 and 8, See Figs. 3.77 and 3.78




Type B10B generator set equipped with standard 3-stage aspirators
------ Engine cover installed
———— Engine cover removed
Tested 29 November 1955

Fig. 3.83. Effect of engine cover removal; B10B.

Nonfree-field conditions
Orientation II, microphone positions 1 and 8, see Figs. 3.77 and 3.78

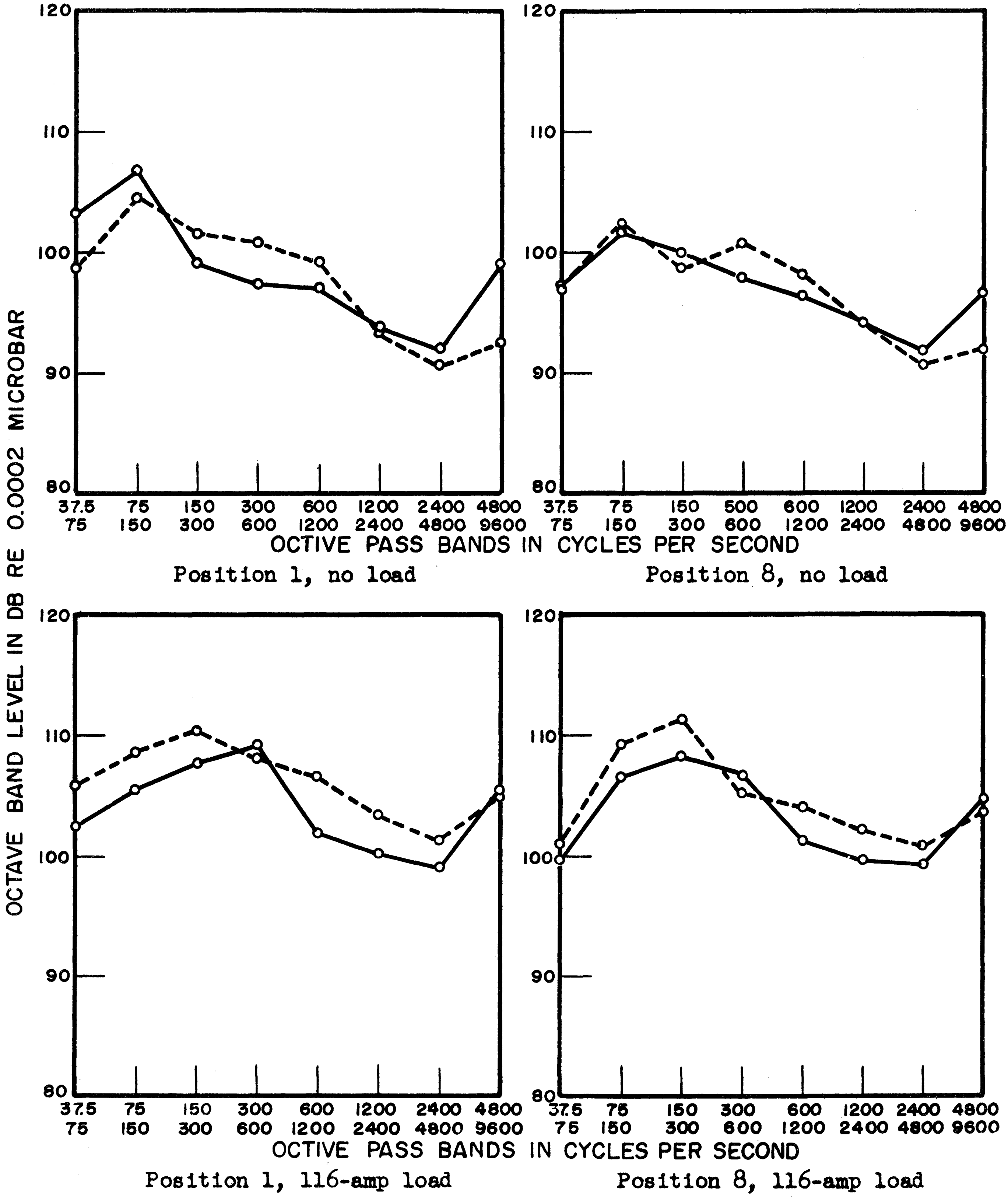



Type B10B generator set with engine cover removed
------ No load
———— 116-amp load
Tested 29 November 1955

Fig. 3.84. Effect of changing load for two different aspirators; B10B.

Nonfree-field conditions

Orientations II, microphone positions 1 and 8, see Figs. 3.77 and 3.78




Type B10B generator set with engine cover removed

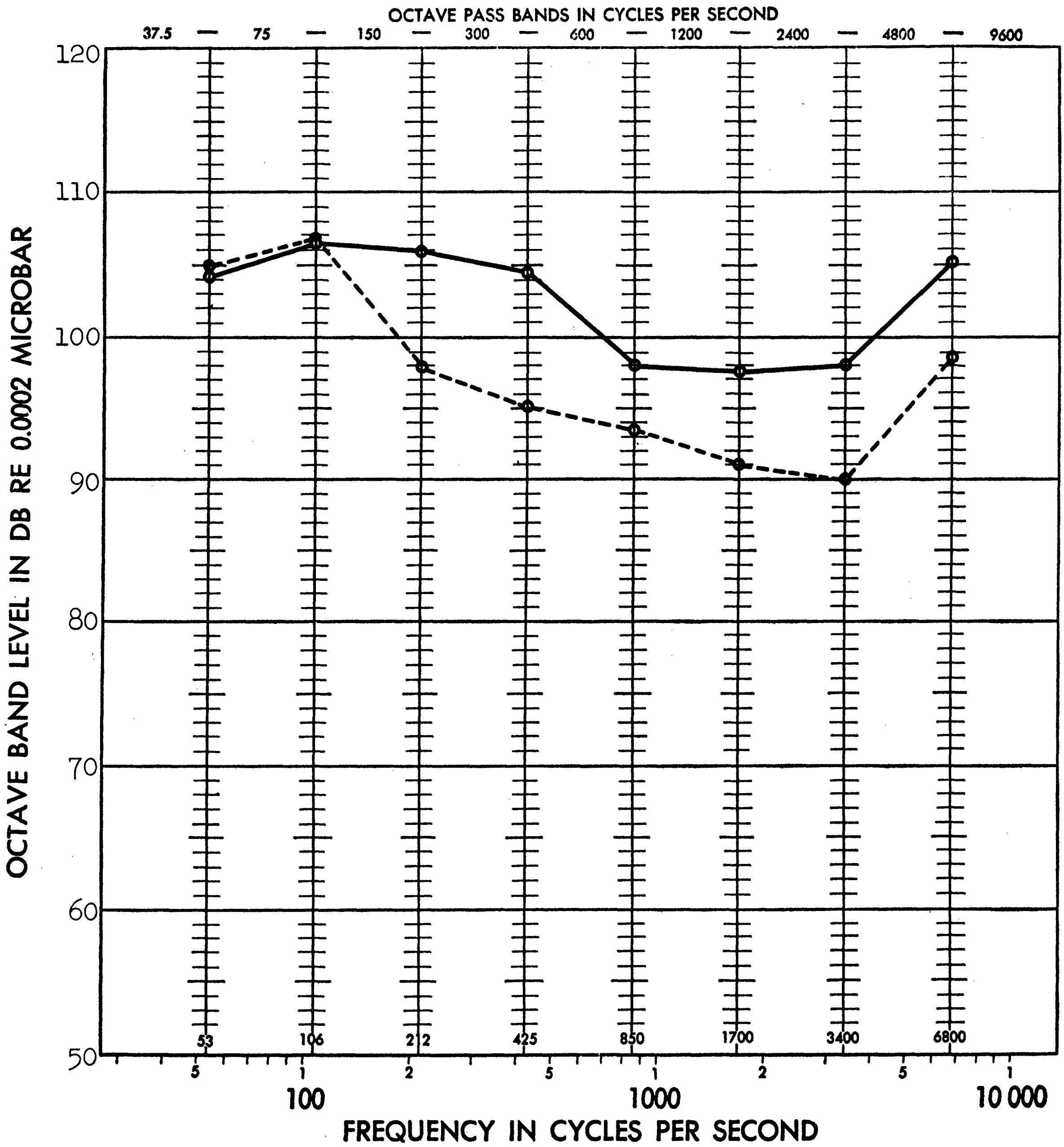
----- Standard 3-stage aspirators

——— New 4-stage aspirators

Tested 29 November 1955

Fig. 3.85. Comparison of two different aspirator designs; B10B.

Nonfree-field conditions
Microphone position 1 (distance 5', height 6'3"), orientation II, see Figs. 3.79 and 3.80




MDX generator set equipped with 4-stage aspirators but cart enclosure not complete

--------- No load

————— 116 amp, 208 volt a-c load

Tested 22 November 1955

Fig. 3.86. Effect of load on noise generation, orientation II, MDX unit.

microphone was raised 10 in. to 6 ft 3 in. above the ground.

Because a new design of aspirator was being tested, it is natural to ask how it differs in acoustic performance from the original three-stage design, and if one type is to be preferred over the other for noise-reduction purposes. Unfortunately, an unequivocal answer cannot be obtained on the basis of the limited non-free-field data obtained at the Wolverine Diesel Power Company. (Sometimes it is possible to make such comparisons on a rigorous basis under such non-free-field conditions, but a very much more elaborate, statistically controlled experimental procedure must be used.) Since no very dramatic changes in noise occurred during tests involving the two different aspirator designs, one is led to the conclusion that either the acoustical performance of these two aspirators is actually very similar or the unavoidable alterations of the acoustic environment concealed the real differences at the microphone positions employed. The fact that most tests had to be run with one stage of each aspirator removed further invalidates comparisons of the overall effectiveness of these aspirators.

## GENERALIZED ENCLOSURE CONCEPT

In the course of the palliative noise-reduction research described above, emphasis has been placed on containing noise energy and harmlessly dissipating it in the form of heat. The discussions, however, have been presented principally in terms of specific treatments applied to specific machines. Therefore, it is appropriate to develop here the pertinent aspects of the enclosure concept in general terms unencumbered by numerous extraneous details.

Throughout this general discussion, various basic acoustical control mechanisms, e.g., sound absorption, will be mentioned. An extensive technology regarding each of these mechanisms exists, and is thoroughly discussed in the literature (see Section V of this report). No attempt will be made to duplicate such discussions or even to summarize them, for the most rudimentary concepts concerning these several basic acoustical control mechanisms will suffice for this presentation. The sole aim here will be to develop a line of reasoning which will explain when and to what extent each of these control mechanisms is involved in achieving a balanced and consistent palliative treatment.

An immediate thought when dealing with an offending noise source is to enclose it in a "soundproof" box. By doing so, all acoustical space is divided into two regions: a small region enclosing the offending source, and the vast exterior region now free of offending noise. However, the practical attainment of a "soundproof" enclosure requires the consideration of several basic mechanisms of acoustical control and their interrelationships.

Actually, a noise source can radiate either directly into the air or indirectly by communicating mechanical vibration to some remote radiating surface. Thus, to design a satisfactory acoustic enclosure, it is necessary to consider means of providing both airborne sound isolation and structure-borne sound (vibration) isolation. For convenience, these two aspects of the problem will be discussed separately, although, of course, in any real problem both types of isolation must be provided concurrently.

Airborne Sound Isolation.—Airborne sound isolation (or insulation) as used here implies the control of the airborne sound by means of a wall or partition impermeable to the airborne sound. The corresponding measure of effectiveness is expressed as the transmission loss (usually in decibels) of the wall and is the difference in sound-pressure level which can be maintained across the wall. Single, homogeneous partitions behave according to "weight law" to the first approximation.[2] That is, the transmission loss of such a partition is proportional to the mass per unit area of the partition and also is proportional to frequency, an increase in either parameter resulting in enhanced transmission loss. Thus by estimating the amount of attenuation desired, it is possible to select the appropriate weight of material from which to construct the enclosure.

When the noise source is placed within an enclosure so designed, the noise measured outside is apt to be found just about as loud as without the enclosure. The walls of the enclosure do indeed provide the predicted transmission loss but the noise level has increased inside the enclosure until that level diminished by the transmission loss of the walls is just as high outside the enclosure as was the case without the enclosure. This occurs because no mechanism for the conversion of sound energy into heat energy has been included within the enclosure. When such a mechanism of dissipation is provided, then the internal buildup of noise will not occur and a reduction of noise will be observed in the exterior space. Thus the inclusion of sound-absorbing materials within a "soundproof" enclosure is established as essential.

It may be found, however, that even wnen an appropriate weight of wall material is used and when adequate sound absorption has been provided within the enclosure, certain sound frequencies leak out and are not attenuated as much as expected. This is the result of resonances in the wall panels of the "soundproof" enclosure, a phenomenon not taken into account in the simplified "weight law" theory of partition transmission loss. At such resonant frequencies, the partitions become essentially transparent to sound. These resonances, however, may be controlled by incorporating a damping mechanism into the partition structure.

Since most structural materials do not possess sufficient inherent damping to suppress resonant transmission of sound, vibration damping treatments applied to their surfaces are required. Just as in the case of sound absorption, an extensive technology exists concerning damping treatments.[2] Sometimes it is possible to combine advantageously the acoustic control functions of sound absorption and vibration damping into a single composite treatment.

Structure-Borne Sound Isolation.—To provide effective enclosure of a noise source, it is necessary to block the indirect escape of noise which occurs when noise energy is transmitted as structure-borne vibration to remote radiating surfaces. This vibration isolation is accomplished by means of resilient mounting which effectively reflects the vibrational energy back to the source.[2]

It is particularly important to ensure that the vibration does not reach the enclosure's walls since they represent almost ideal radiating surfaces. Although it would be best to avoid any mechanical contact, even that contributed by soft resilient mounts, between the machine and its enclosure, this is hardly practical in most real cases. Thus it remains necessary to mount the noise source and/or the enclosure resiliently.

Even with well-designed resilient isolation, certain frequencies may leak through and be radiated from the enclosure panels. Generally this too is the result of panel resonances (assuming undamped panels) excited by the small but nevertheless finite energy transferred by the resilient mounts. Just as in the case of airborne sound excitation, these resonances can be controlled by appropriate damping treatments.

Integrated Enclosure Design.—The above arguments have established that to frustrate the escape of direct airborne sound by means of a "soundproof" enclosure, satisfactory transmission loss can be assured only if adequate sound absorption and vibration damping treatments are employed concurrently. At the same time, it is also necessary to disrupt the indirect escape path by means of resilient mounting also assisted by vibration damping. Thus all four basic control mechanisms, sound isolation, sound absorption, vibration isolation, and vibration damping, must be utilized to achieve an effective enclosure design. Furthermore, the omission of any one of these four control mechanisms could, in any given case, seriously jeopardize the effectiveness of the enclosure. The extent and the effectiveness required of each treatment is, of course, precisely the problem of achieving a balanced design just as in any other engineering design endeavor.

There are other factors which in practical cases may constitute limitations to the acoustic effectiveness of an enclosure. Real machines, if they are to perform useful work, may require openings in the enclosure to permit them to function properly. For example, an internal-combustion engine requires both intake air and exhaust openings. The problem thus arises how to allow these essential openings through the enclosure walls without causing acoustical leaks. In such cases, various combinations of intake silencers, exhaust mufflers, lined ducts, absorptive baffles, etc., may be used to maintain acoustical control over the physical openings through the enclosure walls.

With current technology, however, it appears that openings into the enclosure, even when acoustically closed by means of silencers, mufflers, etc., will probably establish the upper limit of enclosure effectiveness in many cases. Thus, to achieve a well-balanced enclosure design, it is probably necessary to use the most effective mufflers available and to design the enclosure itself to provide comparable attenuation.

Enclosure Applied to Aircraft Ground-Support Equipment.—The parallel between the concept of a machine in an acoustical enclosure and an item of ground-support equipment in its housing is fairly obvious. Although the housings of the ground-support equipment now constitute only physical protection for the internal machinery, the idea exists that perhaps with moderate alteration or redesign these housings might be transformed into true acoustical enclosures. Therefore, by suitable housing design, the noise problem might be alleviated without drastic changes in the size, weight, or function of the offending machine. There appears to be no fundamental reason why this cannot be accomplished, although a comparison of the acoustic enclosure as described above with the existing physical housing of an item of ground-support equipment shows very clearly that it is not simple to transform an existing housing into a highly effective acoustic enclosure.

Actually to realize an enclosure of highest acoustical effectiveness, it is probably necessary to design a completely new housing to enclose the machine, even though the same general external appearance is maintained. Even so, this palliative approach is far simpler and more economical than noise-reduction-at-the-source techniques since the machinery can be transferred as a whole into the new acoustical enclosure. Thus the problem of redesigning an existing and proven machinery mechanism or of altering its function does not arise.

A point of considerable design interest is the one of resiliently mounting the machine and/or its housing. The question may be raised as to whether or not an integrally framed cart is the best enclosure design from the acoustical viewpoint regardless of its other advantages. With such integral framing, the enclosure becomes attached at some intermediate location between the machine and the wheels of the cart, thus subjecting the enclosure to a rather indeterminate amount of vibrational excitation. It might be more advantageous to mount the machine on a chassis of minimum acoustical radiating area and to utilize a complete acoustical enclosure independently resiliently mounted to the chassis. However, further research will be required on a number of aspects of this problem to provide definitive answers.

It can be appreciated that the experimental noise-reduction work reported earlier in this section constitutes only the first steps toward complete acoustic enclosure. Most of the noise reductions were achieved by using exhaust mufflers and intake silencers, i.e., techniques for controlling acoustical leaks. Only limited demonstrations of the applicability of the acoustic control mechanisms of sound isolation, sound absorption, vibration isolation, and vibration damping were possible without becoming involved in drastic alterations to the housings far outside the scope of this present research program. Nevertheless, those experimental noise reductions achieved constitute significant progress toward alleviating a serious noise problem. The concept of effective acoustic enclosure holds promise of even more accomplishment.

REFERENCES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2. Geiger, P. H., Noise-Reduction Manual, Ann Arbor, University of Michigan Press, 1953.

3. Tyler, J., and Towel, G., "A Jet Exhaust Silencer," Noise Control, 1 (July, 1955), p. 37.

## SECTION IV

## EXHAUST-MUFFLER STUDY

The noise generated by the unmuffled exhaust of a reciprocating internal-combustion engine is generally the major contributor to the noise level of the engine and the driven machinery. That this is true of the exhaust noise of ground-support equipment is indicated by the fact that the average broad-band noise level of the A-1 unit was reduced 8.9 db by muffling the exhaust. Similarly, the average broad-band noise level of the C-26 unit was reduced 12.2 db by the use of exhaust mufflers. Considering that these reductions were obtained merely by using the most effective of several readily available automotive-type mufflers, and also that the overall noise of these units was still set by the exhaust noise even after muffling, it appeared that a muffler designed specifically for these units might achieve an even greater quieting. For that reason, and because of the apparent need for knowledge about muffler action and characteristics in any general program directed toward quieting ground-support equipment, a muffler study was initiated. This study was intended, first, to accumulate available information about the mechanisms of the quieting action of exhaust mufflers; second, to evaluate the applicability of this information to the design of practical mufflers; and third, to demonstrate, if possible, the feasibility of constructing a muffler which would provide adequate silencing in a specific application. The study was carried on in three phases. The first phase consisted of a survey of the literature pertaining to acoustic filter and muffler design. The second phase was an attempt to correlate the various design data and formulae with actual measurements on a commercial muffler. The third phase consisted of the construction and experimental optimization of a muffler for use on the C-26 unit.

### EXHAUST-NOISE GENERATION

Before considering the methods for reducing exhaust noise, it may be appropriate to review the mechanism by which exhaust noise is generated in a reciprocating internal-combustion engine. When the engine is operating, each time an exhaust valve opens, the slug of high-pressure gas in the cylinder is released into the exhaust system in which the pressure is comparatively low. The pressure wave thus produced then travels through the exhaust pipe with the result that a large portion of the energy is eventually radiated as sound into the air at the exhaust outlet. Because the explosions in the cylinders and the opening of the exhaust valves follow in rapid periodic succession, there is a tone in the exhaust noise which corresponds to the firing rate of the engine. In addition, because the burned gases are released in sharp repetitive pulses rich in harmonics, a number of tones corresponding to multiples or harmonics of the firing rate are also present with differing amplitudes. Frequently, the second or one of the higher harmonics of the firing frequency is the strongest, with the fundamental and first ten or so harmonics containing most of the acoustic energy present in the exhaust noise. Although the firing frequency and its harmonics generally dominate the discrete or pure-tone spectrum of exhaust noise, the exhaust noise of some engines contain discrete components which correspond to the crankshaft rotational speed and its harmonics. It suffices to say, however, that exhaust noise is characterized by a relatively large number of pure tones which fall in a harmonic series and that when the engine speed is changed, the frequencies of the exhaust tones change proportionally.

In order for an exhaust muffler to be effective, it must reduce or attenuate the amplitude of the gas pulses and average them into a steady flow of gas. If any of the passages through the muffler or the exhaust pipes are so small as to restrict the gas flow, there will be a consequent increase in the pressure against which the engine must exhaust. This pressure results in an additional load on the engine in expelling the burned gases, and in addition, the burned gases trapped in the cylinders dilute the incoming charge of fuel, thus tending to reduce engine efficiency. However, because moderate amounts of backpressure do not significantly impair engine performance, it is possible with only a negligible loss of efficiency to pass the exhaust gases through the sometimes rather tortuous paths of the acoustic filters comprising a muffler. These acoustic filters reduce the amplitude of the pressure variations in the exhaust pipe and so reduce the sound level at the exhaust outlet.

## ACOUSTIC FILTERS AND ATTENUATORS

A survey of the literature relevant to the design of mufflers and other acoustic filters indicated that design data and formulae were available for a number of simple configurations. However, very little information could be found for the less idealized and more complex configurations, or even for configurations in which several simple filters are used in combination. In addition, the design formulae applied only for certain ideal conditions, e.g., with the acoustic filter attached to a nonreflective terminations such as an infinitely long pipe. No data were found that pertained to the departures from normal acoustic behavior that might be encountered under the high flow velocities and high temperatures that exist in mufflers under normal operating conditions. Consequently, it was apparent from the outset that the published information was entirely inadequate for a completely on-paper design of an exhaust muffler without the experimental optimization of critical dimensions and shapes. Nevertheless, it is believed that the data and formulae presented below provide considerable insight into the effect of the various parameters which determine a muffler's effectiveness.

The basic configurations of a number of the types of filters commonly used in exhaust mufflers are discussed below with particular emphasis on the attenuation characteristic which they provide. These attenuation characteristics have been derived elsewhere (see References 1-5) using simple acoustic theory that assumes the physical behavior of the acoustic system to be adequately represented by linear differential equations. The basic assumptions of this theory are: (1) the acoustic medium is homogeneous and isotropic, (2) there are no dissipative forces such as those due to viscosity, and (3) the acoustic waves are of a relatively small amplitude within the linear range of the medium. All these assumptions are violated to one extent or another within an exhaust system.

<u>Changes in Area</u>.—Sound energy passing along through two pipes, joined as shown in Fig. 4.1A, is attenuated because of the change in cross-sectional area at the junction (A). Only part of the incident sound energy is transmitted across the change in area because a reflection occurs which essentially redirects some energy back to-

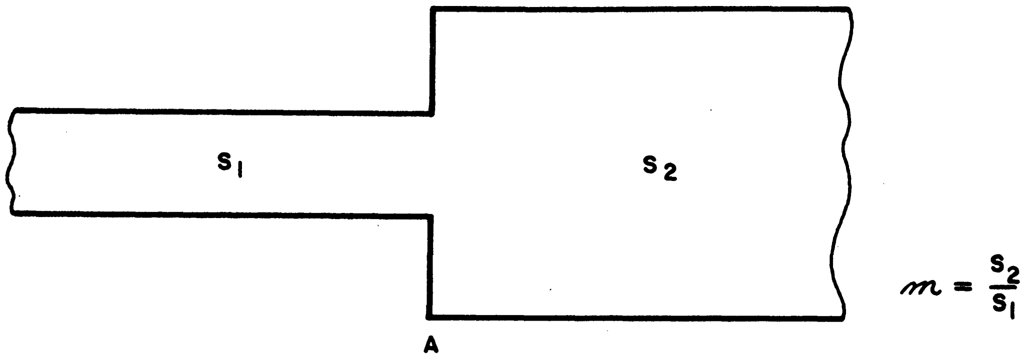



Fig. 4.1A. Change in area of duct.

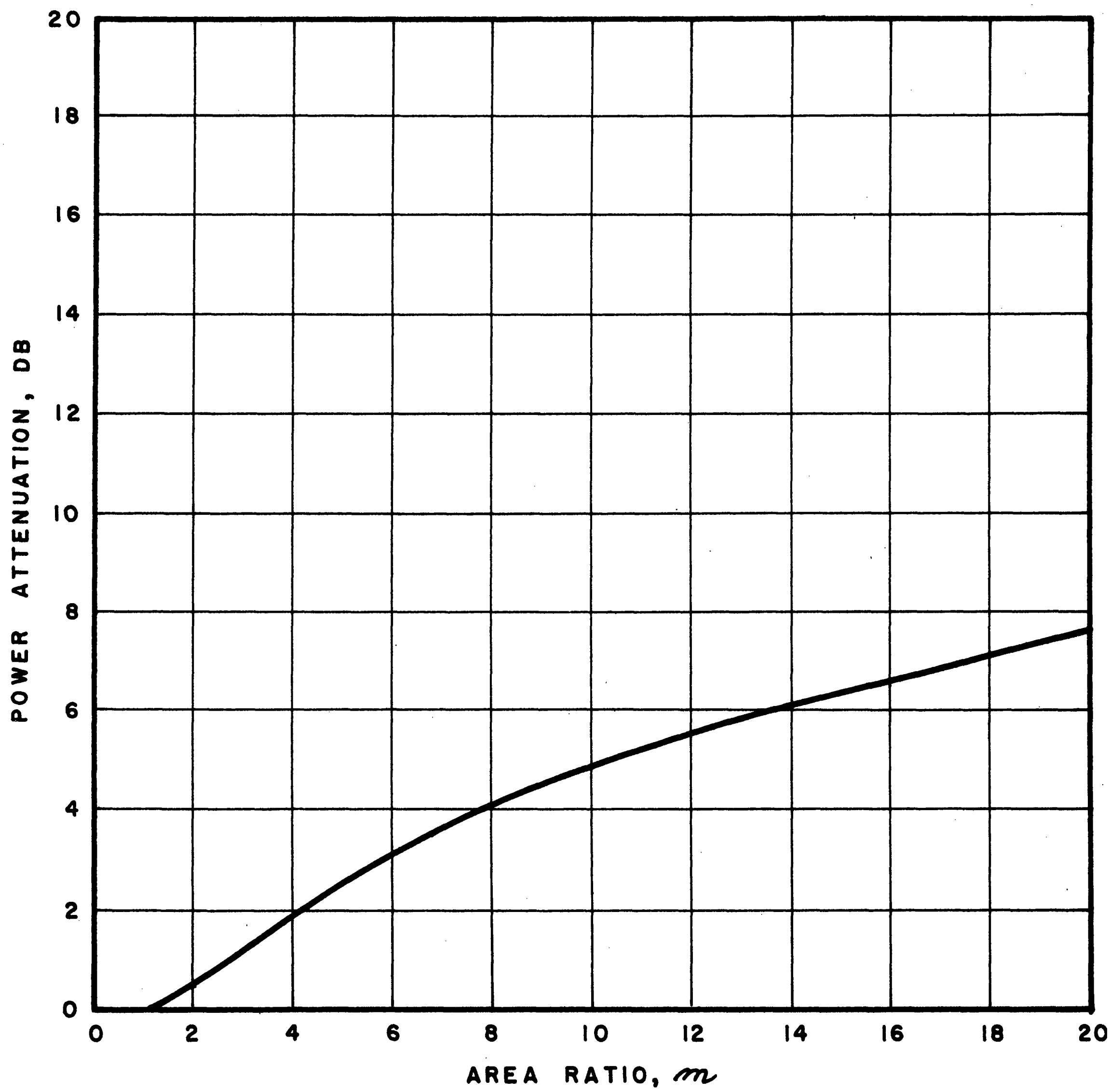



Fig. 4.1B. Effect of area ratio on attenuation.

ward the source. The attenuation* of the sound energy in passing the junction is:

$$\text{Attenuation (db)} = 10 \log_{10} \left[\frac{(m + 1)^2}{4m}\right], \tag{4.1}$$

where $m = S_2/S_1$.

The attenuation depends only on the ratio of the cross-section areas $S_2$ and $S_1$, and not on the frequency of the sound. It should be noted that attenuation is independent of the direction from which the sound arrives since $1/m$ may be substituted for $m$ without changing the attenuation. Figure 4.1B shows the attenuations that are obtained for various area ratios, $m = 1$ to 20. For $m = 20$ the attenuation is only 7.4 db, so it is apparent that the change in area must be relatively large to obtain significant attenuations.

Equation (4.1) assumes the wavelength of the sound is long compared to the diameter of the duct and that the outlet duct is terminated without reflections. In addition, it is assumed that the change in area is abrupt inasmuch as gradual transitions do not provide attenuations.

Single Expansion Chambers.—If two pipes, $S_1$, are connected by a third pipe, $S_2$, as shown in Fig. 4.2A, a single expansion chamber is formed. The sound power** attenuation resulting from this configuration is given by:

$$\text{Attenuation (db)} = 10 \log_{10} \left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 \sin^2 \frac{2\pi f \ell}{c}\right], \tag{4.2}$$

where

$m$ = ratio of the cross-sectional areas $S_1$ and $S_2$,
$\ell$ = length of the chamber,
$c$ = velocity of sound, and
$f$ = frequency in cps.

The attenuation obtained with a single expansion chamber depends on the ratio of the cross-sectional areas, the length of the expanded portion, on the velocity of sound, and on the sound frequency.

*The attenuation referred to here applies only to the reduction of the sound power and cannot be directly determined from sound-pressure measurements made on the inlet and outlet side of the filter. The sound pressure on the outlet side may be higher or lower than on the inlet side depending, respectively, on whether the sound travels from the larger cross section to the smaller or from the smaller to the larger. This can be easily shown if the acoustic resistance of each cross section is considered. For sound arriving as shown in Fig. 4.1A, the db reduction in sound pressure across the junction equals $10 \log_{10}(S_2/S_1)$. If the sound were arriving from the opposite direction, the sound pressure would be increased by that amount.

**In this case, where the inlet and outlet duct have the same diameter, the sound power attenuation corresponds to the decibel reduction in the sound pressure in the outlet pipe when the chamber is in position as compared to the sound pressure in the outlet pipe for the condition where the expansion chamber is replaced by a continuous pipe of the same diameter as the inlet and the outlet pipe. This also applies to the attenuations of the other acoustic filters yet to be discussed.

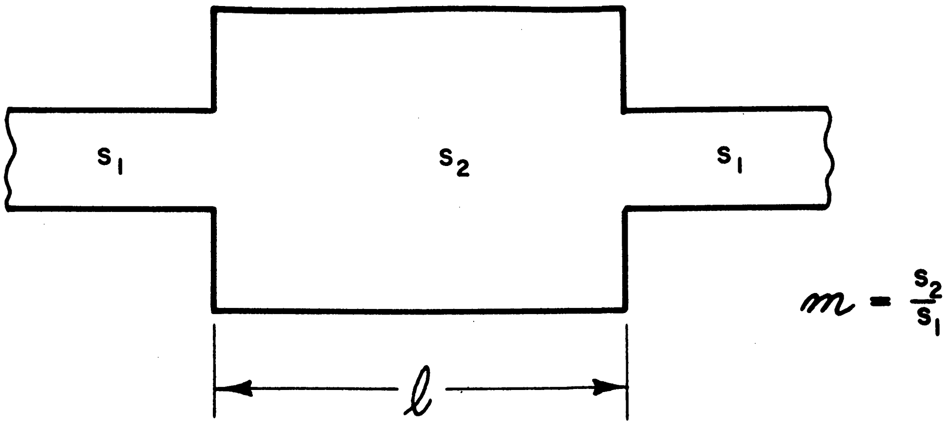



Fig. 4.2A. Single expansion chamber.

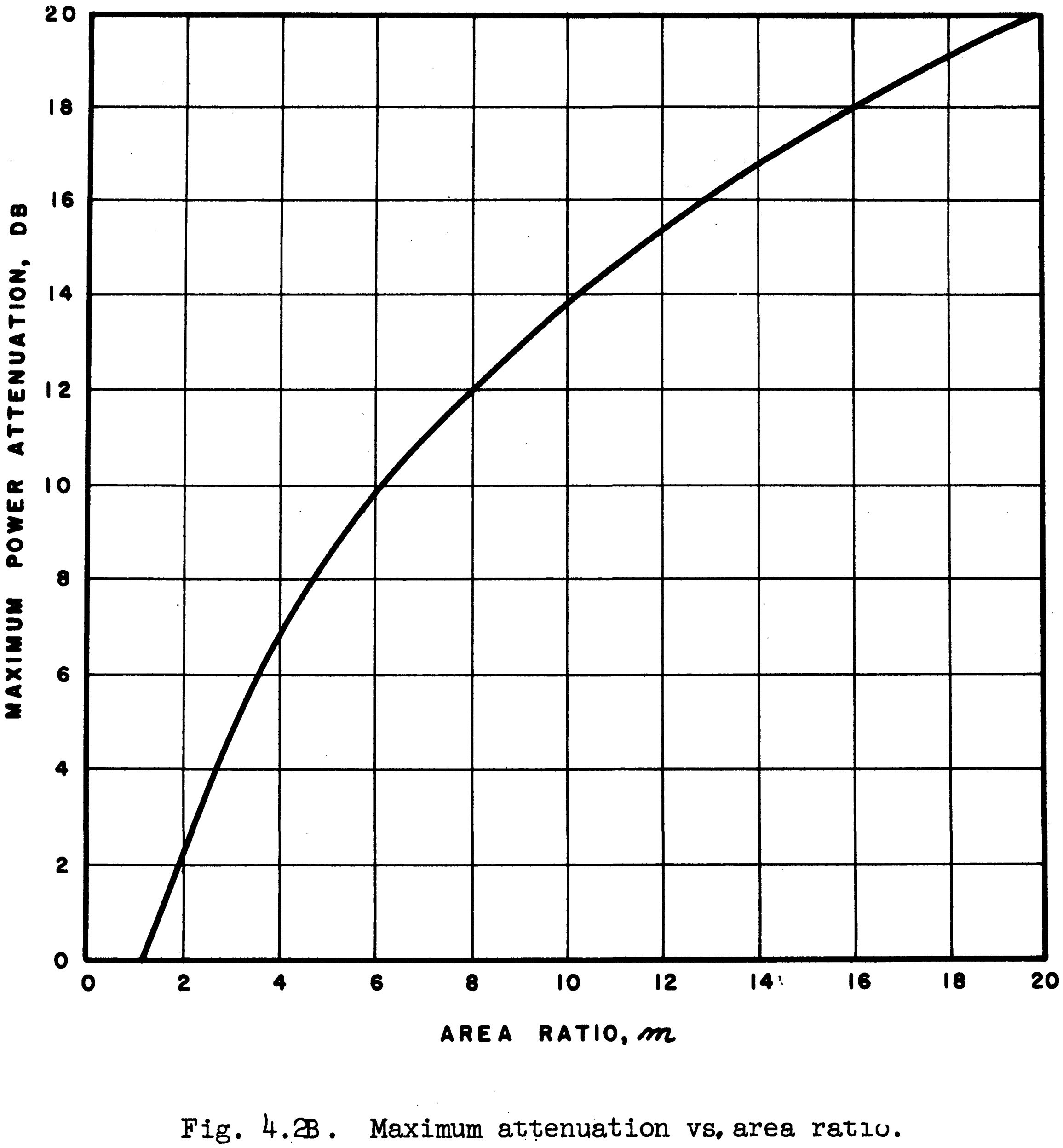



Fig. 4.2B. Maximum attenuation vs. area ratio.

Typical attenuation curves for single expansion chamber filters are shown in Fig. 4.3. It can be seen that for chambers having the dimensions shown the attenuation is relatively high over a broad range of frequencies so that these configurations are useful as noise-control filters. The attenuation reaches a maximum at those frequencies for which the chamber length corresponds to an odd multiple of the quarter wavelength. At these frequencies, the attenuation is determined by the equation:

$$\text{Maximum Attenuation (db)} = 10 \log_{10}\left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2\right] . \qquad (4.3)$$

The maximum attenuations obtained for values of $m = 1$ to 20 are shown in Fig. 4.2B.

The attenuation is zero at zero frequency and at those frequencies for which the length of the chamber is equal to a half wavelength or a multiple of the half wavelength.

Equations (4.2) and (4.3) do not apply for extreme changes of area at the junction or at high frequencies where the wavelength of sound is small compared to the diameter of the pipe. A more detailed theory is applicable for pipe diameters that are so small that the attenuation due to viscous damping forces is large.

Double Expansion Chambers.—It is reasonable to expect that connecting two single expansion chambers in series as shown in Fig. 4.4 would provide an attenuation approximately equal to the sum of the attenuations of the two individual chambers. This would be highly desirable since it might then be possible to eliminate the regions of zero attenuation by the proper selection of the chamber lengths $\ell_1$ and $\ell_2$. However, the series connection of two expansion chambers results in an acoustic system that is far more complex than a single expansion chamber. Additional pass and attenuation bands are introduced at the characteristic frequencies of the connecting pipe, $\ell_3$. Pass frequencies occur when

$$f = n\frac{c}{2\ell_3} , \qquad (4.4)$$

where

$n = 1, 2, 3, 4$ -----,
$c$ = velocity of sound, and
$f$ = frequency.

Increases of attenuation occur when

$$f = (1 + 2n)\frac{c}{4\ell_3} . \qquad (4.5)$$

In addition, the connecting pipe, $\ell_3$, and each expansion chamber combine to form two Helmholtz resonators which in this arrangement result in a loss of attenuation at the respective resonant frequencies. These frequencies are given by the equations:

$$f_{r_1} = \frac{c}{2\pi}\sqrt{\frac{S_1}{\ell_a V_1}} \qquad (4.6)$$

and

$$f_{r_2} = \frac{c}{2\pi}\sqrt{\frac{S_1}{\ell_a V_2}} . \qquad (4.7)$$

Here

$S_1 = \pi r_3^2$, the cross-sectional area of the neck of the resonator, in this case the connector pipe $\ell_3$;

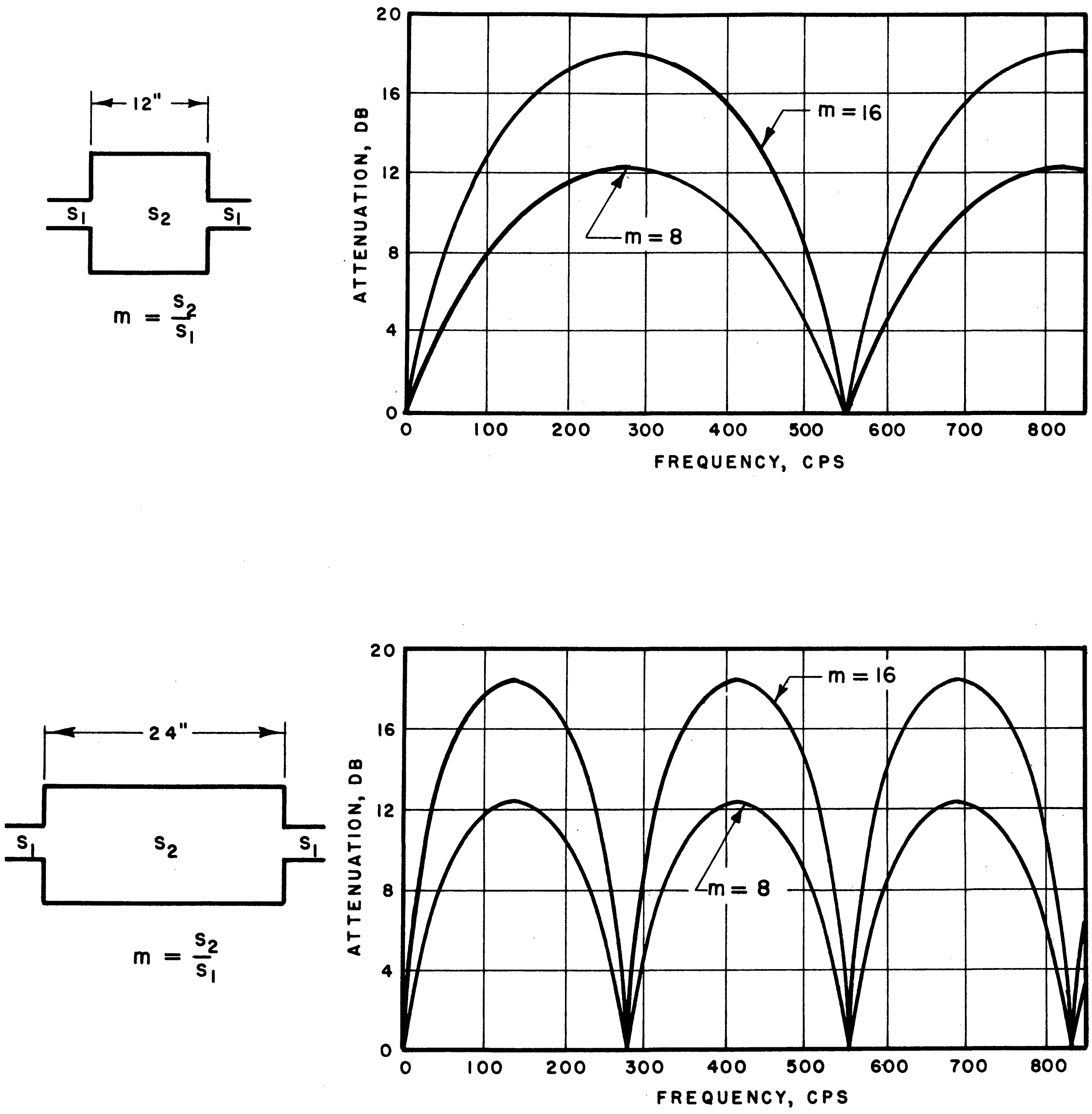



Fig. 4.3. Effect of length and area ratio on the atten uation of a single expansion chamber.

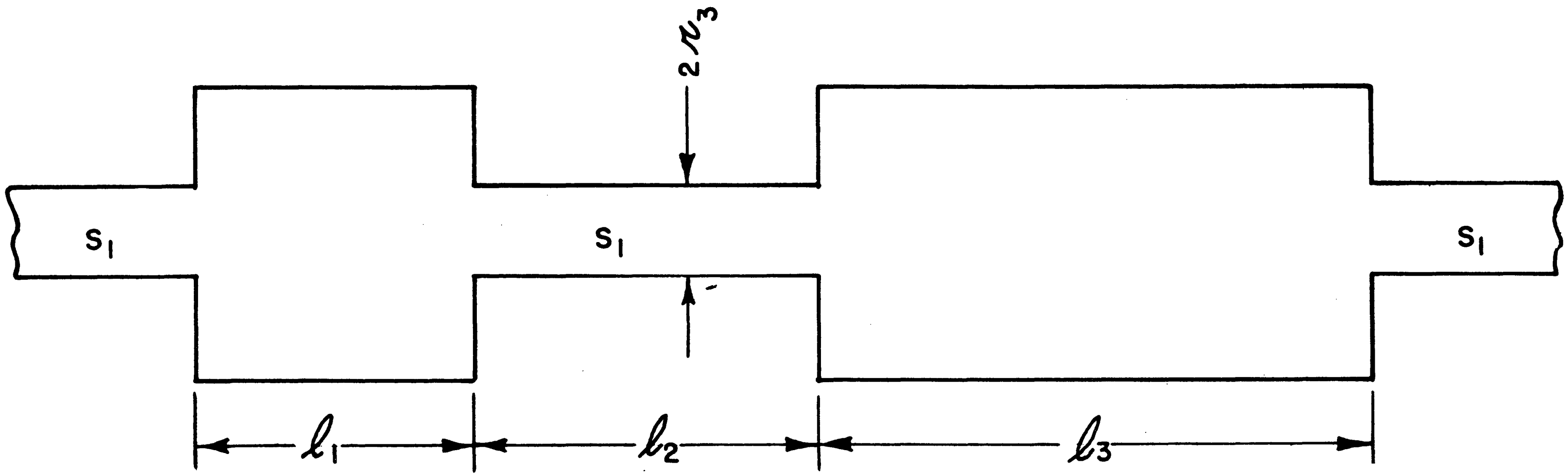



Fig. 4.4. Double expansion chamber.

$\ell_a$ = $\ell_3 + \pi/2\ r_3$, the effective length of the neck (actual length plus end correction),
$V_1$ = volume of chamber having length $\ell_1$, and
$V_2$ = volume of chamber having length $\ell_2$.

Above these resonant frequencies, the attenuation of this type of resonator increases approximately according to the equation:

$$\text{Attenuation (db)} = 17 \log_e f/f_r \ . \tag{4.8}$$

Because of the many pass and attenuation regions found in this filter configuration, the attenuation versus frequency characteristics cannot be accurately computed. Nevertheless, experimental results reported by Bentele and shown in Fig. 4.5 indicate some correlation with predicted values. In these figures, the curve of the combined attenuations of the two expansion chambers, the attenuation curves for the resonators, and the numerous pass frequencies of the various elements are shown. For a long connecting pipe, $\ell_3$, it is seen that high attenuations can be achieved at relatively low frequencies, but that the first pass frequency of the connecting pipe results in a significant drop in attenuation. For the short connecting pipe, the higher Helmholtz resonant frequency results in a complete loss of attenuation over a relatively wide band of frequencies near 200 cps. However, at higher frequencies the attenuation corresponds quite closely with that expected from the double expansion chamber.

Comparing the attenuations achieved with a double expansion chamber muffler with those of a single expansion chamber, it appears that only little is gained by going to the more complicated system. Indeed, unless the connecting pipe length is carefully chosen, there is a possibility of losing attenuation at the low frequencies where it is generally needed most.

<u>Resonators</u>.—Helmholtz resonators attached as side-branches to an exhaust pipe are commonly used in exhaust mufflers. In contrast to the Helmholtz resonators produced incidentally in forming a double expansion chamber, and which resulted in a loss of attenuation because the resonance occurred in the direction of sound travel, a Helmholtz resonator attached to a pipe as shown in Fig. 4.6 is capable of providing considerable attenuation. The action of such a resonator may be explained as follows. During the excess pressure phase of the acoustic wave in the pipe the gas in the chamber is compressed. This compression, in return, pushes the gas back into the pipe during the reduced pressure phase of the acoustic wave in the pipe. Hence, the gas in the chamber acts as a spring or a capacitance, while the gas in the neck of the chamber acts as a mass or an inductance. Because in passing through the neck the sound is radiated into the surrounding medium, energy losses result so that in effect a damper is attached to the mass or a resistor is connected to the inductance. Consequently, in principle, the resonator acts like a damped dynamic vibration absorber or like a series resonant RLC circuit which shunts the energy source.

The attenuation provided by a resonator attached as a side-branch to the main pipe is given by:

$$\text{Attenuation (db)} = 10 \log_{10} \left[ 1 + \frac{c^2}{4S^2 \left( \frac{2\pi f}{C_o} - \frac{c^2}{2\pi f V} \right)^2} \right] , \tag{4.9}$$

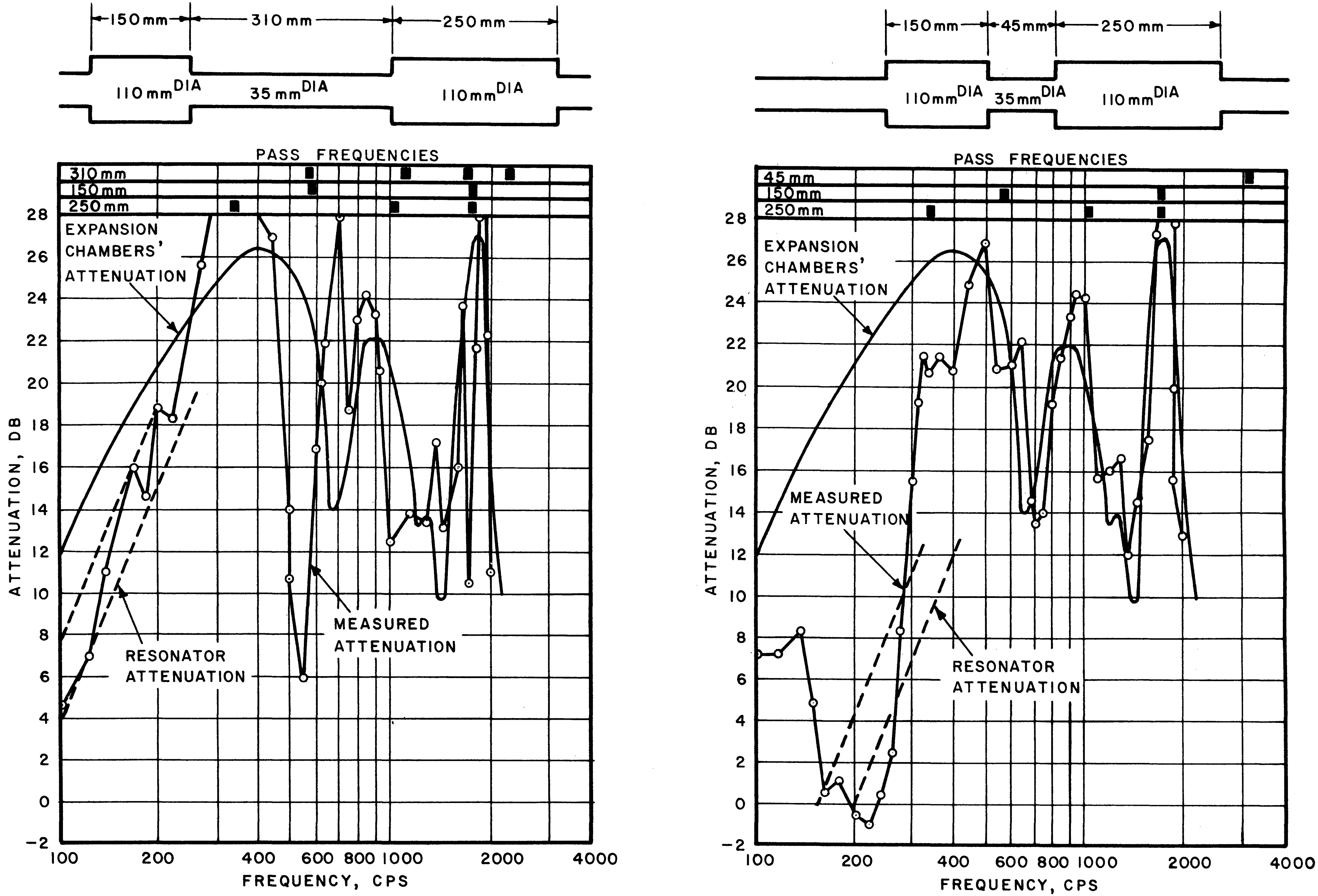



Fig. 4.5. Effect of length of connecting pipe on attenuation of double expansion chamber (after Bentele, Ref. 3).

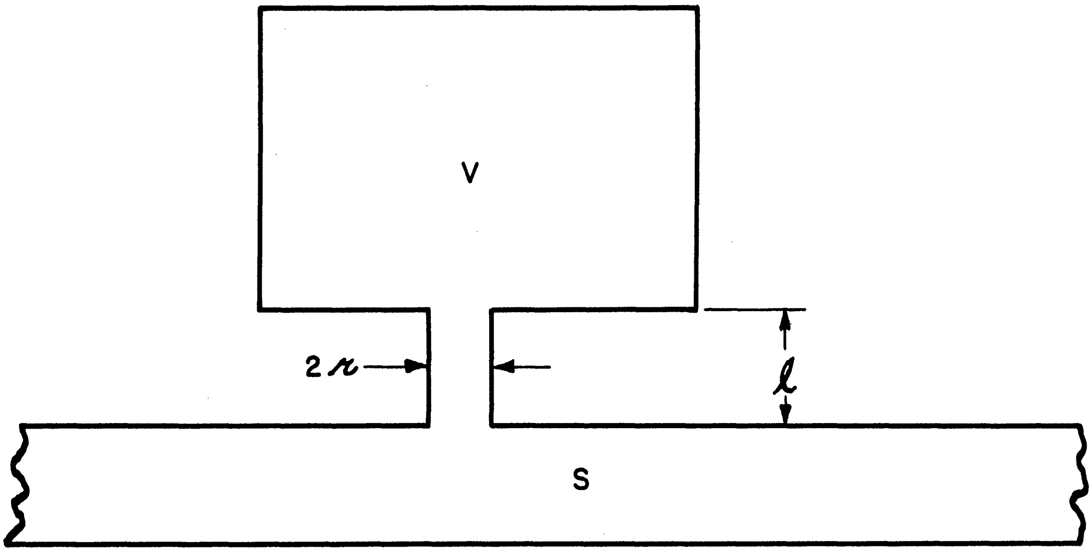



Fig. 4.6. Helmholtz resonator.

where

$c$ = velocity of sound,
$S$ = cross-sectional area of the main pipe,
$f$ = frequency in cps,
$C_o$ = $(\pi r^2)/(\ell + \pi/2\ r)$, the conductivity of the opening to the chamber, and
$V$ = volume of the chamber.

It is seen that the attenuation is infinite when

$$\frac{2\pi f}{C_o} = \frac{c^2}{2\pi f V} .$$

This defines the resonant frequency, $f_r$, where

$$f_r = \frac{c}{2\pi}\sqrt{\frac{C_o}{V}} . \tag{4.10}$$

Equation (4.9) can be written in terms of the resonant frequency $f_r$, as

$$\text{Attenuation (db)} = 10 \log_{10}\left[1 + \frac{C_oV/4S^2}{(f/f_r - f_r/f)^2}\right] . \tag{4.11}$$

It is apparent from Eq. (4.11) that resonators act as band suppression filters which provide attenuation over a frequency bandwidth which increases as the quantity $C_oV/4S^2$ increases. Rewriting the term $C_oV/4S^2$ as

$$\frac{\left(\frac{\pi r^2}{1 + \pi/2\ r}\right)V}{4S^2} ,$$

it can be seen that the attenuation range can be increased by increasing the chamber volume V; by decreasing the cross section, S, of the main pipe; and by eliminating the neck, $\ell$, of the resonator, and replacing it by a hole or slit in the side of the main pipe.

In practice, the attenuation provided by resonator-type filters has been found to depend not only on the variables given in Eq. (4.9) but also on the relative proportions of the resonator chamber, on the position of the opening of the resonator with respect to the walls, and on the shape of the opening into the resonator. The effects of these variables are not considered in the elementary theory and can be studied best experimentally.

Figure 4.7 shows the attenuation characteristics of several experimental resonators tested by Bentele.[3]

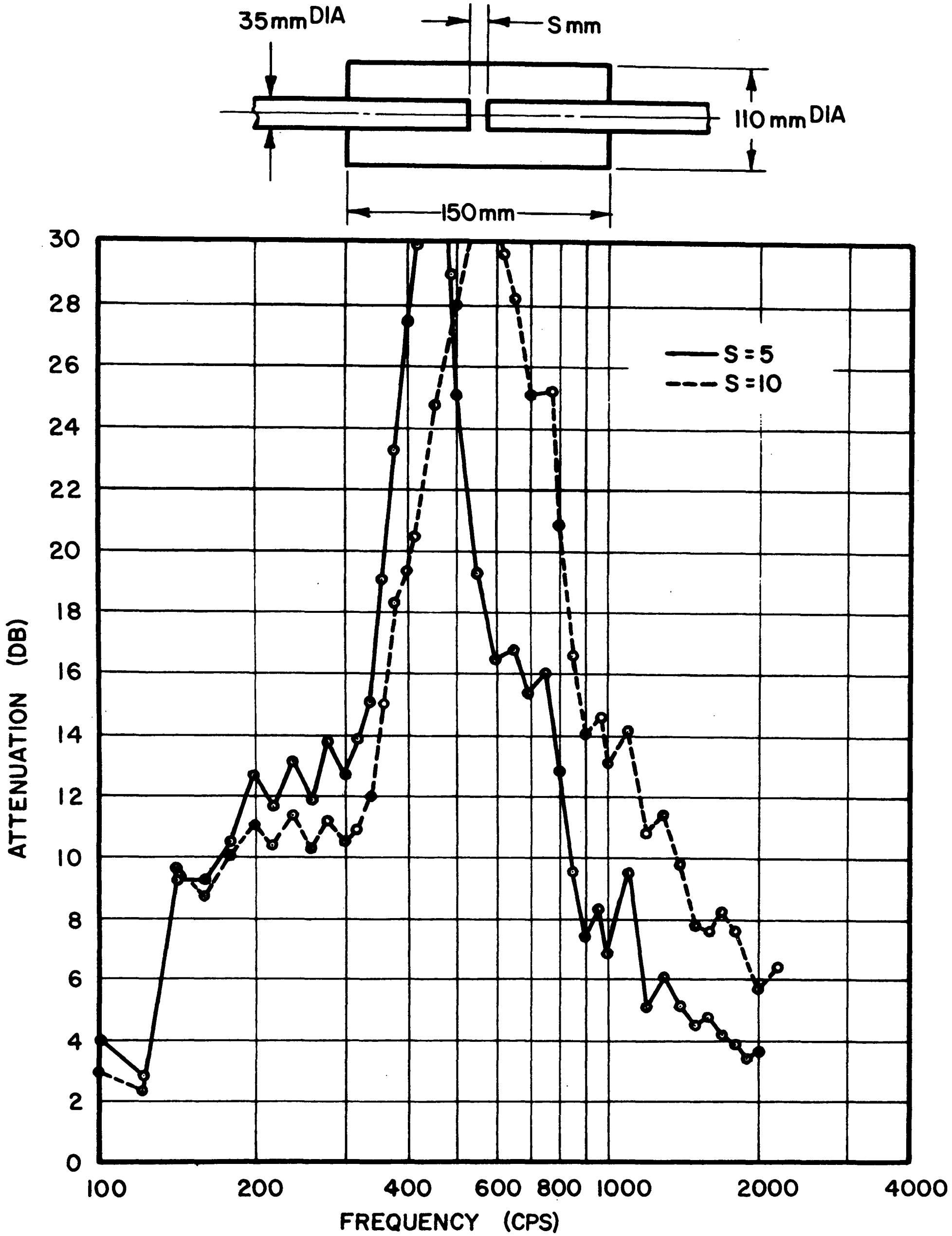
Openings in Side of Duct.—An opening in the side of a pipe as shown in Fig. 4.8, converts the pipe into a high pass filter. The attenuation of the sound in passing the opening is given by the equation

$$\text{Attenuation (db)} = 10 \log_{10}\left[1 + \left(\frac{C_oc}{4\pi fS}\right)^2\right] , \tag{4.12}$$

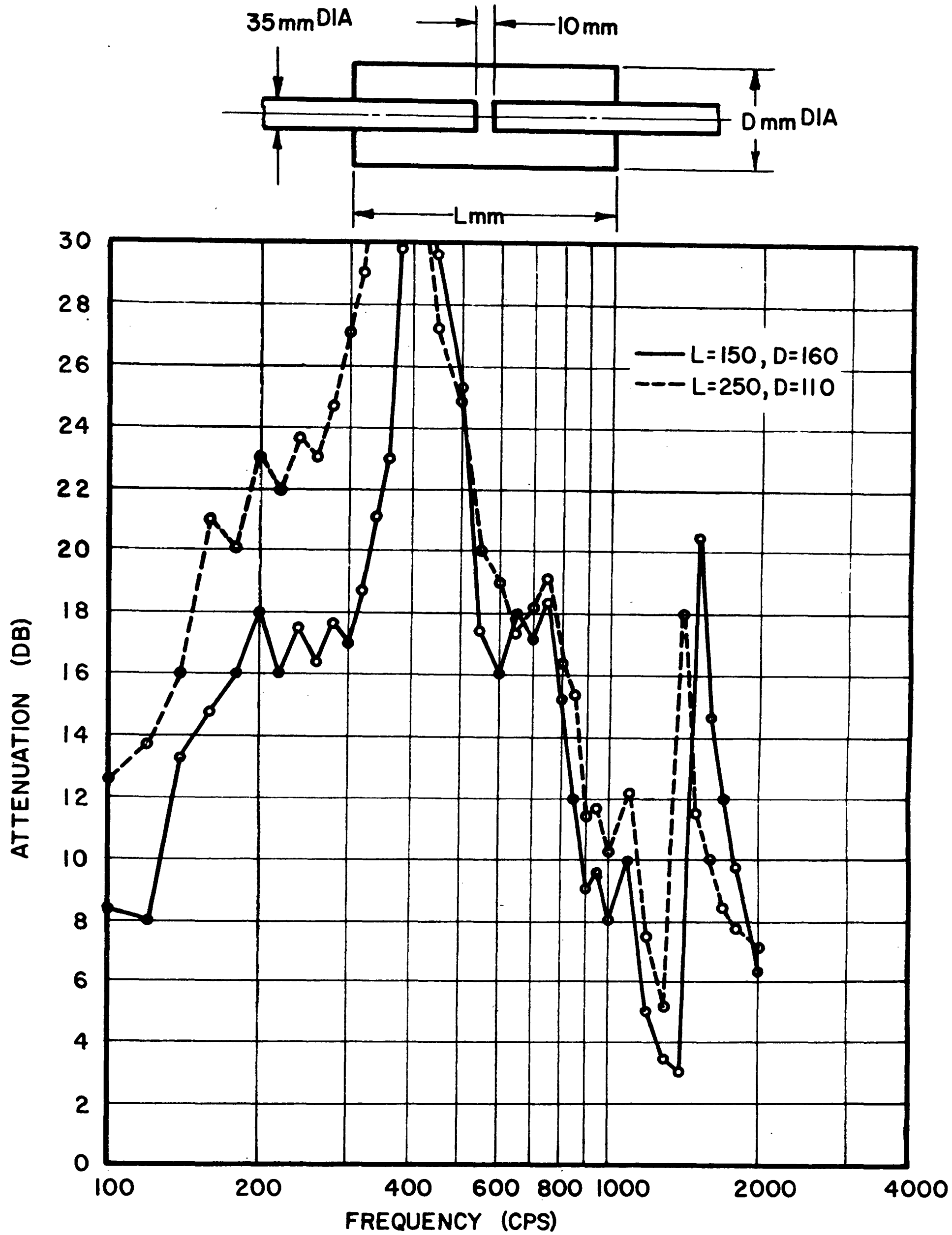
where

$C_o$ = $\dfrac{\pi r^2}{\ell + \pi/2\ r}$ , the conductivity of the opening,

$c$ = velocity of sound,




(a) EFFECT OF SLIT WIDTH




(b) EFFECT OF VOLUME

Fig. 4.7. Measured resonator attenuation (after Bentele, Ref. 3).

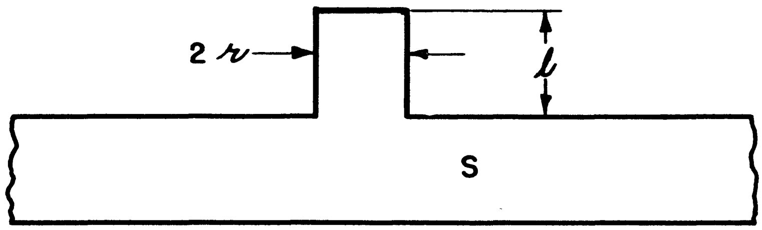



Fig. 4.8. Side-branch opening in duct.

$f$ = frequency in cps, and
$S$ = cross-sectional area of the pipe.

This equation, which applies only if the radius of the opening and the length, $\ell$, are small compared to the wavelength, indicates that as the area of the opening is increased relative to the cross-sectional area of the main pipe, the attenuation increases and the attenuation range extends to higher frequencies. If the pipe has several openings which are separated by only a small fraction of a wavelength, the effect of the multiplicity of holes is to increase the conductivity, $C_o$, in proportion to the number of holes, thus increasing the attenuation. If, however, the holes are more widely spaced, this equation is inapplicable and a more detailed theory must be used. The attenuation provided by an opening in the side of a duct is shown in Fig. 4.9.

Apparently side openings would have only little value in practical mufflers because it seems that appreciable sound would be radiated from the openings themselves. However, under certain conditions very little sound energy is transmitted through the opening inasmuch as the sound attenuation in the main pipe occurs because the side opening reflects the sound back to the source with only a small portion of the sound energy being transmitted out of the side opening.[2] If this is also true in the case of openings in the side of a pipe which is contained within a muffler housing, this filter configuration would provide a convenient method of obtaining low-frequency attenuation. Although this is not discussed in the literature, it is known that the pipes in many commercial mufflers do have side openings; thus it is not unreasonable to expect that they do provide some low-frequency attenuation.

<u>Orifices in Ducts</u>.—The attenuation resulting from an orifice placed in a duct as shown in Fig. 4.10 is given by the equation:

$$\text{Attenuation} = 10 \log_{10} \left[ 1 + \left( \frac{\pi f}{c} \cdot \frac{S}{C_o} \right)^2 \right] , \qquad (4.13)$$

where

$f$ = frequency in cps,
$c$ = velocity of sound,
$S$ = cross-sectional area of duct, and
$C_o$ = $2\pi r$, the conductivity of the orifice.

The attenuation characteristic is that of a low pass filter, the attenuation at any frequency increasing as the diameter of the orifice decreases relative to the diameter of the pipe. This method of obtaining high-frequency attenuation is usable in practical mufflers only if the cross section of the duct is sufficiently large so that a smaller orifice area can be used without restricting the gas flow and increasing the backpressure.

## CORRELATION OF COMPUTED AND MEASURED ATTENUATIONS

The attenuation characteristics of the acoustic filters discussed above were derived for each type of filter acting independently under idealized conditions. It was realized that the assumptions made in the derivation of these formulae were not necessarily consistent with actual conditions encountered when several filters are combined into a single muffler unit. For this reason, it was decided to compare the attenuation of a specific muffler as measured in actual operation with the total of

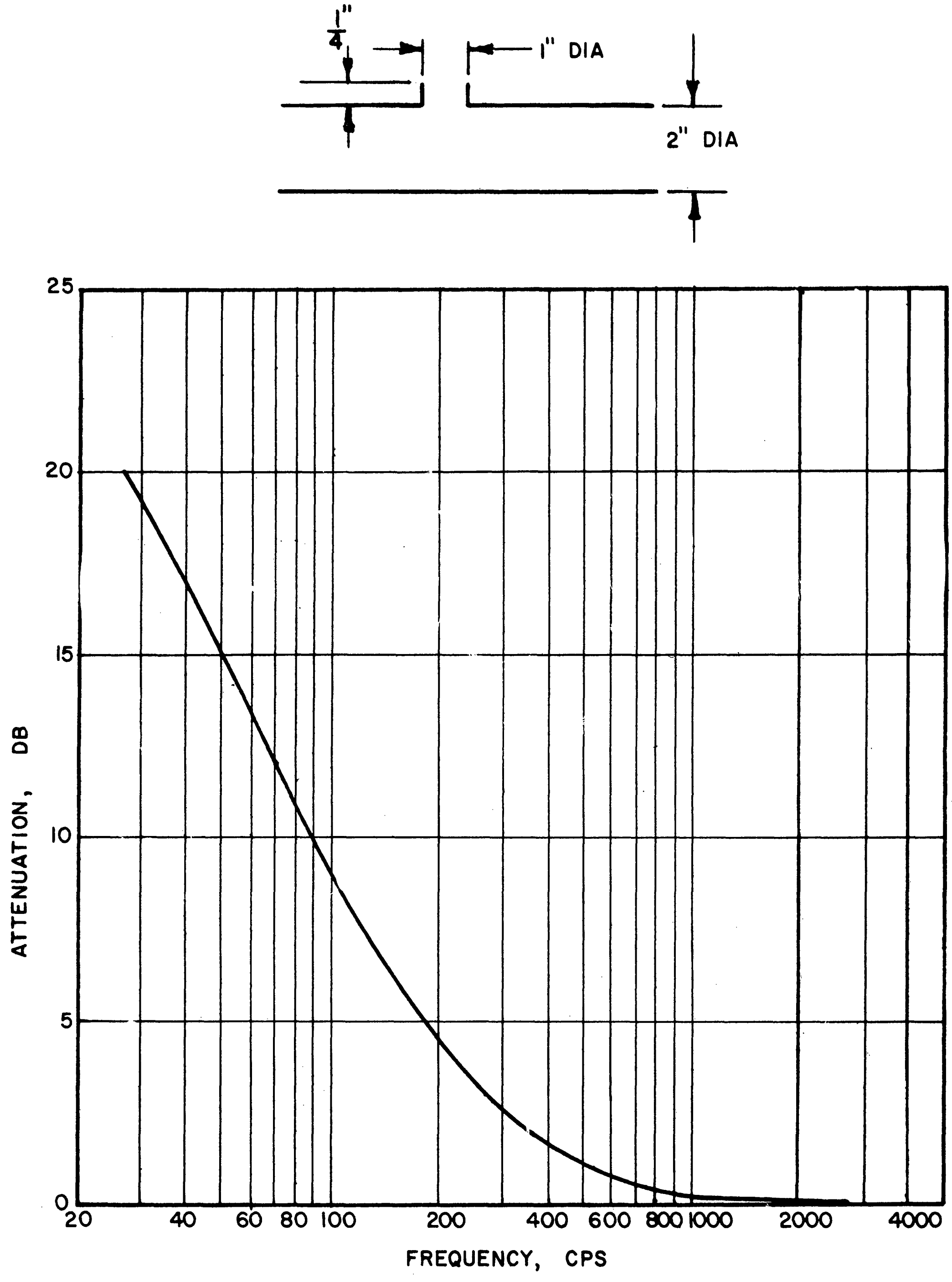



Fig. 4.9. Orifice side-branch attenuation.

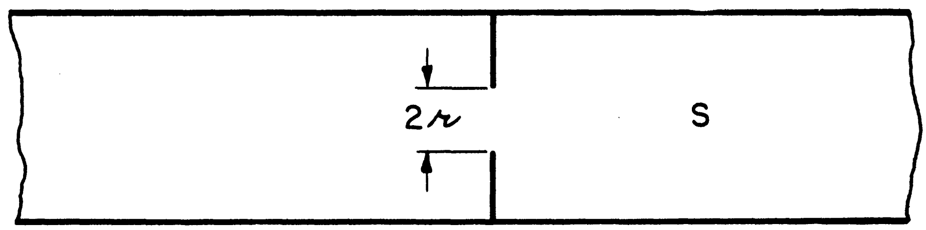



Fig. 4.10. Orifice in duct.

the attenuations computed independently for each of the several elements comprising the muffler.

A commercial muffler, a Walker 582, had previously been tested on the A-1 Generator Set using octave-band measurements to determine the muffler attenuation characteristics. This muffler was disassembled and a scale drawing, Fig. 4.11, was made. (Although not evident from the drawing, the housing and all tubes are cylindrical.) Working from this drawing the elements of the muffler were laid out schematically in Fig. 4.12 which indicates that the muffler basically consists of two expansion chambers and a resonator of the approximate dimensions shown. Using these dimensions, it was possible to compute the attenuation characteristics of each of these elements, assuming that they were operating independently under the idealized conditions for which the formulae applied. The velocity of sound, c, was taken as 1500 ft/sec inasmuch as the temperature of the exhaust gases in the muffler was estimated to be 500°F. The attenuation versus frequency curves computed for the two expansion chambers and the resonator, shown in Fig. 4.13, were then simply added at each frequency to obtain the overall attenuation curve.

It is admitted that the experimental curve should be derived from narrow band information to check accurately the correlation between a computed attenuation curve and the measured attenuation. However, in this case only octave-band attenuation data were available for comparison with the computed results. These data, also plotted in Fig. 4.13, show the general character of the computed overall attenuation curve and that of the measured octave-band attenuation curve to be somewhat similar. However, regardless of this qualitative similarity, discrepancies between the computed and measured results in the low-frequency region below 150 cps and in the region near 1000 cps indicate the inadequacy of this method of predicting muffler characteristics.

Another muffler, a Walker 639, was also disassembled with the intention of studying its internal construction which, as can be seen in Fig. 4.14, was considerably more complicated than that of the Walker 582. Although a brief study indicates that basically this construction embodies a single expansion chamber with baffle plates and a resonator, the filtering characteristics of this muffler cannot be readily computed because of the parallel but dissimilar paths for gas flow between the inlet and outlet tubes. The simple analysis of acoustic filtering action presented here makes no provision for parallel paths of this type.

After considering these two commercial muffler configurations, it is apparent that the formulae discussed earlier in this section are certainly not adequate for quantitatively determining the attenuation characteristics of commercial mufflers. Indeed, it seems extremely doubtful that the filtering action of such complicated shapes and configurations can be accurately determined using any practical analytical approach. For example, the effects of gas flow reversals, of orifice location, and of gas viscosity are not readily analyzed mathematically. The value of the formulae presented lies in the fact that they give a qualitative indication of the effect of the various parameters of the basic acoustic filter configurations. For that reason they may be helpful in refining or optimizing a muffler design inasmuch as they give some insight into those parameters which must be varied to obtain a desired effect.

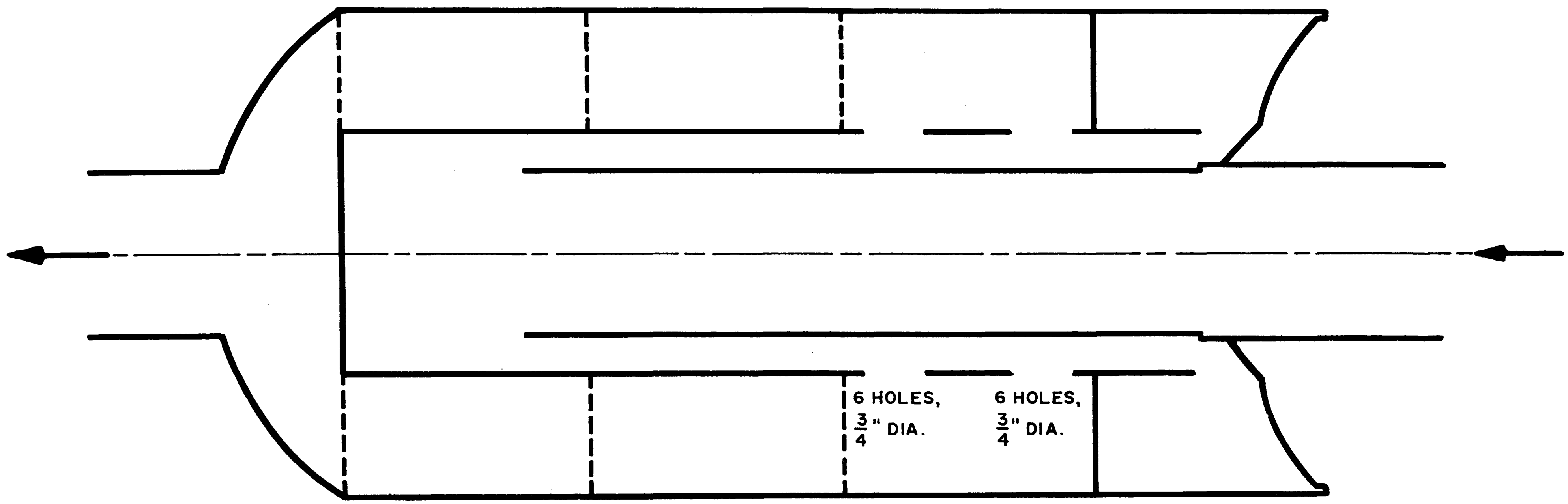
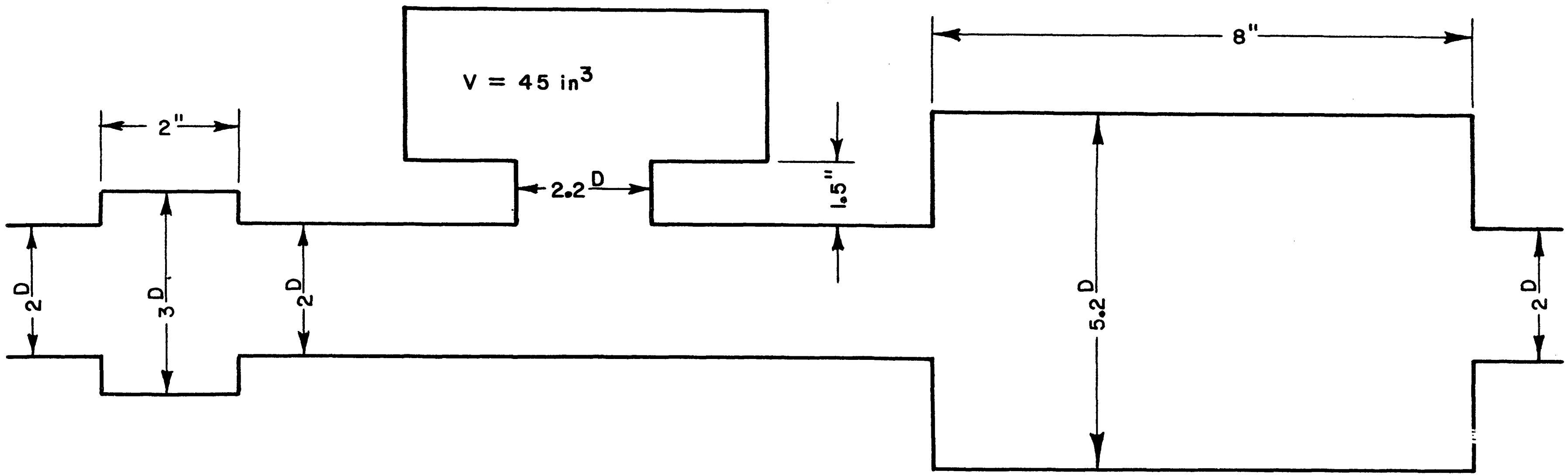



Fig. 4.11. Walker 582 muffler.




EXPANSION CHAMBER NO. 1

$m = \left(\frac{3}{2}\right)^2$

$\ell = 2''$

$c = 18{,}000$ in/sec

RESONATOR

$V = 45 \text{ in}^3$

$C_o = \dfrac{\pi (1.1)^2}{1.5 + \frac{\pi}{2}(1.1)}$

$S = \pi \text{ in}^2$

EXPANSION CHAMBER NO. 2

$m = \left(\frac{5.2}{2}\right)^2$

$\ell = 8''$

$$\text{EXPANSION CHAMBER ATTENUATION} = 10 \log_{10}\left\{1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 \sin^2 \frac{2\pi f \ell}{c}\right\}$$

$$\text{RESONATOR ATTENUATION} = 10 \log_{10}\left\{1 + \frac{c^2}{4S^2\left(\frac{2\pi f}{C_o} - \frac{c^2}{2\pi f V}\right)^2}\right\}$$

Fig. 4.12. Diagram used in derivation of Walker 582 muffler attenuation.

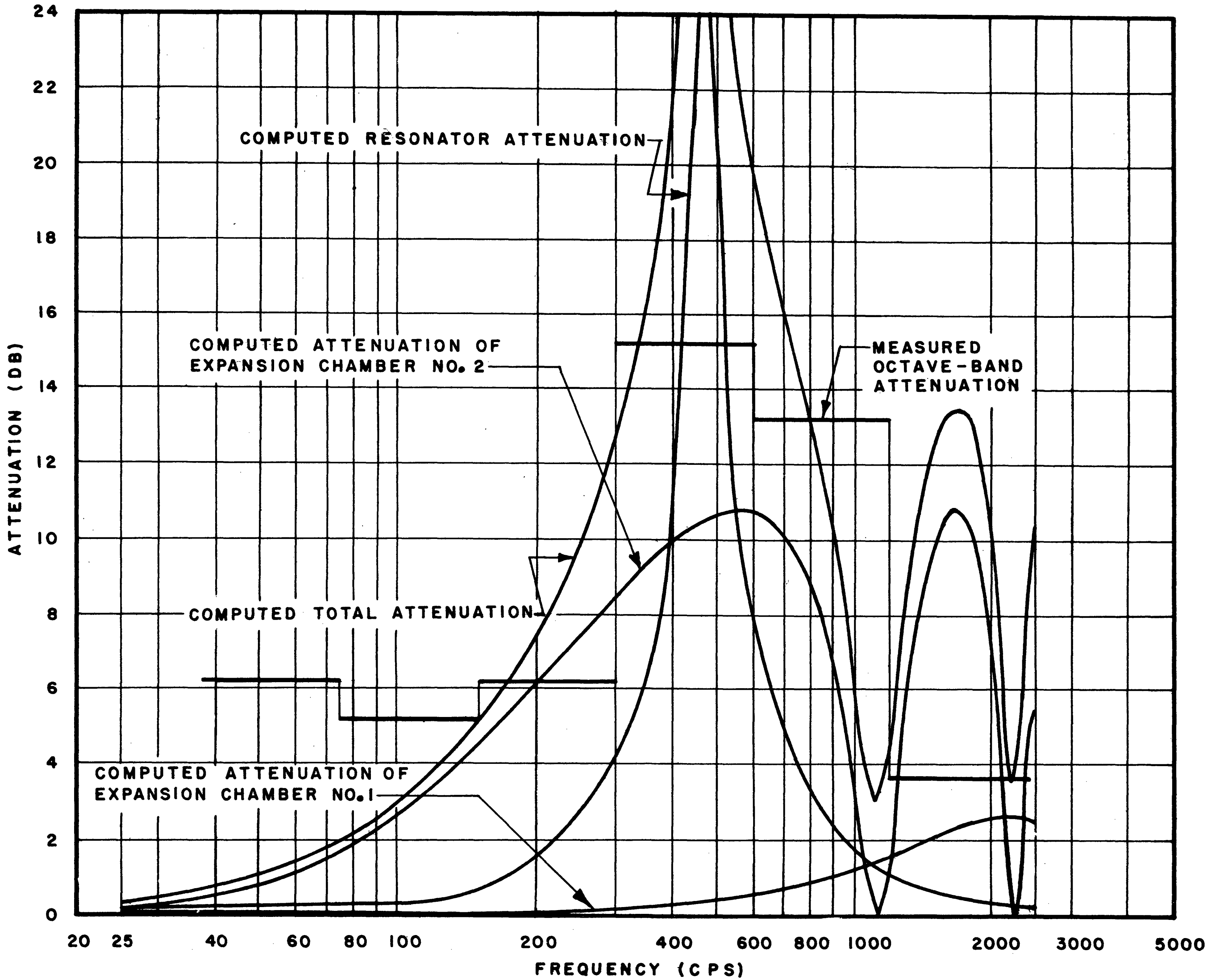



Fig. 4.13. Comparison of computed and measured attenuation of Walker 582 muffler.

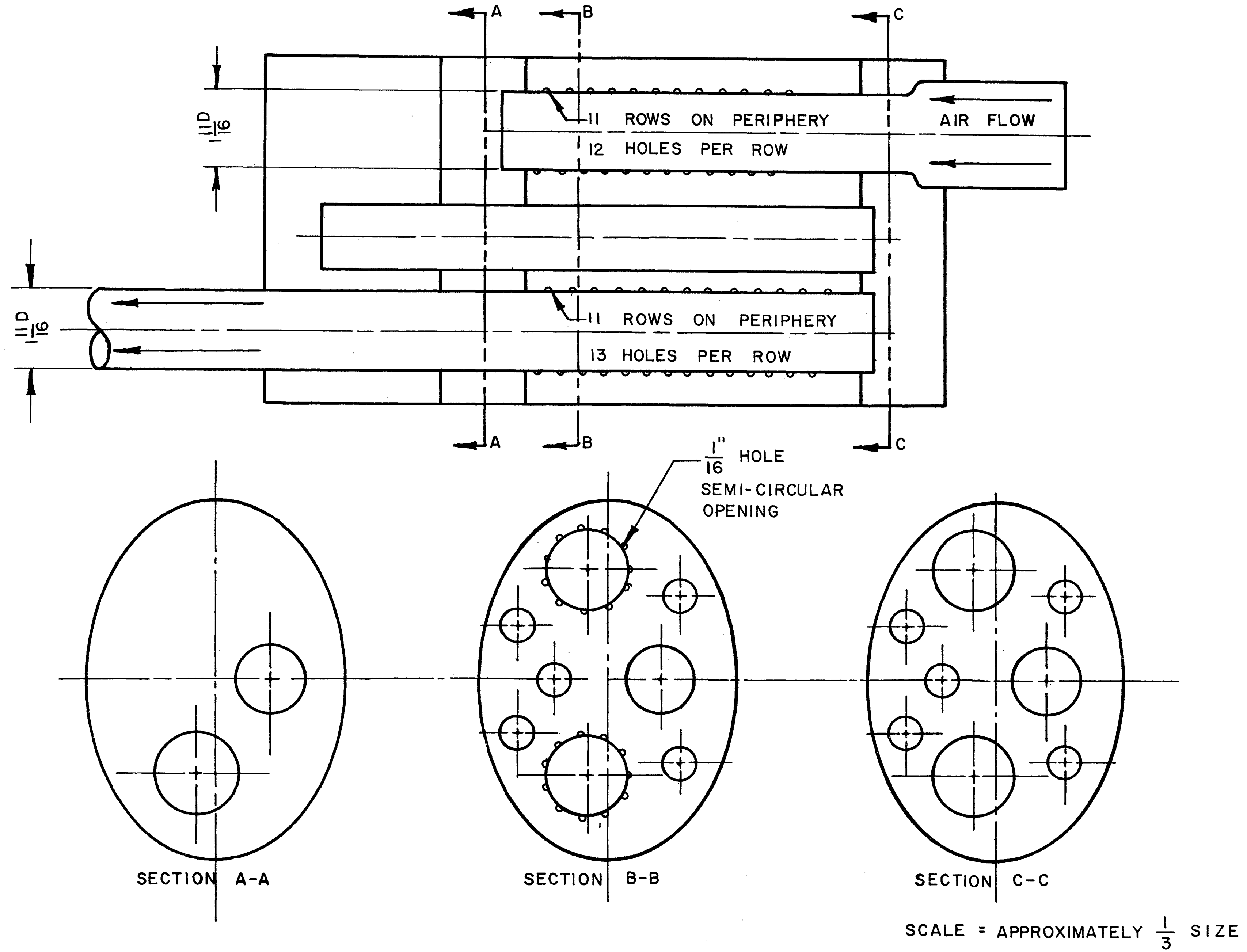



Fig. 4.14. Walker 639 muffler.

Bentele[3] designed a muffler for large rotary blowers which was a combination resonant and expansion chamber. This design, shown schematically in Fig. 4.15A, is especially practical for use in experimental muffler applications because it potentially incorporates the desirable features of both the resonator and the expansion chamber, and because by adjusting three parameters (length and volume of chamber, and gap width) the attenuation characteristic can be varied to fit the application. Figure 4.15B shows an attenuation curve for this type of arrangement as reported by Bentele. The shape of the attenuation peak and the frequency at which it occurs are, for small gap widths, determined, respectively, by the two resonator parameters $C_oV/4S^2$ and $\sqrt{C_o/V}$ as given by Eqs. (4.11) and (4.10). For wide gap widths the attenuation characteristic of this configuration approaches those of the single expansion chamber, and is determined primarily by the area ratio and the length of the chamber.

The expansion chamber—resonator muffler appeared feasible for use as an experimental muffler for several reasons. First, the possibility of changing the attenuation characteristic of this type of muffler by simply varying the gap width and the chamber length permitted easy "tuning" for maximum exhaust quieting. Second, the straight-through design of this muffler would result in a low amount of backpressure, a factor which might be particularly important in military applications. Third, the simple design made it easy to fabricate.

It was convenient to use the C-26 Generator Set as the test vehicle on which the characteristics of the experimental muffler would be measured. The exhaust from one of the two banks of three cylinders was ducted through an eight-foot length of 2-1/8-in. ID flexible metal pipe thus providing some isolation of the exhaust noise from direct motor noise, and from the exhaust noise of the other bank of cylinders. The physical and electrical arrangement of the equipment used is shown in Fig. 4.16. All measurements were made with the C-26 running at 2000 rpm with a 28 volt d-c load of 1000 amperes to simulate normal operating conditions.

The first measurements involved the determination of the discrete or pure-tone spectrum of the C-26 exhaust noise at the outlet of a two-inch ID pipe which replaced the muffler shown in Fig. 4.16. The spectrum shown in Fig. 4.17 indicates that most of the harmonics of the 50-cps firing rate were detectable up to 900 cps and that all but one of the strongest components fell at frequencies below 350 cps.

The combination resonator—expansion chamber muffler shown schematically in Fig. 4.18 was fabricated in a manner which permitted both shortening the length A of the chamber and varying the gap width B. In addition, the muffler could be disassembled to permit lining the interior of the chamber with fibrous glass blanket materials. At the outset, the chamber length was set arbitrarily at 48 in. to permit a wide range of variation. The chamber diameter was chosen at 10 in. to create a relatively large volume even at short chamber lengths and, in addition, provide a high ratio of chamber cross-sectional area to duct area.

The first measurements were made with the muffler length set at 48 in. and with gap widths which were varied from 0 to 40 in. Significant attenuations were achieved for gap widths greater than one inch but maximum attenuation occurred at the 30-in. gap setting. For this latter setting, the broad-band attenuation was about 12 db, with the limit being set by the exhaust-noise components in the frequency band from 150 to 300 cps.

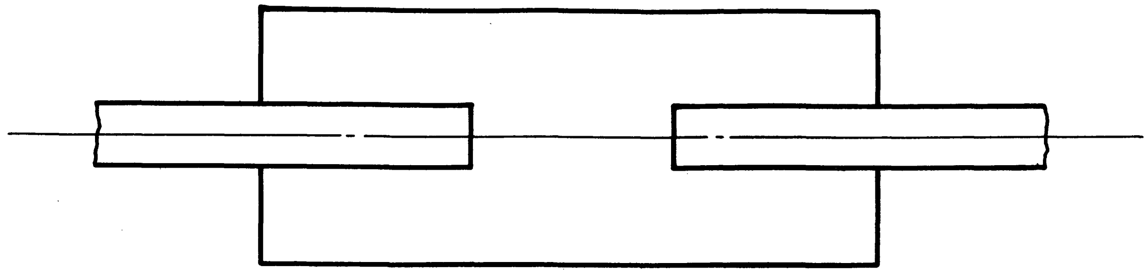


Fig. 4.15A. Combination resonator and expansion **chamber** muffler.

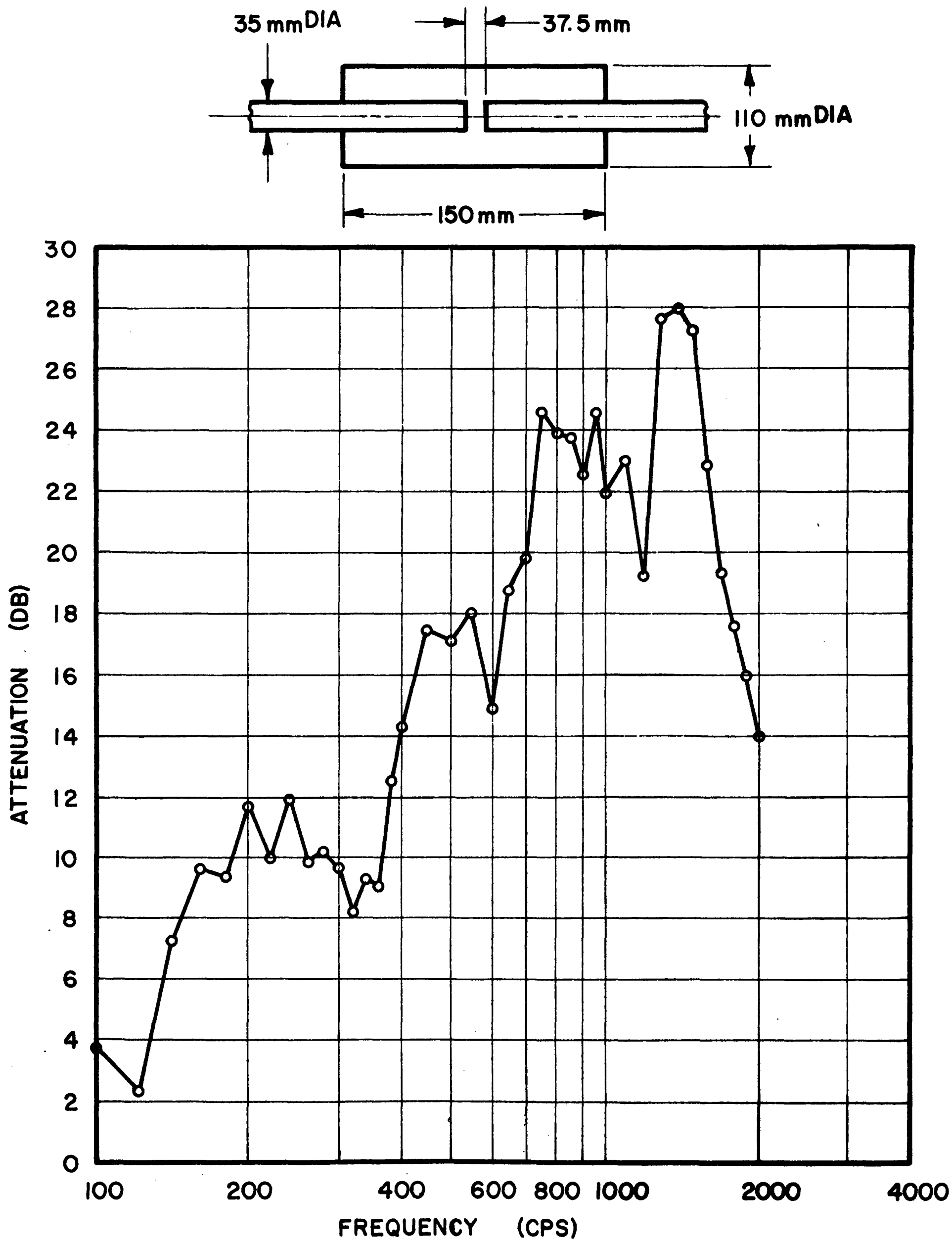



Fig. 4.15B. Measured attenuation of resonator and expansion chamber muffler (after Bentele, Ref. 3).

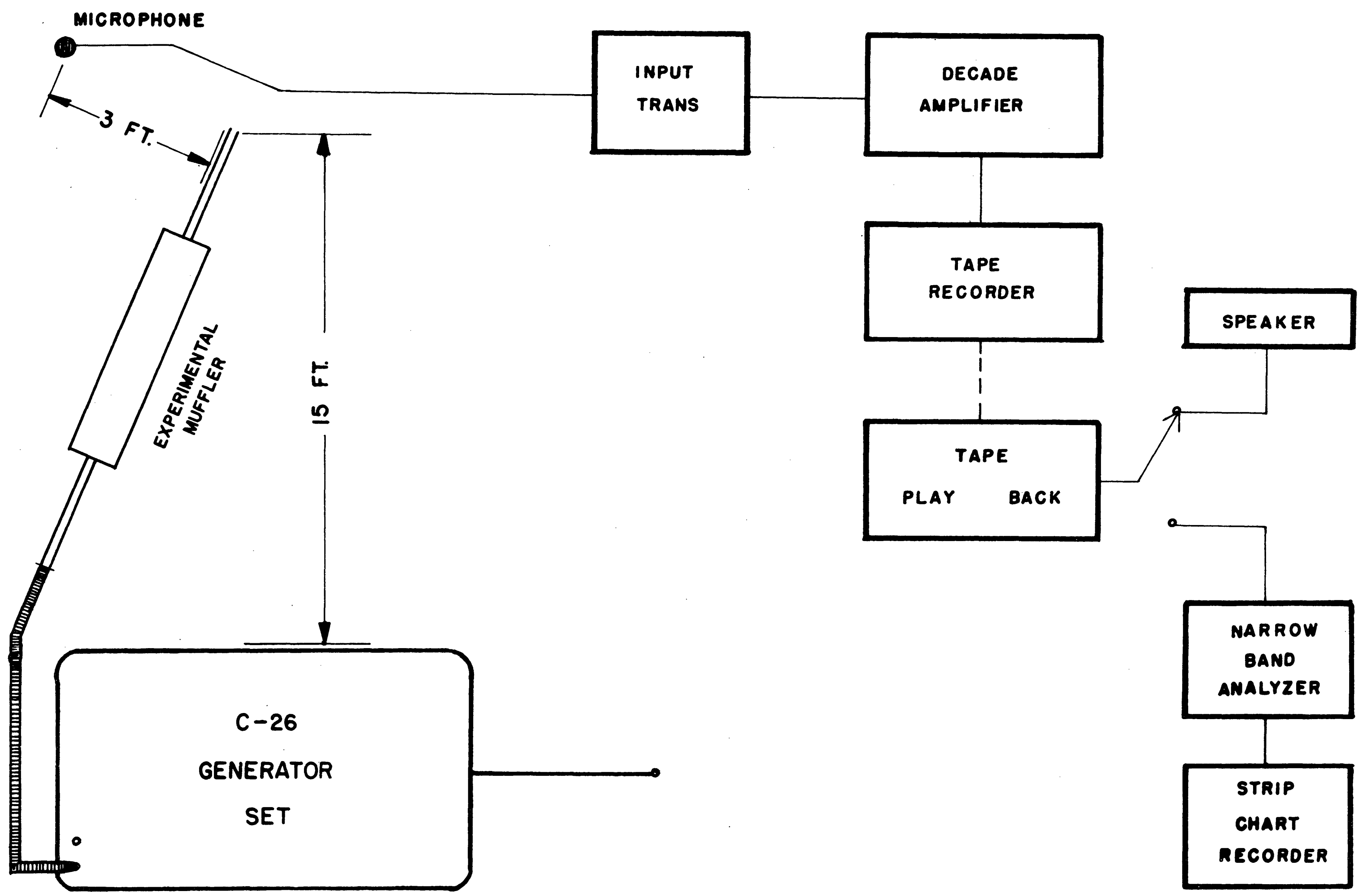



Fig. 4.16. Muffler evaluation measurement apparatus.

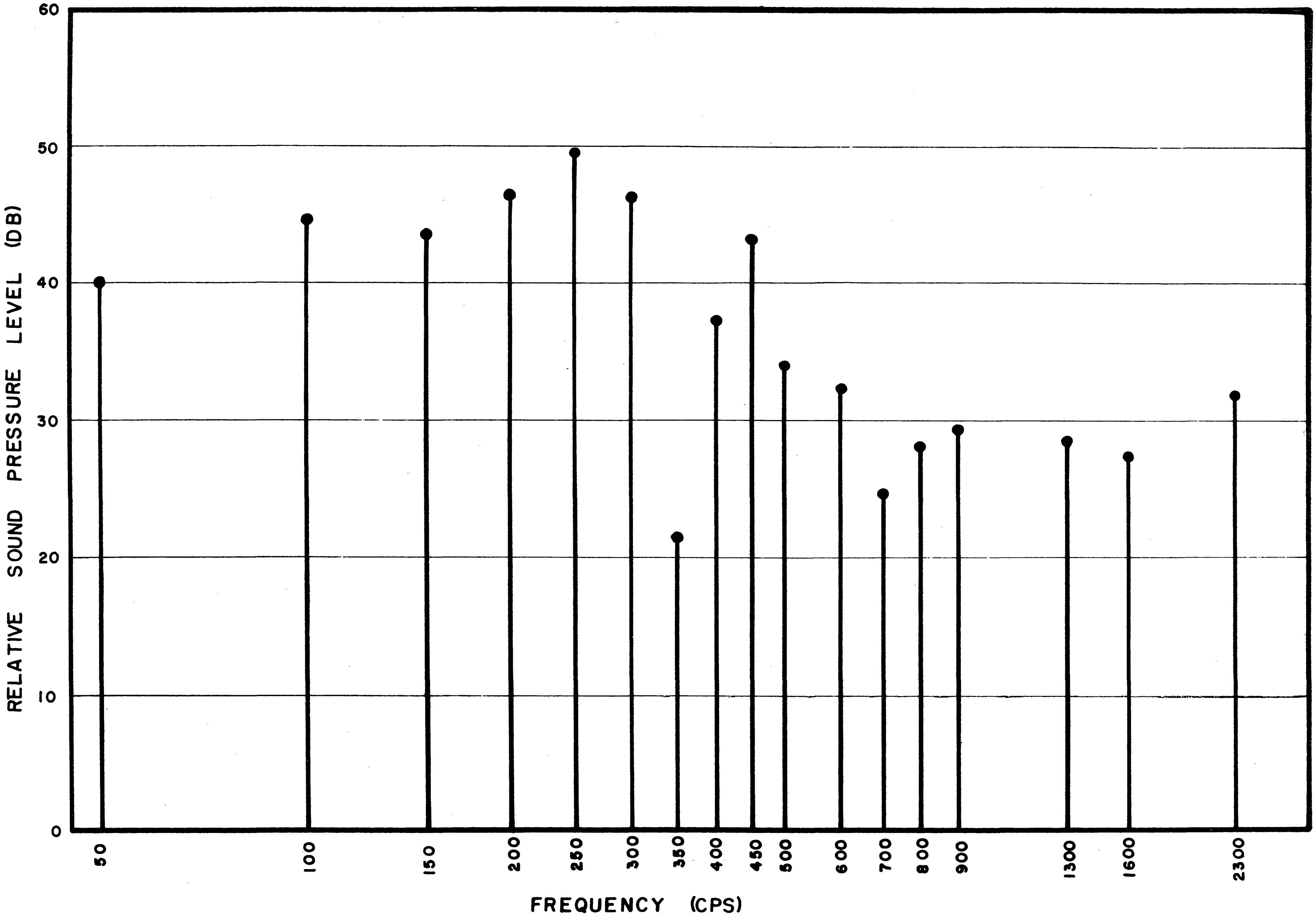
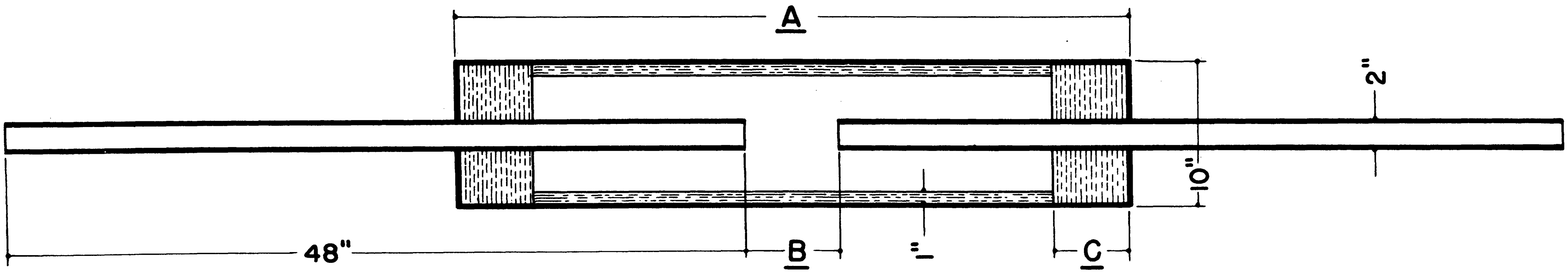



Fig. 4.17. Unmuffled exhaust noise spectrum.




A: CHAMBER LENGTH VARIED FROM 30" - 48"
B: GAP LENGTH VARIED FROM 1" - 28"
C: FIBERGLAS PACKING VARIED FROM 0" - 10"

Fig. 4.18. Experimental muffler (expansion-chamber-resonator combination).

The 48-in. muffler was tested after being lined with a 1-in.-thick blanket of a fibrous glass material having a density of 1/2 lb/cu ft and a fiber diameter of 4 microns. For gap widths less than 4 in., the addition of this lining increased the attenuations observed in the 150-to 300-cps band; however, as the gap width was increased from 4 in. the attenuation in this band decreased rapidly. The maximum overall attenuation, comparable to that obtained with the unlined chamber, was measured with a gap width of only 2 in. No significant change in the muffler's attenuation was observed after shortening the length of the chamber to 42 in. and repeating the measurements at different gap widths.

To obtain a standard of quality for making comparisons, the characteristics of the Walker 639 muffler which had previously been found satisfactory were measured. Then comparing these attenuations with those obtained with the 48-in. and 42-in. fiberglass-lined mufflers at a 2-in. gap setting, it was found that the experimental mufflers had a 4-db greater attenuation in the frequency range below 200 cps. However, above about 200 cps the attenuation of the Walker 639 was higher on the average by about 6 db so that in listening tests, using tape recordings, the Walker muffler sounded much better.

The chamber length was then shortened to 36 in. and the measurements were repeated using several different gap widths. When lined with a 1-in. fiberglass blanket and with 4 in. of fiberglass packing at the ends of the chamber, the 36-in. muffler produced maximum overall attenuation at a gap width of 20 in. Figure 4.19 shows the attenuations at this setting and permits a comparison with those obtained at gap settings of 2 and 8 in. In this figure, the attenuation is the db difference at each frequency between the level of the "no muffler" curve and the point, at that frequency, corresponding to the gap width being considered. It can be seen that nearly all the components lying above 200 cps were attenuated considerably more for the 20-in. gap setting than for the other two settings. Although some loss of attenuation at 50, 100, and 200 cps was observed at the 20-in. setting, only the 200-cps component sound-pressure level was appreciably higher than that of the average component level.

In an effort to increase the attenuation at 200 cps, two tests were made at the 20-in. gap setting, in one case increasing the thickness of the fiberglass packing at the ends of the chamber from 4 to 20 in., and in the other removing the fiberglass completely. The results of these tests, shown in Fig. 4.20, indicated that the 20-in. packing decreased the attenuation at 200 cps, and that removing all packing considerably increased it. However, without fiberglass the attenuation was sufficiently decreased at the high frequencies so the muffler sounded considerably worse than with 4 in. of packing.

The muffler was then shortened from 36 to 30 in. and tested. For this condition, the attenuation was decreased at 100, 250, 400, and 600, but was increased at all other frequencies. However, audibly the 30-in. muffler seemed to be much less effective because of the increased level of these several tones.

The experimental muffler program was discontinued at this point because it was felt that further testing was beyond the scope of the present study. The feasibility of constructing a muffler for a given engine had been adequately demonstrated as indicated by the comparison shown in Fig. 4.21 of the attenuations obtained using the Walker 639 muffler, and the experimental 36-in. muffler in the optimum arrangement, a 20-in. gap setting with 4 in. of fiberglass packing. It can be seen from the figure that comparable or even higher attenuations were obtained with the experimental muffler

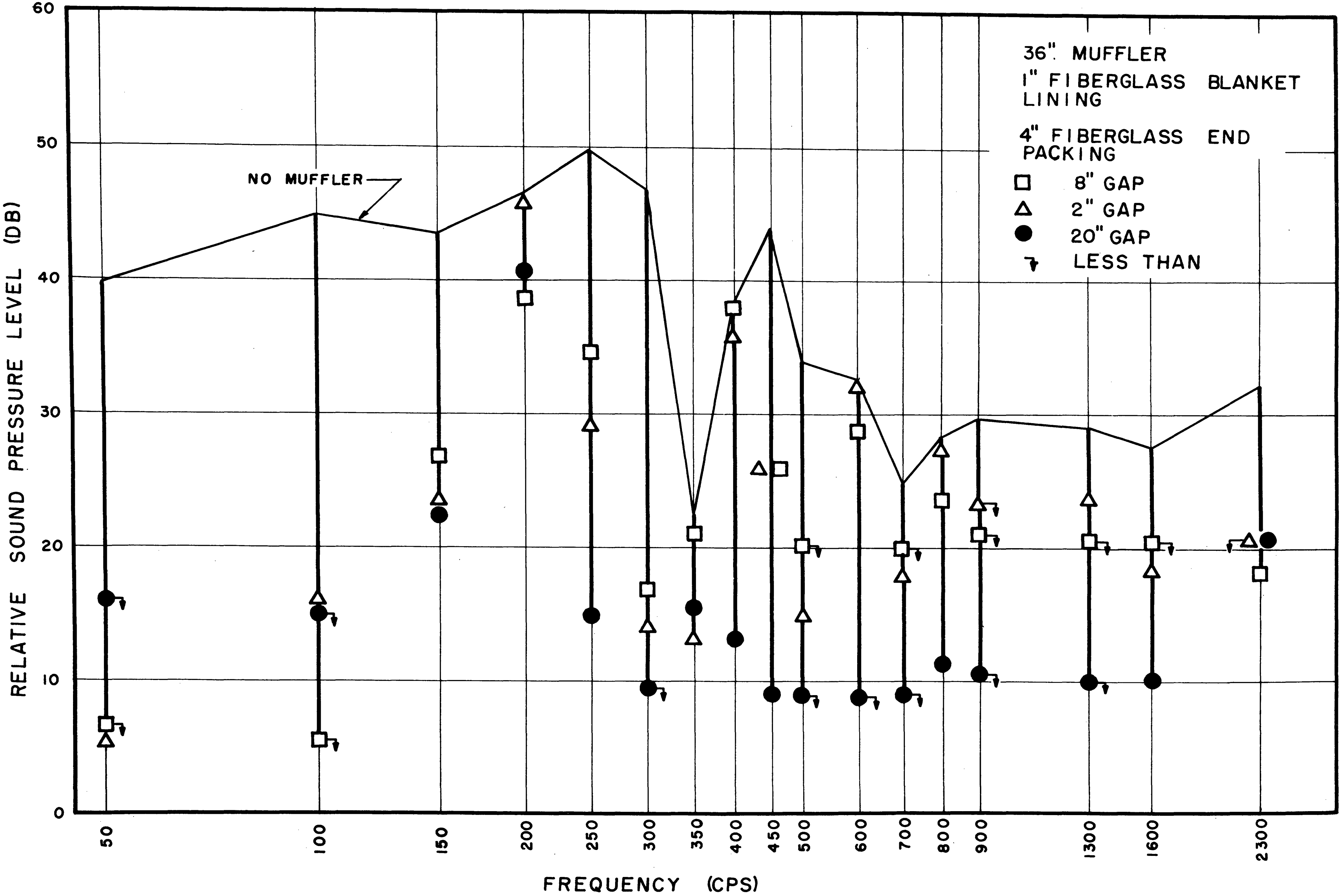



Fig. 4.19. Experimental muffler—effect of gap width.

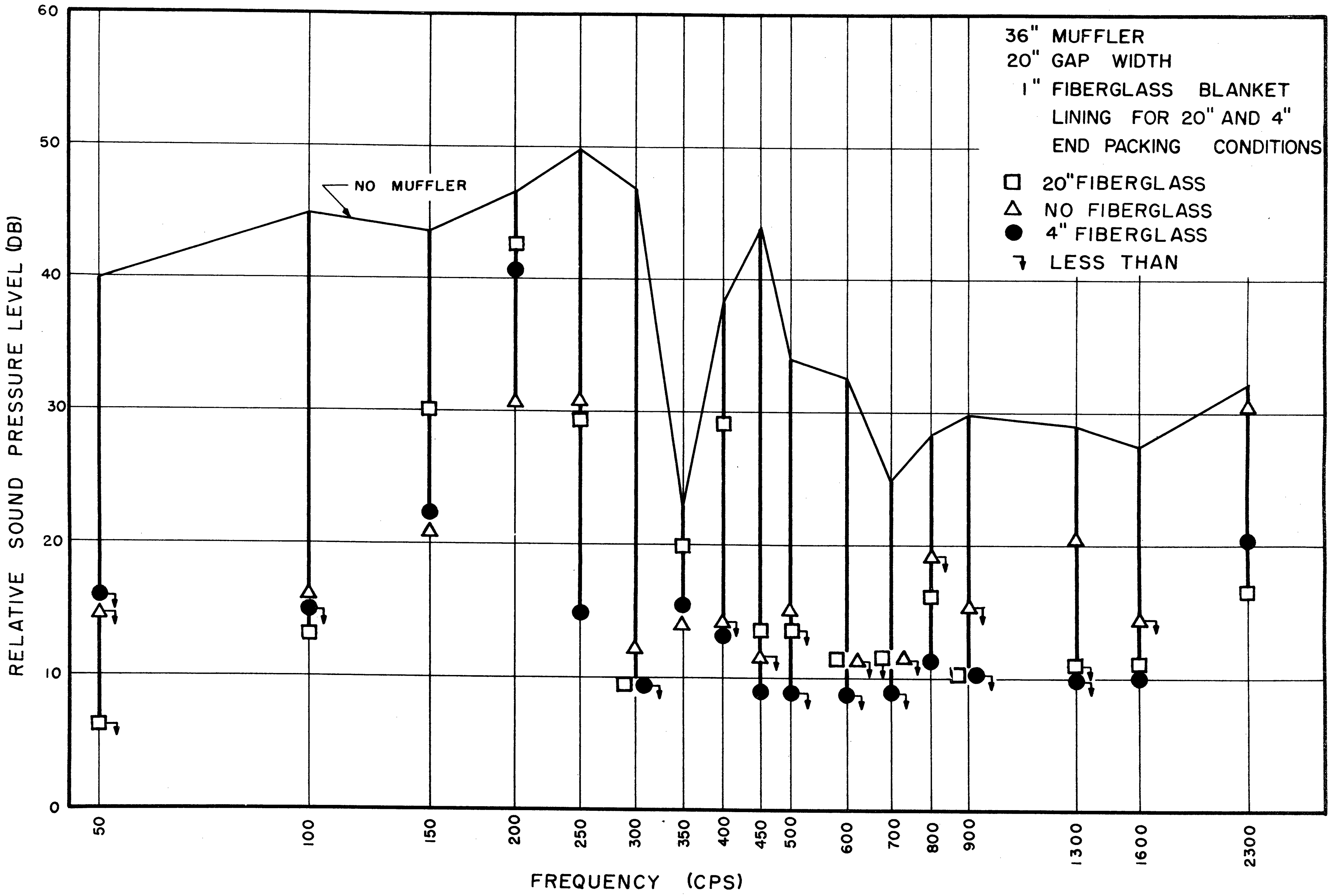



Fig. 4.20. Experimental muffler—effect of fiberglass blanket and packing.

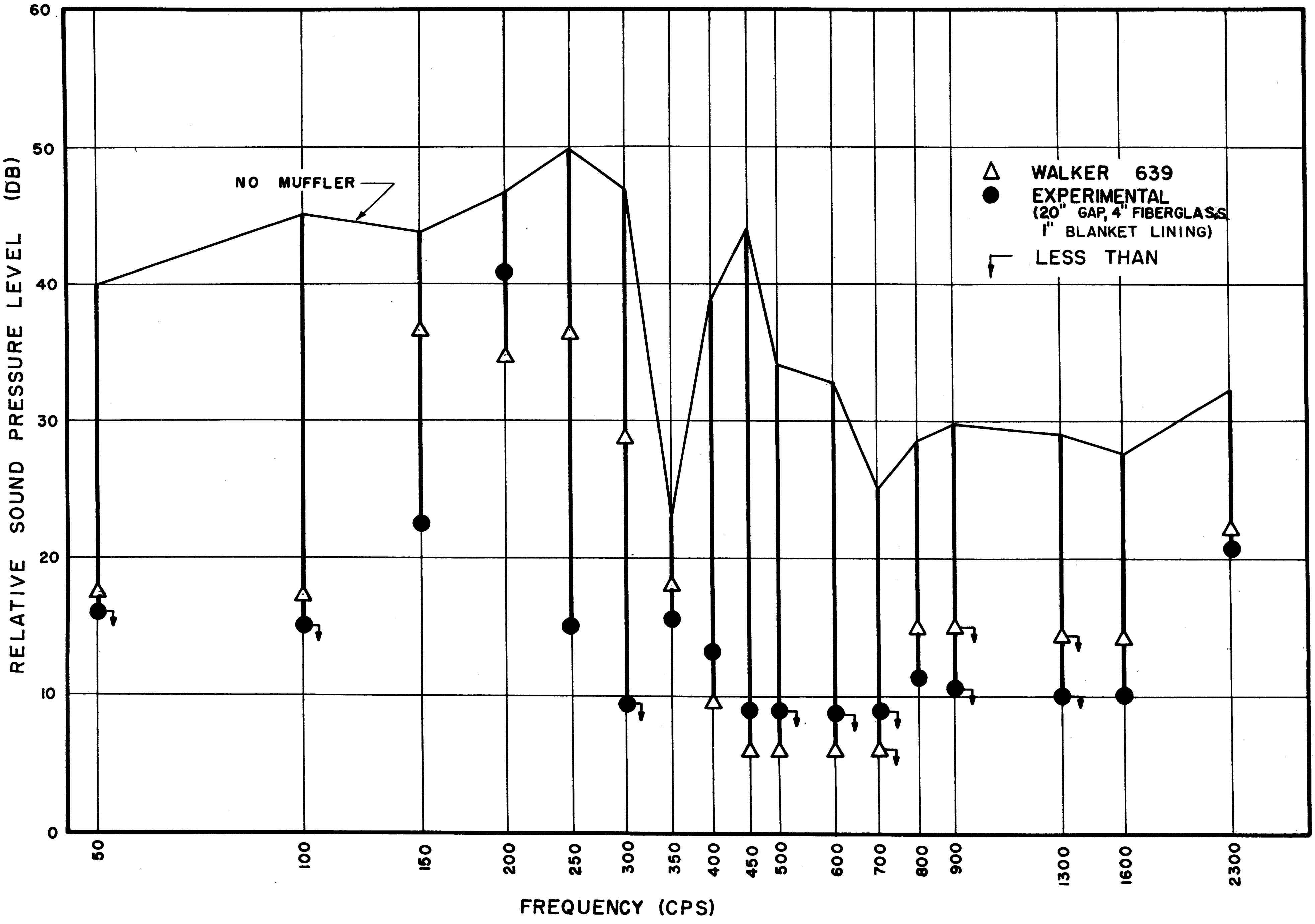



Fig. 4.21. Experimental muffler—comparison of Walker 639 muffler and optimum experimental muffler.

over the entire frequency range. Although not evident from the figure which shows only that somewhat higher attenuations were achieved with the experimental muffler, the exhaust noise with the experimental muffler was audibly much less objectionable than was the noise of the Walker muffler.

Had time permitted, an attempt would have been made to attach a resonator to the experimental muffler further to reduce the level of the 200-cps component. This component was not adequately attenuated by the experimental muffler because in this configuration the muffler had its first pass band at about 185 cps.

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## SECTION V

## NOISE-REDUCTION TECHNOLOGY APPLICABLE TO AIRCRAFT GROUND-SUPPORT EQUIPMENT

At the inception of work under this contract, the need was expressed for a survey of the literature of noise reduction for application to the ground-support-equipment nuisance-noise problem. When it was learned that an extensive literature survey had previously been supported by the Office of Naval Research with respect to noise-reduction techniques applicable aboard ship, the survey task was modified to that of reviewing, screening, and reorienting the Navy literature survey for application to noise problems related to aircraft ground-support equipment. To minimize the duplication of effort, the task was executed as the compilation of an annotated topical outline making explicit reference to the relevant items of technology drawn from their shipboard context and reorganized for the application at hand.

As such, the following review of noise-reduction technology is not intended to stand alone, not even as an exhaustive listing of the many considerations involved in the complete application of noise-reduction techniques to any given noise problem. Its use is postulated on explicit reference to two volumes available from the Undersea Warfare Branch (Code 466) of the Office of Naval Research, entitled, respectively, "Noise-Reduction Manual," by P. H. Geiger, and "Noise-Reduction Manual II, Machinery Noise," by N. E. Barnett, H. F. Reiher, and R. N. Hamme, both prepared at the Engineering Research Institute of The University of Michigan under the auspices of Office of Naval Research Contract N6onr-23219. Throughout the topical outline that follows, itemized reference is made to these manuals by the numerals in parentheses behind each entry, the Roman numeral referring to the appropriate volume and the arabic to the appropriate pages. Notations are included making reference to specific items of ground-support equipment wherever the application of a given entry in the outline has been demonstrated experimentally in the course of work under this contract, although such identifications have not been included exhaustively because of the detail of the organization of the outline. Paragraph annotations are included regarding the applicability of classes of noise-reduction techniques.

The content of the outline has been organized into the basic categories of "Noise Reduction at the Source" and "Palliative Noise Reduction," the distinction being between techniques which reduce the generation of disturbances, as contrasted to those that seek their confinement or dissipation without regard for their mode of generation. The latter are subdivided into four basic mechanisms of noise confinement, representing both the isolation and the dissipation of airborne and structure-borne sound, respectively. Each of these techniques represents a complete technology for the application of a separate class of acoustical materials, interdependent in practical usage but conceptually distinct. In this respect the outline is intended to suggest a broad philosophy of noise reduction, its specific application to aircraft ground-support equipment being almost incidental to this thesis. The wide variety of noise-producing elements represented in ground-support equipment makes this approach to noise control but only understandable but also indispensable.

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      (3) Shaft whirl (II, 159)
   b. High-frequency noise due to blades passing steam nozzles or stator blades (II, 159)
   c. Unpitched noise of turbine journal bearings (II, 160)
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      (7) Damping by shroud ring (II, 163)
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      (3) Splitters and sound absorbing material in ducting (II, 166)
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         (ii) Shape (II, 168-169)
      (4) Turning vanes (II, 169) <u>MA-1 exhaust</u>
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   c. Mechanical balance (II, 170-171)
      (1) Minimum of six cylinders required (II, 170)
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---

*This technology is applicable to the selection, manufacture, design, and assembly of ground-support equipment and components, emphasis being directed toward systematic and consistent specification of noise-reduction requirements with respect to (1) the relative noise contribution of each component during a representative service cycle, and (2) the relative applicability of palliative noise-reduction techniques.

(3) Centrifugal pendula (II, 170)

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e. Engine component noise (II, 171-177, 240-242)

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b. Factors in generation of gear noise (II, 194-197, 201, 204-217, 227)

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(5) Three-ball balancer - self-balancing (II, 277-278)

II. Palliative Noise Reduction*

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a. Air friction (I, 27)
b. Fiber friction (I, 28)

*This technology is applicable in all cases where confinement of a disturbance in the immediate neighborhood of the source will constitute a reduction of nuisance. Economically it is preferable to extensive noise reduction at the source in initiating a noise-control program.

**The absorption of airborne sound, or its dissipation in the form of heat, is palliative in the sense that (1) noise levels within source enclosures are reduced, (2) airborne sound isolation can be fully realized by reactive enclosures, (3) airborne flanking paths and leaks can be sealed without mechanical closure, and (4) environmental conditions are improved. The technology is directly applicable to the lining of the cart enclosures of ground-support equipment.

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    b. Frequency dependence (I, 28)
    c. Noise-reduction coefficient, NRC (I, 29)
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4. Effect of Mounting (I, 29)
5. Effect of Painting (I, 32)
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    b. Doors (I, 49)
    c. Ventilating ducts (I, 49)
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    b. Absorbing linings (I, 51)
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    b. Need for absorption (I, 53)
6. Sound Transmission Through Ducts (I, 54) <u>A-1 intake blower</u>
    a. Structure-borne sounds (I, 54)

---

*Insulation against airborne sound, or its confinement by barriers in the propagation path from the source to the point of disturbance, is palliative in the sense that reactive attenuations manifest themselves as general noise reduction whenever concurrent provision is made to absorb the energy of the source. The barrier is a vehicle for redirecting noise to the locations of convenient absorption, as well as reacting on the noise source. The technology is directly applicable to the cart enclosure of components of ground-support machinery.

(1) Resilient duct couplings (I, 54)

b. Sound-absorbing linings (I, 54-56)

(1) Mounting (I, 55)

(2) Frequency dependence (I, 56)

(3) Attenuation per foot (I, 54, 55)

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(1) Right angle bend (I, 56)

(2) Effect of splitters (I, 57) MA-1 intake

(3) Use of baffles (I, 57) C-26 vent louvers

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(2) Fabrication (I, 75-76; II, 610-611)

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(3) Fibrous blankets (I, 90-93) C-26 cart lining

(i) Surface loading (I, 90, 92)

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(iii) Airborne sound absorption (I, 91)

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(v) Thickness and weight (I, 91, 93)

(4) Miscellaneous deadeners (I, 94, 95)

(5) Frequency-selective spot deadener (I, 96)

(i) Effectiveness vs frequency (I, 96)

(ii) Principle of operation (I, 96, 98, 99)

---

*Vibration damping, or the conversion of mechanical vibratory energy into heat, is palliative in the sense that (1) resonant amplifications of source vibrations are avoided before their radiation as noise, (2) the average noise levels of recurrent impact are reduced by elimination of ringing, and (3) the character of the noise of variable speed sources is rendered more uniform. The technology is uniquely applicable to sheet-metal enclosures such as ground-support equipment carts.

        (iii) Tunability (I, 98)
        (iv) Antinodal location required (I, 99)

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      (1) Force transmissibility (I, 103-105)
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      (1) Amplitude limiter (I, 109)
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*Vibration isolation, or the confinement of structure-borne vibratory energy to the source, is palliative by preventing the excitation of efficient radiating surfaces and by mechanically decoupling sound sources and sound barriers. The technology is applicable directly to the mounting of small components in ground-support equipment carts and the mounting of hoods and panels, and applicable with refinement to the mounting of main components.

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(3) Machine shock protection (I, 109; II, 460-464)
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(1) Drift in rubber (II, 444, 498-500)

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(3) Effect of paint on rubber (II, 500-501)
(4) Temperature effects (II, 503)
e. Acoustical shorting (II, 399, 435, 504-507, 511, 542, 549-550)
(1) Mechanical shorting (II, 435, 503, 511, 549-550)
(2) Airborne feedover (II, 504-507)
(3) Liquid-borne feedover (II, 506-507, 542)
3. Commercial mounts (I, 111; II, 413)
a. Shear type (I, 111-113, 117; II, 413-416, 426-431, 448-451)
(1) Springs (I, 111-112; II, 413-415, 427-430)
(i) Load and clearance limitations (II, 415)
(ii) Buckling (II, 415)
(iii) Damping (I, 111-112; II, 428-430)
(2) Rubber (I, 111, 113, 117; II, 416, 420-423, 426, 431, 448-451)
(i) Rail-type mountings (I, 113; II, 420-422, 431)
(ii) Ribbed pads (I, 117; II, 423)
(iii) Equi-stiffness mounts (II, 426)
(iv) U. S. Rubber "3-Angle Safety" mount (II, 448)
(v) Lord "Plate-Form" mounts (II, 450)
(vi) U. S. Rubber "Finger-Flex" mount (II, 451)
b. Compression-type mounts (I, 114-117; II, 417, 424-425, 433)
(1) Bonded vs unbonded pads (I, 115-117; II, 424-425)
(2) Extruded rubber washers (I, 117)
(3) Cork, felt, fibrous glass boards (II, 425)
(4) Multilayer (II, 433)
4. Shaft Couplings (I, 134; II, 511)
a. Flexible shaft couplings (I, 134; II, 512, 517)
b. Resilient shaft couplings (I, 134; II, 512-517)
(1) Light-duty rubber (II, 512)
(2) Medium-duty rubber (II, 513)
(3) Airflex coupling, (II, 514)
(4) Toothed-type or Bibby-type couplings (II, 516-517)
c. Belt drives (I, 134; II, 519-520)
(1) Flat belt (I, 134; II, 520)
(2) V-Belt (I, 134; II, 520)
(3) Tensioning arm (II, 520)
(4) Isolating rubber bushings (II, 520)
d. Automotive torque-tube drive (I, 135)
5. Other Connections to Machinery
a. Pipe connections (I, 130-133; II, 522-525, 529, 532-535, 544-549)
(1) Points of attachment (I, 130; II, 549)
(2) Resilient connectors (I, 130-133; II, 522-525, 529, 532-535, 544-548)
(i) Rubber tubing (II, 522, 525)
(ii) Long flexible loops (I, 131; II, 548)
(iii) Strong tubing coils (I, 130; II, 532, 535)
(iv) Commercial reinforced hose (I, 132; II, 523, 529)
(v) Plastic hose connectors (II, 547)
(vi) Flexible metal tubing (I, 132-133; II, 524-525)
(vii) Interlocked flexible metal tubing (I, 132; II, 524, 534)

(viii) Vulcanized rubber-metal pipe couplings (II, 529, 545)
(ix) V-configuration (II, 544-547)
(3) High-pressure connections (II, 532, 544)
(4) Ventilation duct connectors (II, 540)
(5) Rubber-lined pipe hangers (II, 535, 548-549)

b. Electrical connections (II, 551-555)
(1) Cables (II, 551-554)
(i) Flexible configurations (II, 551, 553-554)
(ii) Rubber-lined clamps (II, 552)
(2) Grounding (II, 554-555)
(i) Conducting rubber in mount (II, 555)
(ii) Flexible braided strap (II, 555)

c. Control attachments (II, 555-557)
(1) Flexible shafts (II, 556)
(2) Flexible push-pull controls (II, 556)

# APPENDICES

## APPENDIX A

## PROPOSAL TO WRIGHT AIR DEVELOPMENT CENTER ON NOISE REDUCTION IN GROUND-SUPPORT EQUIPMENT (P.R. No. 207969)

### INTRODUCTION

It is requested that Wright Air Development Center review the merits of the engineering study described herein and consider the advisability of sponsoring the project at The University of Michigan. This proposal is submitted in reply to the WADC Request for Proposal - Purchase Request No. 207969, dated 25 October 1954. The study is planned in accordance with WADC Exhibit WCLEF-385 dated 17 September 1954, and in accordance with discussions between Mr. James Payne, WCLEF-2, and Messrs. R. N. Hamme and W. E. Quinsey, The University of Michigan, during inspection of one of the items of aircraft ground-support equipment involved.

### INTERPRETATION OF PROBLEM

The statement of the requirements, Paragraph 3 of Exhibit WCLEF-385, and the obvious urgency of noise reduction in ground-support equipment indicate that a distinction should be made explicit at the outset between what is called noise reduction at the source and the various techniques of sound isolation. By noise reduction at the source is meant the process of fundamental machinery redesign for the purpose of reducing the periodic mechanical forces which are responsible for the generation of noise; e.g., skewing armature slots in rotating electrical equipment, smoothing flow, and eliminating blade resonances in turbine wheel designs, etc. This process is both time-consuming and expensive out of all proportion to the spirit of the Exhibit, primarily because of the elaborate precision noise measurements that are required at each stage for the identification of the origin of each of the individual noise components. For example, octave-band noise measurements are virtually useless as the point of departure for noise reduction at the source.

For these reasons, noise reduction at the source will not constitute an essential element of this proposal, except insofar as reference will be made to the extensive experience and published studies of this laboratory in the field of noise reduction at the source. Rather, the emphasis will be laid on exploiting the methods of noise isolation with a view to dramatic noise reductions at reasonable cost.

By noise isolation is meant the process of impeding the escape and/or radiation of noise from the machinery components and their housing with little regard for the detailed mechanisms by which the noise components are generated. To this end the various techniques of vibration damping, sound absorption, vibration and sound isolation, etc., are exploited to their utmost. Specifically, vibration damping treatments are employed to minimize resonant response of sheet-metal housings, etc. Airborne sound isolation techniques are employed to confine high levels within the housings; e.g., transmission loss of the housing is enhanced and maintained by eliminating "leaks," by utilizing resonant-chamber and absorptive exhaust and intake mufflers and ducting, and by exploiting the techniques of surpassing "weight-law" sound transmission loss. Resilient mount methods of vibration isolation are used to prevent the transfer of vibratory energy to potentially radiating surfaces, and sound absorption treatments are used to reduce source levels within enclosures in order to take maximum advantage of the enclosure's transmission loss properties. For the application and evaluation of these techniques, the octave-band noise measurements specified in the Exhibit are entirely adequate.

On the basis of inspection and demonstration of the Type C-26 Generator Set, and on the basis of past experience with the Type MA-1 Air Compressor, it is judged that the present status of these items of ground-support equipment is such that enormous noise reductions can be accomplished by the techniques of noise isolation alone, to such an extent indeed where long-term noise reduction at the source undertaken afterward might well be regarded as practically and economically unfeasible by comparison.

BACKGROUND EXPERIENCE OF THIS LABORATORY

Under the supervision of the late Dr. Paul H. Geiger, this laboratory has pioneered the systematization of noise reduction and noise measurement since 1929, being actively and continuously engaged in all phases of this work since that time. The present supervisor and the nucleus of the present research group have been associated with the laboratory for more than ten years, participating during that time in the following WADC acoustical investigations for the Aircraft Laboratory:

| Date | Contract |
|---|---|
| 1943 | (33-038)44-1475E |
| 1944 | (33-038)44-4300E |
| 1947 | W33-038-ac-14775 |
| 1949 | AF33(038)-2652 |
| 1952 | AF18(600)-56 |
| 1954 | AF33(616)-2333 |

In addition to numerous industrial noise-reduction studies (automotive, household appliance, industrial machinery, office equipment, etc.), the laboratory has been under contract since 1948 to the Undersea Warfare Branch of the Office of Naval Research for the experimental and analytical study of noise reduction in ship and submarine machinery and components (ONR Contracts N6onr-23211 and N6onr-23219). In connection with this work, there has been completed an exhaustive survey of American and foreign literature on noise reduction which culminated in publication of a 552-page volume entitled "Machinery-Noise Reduction." This volume, which deals in detail with both noise reduction at the source and techniques of noise isolation is currently undergoing complete revision into three volumes entitled "Noise-Reduction Manual I, II, III." A copy of Volume I has been submitted to WADC WCLEF-2, and Volume III entitled "Noise-Reduction Manual III, Propeller Noise" must, because of its classification, be obtained directly from ONR, Code 466, Washington 25, D. C. Volume II entitled "Noise Reduction Manual II, Machinery Noise" will be issued shortly. Our work in surveying and interpreting the noise-reduction literature is emphasized because of its direct applicability to Paragraph 3.3.1 of the Requirements of the Exhibit.

Wide contact has been established and maintained through the years with acoustical materials manufacturers, and extensive materials surveys have been conducted here on the sound absorption, vibration damping, and sound transmission characteristics of acoustical materials. Complete apparatus is maintained for rapid and precise determination of sound absorption coefficients by the reverberation room method, vibration damping effectiveness by the thick-plate decay-rate method, and sound transmission loss by both the two-room method and the tube method. Apparatus is maintained for making octave-band sound-level measurements in accordance with American Standards Association specifications, and the requisite analyzers and recorders are available for making the discrete-frequency noise analyses required for noise-reduction-at-the-source studies.

University ownership of the nearby Willow Run Airport and the close proximity of the Engineering Research Institute's Willow Run Research Center make facilities readily available for the "free-field" measurements required for the work proposed.

PROPOSED RESEARCH PROCEDURE

The review of literature pertinent to the concepts of hazardous nuisance levels (a subject not specifically included in our ONR studies inasmuch as detectability of radiated noise, rather than hazardous nuisance level, establishes the naval noise-reduction criteria) would be undertaken immediately and continued throughout the course of the work in an effort to establish allowable levels in the various frequency bands in order to direct and evaluate the noise-reduction effort. Review of literature covering noise-reduction techniques, methods, materials, etc., is rendered largely unnecessary by our ONR survey which would be abstracted and pointed up directly toward the ground-support-equipment noise-reduction problem. No specific study would be undertaken toward basic extension of the theory and practice of resonant chamber noise-reduction principles, these being extensive studies in themselves beyond the scope or need of the present work, inasmuch as the status of commercial exhaust and intake technique and resonant absorption techniques are adequate for the present needs. No analytical investigations beyond those currently on record are believed to be fruitful in the approach to the present noise-reduction problem.

Immediately upon delivery of one or both of the machinery items a) and b), Paragraph 3.5.1 of the Exhibit, free-field measurements would be completed as specified in Paragraph 4.4 to establish the point of departure, and the equipment would then be moved to the laboratory, if possible, for modification. Noise isolation would proceed consistent with the principles hitherto established here, with repeated measurements after each major modification in order to assess the relative value of each change. Concurrently, an inspection would be made, and if feasible noise measurements would be completed on the items of equipment listed in Paragraph 3.4.1 of the Exhibit, with a specific view toward establishing the general applicability of the laboratory modifications.

With exception of the backpressure inherent to an exhaust muffler on the internal-combustion engine, it is not anticipated that any of the noise-isolation techniques proposed will affect environmental or check-run data appreciably, though provision will be made for verification of performance characteristics where deemed necessary by the contractor.

## APPENDIX B

## INSTRUMENTATION FOR NOISE MEASUREMENT AND CALIBRATION PROCEDURE

### FREE-FIELD MEASURING SITE

The free-field octave-band survey has been one of the major acoustical measurement techniques employed in the experimental noise-reduction phase of this research program. Such measurements were used for diagnostic purposes and they constituted a standard measurement in terms of which all noise reductions were evaluated. Moreover, because it is important to make standardized free-field measurements which, within the stability of the source and the accuracy of the instrumentation, can be repeated by any other laboratory, considerable care was exercised in selecting the measuring site, in calibrating the instrumentation, and in developing the procedures used during the free-field surveys.

An acoustically free field means that the test site is free of anything which might reflect or distort the sound field radiated from the source (other than the ground plane itself) in any way which would alter the sound levels measured at the selected microphone locations. Obviously this implies a flat open field devoid of obstructions such as buildings, trees, steep banks, etc., which could reflect sound back to the microphone. It is also tacitly assumed that the area is remote from man-made noises which could interfere with the measured noises. For convenience, the area must be readily accessible, free of maintenance difficulties, and close enough to the laboratory base of operations to make transit times reasonable.

In the present case, a free-field test site was selected in the North Campus area situated to the northwest of the Cooley Building where the laboratory activities were concentrated. This site satisfied the general requirements stated above and assurance was obtained from the cognizant University authorities that this site would remain available and unchanged for the duration of this contract. A portion of the test site is illustrated in Fig. B.1 in the form of a topographic map.

The usual method of free-field testing requires that the source be placed near the center of the test site and that the microphone be moved to various locations around the source. While this procedure could have been utilized at the selected site, it was felt that a somewhat altered procedure had obvious advantages and would yield just as reliable if not actually better free-field data. In this case, the machine under test was located as shown in Fig. B.1 at the intersection of two dirt roads. The microphone was located at the appropriate distance away to the east, also on the dirt road. The microphone was maintained fixed at this location and the machine under test was rotated at its location to produce the desired angular bearing relationship between the machine and microphone.

There were a number of reasons for adopting this altered procedure. First, the use of an existing road meant that the test site would be accessible all year around. Operating in the middle of a field would have required building a road to avoid sinking into the mud and plowing the snow in the winter, both undesirably expensive and time-

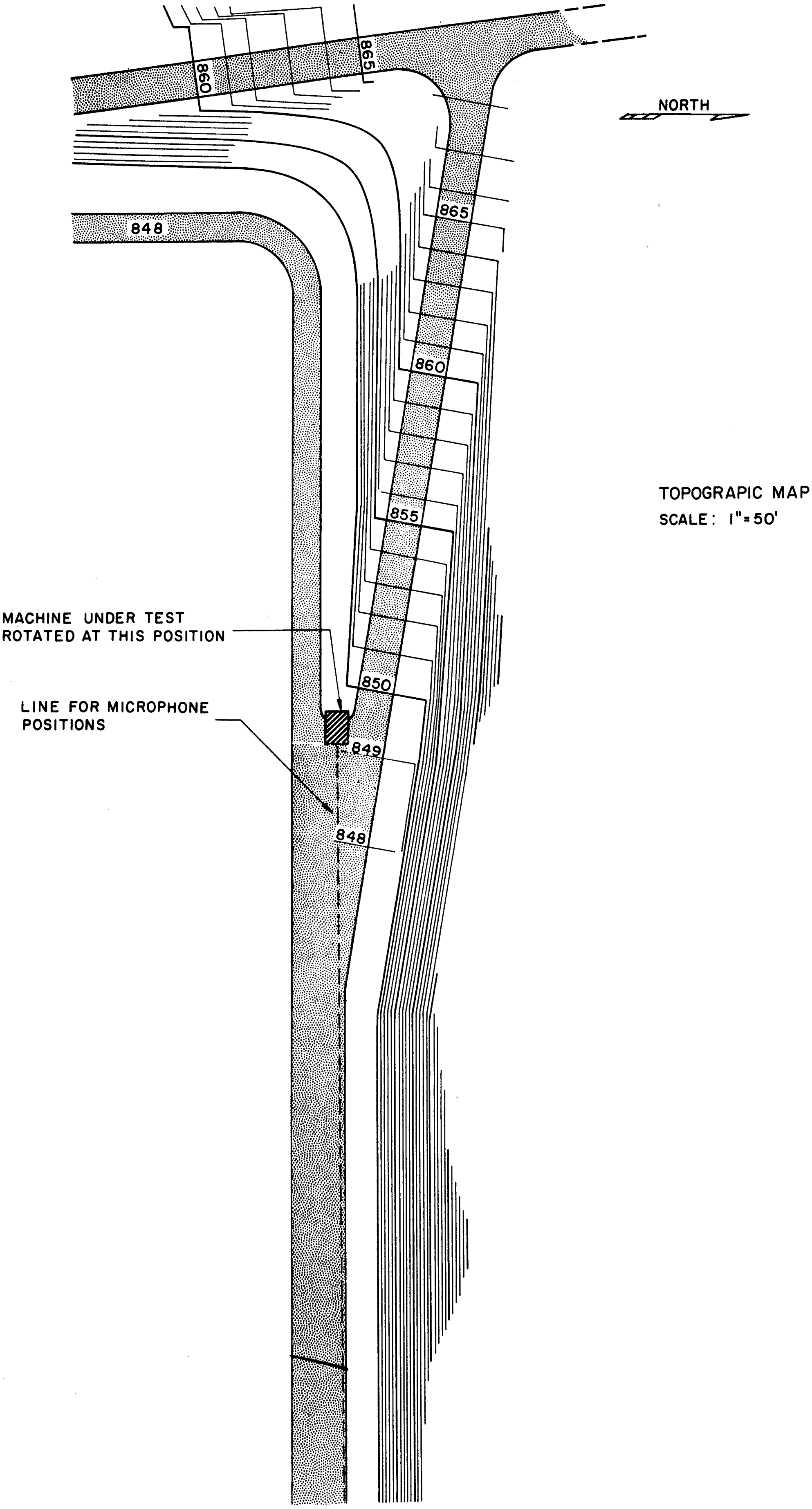



Fig. B.1. Topographic map of free-field test site.

consuming operations. Secondly, the dirt road provided the most invariant acoustic path available between the source and the microphone both in summer and in winter. Sound received at a distant microphone is somewhat dependent upon the acoustical characteristics of the ground path between it and the source. Thus the measurements obtained over a path that varies from high grass to mowed grass to snow-covered can be expected to be somewhat variable also. Consequently the dirt road utilized here constituted the most acceptable and invariant acoustical path available. Finally, the technique of rotating the source and fixing the microphone location permitted optimum use of the available space. In the direction of the microphone a level path free of acoustical obstruction at least 1000 feet long was obtained and even then the obstructions were oriented in such a way that the directly reflected sound could not find its way back to the microphone. In other directions, the ground was not always level enough to provide satisfactory microphone locations, but sloped gently enough to avoid acoustic influence at the measuring microphone. Furthermore, although the acoustic nature of the surface covering in these other directions, i.e., grass or snow, could not be controlled, it could at most exert a second-order influence on the measurements.

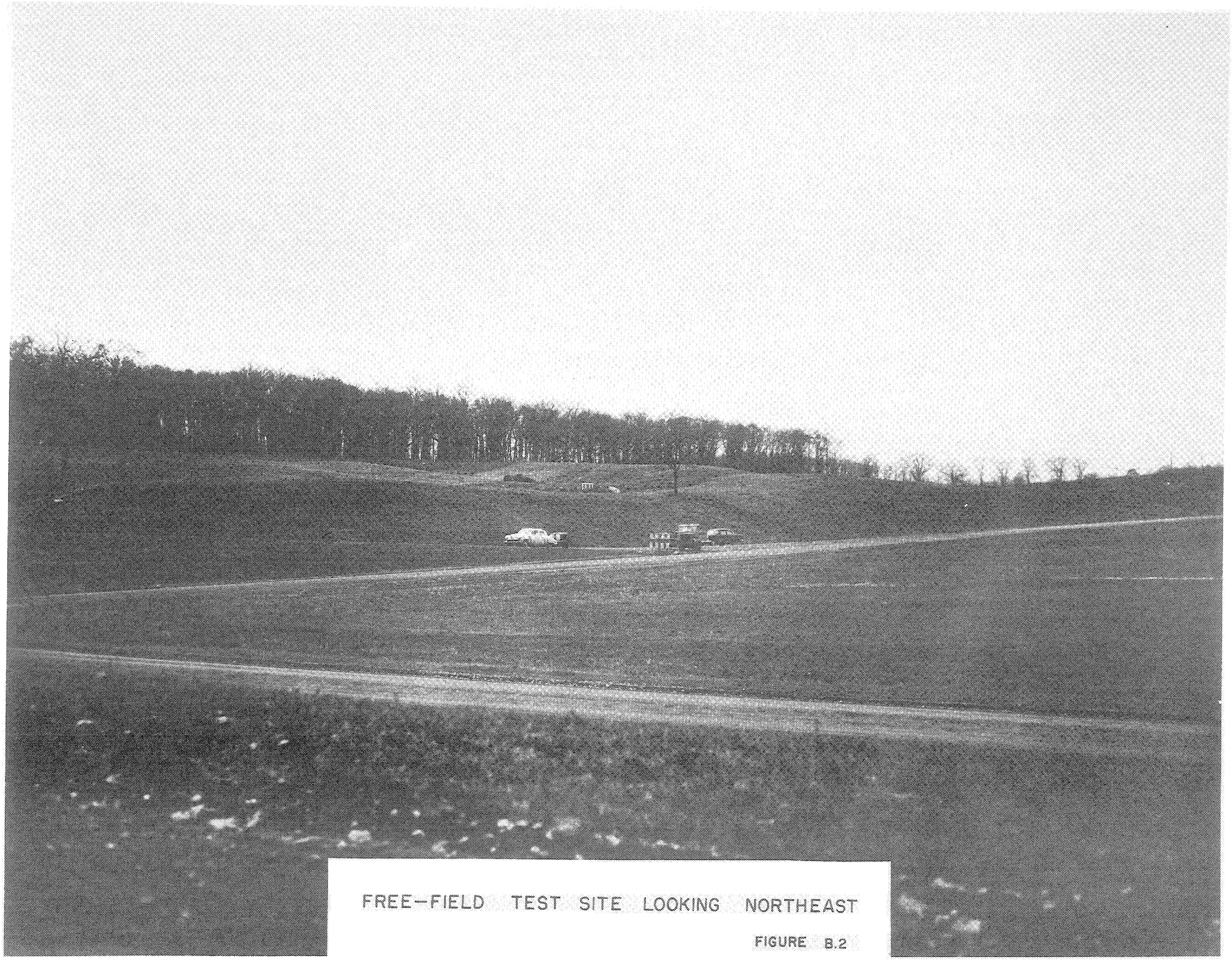
Figures B.2, B.3, and B.4 are photographs of the free-field test site looking northeast, due east, and due west, respectively, across the measuring area, taken during measurements on the Consolidated MA-1 Multipurpose Unit. Figure B.5, taken facing northwest, shows the relation of the microphone (placed 40 feet from the Consolidated MA-1) to the item of ground-support equipment under test. The closest obstructions are the trees, which are over 500 feet away.

For all free-field measurements the microphone was placed at least five times the major dimension of the source machine away from the center of the source. This placed the microphone well beyond the source's near field sound. On the other hand, extreme microphone distances were avoided to minimize effects of differential air absorption and scattering due to humidity and wind, and to minimize interference due to background noise originating from either human activity or wind noise at the microphone. In actual practice, a measuring range of 40 feet from the geometrical center of the source was used for all tests except those on the A-1 Generator Set, where because of the smaller physical size, a range of 30 feet was selected. In all cases, the microphone was placed at a height of 5 ft 5 in. above the ground which approximates the ear level of a standing man and is reasonably far above the reflecting ground plane.

## MICROPHONE CALIBRATION

All microphone calibrations were accomplished by a substitution method employing a Kellogg Condenser Microphone, serial No. 1289, as a secondary standard. Two different Altec 633A microphones, serial Nos. A1927 and A1933, were used for the octave-band sound-pressure measurements during the course of this research. The calibrations were accomplished in the laboratory utilizing the termination chamber, illustrated in Fig. B.6, of the sound transmission test facility as a small anechoic room. For this purpose, the open end of the chamber was closed with an absorptive baffle containing a loud-speaker source.

Calibration is accomplished by first adjusting the source to furnish a particular sound level as indicated by the secondary standard microphone. The microphone being calibrated is then substituted for the secondary standard and its output measured while



FREE-FIELD TEST SITE LOOKING NORTHEAST

FIGURE B.2

FREE-FIELD TEST SITE LOOKING DUE EAST

FIGURE B.3

FREE—FIELD TEST SITE LOOKING DUE WEST

FIGURE B.4

FREE-FIELD TEST SITE LOOKING NORTHWEST

FIGURE B.5

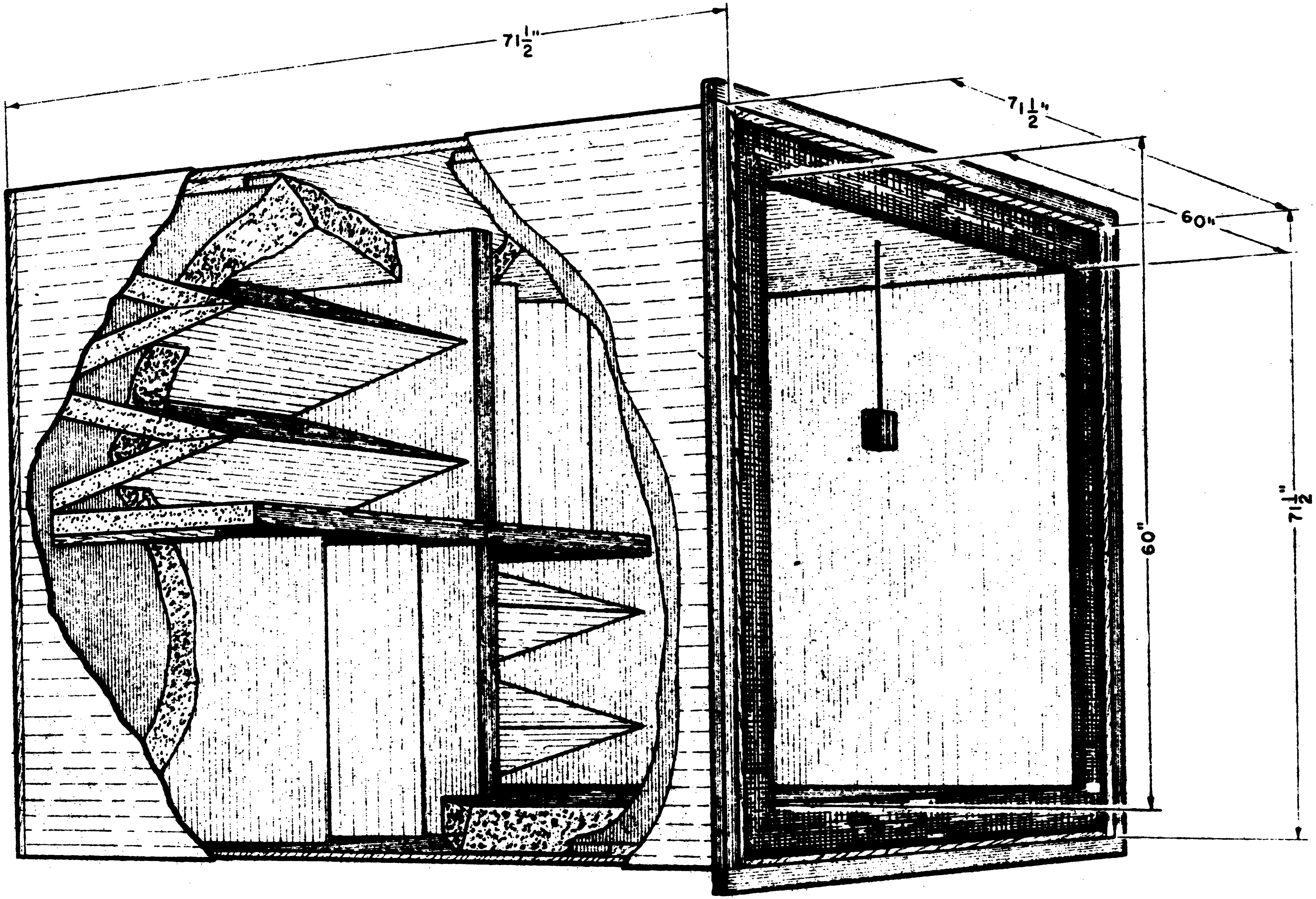



Fig. B.6. Absorptive termination enclosure for microphone calibration.

maintaining the source at the selected level. Then the microphone's absolute sensitivity can be computed using the known sensitivity of the secondary standard microphone. Figure B.7 shows a block diagram of the instrumentation involved. Calibration was accomplished in the above manner at 23 individual frequencies ranging from 45 to 10,800 cps. The microphone sensitivities derived in this manner are shown in Figs. B.8 and B.9. Repeated calibrations spread over considerable periods of time have shown excellent reproducibility of calibration both for the two microphones used on this project and other microphones used in this laboratory.

SOUND-LEVEL-METER CALIBRATIONS AND COMPUTATION OF CORRECTION FACTORS

The sound-level meter and associated octave-band filter set were carefully calibrated and were determined to conform to the specifications of the American Standards Association Bulletins Z24.3 - 1944 and Z24.10 - 1953. The flat frequency range and slow meter settings of the sound-level meter were used for all measurements.

The octave-band filter employed was an Allison Laboratories' Continuously Variable High Pass and Low Pass Model 2A Filter Set, serial No. 2010. Using normal electrical measurement techniques, the frequency response characteristics were measured for each of the octave bands and appropriate settings of the continuous control dials were selected which yielded the optimum conformance with standards. During the course of the research program, this filter set was severely damaged. After factory repair, this filter set and the entire instrumentation system was recalibrated.

The sound-level meter utilized was a battery-operated, laboratory-built unit which has given many years of excellent service. The instrument-type copper-oxide rectifier used in this sound-level meter has an appreciable temperature coefficient, and therefore it was necessary when standardizing this instrument by means of its internal standardizing oscillator to make the proper setting corresponding to the instrument's temperature. Long experience has shown this particular sound-level meter to be very stable and reliable. In addition, a small battery-operated cathode-follower matching stage was required between the sound-level meter and the octave-band filter set to secure proper operation of the instrument system.

Because the sensitivity of the microphone varies slightly with frequency and because the octave-band filters do not have ideal, square-topped characteristics, it is necessary to determine appropriate correction factors to be added or subtracted from the directly indicated octave-band sound-pressure levels. Of course, a variable sensitivity network could have been switched simultaneously with the octave-band filter, but it was more expedient to apply a correction factor later.

The method of computation of the correction factors to be applied to the various octave-band sound-pressure-level readings is essentially the same as the alternative method given in Leo L. Beranek's Acoustic Measurements (New York, John Wiley and Sons, 1949, p. 561). To do this, the sensitivity characteristics are determined for the complete instrumentation system including the microphone for each octave band. Then a straight line corresponding to the ideal flat-topped octave-band response is located graphically for each octave in such a way that equal areas of deviation lie above and below this line. The instrument sensitivity represented by each such line is compared to the sensitivity of the instrumentation system at 1000 cps (re 0.0002 dynes/cm$^2$) and the difference in sensitivity found for each octave band used as the correction

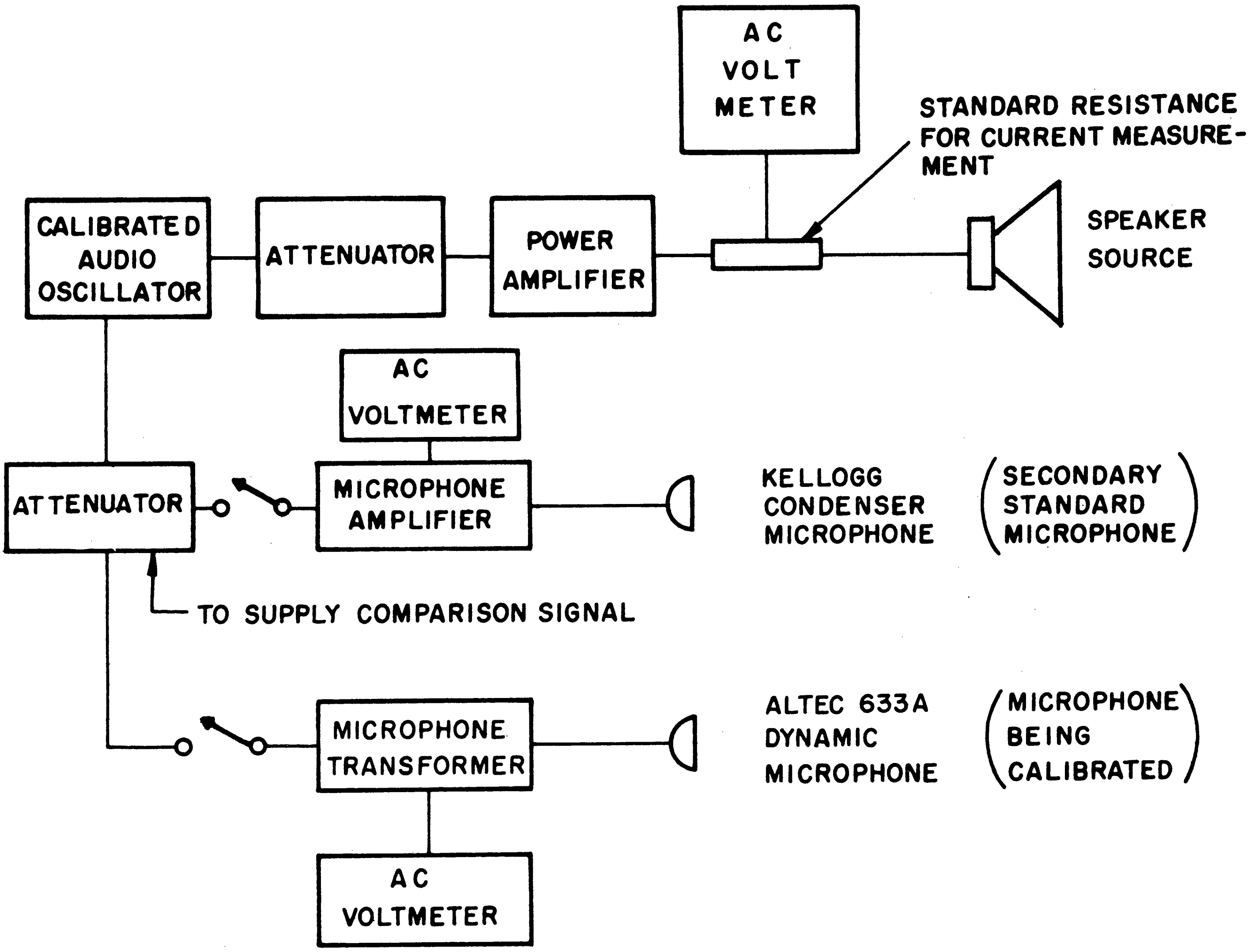
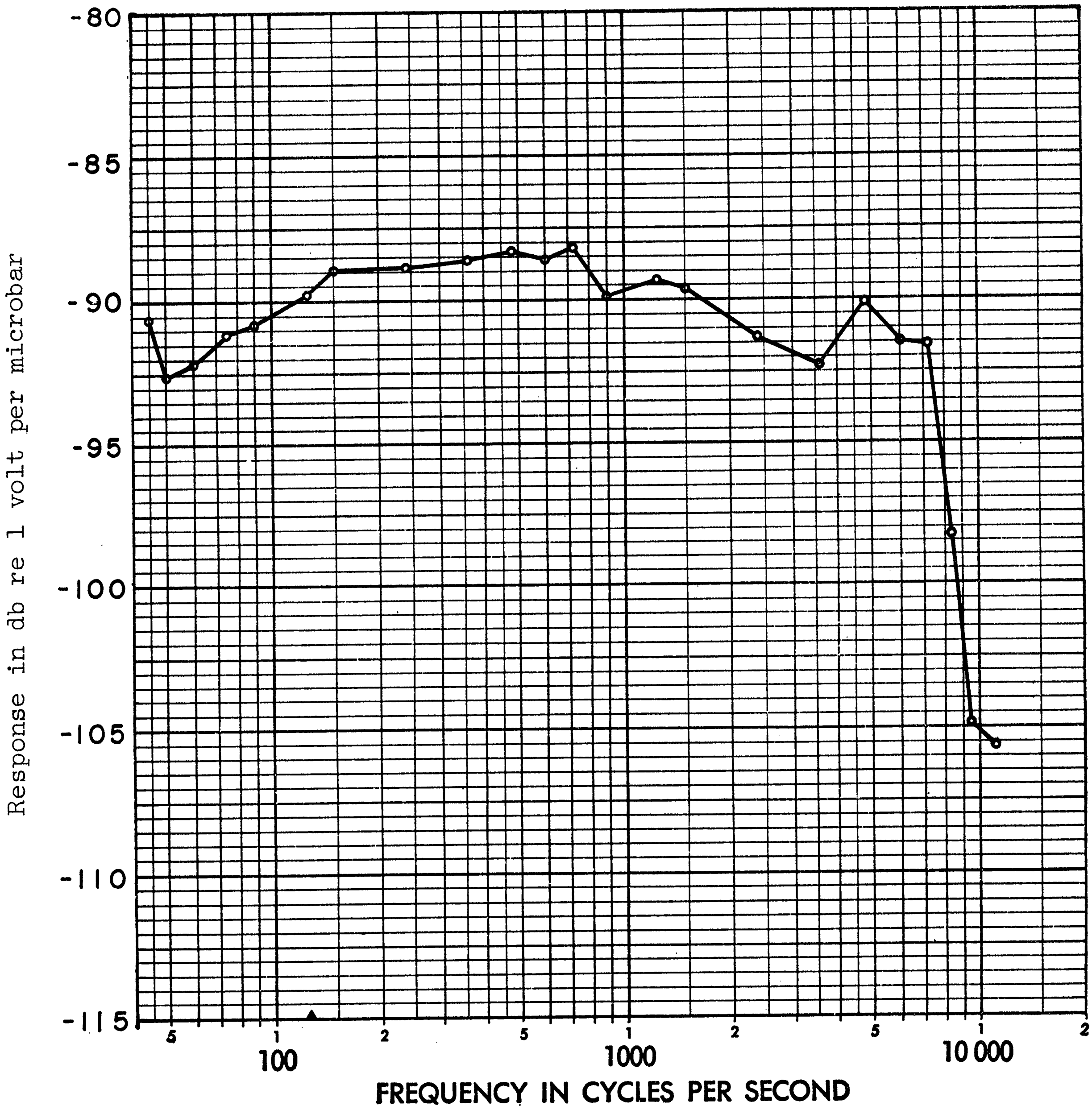
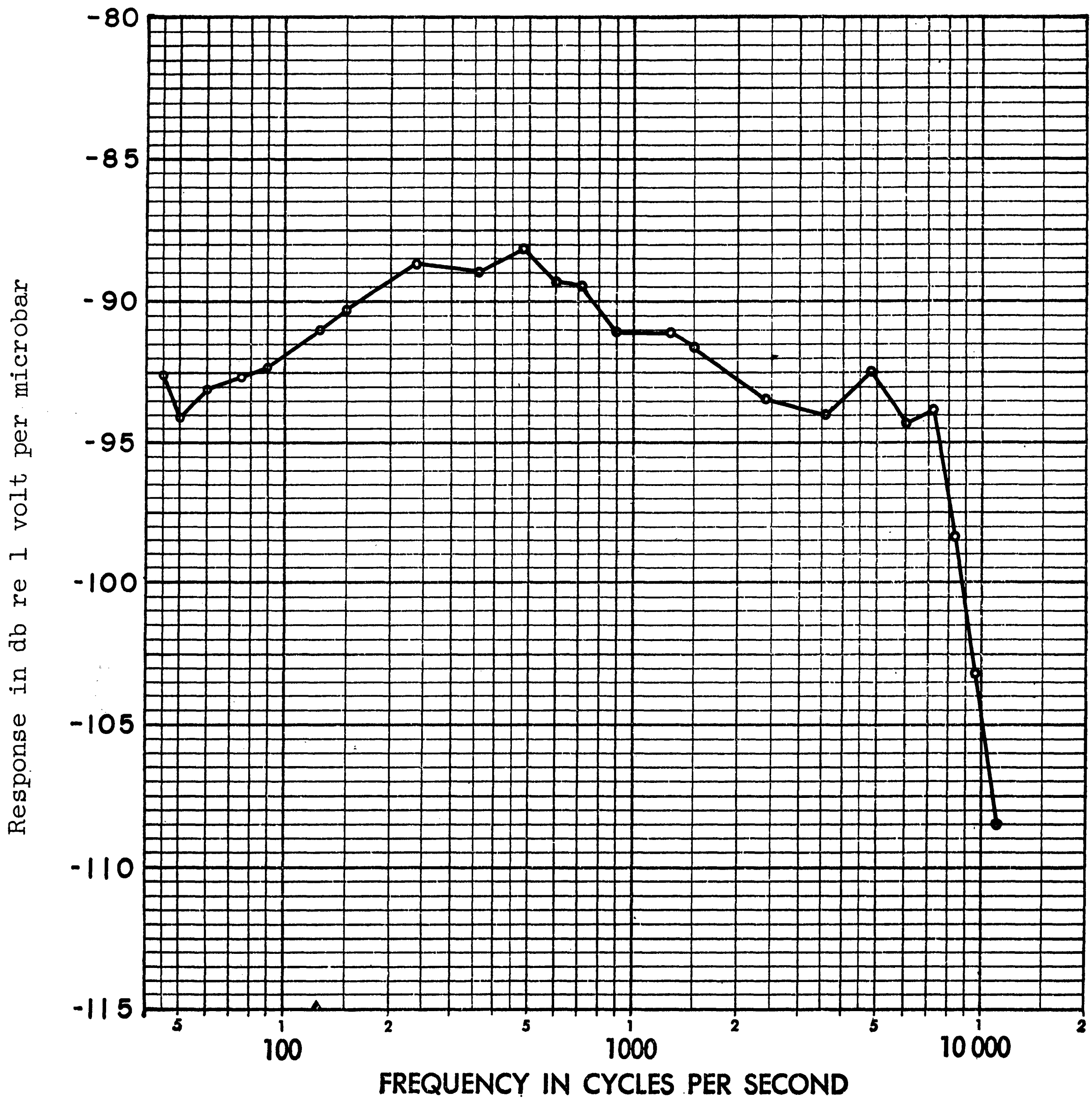



Fig. B.7. Block diagram of microphone calibration instrumentation.




Calibrated 24 February 1956 against secondary standard microphone—Kellogg condenser microphone serial No. 1289. See Fig. B.7.

Fig. B.8. Sensitivity of Altec 633A microphone, serial No. A1927.




Calibrated 21 June 1955 against secondary standard microphone—
Kellogg condenser microphone serial No. 1289. See Fig. B.7.

Fig B.9. Sensitivity of Altec 633A microphone, serial No. A1933.

factor for that octave. Thus in terms of final calibration, the instrumentation system allows the determination of the octave-band sound-pressure levels in terms of the standard sound-pressure levels of an equivalent single pure tone lying anywhere within the particular octave in question.

Such a calibration procedure may seem overly complex and perhaps somewhat lacking in accuracy; however, in the absence of ideal square-topped filter-response characteristics and microphone sensitivity, which is ideally flat with frequency, the above procedure, or an equivalent one, is about the best that can be done. Of course, if one has detailed a priori knowledge of the characteristics of the sound being measured, more appropriate correction factors could be computed, but with such a priori knowledge there would be no need to make the octave-band measurements.

It is difficult to estimate the overall calibration accuracy of the instrumentation system. It is felt, however, that microphone calibration is probably accurate to ± 1.5 db over the frequency range from 37.5 to 10,000 cps and the repeatability is much better, being perhaps of the order of ± 0.3 db. The excellent repeatability of the standardization of the instrumentation system under field measuring conditions also suggests that a reproducibility of about ± 0.2 db was realized; a value considerably smaller than the normal variability of the machinery sources under test.

Because of the accident to the octave-band filter set and of a microphone failure, three different calibrations were required throughout the course of this research project, each applying to data taken within definite time periods. The three sets of octave-band correction factors are given in Table B.1 below.

TABLE B.1. CORRECTION FACTORS FOR OCTAVE-BAND SOUND-PRESSURE-LEVEL MEASUREMENTS

| Octave Band cps | Correction Factors in db (Add or subtract from raw data according to indicated sign.) July 22, 1955, to Oct. 1, 1955 | Oct. 25, 1955, to Feb. 20, 1956 | Feb. 21, 1956,* to July, 1956 |
|---|---|---|---|
| 37.5 - 75 | + 3.1 | + 3.0 | + 2.9 |
| 75 - 150 | - 0.3 | + 0.4 | + 0.4 |
| 150 - 300 | - 1.6 | - 1.7 | - 0.7 |
| 300 - 600 | - 2.0 | - 1.8 | - 0.5 |
| 600 - 1200 | - 0.1 | + 0.3 | + 0.3 |
| 1200 - 2400 | + 1.6 | + 2.0 | + 1.4 |
| 2400 - 4800 | + 4.2 | + 3.3 | + 2.7 |
| 4800 - 9600 | +11.6 | + 7.2 | + 7.6 |

*Between Feb. 21, 1956, and April 27, 1956, subtract 1.4 db from all raw data in addition to applying correction factor.

The large correction factor for the 4800-9600 cps octave band is the direct result of the rapidly falling microphone sensitivity toward the higher frequency side of this octave.

A calibration correction factor for the next lower octave band from 18-3/4—37-1/2 cps could not be determined since the microphone calibration could not be extended to low enough frequencies with present instrumentation. Field measurements were sometimes recorded from this octave band for comparative purposes only. However, since no special acoustic features were observed in this frequency range which were not better displayed in the higher frequency octave bands, and since correction to standard sound-pressure level could not be accomplished, no data from the 18-3/4—37-1/2 cps octave band have been used in this report.

## FIELD OPERATION OF SOUND-LEVEL METER AND OCTAVE-BAND FILTER

Because the entire octave-band instrumentation system was battery-operated, no modifications of procedure were required for field measurements. The measuring system was standardized at frequent intervals by means of the sound-level meter's internal standardization oscillator, making due allowance for rectifier temperature. The fluctuating octave-band sound-pressure levels were recorded manually to the nearest tenth of a decibel by averaging the fluctuations of the slow meter by eye.

During the free-field surveys, the noise at each bearing was measured as rapidly as the sound-level meter could be read and the filter switched through the eight octave bands. Engine data and other source parameters were also recorded corresponding to each bearing at which an octave-band analysis was taken.

Beyond this, the usual precautions of good acoustical technique were observed. The instrumentation system was kept away from the intense sound fields as much as possible to minimize microphonics. Instrumentation standardization and battery condition were checked frequently. The microphone was disconnected at frequent intervals, and the output meter checked for a drop of 10 db or more of indication as protection against false data. Checks were made of the ambient noise level to ascertain at what measurement levels it would begin to interfere. Also, a careful surveillance was maintained of both the noise source and of all activity near the test site to detect any unusual occurrences, and to avoid the collection of invalid data.

In several instances, and particularly when interpretational difficulties were encountered during some phases of the MA-1 Gas Turbine measurement program, an independent check was made of the complete instrumentation system from microphone to output meter. For this purpose, a General Radio Company's Type 1552-B Sound Level Calibrator and Type 1307A Transistor Oscillator were used to supply a known acoustic signal of 116 db at 400 cps to the 633A microphone. Agreement within a fraction of a decibel was always obtained. In fact, no difficulty due to faulty or erratic instrument behavior ever occurred during this research program. The few instrument failures which did occur were sudden and obvious.

## REDUCTION OF FREE-FIELD DATA

As far as the octave-band measurements are concerned, the only operation necessary to convert the raw data into standard sound-pressure levels above 0.0002 dynes/cm$^2$ is to apply the appropriate correction factors given in Table B.1 above. These corrected data in terms of octave-band sound-pressure level have been tabulated in Appendix C and used for the preparation of the various graphs found in the main text.

The average sound-pressure level for each octave band was computed by arithmetically averaging the levels measured in that octave band at each of the twelve bearings.

These computations were made by the usual methods and require no special explanation.

At each bearing, the overall sound-pressure level for a frequency range from 37.5 to 9600 cps has been computed. This can be done in the usual straightforward manner of converting each octave-band sound-pressure level back into sound pressure. Next, the square root of the sum of the squares of the eight individual octave-band sound pressures is computed, and finally converted again into sound-pressure level. Actually such computations can be made very easily by an alternate method using a conventional "decibel vs voltage and power" chart. One has merely to look up the "power ratio" corresponding to the eight individual octave-band sound-pressure levels, take the arithmetic sum of these eight "power ratios" and then look up the "db" corresponding to the "power ratio" sum. This final "db" value is the computed overall sound-pressure level.

To enable rapid comparisons in the field while collecting data, an overall measurement was made directly. However, because of the frequency variation of microphone sensitivity only the computed values described above have been tabulated and used for final data analysis. The various graphs and other presentations can be derived directly from the tabulated data reduced as described above, to be found in Appendix C.

## SPECIAL PURPOSE ACOUSTIC MEASUREMENTS

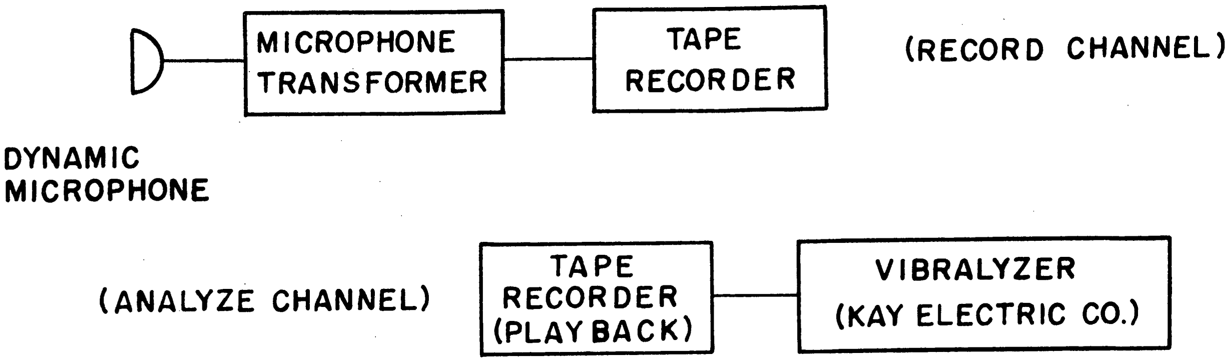
Various close-in measurements, often under non-free-field conditions, were made for particular reasons. When the octave-band instrumentation was used, the data could still be expressed in terms of sound-pressure level although, of course, only certain very limited comparisons are valid because of the non-free-field conditions.

Discrete-frequency analyses were carried out, using two different instrumentation schemes as illustrated by the block diagrams in Fig. B.10. In both cases, the noise to be investigated was tape-recorded and played back into the analysis system. These tape recordings were taken under non-free-field conditions so that as far as acoustic levels are concerned, the analyzed data are only qualitative in most cases. The frequency composition, of course, is quantitative.

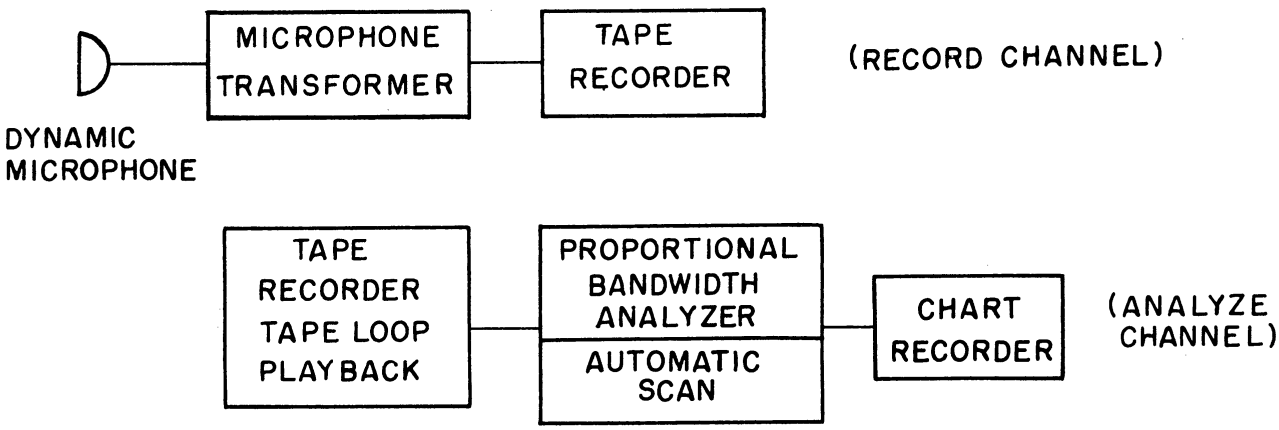
The Kay Vibralyzer was used for discrete-frequency analysis when a quick display and only moderate resolution were required. When higher resolution was necessary, a laboratory-built ± 1% proportional bandwidth analyzer, equipped with an automatic frequency sweep and large chart recorder, was employed. Also much longer tape playback loops could be used with this system, thus enabling the determination of the time average stability of the noise being analyzed.

Since, for the most part, the results of these discrete-frequency analyses were merely qualitative and used mainly as diagnostic aids, the raw data have not been tabulated in Appendix C; however, the important features or deductions have been mentioned in the text where applicable. In particular, a considerable amount of the experimental muffler studies was accomplished using the ± 1% proportional bandwidth discrete-frequency-analysis instrumentation.

A few wide-band comparative noise measurements were taken during the time when the octave-band filter set was being repaired. Makeshift instrumentation systems were employed during that period of time and therefore an accurate calibration of this instrumentation was not attempted.




A. KAY VIBRALIZER INSTRUMENTATION




B. ±1% PROPORTIONAL BANDWIDTH ANALYZER INSTRUMENTATION

Fig. B.10. Block diagram of discrete frequency analysis instrumentation.

## SIGHTING ARRANGEMENTS

As mentioned above, free-field surveys were conducted by using a fixed microphone location and by rotating the item of ground-support equipment to establish the proper relative bearings at 30-degree intervals. To locate properly the machine under test, a plumb bob was suspended under the geometrical center of the machine. Then as the machine was rotated, the plumb bob was kept centered above a metal locating stake driven in the road.

The angular orientation of the machine was determined with a special sighting device consisting of an 8-in.-diameter compass rose and a pivoted sighting rod. The compass rose was attached to the top of the machine's housing directly above its geometrical center. The pivoted sighting rod, much like an overgrown alidade, had sighting arms spaced six feet apart, and it was kept aimed directly at the microphone while the test machine was rotated to establish the desired relative bearing. In all cases, a bearing accuracy of ± 2 degrees or less was maintained.

## ELECTRICAL LOADING

Most of the items of ground-support equipment tested produced useful output in the form of electrical power. To provide representative loading of the engines, a resistor bank was used to dissipate the power.

For laboratory purposes, it was found expedient to construct resistors from standard 50-foot coils of 3/8-in. diameter, 1/32-in. wall copper refrigerator tubing. Each 50-ft length of tubing was arranged into a helix about 18 in. in diameter by 3 ft long and supported by Transite strips. Eight similar coils, mounted four per unit, could be interconnected in various series-parallel arrangements to provide some adjustment of the total load resistance available.

For comparatively light loads, normal air circulation over the copper tubing provided satisfactory cooling but at high loading, water had to be circulated through the coils to dissipate the heat. At the free-field test site, the water was supplied from two 55-gallon drums by means of a 28-volt d-c aircraft-type fuel pump. When operating near the laboratory under non-free-field conditions, the cooling water was supplied directly from the building's water lines.

Except for the large changes in resistance available by reconnection of the eight coils, the resistance of the load bank was not adjustable directly. The actual resistance varied slightly depending on ambient air temperature or the temperature and flow rate of the cooling water. These changes in resistance account for most of the variability in actual load current observed under the same nominal operating conditions. However, even though some variability occurred as a consequence of these slight resistance changes, the variability was still satisfactorily small for the present research purposes.

## BACKPRESSURE MEASUREMENTS

It was desirable to measure the increase in exhaust backpressure resulting from the attachment of several mufflers. The advice of the University Automotive Laboratory was followed in instrumenting for and in taking these backpressure measurements.

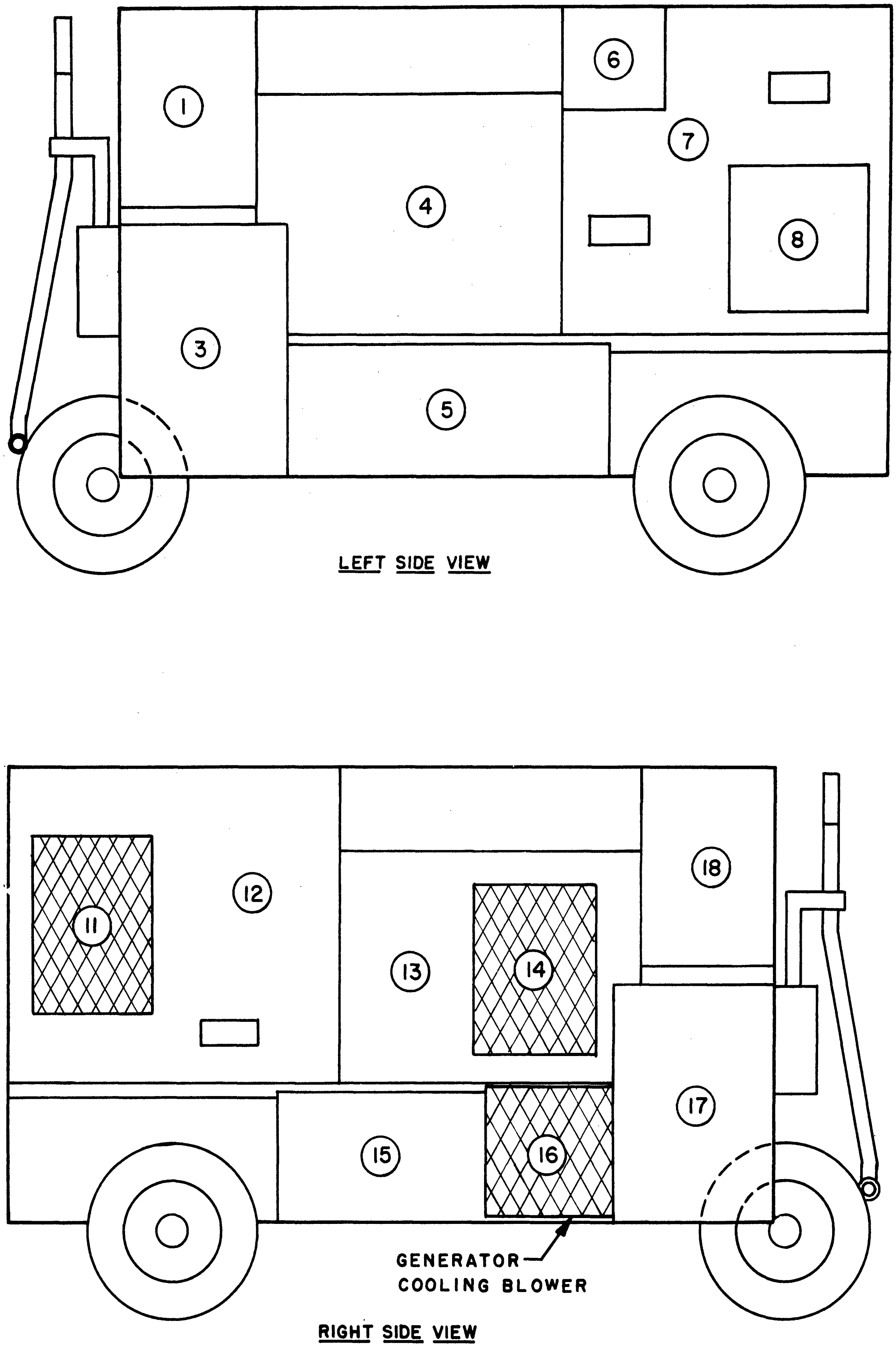
In the case of the A-1 Generator Set, a static pressure tap was inserted in the exhaust line at a point about two inches beyond the junction of the exhaust pipes from the two cylinders. The tap itself was a No. 59 drill hole (0.041-in. diameter) terminating flush with the inside surface of the exhaust pipe. The static pressure existing in the exhaust line was then measured relative to atmospheric pressure by means of a U-tube manometer filled either with water or mercury depending on the pressure range required. To evaluate the effect of the several intake silencers on the cooling-air blower performance, the static pressure was monitored in one of the several distribution ducts leading away from this blower.

Exhaust backpressure was also measured on the C-26 Generator Set, and the same experimental arrangement described above was used. In this case, a static pressure tap was inserted in the exhaust line of the one-cylinder heated-air-blower engine just above the point where this exhaust line joins the large exhaust line from one bank of cylinders of the main engine.

## APPENDIX C

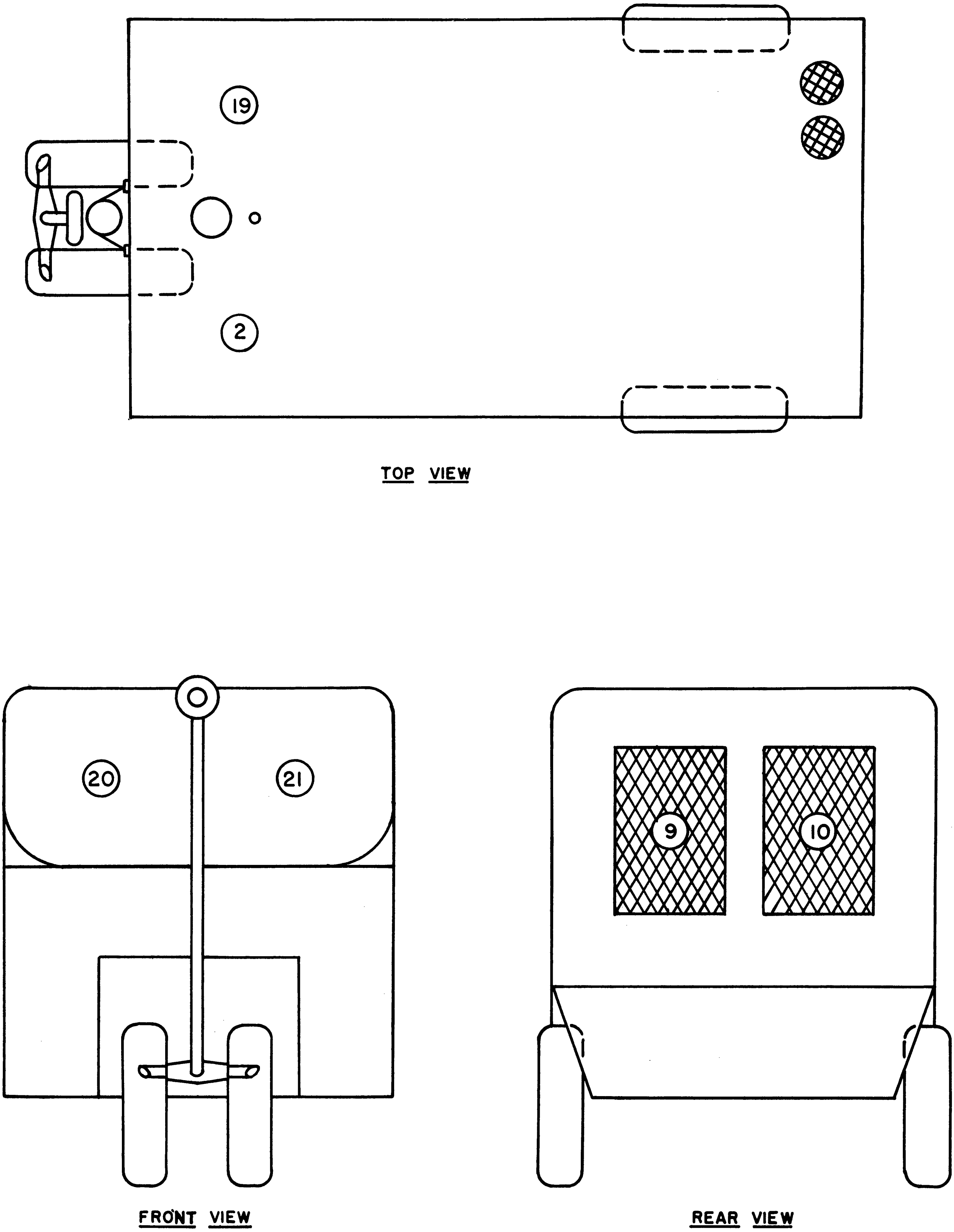
## COMPILATION OF DATA

The principal data obtained during the experimental noise-reduction phases of this research program are presented in tabular form below. In all cases, the machinery operating parameters were thought to remain sufficiently constant so that their residual variations did not significantly affect the acoustical data. Therefore, only the numerical ranges of these parameters which occurred during each test are presented at the top of the corresponding table. The octave-band sound-pressure levels presented have been corrected by means of the appropriate calibration factors (see Table B.1) except where noted. Attention is called to the fact that some tables present non-free-field data, and that even though corrected sound-pressure levels are tabulated, only comparisons between corresponding test conditions are valid.




C-26 GENERATOR SET

Fig. C.1. Microphone positions for panel measurements.




C-26 GENERATOR SET

Fig. C.2. Microphone positions for panel measurements.

TABLE C.1

Unit: A-1
Date: July 29, 1955
Treatment: As-received
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 37.5-75 | 87.6* | 87.3 | 87.2 | 86.5 | 87.8 | 89.4 | 90.1 | 89.8 | 90.9 | 89.8 | 89.8 | 88.1 | 88.7 |
| | 87.4 | 87.1 | 87.5 | 86.8 | 87.3 | 88.4 | 89.8 | 89.3 | 90.1 | — | 89.4 | 88.2 | 88.3 |
| 75-150 | 92.0 | 91.2 | 91.8 | 92.4 | 93.9 | 93.9 | 94.5 | 93.7 | 93.2 | 93.3 | 92.7 | 92.7 | 92.9 |
| | 91.7 | 91.4 | 92.1 | 92.0 | 94.1 | 94.1 | 95.3 | 93.5 | 93.2 | — | 92.4 | 91.7 | 92.9 |
| 150-300 | 93.7 | 89.6 | 91.1 | 93.7 | 95.2 | 95.0 | 94.4 | 96.3 | 95.6 | 96.6 | 97.0 | 96.2 | 94.5 |
| | 93.8 | 90.2 | 91.5 | 93.1 | 94.7 | 94.2 | 93.5 | 96.2 | 95.4 | — | 96.9 | 96.2 | 94.2 |
| 300-600 | 83.5 | 83.3 | 87.8 | 84.0 | 85.2 | 86.0 | 87.1 | 88.5 | 88.0 | 85.5 | 87.8 | 89.0 | 86.6 |
| | 84.0 | 83.4 | 87.7 | 83.8 | 84.0 | 85.1 | 86.2 | 88.2 | 88.2 | — | 87.1 | 89.0 | 86.1 |
| 600-1200 | 77.6 | 77.5 | 79.4 | 81.3 | 83.9 | 82.7 | 84.1 | 83.0 | 89.4 | 82.1 | 81.2 | 78.9 | 81.8 |
| | 77.2 | 77.8 | 79.5 | 80.9 | 83.1 | 81.9 | 82.2 | 84.4 | 86.4 | — | 81.1 | 78.1 | 81.1 |
| 1200-2400 | 72.6 | 75.6 | 77.0 | 78.5 | 83.1 | 88.5 | 95.1 | 90.2 | 86.1 | 78.1 | 79.1 | 79.1 | 81.9 |
| | 73.6 | 75.6 | 76.8 | 77.9 | 85.6 | 87.8 | 94.6 | 87.6 | 85.6 | — | 78.4 | 77.7 | 81.9 |
| 2400-4800 | 66.2 | 74.2 | 77.2 | 78.6 | 81.9 | 83.2 | 86.8 | 84.8 | 83.7 | 77.6 | 80.7 | 75.7 | 79.2 |
| | 67.7 | 73.7 | 77.2 | 78.4 | 82.0 | 82.6 | 86.2 | 83.2 | 83.1 | — | 77.0 | 74.9 | 78.7 |
| 4800-9600 | 69.6 | 79.6 | 85.1 | 85.9 | 89.0 | 92.6 | 94.6 | 94.6 | 94.8 | 81.5 | 75.1 | 78.1 | 85.0 |
| | 69.6 | 80.0 | 83.2 | 87.1 | 88.6 | 92.2 | 94.0 | 94.6 | 89.6 | — | 80.4 | 77.9 | 85.2 |
| Computed overall | 96.8 | 95.0 | 96.5 | 97.4 | 99.1 | 99.9 | 101.4 | 101.0 | 100.8 | 99.4 | 99.5 | 98.9 | 98.6 |
| | 96.8 | 95.3 | 96.6 | 97.1 | 98.9 | 99.5 | 101.1 | 100.7 | 99.6 | — | 99.2 | 98.6 | 98.5 |
| Order in which measured | 1 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | |

*Two consecutive complete measurements performed.

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 190-204 | 28.5-29.2 | 33-41 | 300-325 | 70-203 | 2450-2650 | 77 |

TABLE C.2

Unit: A-1
Date: August 29, 1955
Treatment: Muffler evaluations
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Bearing (degrees)* | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 215* | | | | 225 | | | | 235 | | | |
| | Muffler Used | | | | Muffler Used | | | | Muffler Used | | | |
| | None | Riker | Tractor | Walker 629 | None | Riker | Tractor | Walker 629 | None | Riker | Tractor | Walker 629 |
| 37.5-75 | 93.0 | 92.2 | 92.6 | 91.0 | 95.0 | 93.1 | 91.8 | 91.3 | 92.7 | 92.7 | 90.9 | 91.3 |
| 75-150 | 94.4 | 94.5 | 96.4 | 85.9 | 94.2 | 93.6 | 94.0 | 85.5 | 93.1 | 92.7 | 92.0 | 84.9 |
| 150-300 | 96.8 | 90.5 | 93.6 | 81.3 | 94.5 | 90.2 | 93.2 | 80.3 | 92.7 | 92.9 | 91.7 | 83.0 |
| 300-600 | 92.4 | 84.6 | 80.5 | 74.2 | 92.0 | 83.5 | 79.7 | 74.4 | 89.9 | 87.5 | 81.0 | 74.5 |
| 600-1200 | 88.1 | 81.6 | 79.2 | 77.1 | 87.6 | 80.6 | 78.7 | 77.7 | 88.4 | 82.2 | 78.4 | 76.9 |
| 1200-2400 | 83.8 | 80.1 | 83.1 | 82.1 | 83.0 | 84.4 | 86.3 | 82.2 | 84.9 | 85.9 | 84.8 | 83.6 |
| 2400-4800 | 82.8 | 81.0 | 81.2 | 80.9 | 82.5 | 81.7 | 81.6 | 80.6 | 83.8 | 81.9 | 81.7 | 81.7 |
| 4800-9600 | 95.5 | 94.3 | 95.1 | 95.9 | 94.5 | 93.4 | 92.9 | 93.6 | 93.9 | 96.8 | 96.1 | 94.6 |

*See Fig. 3.6.

b) Engine Performance (Range) Data

| Muffler Used | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Backpressure, oz water | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | | |
| None | 200-205 | 28.8 | 36-37 | 300-305 | 207-235 | 2470-2500 | 0 | 82 |
| Ricker | 198-205 | 28.7-28.9 | 36 | 300-310 | 204-208 | 2430-2500 | 1 | 88 |
| Tractor | 198-200 | 28.6-28.7 | 36-38 | 310 | 205-217 | 2410-2500 | 0 | 88 |
| Walker 629 | 195-197 | 28.6-28.7 | 37-38 | 310 | 210-216 | 2470-2480 | $<5$ | 88 |

TABLE C.3

Unit: A-1
Date: September 9, 1955
Treatment: Intake silencer No. 1, evaluation
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | | 4 | | 5 | | 6 | |
| | Intake Silencer | | Intake Silencer | | Intake Silencer | | Intake Silencer | | Intake Silencer | | Intake Silencer | |
| | On | Off | On | Off | On | Off | On | Off | On | Off | On | Off |
| 1200-2400 | 85.6 | 85.5 | 86.6 | 86.5 | 85.9 | 85.6 | 82.9 | 91.2 | 84.6 | 85.1 | 88.4 | 82.6 |
| 2400-4800 | 85.6 | 87.7 | 85.7 | 84.9 | 84.2 | 84.0 | 81.7 | 88.2 | 83.7 | 82.0 | 86.6 | 87.0 |

*See Fig. 3.9.

b) Engine Performance (Range) Data

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| On | 197-200 | 28.2-28.3 | 37-38 | 300-325 | 167-192 | 2460-2480 | 75 |
| Off | 198-200 | 28.3-28.5 | 37-38 | 325 | 198-200 | 2440-2460 | 75 |

TABLE C.4

Unit: A-1
Date: September 14, 1955
Treatment: Intake silencer, liner evaluation
Type of Measurement: Non-free-field

Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Measurements at Position 4* | | | |
|---|---|---|---|---|
| | Bare | Baffle On | Liner On | Baffle and Liner On |
| 1200-2400 | 101.7 | 94.1 | 95.6 | 87.4 |
| 2400-4800 | 99.3 | 92.0 | 94.7 | 85.9 |

*See Fig. 3.11.

TABLE C.5

Unit: A-1
Date: September 26, 1955
Treatment: Intake silencer No. 2 evaluation,
Walker 629 installed
Type of Treatment: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Measurements at Position 4* | | |
|---|---|---|---|
| | Silencer On, Muffler On | Silencer Off, Muffler On | Lined Silencer On, Muffler On |
| 75-150 | 85.2 | 88.2 | 89.7 |
| 150-300 | 85.8 | 85.4 | 88.3 |
| 1200-2400 | 84.0 | 91.1 | 84.6 |
| 2400-4800 | 79.9 | 89.2 | 79.5 |

*See Fig. 3.11.

b) Engine Performance (Range) Data

| Condition | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| No silencer, muffler on | 199-200 | 27.9-28.2 | 36 | 320-330 | 153-208 | 2410-2420 | 67 |
| Muffler on, silencer on | 197-201 | 28.2-28.4 | 38-41 | 320-325 | 163-186 | 2410-2480 | 63 |
| Muffler on, lined silencer on | 199-207 | 28.3-28.4 | 34-38 | 320-400 | 118-135 | 2480-2520 | 63 |

TABLE C.6

Unit: A-1
Date: November 1, 1955
Treatment: Walker 629 muffler and intake silencer No. 3 installed
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 81.9 | 83.1 | 83.9 | 84.5 | 85.8 | 87.6 | 89.4 | 89.4 | 87.1 | 84.1 | 86.6 | 81.7 | 85.4 |
| 75-150 | 76.8 | 77.3 | 76.9 | 72.9 | 78.5 | 84.6 | 86.6 | 84.7 | 83.3 | 78.9 | 76.8 | 73.0 | 79.2 |
| 150-300 | 77.4 | 78.0 | 82.3 | 79.9 | 79.7 | 84.0 | 84.8 | 84.3 | 81.4 | 81.3 | 82.5 | 79.5 | 81.2 |
| 300-600 | 70.9 | 74.7 | 72.5 | 74.2 | 74.5 | 76.8 | 81.7 | 77.0 | 75.2 | 73.0 | 72.7 | 74.4 | 74.8 |
| 600-1200 | 73.6 | 73.5 | 74.8 | 77.8 | 79.5 | 78.5 | 79.2 | 79.0 | 78.8 | 75.4 | 75.0 | 74.0 | 76.6 |
| 1200-2400 | 74.6 | 77.1 | 83.2 | 83.0 | 83.9 | 79.6 | 77.9 | 81.9 | 83.5 | 79.2 | 78.0 | 78.6 | 80.0 |
| 2400-4800 | 71.2 | 74.7 | 78.2 | 79.8 | 79.7 | 76.6 | 75.1 | 78.5 | 78.0 | 76.4 | 76.1 | 75.2 | 76.6 |
| 4800-9600 | 72.0 | 76.1 | 78.7 | 83.7 | 82.2 | 78.6 | 78.1 | 79.9 | 79.6 | 76.8 | 78.9 | 76.6 | 78.4 |
| Computed overall | 85.5 | 86.9 | 89.4 | 90.1 | 90.7 | 91.6 | 93.1 | 92.8 | 91.3 | 88.4 | 89.6 | 86.6 | 89.7 |
| Order in which measured | 7 | 8 | 9 | 10 | 11 | 12 | 1 | 2 | 3 | 4 | 5 | 6 | |

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 205-215 | 27.9-29.0 | 33-39 | 325-340 | 134-184 | 2490-2570 | 64 |

TABLE C.7

Unit: C-26
Date: August 4-5, 1955
Treatment: As-received
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (Cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 88.1* | 85.9 | 85.7 | 85.4 | 85.4 | 87.8 | 90.3 | 89.6 | 88.2 | 87.9 | 88.9 | 89.0 | 87.7 |
| | 87.8 | 85.9 | 84.9 | 85.3 | 85.1 | 87.3 | 90.1 | 89.4 | 87.9 | 88.0 | 88.9 | 88.4 | 87.4 |
| 75-150 | 100.4 | 98.2 | 97.6 | 97.7 | 97.8 | 100.3 | 102.6 | 102.2 | 100.4 | 100.5 | 100.3 | 100.6 | 99.9 |
| | 100.0 | 97.9 | 97.1 | 98.0 | 97.6 | 100.0 | 102.8 | 101.8 | 100.1 | 100.3 | 100.4 | 100.5 | 99.7 |
| 150-300 | 87.3 | 89.2 | 91.0 | 90.2 | 91.1 | 92.7 | 96.2 | 94.9 | 90.6 | 93.8 | 93.9 | 93.0 | 92.0 |
| | 87.0 | 88.9 | 90.8 | 89.9 | 90.8 | 92.6 | 96.4 | 94.7 | 90.4 | 93.7 | 94.1 | 92.6 | 91.8 |
| 300-600 | 78.7 | 83.9 | 86.4 | 84.0 | 86.0 | 84.4 | 83.5 | 82.7 | 80.3 | 77.6 | 79.2 | 78.7 | 82.1 |
| | 79.8 | 83.5 | 86.0 | 83.8 | 85.9 | 84.5 | 83.6 | 82.6 | 80.8 | 77.6 | 79.0 | 79.0 | 82.2 |
| 600-1200 | 75.4 | 80.4 | 78.5 | 80.4 | 81.2 | 80.7 | 81.5 | 80.7 | 79.5 | 77.4 | 75.7 | 77.3 | 79.0 |
| | 75.6 | 79.6 | 78.7 | 82.0 | 81.0 | 80.6 | 80.6 | 80.2 | 79.1 | 77.0 | 76.0 | 76.1 | 78.9 |
| 1200-2400 | 70.1 | 74.6 | 76.9 | 75.5 | 78.8 | 76.5 | 74.5 | 75.6 | 72.0 | 68.5 | 70.0 | 69.9 | 73.6 |
| | 68.4 | 75.0 | 76.8 | 75.5 | 79.4 | 76.4 | 74.4 | 75.5 | 73.1 | 69.7 | 71.6 | 72.4 | 74.0 |
| 2400-4800 | 56.7 | 72.2 | 71.0 | 73.2 | 74.9 | 73.0 | 64.3 | 64.7 | 68.5 | 55.7 | 64.0 | 54.6 | 66.1 |
| | 56.9 | 73.0 | 71.9 | 72.0 | 75.2 | 72.4 | 66.2 | 67.6 | 68.2 | 54.7 | 66.8 | 56.2 | 66.8 |
| 4800-9600 | <56.6 | 61.6 | 65.9 | 69.1 | 68.6 | 62.3 | <56.6 | <56.6 | <56.6 | <56.6 | 56.6 | <56.6 | <60.3 |
| | <56.6 | 63.9 | 67.6 | 68.4 | 67.5 | 61.6 | <56.6 | <56.6 | <56.6 | <56.6 | 56.6 | <56.6 | <60.4 |
| Computed overall | 100.9 | 99.2 | 99.0 | 98.9 | 99.2 | 101.3 | 103.8 | 103.2 | 101.1 | 101.7 | 101.5 | 101.6 | 101.0 |
| | 100.5 | 98.9 | 98.6 | 99.1 | 99.0 | 101.1 | 103.9 | 102.9 | 100.9 | 101.4 | 101.6 | 101.4 | 100.8 |
| Order in which measured | 1 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | |

*Two consecutive complete measurements performed.

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 800-955 | 29.0-30 | 42-50 | 175-197 | 100-130 | 1890-1895 | 92-93 |

TABLE C.8

Unit: C-26
Date: October 28, 1955
Treatment: 2 Walker 629 mufflers installed
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 78.7 | 73.5 | 76.4 | 76.1 | 77.3 | 77.0 | 75.2 | 79.3 | 75.8 | 75.5 | 77.0 | 78.2 | 76.7 |
| 75-150 | 88.9 | 87.7 | 89.2 | 88.1 | 86.7 | 88.3 | 85.1 | 86.6 | 88.1 | 86.3 | 86.2 | 87.6 | 87.4 |
| 150-300 | 78.1 | 79.7 | 81.9 | 77.4 | 74.3 | 76.3 | 77.5 | 80.2 | 77.9 | 77.5 | 76.3 | 77.5 | 77.9 |
| 300-600 | 71.9 | 71.3 | 69.8 | 69.6 | 70.5 | 70.2 | 72.3 | 71.3 | 72.7 | 73.0 | 71.7 | 71.6 | 71.3 |
| 600-1200 | 73.1 | 73.2 | 72.0 | 72.6 | 70.3 | 71.6 | 70.7 | 72.7 | 74.1 | 73.8 | 74.2 | 75.7 | 72.8 |
| 1200-2400 | 72.8 | 71.8 | 71.1 | 71.6 | 72.4 | 72.6 | 70.0 | 76.2 | 76.1 | 76.6 | 77.0 | 74.4 | 73.6 |
| 2400-4800 | 70.8 | 65.1 | 65.6 | 66.7 | 66.3 | 65.6 | 67.5 | 69.6 | 70.5 | 70.7 | 71.3 | 70.0 | 68.3 |
| 4800-9600 | 64.6 | 59.4 | 57.3 | 57.5 | 57.2 | 57.2 | 57.8 | 63.8 | 65.0 | 66.5 | 66.2 | 66.5 | 61.6 |
| Computed overall | 89.9 | 88.8 | 90.3 | 89.0 | 87.7 | 89.1 | 86.6 | 88.6 | 89.3 | 88.0 | 87.9 | 89.0 | 88.7 |
| Order in which measured | 1 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | |

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Backpressure, oz water | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| Amperes | volts | | | | | | |
| 910-990 | 29.5-30 | 45-52 | 121-192 | 60-117 | 1880-1960 | 5.25-6.0 | 68-71 |

TABLE C.9

Unit: C-26
Date: November 7, 1955
Treatment: Panel tests, blower rigid, high load
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Positions* 7 | 16 | 19 |
|---|---|---|---|
| 37.5-75 | 88.3 | 90.3 | 92.5 |
| 75-150 | 100.2 | 104.2 | 106.3 |
| 150-300 | 96.4 | 97.7 | 98.4 |
| 300-600 | 93.1 | 95.4 | 85.5 |
| 600-1200 | 93.4 | 97.0 | 89.2 |
| 1200-2400 | 92.4 | 100.4 | 87.0 |
| 2400-4800 | 89.6 | 97.6 | 83.8 |
| 4800-9600 | 85.9 | 95.5 | 79.4 |

*See Section III, C-26 discussion.
Also, Figs. C.1 and C.2.

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 960-1030 | 28.7-29.5 | 51-56 | 145-169 | 85-118 | 1860-1900 | 70-74 |

TABLE C.10

Unit: C-26
Date: November 10, 1955
Treatment: As-received, load variation
Type of Measurement: Free-field (180° bearing)

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Load (amperes) | | | | |
|---|---|---|---|---|---|
| | 0 | 100 | 300 | 500 | 900 |
| 37.5-75 | 83.7 | 83.6 | 87.2 | 89.9 | 92.2 |
| 75-150 | 90.8 | 91.8 | 95.0 | 96.7 | 103.8 |
| 150-300 | 90.4 | 91.8 | 94.7 | 96.3 | 96.6 |
| 300-600 | 80.8 | 80.5 | 79.9 | 80.9 | 89.6 |
| 600-1200 | 73.1 | 73.4 | 73.5 | 74.5 | 82.6 |
| 1200-2400 | 68.5 | 70.7 | 71.1 | 71.6 | 76.2 |
| 2400-4800 | 61.5 | 62.4 | 63.8 | 62.0 | 70.1 |
| 4800-9600 | < 52.5 | 52.2 | 52.2 | 52.7 | 54.2 |
| Computed overall | 94.3 | 95.3 | 98.3 | 100.0 | 105.0 |

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 0 | 28.1 | 48 | 144 | 95 | 1800 | 58 |
| 100 | 28.1 | 50 | 146 | 96 | 1780 | 58 |
| 300 | 28.5 | 50 | 150 | 98 | 1750 | 58 |
| 500 | 28.5 | 50 | 150 | 97 | 1725 | 58 |
| 900 | 29.3 | 57 | 160 | 90 | 1890 | 58 |

TABLE C.11

Unit: C-26
Date: December 19, 1955
Treatment: Panel tests, blower resiliently mounted, mufflers on
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | 7 | | 16 | | 19 | |
| | Blower | | Blower | | Blower | |
| | On | Off | On | Off | On | Off |
| 37.5-75 | 83.1 | 82.2 | 83.4 | 83.6 | 85.7 | 84.0 |
| 75-150 | 86.6 | 85.6 | 88.7 | 86.9 | 91.7 | 91.1 |
| 150-300 | 89.9 | 88.5 | 87.7 | 86.2 | 85.1 | 84.7 |
| 300-600 | 87.0 | 86.9 | 91.0 | 89.3 | 91.9 | 82.8 |
| 600-1200 | 88.9 | 88.4 | 93.2 | 92.5 | 84.4 | 84.7 |
| 1200-2400 | 84.3 | 82.8 | 94.7 | 89.9 | 81.3 | 80.2 |
| 2400-4800 | 77.1 | 76.9 | 91.7 | 84.0 | 74.9 | 73.2 |
| 4800-9600 | 70.8 | 72.1 | 92.5 | 79.9 | 73.3 | 70.9 |
| Computed overall | 95.1 | 94.3 | 100.5 | 97.1 | 94.5 | 93.9 |

*See Section III, C-26 discussion; see Figs. C.1, C.2.

b) Engine Performance (Range) Data

| Blower | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| On | 105-115 | 27-27.6 | 51 | 115-121 | 70-76 | 1710-1720 | 15 |
| Off | 0 | 27-27.8 | 50 | 105-115 | 65-75 | 1710-1720 | 15 |

TABLE C.12

Unit: C-26
Date: December 21, 1955
Treatment: Panel tests, mufflers off, blowers resiliently mounted
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | 7 | | 16 | | 19 | |
| | Blower | | Blower | | Blower | |
| | On | Off | On | Off | On | Off |
| 37.5-75 | 97.6 | 98.8 | 99.3 | 97.6 | 94.4 | 91.7 |
| 75-150 | 103.6 | 102.6 | 105.6 | 104.7 | 101.2 | 98.5 |
| 150-300 | 108.4 | 106.5 | 105.0 | 103.1 | 101.2 | 101.0 |
| 300-600 | 98.5 | 98.3 | 99.8 | 98.5 | 96.5 | 95.1 |
| 600-1200 | 92.0 | 94.8 | 96.1 | 95.0 | 89.7 | 89.3 |
| 1200-2400 | 86.4 | 86.3 | 95.4 | 91.1 | 85.7 | 85.2 |
| 2400-4800 | 77.8 | 76.5 | 92.3 | 86.1 | 75.6 | 73.5 |
| 4800-9600 | 63.6 | 63.3 | 93.8 | 80.6 | 68.7 | 66.0 |
| Computed overall | 110.3 | 109.1 | 109.9 | 108.3 | 105.4 | 104.1 |

*See Section III, C-26 discussion; see Figs. C.1, C.2

b) Engine Performance (Range) Data

| Blower | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| On | 100-105 | 27.5-27.8 | 51 | 112-120 | 82-85 | 1740 | 75 |
| Off | 0 | 0 | 50 | 112-118 | 80-86 | 1750-1780 | 75 |

Unit: C-26
Date: December 23, 1955
Treatment: Panel tests, mufflers on, blower rigidly mounted
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | 7 | | 16 | | 19 | |
| | Blower | | Blower | | Blower | |
| | On | Off | On | Off | On | Off |
| 37.5-75 | 85.6 | 84.8 | 84.7 | 84.3 | 83.6 | 82.6 |
| 75-150 | 99.2 | 84.6 | 107.1 | 85.5 | 97.2 | 89.7 |
| 150-300 | 93.0 | 88.1 | 100.4 | 84.7 | 92.1 | 84.6 |
| 300-600 | 87.8 | 86.9 | 92.4 | 88.3 | 88.8 | 82.1 |
| 600-1200 | 89.7 | 88.2 | 94.4 | 90.0 | 85.4 | 84.9 |
| 1200-2400 | 84.2 | 84.0 | 98.2 | 90.2 | 82.2 | 80.7 |
| 2400-4800 | 78.4 | 77.1 | 91.9 | 84.5 | 75.9 | 74.1 |
| 4800-9600 | 76.5 | 72.1 | 93.2 | 79.7 | 76.9 | 71.6 |

b) Engine Performance (Range) Data

| Blower | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| On | 100-110 | 27.5-28 | 50-51 | 120-130 | 68-82 | 1700-1750 | 76-78 |
| Off | 0 | 0 | 48-50 | 115-128 | 55-83 | 1710-1750 | 76-78 |

TABLE C.14

Unit: C-26
Date: December 23, 1955
Treatment: Panel tests, mufflers off, blower rigidly mounted
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | 7 | | 16 | | 19 | |
| | Blower | | Blower | | Blower | |
| | On | Off | On | Off | On | Off |
| 37.5-75 | 97.5 | 98.2 | 97.4 | 101.8 | 93.3 | 92.2 |
| 75-150 | 104.6 | 102.6 | 106.3 | 103.5 | 103.3 | 97.8 |
| 150-300 | 108.9 | 108.1 | 103.5 | 103.7 | 100.1 | 100.4 |
| 300-600 | 97.9 | 97.5 | 96.6 | 94.7 | 94.3 | 95.0 |
| 600-1200 | 89.6 | 89.7 | 96.5 | 91.0 | 89.0 | 88.4 |
| 1200-2400 | 84.6 | 84.0 | 98.4 | 89.2 | 85.8 | 84.1 |
| 2400-4800 | 74.9 | 73.2 | 92.9 | 84.9 | 75.6 | 72.6 |
| 4800-9600 | 64.0 | 62.9 | 94.3 | 77.0 | 72.4 | 63.4 |
| Computed overall | 110.8 | 109.8 | 109.6 | 108.2 | 105.8 | 103.6 |

*See Section III, C-26 discussion; see Figs. C.1 and C.2.

TABLE C.15

Unit: C-26
Date: December 30, 1955
Treatments: Load variation, mufflers on,
blower rigidly mounted
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | | | 90 | | | 180 | | | 270 | | |
| | 1* | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 |
| 37.5-75 | 71.4 | 72.0 | 70.7 | 75.5 | 75.4 | 72.2 | 73.2 | 74.6 | 74.2 | 68.4 | 69.2 | 73.8 |
| 75-150 | 73.6 | 79.2 | 83.4 | 75.8 | 80.5 | 86.3 | 75.1 | 84.6 | 88.2 | 74.2 | 80.2 | 86.2 |
| 150-300 | 66.0 | 73.2 | 72.5 | 71.6 | 77.4 | 81.1 | 71.9 | 78.0 | 80.0 | 67.9 | 73.9 | 77.2 |
| 300-600 | 65.4 | 67.4 | 69.0 | 67.3 | 68.9 | 73.8 | 70.0 | 71.4 | 75.3 | 65.3 | 66.2 | 69.5 |
| 600-1200 | 65.7 | 68.7 | 70.0 | 70.6 | 71.7 | 75.6 | 70.7 | 71.2 | 76.9 | 70.9 | 69.4 | 72.6 |
| 1200-2400 | 61.3 | 64.5 | 68.3 | 68.0 | 74.2 | 75.1 | 66.7 | 66.0 | 72.2 | 64.3 | 64.7 | 71.4 |
| 2400-4800 | 52.0 | 56.4 | 61.1 | 63.1 | 65.9 | 71.1 | 59.9 | 60.9 | 68.4 | 56.7 | 58.7 | 64.0 |
| 4800-9600 | <52.2 | <52.2 | 56.2 | 55.7 | 62.3 | 66.3 | 53.2 | 54.2 | 63.6 | <52.2 | <52.2 | 59.0 |
| Computed overall | 76.9 | 81.3 | 84.4 | 80.6 | 84.1 | 88.3 | 79.8 | 86.2 | 89.5 | 77.7 | 81.9 | 87.3 |

*1 - No load, blower off. 2 - 100-amp load, blower on. 3 - 1000-amp load, blower on.

b) Engine Performance (Range) Data

| | Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| 1* | 0 | 0 | 48-51 | 125-130 | 89-100 | 1750-1760 | 43-48 |
| 2 | 100 | 28.0 | 48-51 | 130-145 | 85-102 | 1740-1750 | 43-48 |
| 3 | 1000 | 29.3-29.8 | 50-51 | 150-165 | 87-95 | 1990-2000 | 43-48 |

TABLE C.16

Unit: C-26
Date: February 21, 1956
Treatment: Load variation, mufflers on,
blower resiliently mounted
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes $cm^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | | | 90 | | | 180 | | | 270 | | |
| | 1* | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 | 1 | 2 | 3 |
| 37.5-75 | 70.3 | 71.8 | 70.3 | 74.7 | 75.6 | 71.8 | 70.0 | 70.6 | 73.9 | 67.7 | 69.3 | 71.2 |
| 75-150 | 71.5 | 73.8 | 83.3 | 74.2 | 74.9 | 83.9 | 73.3 | 75.2 | 87.0 | 73.1 | 74.9 | 84.5 |
| 150-300 | 64.2 | 67.3 | 71.5 | 69.5 | 70.5 | 75.1 | 68.6 | 69.1 | 75.7 | 65.8 | 67.9 | 71.9 |
| 300-600 | 62.8 | 64.3 | 67.8 | 65.9 | 65.3 | 69.8 | 67.3 | 67.3 | 70.8 | 62.4 | 62.5 | 66.5 |
| 600-1200 | 65.7 | 65.2 | 70.7 | 69.0 | 70.2 | 75.2 | 71.7 | 69.3 | 73.1 | 67.3 | 67.7 | 70.7 |
| 1200-2400 | 64.1 | 62.1 | 69.2 | 66.3 | 68.6 | 73.4 | 66.3 | 67.3 | 73.2 | 64.5 | 65.6 | 71.2 |
| 2400-4800 | 52.0 | 55.1 | 63.9 | 61.0 | 65.7 | 71.9 | 58.5 | 59.9 | 69.7 | 57.5 | 57.4 | 66.2 |
| 4800-9600 | <51.2 | 51.2 | 56.5 | 54.2 | 61.4 | 69.0 | 53.2 | 54.6 | 62.4 | 57.9 | 51.2 | 61.0 |
| Computed overall | 75.6 | 77.2 | 84.3 | 79.2 | 80.2 | 85.8 | 78.1 | 78.6 | 88.0 | 76.2 | 78.4 | 85.4 |

*1 - No load, blower off. 2 - 100-amp load, blower on. 3 - 1000-amp load, blower on.

b) Engine Performance (Range) Data

| | Load | | Oil Pressure psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|---|
| | Amperes | Volts | | | | | |
| 1* | 0 | 28.0 | 56-57 | 120-135 | 76-96 | 1730-1780 | 48-53 |
| 2 | 100 | 28.2-28.4 | 56-57 | 120-145 | 80-100 | 1720-1780 | 48-53 |
| 3 | 1000 | 29.9-30.0 | 56-57 | 145-170 | 85-95 | 2000-2010 | 48-53 |

TABLE C.17

Unit: C-26
Treatment: Panel absorptive and damping treatment

| Panel No. Treated* | Treatment |
|---|---|
| 7 | 1 piece, 2-1/2" thick, PF-334 Fiberglas (.5 lb/ft$^3$), 6" x 24". |
| Top of engine compartment | 2 pieces, 2-1/2" thick, PF-334 Fiberglas, 18" x 36" and 18" x 18". |
| 12 | 1 piece, 2-1/2", PF-334 Fiberglas, 18" x 18". |
| 6 | 2 pieces, 2-1/2", PF-334 Fiberglas, 15" x 26" back to back with 18-gauge galvanized sheet metal between. This composite panel is attached perpendicularly to the engine compartment cover between panels 6 and 7. It is designed to block the straight-through path from the engine compartment out around the engine control panel. |
| Top of generator compartment | 2 pieces, 1" Microlite fibrous glass duct liner 12" x 24" and 12" x 18". |
| 14 | Panel over louvers; 1 piece, 2-1/2", PF-334 and 18-gauge galvanized sheet metal spaced 3" inside housing, 13" x 16". |
| 11 | Panel over louvers; 1 piece, 1" Microlite and 18-gauge galvanized sheet metal placed 3-1/2" inside housing, 14" x 17". |
| 9 and 10 | Panel over louvers; 1 piece, 1" Microlite and 18-gauge galvanized sheet metal placed 3-1/2" outside housing, 21" x 36". |
| 4 | 1 piece, 2-1/2", PF-334, 6" x 20". |
| 12 and 13 | 2 pieces, 1" Microlite with 18-gauge sheet-metal septum between, 6" x 18"; placed perpendicular to housing between panels 12 and 13 to block engine noise. |

*See Figs. C.1, C.2.

TABLE C.18

Unit: C-26
Date: June 27, 1956
Treatment: Absorptively lined, mufflers on, blower resiliently mounted
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 71.6 | 76.1 | 71.0 | 71.0 | 72.4 | 79.9 | 74.1 | 81.4 | 73.7 | 73.1 | 73.0 | 73.5 | 74.2 |
| 75-150 | 85.6 | 84.6 | 85.1 | 86.1 | 86.7 | 88.6 | 89.3 | 89.1 | 88.8 | 87.3 | 86.5 | 87.1 | 87.1 |
| 150-300 | 74.1 | 74.9 | 75.6 | 74.3 | 73.5 | 74.9 | 77.1 | 75.6 | 75.0 | 74.6 | 71.6 | 69.0 | 74.2 |
| 300-600 | 66.8 | 71.8 | 69.8 | 68.1 | 68.6 | 69.3 | 70.8 | 71.1 | 68.6 | 66.6 | 67.3 | 66.3 | 68.8 |
| 600-1200 | 66.4 | 69.0 | 70.0 | 70.0 | 69.6 | 69.0 | 70.7 | 69.2 | 70.5 | 66.8 | 67.2 | 66.7 | 68.8 |
| 1200-2400 | 66.7 | 71.2 | 72.8 | 70.7 | 73.5 | 69.8 | 68.2 | 67.8 | 67.2 | 64.1 | 66.7 | 65.8 | 68.7 |
| 2400-4800 | 58.4 | 64.4 | 63.9 | 63.1 | 74.9 | 60.9 | 58.7 | 59.0 | 57.4 | 57.9 | 60.4 | 57.9 | 61.4 |
| 4800-9600 | < 52.6 | 58.5 | 59.2 | 59.7 | 59.9 | 57.1 | 55.1 | 55.8 | 56.4 | 58.0 | 55.8 | < 52.6 | < 56.7 |
| Computed overall | 86.2 | 86.0 | 86.2 | 86.8 | 87.6 | 89.4 | 89.8 | 90.1 | 89.2 | 87.8 | 87.0 | 87.5 | 87.8 |

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Cylinder Temp, °F | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|---|
| Amperes | Volts | | | | | |
| 930 | 29.5-30.2 | 40-56 | 178-200 | 90-125 | 2000-2030 | 77 |

TABLE C.19

Unit: MA-1 gas turbine
Date: September 14, 1955
Treatment: Effect of air-load silencer, unlined
Type of Measurement: Free-field, wide band, uncorrected

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| | Positions* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | | 4 | |
| | a | b | a | b | a | b | a | b |
| Wide-band levels | 84.5 | 85.0 | 97.5 | 100.2 | 89.3 | 95.4 | 86.4 | 103.8 |

*See Section III, MA-1 discussion; see Fig. 3.46.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 38-42 | 38-42 | 1060-1090 | 35-35.2 | 85 |

TABLE C.20

Unit: MA-1 gas turbine
Date: September 21, 1955
Treatment: Effect of air-load silencer, lined
Type of Measurement: Free-field, wide band, uncorrected

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| | Positions* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | | 4 | |
| | a | b | a | b | a | b | a | b |
| Wide-band levels | 86.4 | 86.7 | 103.8 | 104.2 | 87.2 | 98.5 | 90.8 | 107.2 |

*See Section III, MA-1 discussion; see Fig. 3.46.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 40-43.8 | 38-39 | 1010-1075 | 34.8-34.95 | 65 |

TABLE C.21

Unit: MA-1 gas turbine
Date: October 26, 1955
Treatment: As-received with air-load silencer
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 79.9 | 80.2 | 80.5 | 78.0 | 80.3 | 83.8 | 84.8 | 83.3 | 71.0 | 80.9 | 80.2 | 79.8 | 80.2 |
| 75-150 | 100.7 | 89.4 | 90.6 | 86.9 | 91.2 | 92.4 | 93.8 | 93.7 | 92.0 | 90.8 | 90.7 | 91.3 | 92.0 |
| 150-300 | 92.9 | 91.9 | 91.5 | 90.1 | 88.9 | 95.2 | 97.5 | 96.5 | 93.3 | 92.1 | 90.8 | 90.6 | 92.6 |
| 300-600 | 93.0 | 92.7 | 92.4 | 87.5 | 93.3 | 96.7 | 98.6 | 98.1 | 92.8 | 90.4 | 89.8 | 91.0 | 93.0 |
| 600-1200 | 93.5 | 95.1 | 94.0 | 92.1 | 98.2 | 102.3 | 105.0 | 103.4 | 97.8 | 95.5 | 92.5 | 94.4 | 97.0 |
| 1200-2400 | 93.7 | 95.9 | 93.8 | 88.2 | 93.7 | 98.1 | 102.8 | 98.3 | 93.0 | 91.3 | 92.8 | 96.5 | 94.8 |
| 2400-4800 | 96.7 | 97.3 | 95.5 | 93.2 | 96.3 | 97.3 | 98.6 | 96.8 | 93.8 | 97.0 | 98.3 | 94.9 | 96.3 |
| 4800-9600 | 97.8 | 109.5 | 108.4 | 105.1 | 109.6 | 113.1 | 97.3 | 95.3 | 109.7 | 98.6 | 99.1 | 107.8 | 104.3 |
| Computed overall | 104.9 | 110.3 | 109.1 | 105.9 | 110.4 | 113.9 | 109.0 | 107.0 | 110.4 | 103.3 | 103.4 | 108.8 | 108.0 |
| Order in which measured | 7 | 6 | 5 | 4 | 3 | 2 | 1 | 12 | 11 | 10 | 9 | 8 | |

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 43 | 38 | 1040-1080 | 34.8-35.1 | 76-84 |

TABLE C.22

Unit: MA-1 gas turbine
Date: December 15, 1955
Treatment: Cart-transmission studies, panels damped
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | |
| | Undamped | Damped | Undamped | Damped | Undamped | Damped |
| 37.5-75 | 99.6 | 98.8 | 99.8 | 100.0 | 99.8 | 100.6 |
| 75-150 | 104.7 | 104.7 | 104.7 | 104.2 | 104.6 | 105.7 |
| 150-300 | 100.5 | 100.3 | 100.9 | 100.2 | 100.7 | 101.0 |
| 300-600 | 104.9 | 104.4 | 104.9 | 105.0 | 105.3 | 106.0 |
| 600-1200 | 100.4 | 100.3 | 100.7 | 100.4 | 100.1 | 100.9 |
| 1200-2400 | 94.2 | 94.8 | 94.7 | 94.8 | 94.3 | 95.9 |
| 2400-4800 | 99.6 | 102.9 | 103.5 | 104.3 | 102.6 | 99.0 |
| 4800-9600 | 96.9 | 99.2 | 102.8 | 97.2 | 101.8 | 95.6 |

*See Fig. 3.62

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 45-46.5 | 38 | 985-1040 | 33.9-34.5 | 41-46 |

TABLE C.23

Unit: MA-1 gas turbine
Date: December 17, 1955
Treatment: Air-load silencer evaluation
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | | |
|---|---|---|---|---|---|---|---|
| | 1 | | 2 | | 3 | 3A | 3B |
| | Silencer | | Silencer | | Silencer | | |
| | On | Off | On | Off | Off | On | On |
| 37.5-75 | 83.7 | 83.2 | 94.8 | 93.2 | 96.5 | 91.3 | 96.4 |
| 75-150 | 92.2 | 92.1 | 100.0 | 98.2 | 97.6 | 99.3 | 102.6 |
| 150-300 | 94.5 | 94.1 | 107.5 | 105.0 | 100.6 | 98.7 | 102.7 |
| 300-600 | 103.6 | 100.9 | 113.9 | 112.4 | 109.0 | 98.4 | 98.4 |
| 600-1200 | 101.0 | 101.6 | 110.0 | 108.7 | 117.0 | 97.0 | 98.1 |
| 1200-2400 | 95.3 | 95.4 | 110.0 | 108.1 | 112.9 | 100.1 | 98.4 |
| 2400-4800 | 95.5 | 96.6 | 123.7 | 122.4 | 116.5 | 104.9 | 95.0 |
| 4800-9600 | 96.8 | 100.0 | 125.9 | 123.0 | 117.6 | 103.0 | 96.0 |
| Computed overall | 107.2 | 106.9 | 128.3 | 126.1 | 122.6 | 109.5 | 108.4 |

*See Fig. 3.51.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 44-46 | 38-39 | 990-1070 | 34.0-35.1 | 60-65 |

TABLE C.24

Unit: MA-1 gas turbine
Date: January 18, 1956
Treatment: As-received, air by-passed
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Bearing (degrees) 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 37.5-75 | 86.8 | 86.4 | 83.6 | 83.2 | 81.4 | 82.4 | 82.2 | 82.6 | 81.8 | 82.8 | 83.8 | 85.7 | 83.6 |
| 75-150 | 95.6 | 95.0 | 94.2 | 93.0 | 92.5 | 92.7 | 92.6 | 92.1 | 91.0 | 92.1 | 93.5 | 94.6 | 93.2 |
| 150-300 | 98.7 | 97.7 | 95.7 | 95.1 | 92.7 | 93.3 | 94.7 | 92.9 | 92.6 | 94.4 | 95.0 | 97.6 | 95.0 |
| 300-600 | 102.1 | 100.8 | 97.4 | 94.1 | 94.4 | 96.4 | 97.8 | 97.7 | 95.4 | 95.6 | 97.2 | 100.6 | 97.4 |
| 600-1200 | 107.2 | 106.5 | 102.9 | 99.8 | 99.4 | 102.2 | 102.2 | 102.7 | 99.6 | 99.6 | 101.7 | 105.9 | 102.5 |
| 1200-2400 | 105.7 | 102.4 | 98.1 | 95.8 | 98.3 | 100.1 | 100.3 | 100.0 | 97.7 | 96.6 | 98.5 | 102.8 | 99.7 |
| 2400-4800 | 103.9 | 104.0 | 103.0 | 95.6 | 99.6 | 100.0 | 100.7 | 105.6 | 95.9 | 97.5 | 99.6 | 102.7 | 100.7 |
| 4800-9600 | 108.0 | 107.0 | 102.5 | 99.0 | 99.4 | 102.5 | 101.2 | 107.0 | 98.7 | 97.8 | 101.6 | 106.0 | 102.6 |
| Computed overall | 113.1 | 112.0 | 108.8 | 105.2 | 106.0 | 108.0 | 108.0 | 111.0 | 105.2 | 105.3 | 107.5 | 111.4 | 108.4 |
| Order in which measured | 1 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | |

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 44.5-45.6 | 38 | 900-905 | 33.7-34.2 | 40-50 |

TABLE C.25

Unit: MA-1 gas turbine
Date: January 30, 1956
Treatment: 75-150 cps octave-band study of silencer and by-pass conditions
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | |
|---|---|---|---|---|---|---|
| | A | | B | | C | |
| | Silencer On | By-pass | Silencer On | By-pass | Silencer On | By-pass |
| 75-150 | 89.1 | 88.7 | 106.6 | 107.3 | 97.1 | 94.8 |
| | 89.1 | 101.2 | 107.6 | 115.3 | 97.5 | 106.1 |
| | 90.0 | 100.8 | 106.6 | 115.2 | 97.2 | 105.6 |
| | 99.8 | 88.7 | 114.1 | 107.3 | 112.6 | 94.8 |
| | | | | | 96.6 | |

*See Fig. 3.57.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 44.5-47 | 38 | 940-1060 | 34.8-35.1 | 30 |

TABLE C.26

Unit: MA-1 gas turbine
Date: January 31, 1956
Treatment: Reproducibility of 75-150 cps band
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | |
|---|---|---|
| | C | |
| | Silencer On | By-pass |
| 75-150 | 97.6 | 96.0 |
| | 97.6 | 96.1 |
| | 97.7 | 96.0 |
| | 97.8 | 96.0 |
| | 98.5 | 96.0 |
| | 98.1 | 96.0 |
| | 98.2 | 96.1 |
| | 98.7 | 96.8 |
| | | 96.6 |
| | | 96.5 |
| | | 96.6 |

*See Fig. 3.57.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 44.2-47 | 38 | 920-1050 | 33.3-35 | 23-27 |

TABLE C.27

Unit: MA-1 gas turbine
Date: February 1, 1956
Treatment: 75-150 cps octave-band study of silencer and by-pass conditions
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Box | Positions* | | | | | |
|---|---|---|---|---|---|---|---|
| | | A | | B | | C | |
| | | Silencer On | By-pass | Silencer On | By-pass | Silencer On | By-pass |
| 75-150 | On | 91.0<br>90.8 | 90.7<br>91.1 | 107.1<br>106.5 | 106.8<br>107.1 | 101.7<br>101.7 | 100.2<br>100.6 |
| | Close | 91.1<br>90.8 | 91.5<br>91.2 | 106.7<br>106.2 | 107.0<br>107.1 | 101.1<br>101.2 | 100.7<br>100.9 |
| | Remote | 91.1<br>90.6 | 91.2<br>90.8 | 106.4<br>106.1 | 106.8<br>107.0 | 98.6<br>98.6 | 96.6<br>97.0 |

*See Fig. 3.57.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 46-48 | 38 | 930-105 | 34.5-35 | 17.4-22 |

TABLE C.28

Unit: MA-1 gas turbine
Date: April 9, 1956
Treatment: Exhaust muffler No. 1
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Operator's Station | | | In Line with Shaft 40 Feet from Exhaust | | |
|---|---|---|---|---|---|---|
| | Muffler | | | Muffler | | |
| | Off | On Without Perforated Top | On With Per. Top | Off | On Without Perforated Top | On With Per. Top |
| 37.5-75 | 92.3 | 94.1 | 82.3 | 79.5 | 84.2 | 83.2 |
| 75-150 | 104.7 | 109.2 | 107.5 | 89.6 | 95.1 | 93.3 |
| 150-300 | 108.0 | 106.5 | 107.0 | 93.5 | 89.1 | 88.8 |
| 300-600 | 109.4 | 108.5 | 109.3 | 91.9 | 92.0 | 94.3 |
| 600-1200 | 111.4 | 108.4 | 107.6 | 95.1 | 89.3 | 89.0 |
| 1200-2400 | 111.2 | 108.0 | 108.1 | 93.3 | 86.9 | 83.2 |
| 2400-4800 | 120.2 | 123.4 | 119.1 | 92.0 | 85.1 | 82.4 |
| 4800-9600 | 122.5 | 121.6 | 128.0 | 97.4 | 87.8 | 89.4 |

b) Engine Performance (Range) Data

rpm = 30,000

TABLE C.29

Unit: MA-1 gas turbine
Date: April 30, 1956
Treatment: Exhaust muffler No. 2.
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Positions* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Muffler On | | | | Muffler Off | | | |
| | A | B | C | D | E | F | G | H |
| 37.5-75 | 90.8 | 95.7 | 101.1 | 100.4 | 93.2 | 97.1 | 102.3 | 97.4 |
| 75-150 | 93.1 | 99.1 | 104.7 | 99.1 | 98.6 | 105.0 | 103.8 | 99.1 |
| 150-300 | 103.4 | 104.8 | 110.5 | 108.8 | 106.6 | 119.0 | 111.3 | 106.9 |
| 300-600 | 111.9 | 113.8 | 120.3 | 119.8 | 112.7 | 117.1 | 122.3 | 116.2 |
| 600-1200 | 108.5 | 112.1 | 116.7 | 114.3 | 112.1 | 115.4 | 119.7 | 115.8 |
| 1200-2400 | 110.3 | 114.9 | 116.3 | 114.2 | 112.1 | 117.5 | 120.0 | 116.0 |
| 2400-4800 | 122.0 | 125.8 | 112.3 | 110.0 | 122.0 | 123.9 | 118.0 | 114.3 |
| 4800-9600 | 123.3 | 128.4 | 113.6 | 109.8 | 122.3 | 126.3 | 124.1 | 119.6 |

*See Fig. 3.67.

b) Engine Performance (Range) Data

| Delivery Pressure, psi | Oil Pressure, psi | Exhaust Temp, °F | Speed, rpm (1000) | Ambient Temp, °F |
|---|---|---|---|---|
| 46 | 37-38 | 930-950 | 34.8-35.0 | 55 |

TABLE C.30

Unit: MA-1 Consolidated Multipurpose
Date: November 18, 1955
Treatment: As-received
Type of Measurement: Free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Bearing (degrees) | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 30 | 60 | 90 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | Avg |
| 37.5-75 | 69.8 | 67.1 | 66.9 | 68.3 | 68.8 | 67.4 | 68.8 | 71.1 | 71.0 | 81.9 | 71.4 | 70.2 | 70.2 |
| 75-150 | 83.5 | 79.8 | 78.2 | 77.9 | 78.6 | 77.5 | 78.7 | 78.1 | 75.6 | 76.5 | 81.7 | 83.8 | 79.2 |
| 150-300 | 82.1 | 79.6 | 81.6 | 76.4 | 79.7 | 79.9 | 79.0 | 78.1 | 78.9 | 80.6 | 82.0 | 81.2 | 79.9 |
| 300-600 | 77.2 | 75.1 | 76.8 | 73.2 | 75.1 | 78.3 | 79.1 | 76.1 | 76.6 | 77.6 | 76.8 | 77.4 | 76.6 |
| 600-1200 | 80.8 | 77.0 | 77.7 | 75.5 | 76.5 | 81.7 | 85.1 | 77.7 | 73.8 | 75.1 | 74.6 | 77.9 | 77.8 |
| 1200-2400 | 76.6 | 76.0 | 75.9 | 74.4 | 75.6 | 78.3 | 80.4 | 77.1 | 74.3 | 73.8 | 72.8 | 75.7 | 75.9 |
| 2400-4800 | 72.7 | 73.5 | 72.6 | 73.1 | 75.1 | 75.2 | 75.0 | 75.9 | 73.6 | 68.5 | 69.4 | 72.7 | 73.1 |
| 4800-9600 | 75.9 | 74.8 | 73.5 | 75.0 | 77.6 | 75.9 | 76.1 | 79.2 | 74.9 | 70.1 | 70.2 | 74.4 | 74.8 |
| Computed overall | 88.3 | 85.6 | 86.0 | 83.9 | 85.7 | 87.1 | 88.7 | 86.2 | 84.4 | 86.5 | 86.4 | 87.7 | 86.4 |
| Order in which measured | 1 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | |

b) Engine Performance (Range) Data

| D-c Generator Load | | A-c Generator Load | | Oil Pressure, psi | Cylinder Temp, °F | Compressor After Cooler Service | | |
|---|---|---|---|---|---|---|---|---|
| Amperes | Volts | Volts | Cps | | | Oil Pressure, psi | Air Pressure,* psi | Ambient Temp, °F |
| 880-890 | 30.0-30.1 | 115-116 | 402-406 | 45-47 | 130-150 | 27-36.5 | 2950-3500 | 37-50 |

*Receiver drain opened to maintain pressure below 3500 psi.

Unit: B10B
Date: November 29, 1955
Treatment: No load, as received
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|---|
| 37.5-75 | 99.6 | 99.9 | 105.8 | 105.0 | 104.9 | 104.2 | 106.8 | 101.3 |
| 75-150 | 105.2 | 106.5 | 112.1 | 109.5 | 111.1 | 110.4 | 112.9 | 105.4 |
| 150-300 | 96.9 | 100.5 | 104.8 | 104.6 | 101.1 | 102.1 | 105.6 | 97.5 |
| 300-600 | 99.0 | 96.5 | 96.9 | 99.6 | 100.9 | 98.6 | 101.0 | 96.1 |
| 600-1200 | 99.9 | 95.5 | 100.6 | 98.6 | 106.7 | 99.9 | 102.6 | 97.0 |
| 1200-2400 | 97.3 | 91.0 | 96.8 | 94.6 | 96.8 | 95.1 | 97.1 | 93.9 |
| 2400-4800 | 91.6 | 85.2 | 94.2 | 91.7 | 93.9 | 92.2 | 93.7 | 91.1 |
| 4800-9600 | 94.4 | 86.9 | 94.6 | 91.3 | 93.4 | 92.2 | 94.9 | 92.4 |
| Computed overall | 108.7 | 108.8 | 114.1 | 112.3 | 113.8 | 112.4 | 115.1 | 108.3 |

*See Fig. 3.78.

b) Engine Performance (Range) Data

| Load Amperes | Load Volts | Oil Pressure, psi | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|
| 0, 116 | 201-202 | 68-80 | 80-135 | 2370-2496 | 50 |

TABLE C.32

Unit: B10B
Date: November 29, 1955
Treatment: 116-ampere load, as-received
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Position* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
| 37.5 | 108.8 | 107.3 | 108.3 | 109.9 | 109.4 | 107.3 | 109.7 | 105.7 |
| 75-150 | 113.1 | 114.3 | 117.2 | 114.7 | 114.0 | 112.3 | 117.6 | 112.2 |
| 150-300 | 108.7 | 110.8 | 112.1 | 110.4 | 110.3 | 110.1 | 113.5 | 111.7 |
| 300-600 | 105.1 | 103.0 | 104.1 | 105.7 | 103.6 | 106.5 | 104.5 | 102.8 |
| 600-1200 | 107.9 | 101.9 | 105.3 | 104.0 | 105.2 | 105.7 | 109.0 | 102.9 |
| 1200-2400 | 108.4 | 98.6 | 103.2 | 101.4 | 99.2 | 102.3 | 104.3 | 101.7 |
| 2400-4800 | 104.6 | 95.1 | 102.4 | 99.8 | 98.7 | 101.1 | 103.0 | 101.1 |
| 4800-9600 | 106.0 | 97.2 | 104.6 | 101.1 | 102.4 | 103.5 | 105.8 | 103.8 |
| Computed overall | 117.7 | 116.9 | 119.5 | 117.8 | 117.3 | 116.6 | 120.4 | 116.4 |

*See Fig. 3.78

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|
| Amperes | Volts | | | | |
| 0,116 | 201-202 | 68-80 | 80-135 | 2370-2496 | 50 |

TABLE C.33

Unit: B10B
Date: November 29, 1955
Treatment: Comparison of 3- and 4-stage aspirators, load and no load, cover removed
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/cm$^2$)

| Octave Band (cps) | Positions* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 1 | | 8 | | 1 | | 8 | |
| | 3-Stage Aspirator | | | | 4-Stage Aspirator | | | |
| | 0 load | 116-amp load | 0 load | 116-amp load | 0 load | 116-amp load | 0 load | 116-amp load |
| 37.5-75 | 98.7 | 105.9 | 96.8 | 101.0 | 103.1 | 102.4 | 97.1 | 99.6 |
| 75-150 | 104.4 | 108.5 | 102.3 | 109.2 | 106.7 | 105.4 | 101.6 | 106.5 |
| 150-300 | 101.5 | 110.3 | 98.6 | 111.2 | 99.0 | 107.6 | 99.9 | 108.1 |
| 300-600 | 100.7 | 108.0 | 100.6 | 105.2 | 97.3 | 109.1 | 97.8 | 106.7 |
| 600-1200 | 99.1 | 106.5 | 98.0 | 104.0 | 96.7 | 101.8 | 96.3 | 101.3 |
| 1200-2400 | 93.2 | 103.3 | 94.0 | 102.1 | 93.7 | 100.2 | 94.2 | 99.7 |
| 2400-4800 | 90.5 | 101.3 | 90.6 | 100.8 | 91.9 | 99.0 | 91.8 | 99.4 |
| 4800-9600 | 92.4 | 104.8 | 92.0 | 103.5 | 99.0 | 105.5 | 96.6 | 104.7 |
| Computed overall | 108.7 | 115.9 | 107.2 | 115.3 | 109.9 | 114.1 | 106.8 | 113.5 |

*See Fig. 3.78.

b) Engine Performance (Range) Data

| Load | | Oil Pressure, psi | Oil Temp, °F | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|---|---|
| Amperes | Volts | | | | |
| 0, 116 | 201-202 | 68-80 | 80-135 | 2370-2496 | 50 |

TABLE C.34

Unit: MDX
Date: November 29, 1955
Treatment: As-received with 4-stage aspirator
Type of Measurement: Non-free-field

a) Acoustical Data
(Sound-pressure level, db re 0.0002 dynes/$cm^2$)

| Octave Band (cps) | Position* | |
|---|---|---|
| | 1 | |
| | No Load | 116-Ampere Load |
| 37.5-75 | 105.0 | 104.2 |
| 75-150 | 106.7 | 106.4 |
| 150-300 | 97.9 | 105.9 |
| 300-600 | 95.1 | 104.5 |
| 600-1200 | 93.4 | 98.0 |
| 1200-2400 | 91.0 | 97.5 |
| 2400-4800 | 90.0 | 98.0 |
| 4800-9600 | 98.6 | 105.1 |
| Computed overall | 110.0 | 112.7 |

*See Fig. 3.78; position 1 five feet from unit and 6 feet 3 inches above ground.

b) Engine Performance (Range) Data

| Load | | Engine Speed, rpm | Ambient Temp, °F |
|---|---|---|---|
| Amperes | Volts | | |
| 0, 116 | 209-210 | 2364-2370 | 50 |