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NOISE CONTROL FOR AIRCRAFT ENGINE TEST CELLS AND GROUND RUN-UP SUPPRESSORS

Volume 3: An Engineering Analysis of Measurement Procedures and of Design Date

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AND
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FOREWORD

This report was prepared by the firm Bolt Beranek and Newman Inc., Cambridge, Massachusetts, under Contract Numbers AF 33(616)-3335 and AF 33 (616)-3938 for the Bioacoustics Branch under Project 7210, "The Generation, Propagation, Action and Control of Acoustic Energy," Task 71708, "Investigation of Physical Structures and their Components with Respect to their Characteristics for Acoustic Energy, Reception, Transmission and Reduction." Mr. R. N. Hancock was the task engineer. Technical supervision of the preparation of this report was the responsibility of Mr. R. N. Hancock, Capt. L. O. Hoeft and Dr. H. E. von Gierke, Bioacoustics Branch, Aerospace Medical Laboratory, Aeronautical Systems Division, Wright-Patterson Air Force Base, Ohio.

This is the third of three volumes concerning the physical effects of noise control in aircraft engine test cells. Volume 1 presents recommended procedures for measuring noise control effectiveness and volume 2 deals with design and planning for noise control.

The suggestions and comments of Dr. R. H. Bolt and Mr. A. C. Pietrasanta of Bolt Beranek and Newman Inc. and Capt. L. O. Hoeft of the Bioacoustics Branch have been of great help in the preparation of this report.

The WADC technical report number identifying this series of documents was assigned by Wright Air Development Center before it was redesignated Aeronautical Systems Division.

A companion report, technical documentary report number AMRL-TDR-62-134, Influence of Noise Control Components and Structures on Turbojet Engine Testing and Aircraft Ground Operation, has been written by Bonard E. Morse and the staff of Kittell-Lacy, Inc., El Monte, California, under Contract AF 33(616)-5789, for 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio.

ABSTRACT

This volume is one of three volumes on the physical aspects of noise control in aircraft engine test cells and ground run-up suppressors. The measurement procedures and the noise reduction data that form a technical basis for many of the techniques and ideas presented in the other volumes are analyzed. Errors arising from the measurement equipment, wide-band frequency analysis, random variations of noise level in time and space, the use of artificial noise sources, variations in air flow conditions and different measurement procedures are investigated to obtain an objective measure of the reliability of data obtained from an AF sponsored program of acoustical evaluations of test cells and ground run-up suppressors. Data on impervious barriers and noise control components for air passages are analyzed. The performance of a single wall barrier can be reliably estimated, but the large noise reductions expected from double wall barriers are seldom obtained because of flanking paths. The performance of noise control components for air passages was found to differ significantly from that predicted by theory (first order modes, long treatment). Differences are attributed to the spatially random nature of the noise field. Empirical corrections are presented.

PUBLICATION REVIEW

This report has been reviewed and approved.

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SECTION I
INTRODUCTION

The United States Air Force is conducting a program of acoustical evaluations of aircraft engine test cells and aircraft ground run-up suppressors. Under this program, detailed measurements have been carried out on more than twenty test cells and four ground run-up suppressors. The results of the program obtained to date, together with relevant information from other sources, are summarized in three volumes:

1. Measurement and Analysis of Acoustical Performance.[41]
2. Design and Planning for Noise Control.[42]
3. An Engineering Analysis of Measurement Procedures and of Design Data.

These three volumes deal only with the physical aspects of noise control. Information concerning the psychological and physiological problems of criteria for noise control is contained in other Air Force reports[1-6].

In the first two volumes, no attempt was made to provide a technical justification or basis for the information presented. Where possible, references were made to the literature of acoustics. However, much of the data and many of the procedures in the first two volumes are based on information not available in the literature of acoustics. The present volume provides that information which is not available elsewhere.

The primary objective of this report is to analyze and extrapolate the noise reduction data obtained from the program of acoustical evaluations for incorporation in the second volume. In order to accomplish this objective, several possible definitions of noise reduction for noise control components are reviewed in Section II. The differences between the various "noise reductions" are particularly stressed. Definitions of acoustical effectiveness for use in these volumes are presented and the limitations of the definition are discussed.

The possible sources of error in the measurement of noise reduction are quantitatively analyzed in Section III by use of extensive experimental data. The main objective of the analysis is to determine the magnitude of errors in the data presented in this volume so that analyses and extrapolations of the measured noise reduction data may be carried out in a rational manner. However, the analysis of error is general enough so that quantitative estimations of possible errors in test cell data from other sources can be made, if sufficient information is given about the number of measuring positions, the location of microphones, etc. The analysis of the errors also provides a basis for estimating the reliability of the data contained in this volume and in Volume Two.

The noise reduction data is presented in two parts. The first part, presented in Section IV, deals with the noise reduction of impervious barriers. This part illustrates, by selected examples, the differences between theoretical predictions and field data. Generally the differences between theory and field data are small for single-partition structures. For double-wall structures, the differences are large. The relations between the theory and the measurements are discussed and explained.

The second part, presented in Section V, deals with the noise reduction characteristics of acoustical treatments in air passages. Serious discrepancies between theory and the field data have been found. These discrepancies arise primarily because conditions which obtain in test cells are beyond the scope of present day theories. Analysis of the data shows that the behavior of noise reduction components in test cells is significantly different from the generally accepted theories. Furthermore, the analysis casts serious doubts upon the validity of certain types of data obtained by some field and laboratory measurement techniques. The analysis therefore begins with a qualitative description of the behavior of baffles, ducts, and bends in engine test cells. An analysis is then carried out to generalize the data obtained under the program. Extrapolation procedures are presented, and tested, where possible, for determining the noise reduction characteristics of many structures which were not measured under the program.

The final section is devoted to miscellaneous information which is required for the application of noise reduction data to the design of engine test facilities and ground run-up suppressors.

SECTION II
DEFINITIONS OF ACOUSTICAL EFFECTIVENESS

Insertion loss, transmission loss, SPL difference, transmission coefficient, transmission factor, and attenuation are but a few of the many terms which are used in the literature of acoustics to describe the acoustical effectiveness ("noise reduction") of a noise control component. Unfortunately, there is not a one-to-one correspondence between the terms and their definitions, so that it is essential to define carefully the terms that will be used in this volume to describe the acoustical effectiveness of noise control components.

A. General Discussion

In this section, some of the more commonly used measures of acoustical effectiveness are defined and a simple example is presented to show that:

1. The several definitions yield quantitatively different "noise reductions".

2. The "noise reductions" are not solely physical properties of the noise control component, but, instead, are measures of the physical properties <u>and</u> the interaction of these properties with their environment.*

Some of the terms describing acoustical effectiveness deal with ratios of sound energy or sound power. Others deal

* Some definitions of acoustical effectiveness specify certain characteristics of the environment in the definition. For example, <u>transmission loss</u> is usually defined as a ratio of incident to transmitted energy when the energy is transmitted to a ρc impedance. It is then a matter of semantics as to whether or not transmission loss is a property only of the element.

with ratios of sound pressure. Those terms that deal with ratios of sound pressure are of primary interest because the measurement or calculation of sound energy or power is possible only under a few limited conditions, which are not generally applicable to the evaluation of noise control components in aircraft engine test facilities.

In terms of sound pressures, the acoustical effectiveness of a component can be defined as:

1. The ratio of a sound pressure at some point before the noise control element is inserted, to the sound pressure at the same point after the noise control element is inserted.

2. The ratio of a sound pressure incident on the noise control element to a sound pressure transmitted by the noise control element, or

3. The ratio of a sound pressure on the input side of the element, to a sound pressure at the output side of the element.

An illustrative example showing how the several definitions differ is illustrated in Fig 1. The piston at the end of a rigid tube causes a sound pressure which is given by the product of the velocity of the piston and the characteristic impedance of air, ρc. When the massive plate is introduced in the tube, the sound pressures p_{11} and p_{21} obtain. The expressions for the sound pressure, particle velocity, and the impedance at any point in the tube between the piston and the mass can be expressed by Eqs 1 through 3 below[7].

$$p = 2P_{+} e^{-\pi\alpha} e^{-i\omega t} \sqrt{\cosh^2 (\pi\alpha) - \sin^2 (\pi\beta)} \qquad (1)$$

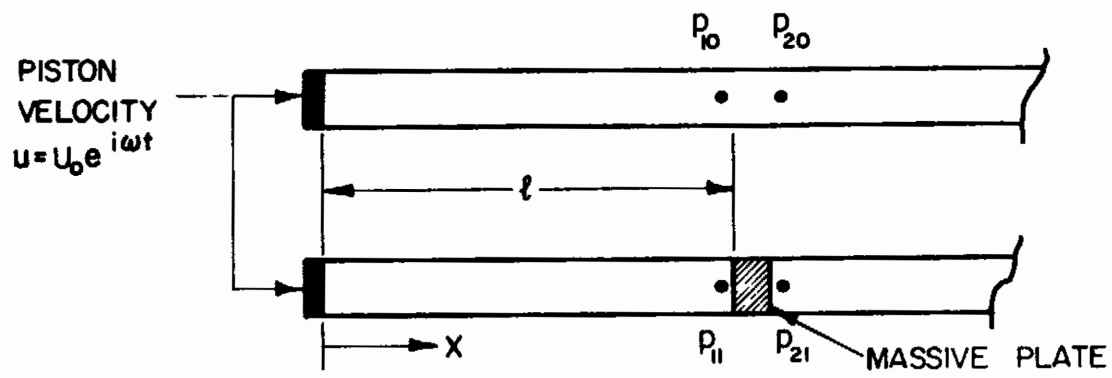

FIG. I RELEVANT TO THE DEFINITION OF ACOUSTICAL EFFECTIVENESS.

$$u = \frac{2P_+}{\rho c} e^{-\pi\alpha} e^{-i\omega t} \sqrt{\cosh^2(\pi\alpha) - \cos^2(\pi\beta)} \tag{2}$$

$$z = \frac{p}{u} = \rho c \frac{\sqrt{\cosh^2(\pi\alpha) - \sin^2(\pi\beta)}}{\sqrt{\cosh^2(\pi\alpha) - \cos^2(\pi\beta)}} \tag{3}$$

in which

p = the sound pressure at any point between $x = 0$ and $x = \ell$

u = particle velocity at any point between $x = 0$ and $x = \ell$

z = the specific acoustic impedance at any point between $x = 0$ and $x = \ell$

P_+ = the sound pressure of a wave propagated from the piston towards the plate

α = a real number which measures the ratio of the magnitudes of the incident and reflected sound pressure or particle velocity waves

β = a real number which measures the phase angle between the incident and reflected waves

Both α and β can be found from the impedance of the massive plate of the tube. For this example, we shall assume that the mass reactance of the plate is much greater than the characteristic impedance of air, and that the impedance of the termination is simply:

$$z_\ell = i\omega m = \frac{p_{11} - p_{21}}{u_{11}} \approx \frac{p_{11}}{u_{11}} \tag{4}$$

where

m is the surface density of the massive plate

ω is the angular frequency

u_{11} is the velocity of the mass

and p_{11} and p_{21} are defined in Fig 1.

Applying the boundary conditions: 1) the particle velocity equals U_0 at $x = 0$, and 2) the impedance of the plate is as given in (4), yields

$$\alpha = 0 \tag{5}$$

$$\beta = k\,(\ell - x) + \phi \tag{6}$$

$$\phi = \tan^{-1} \frac{\omega m}{\rho c} \tag{7}$$

$$P_+ = \frac{U_0\,\rho c}{2 \cos\,[k\ell + \phi]} \tag{8}$$

where

$$k = \frac{\omega}{c}$$

ℓ = the distance from the piston to the mass

Equations 1 and 2 can then be evaluated at $x = \ell$ to yield:

$$p_{11} = U_0 \rho c \quad \frac{\sin \phi}{\cos\,(k\ell + \phi)} \tag{9}$$

and

$$U_{11} = U_0 \; \frac{\cos \phi}{\cos\,(k\ell + \phi)} \tag{10}$$

The transmitted sound pressure, p_{21}, is:

$$p_{21} = \rho c U_{11} = U_0 \rho c \; \frac{\cos \phi}{\cos\,(k\ell + \phi)} \tag{11}$$

and

$$p_{20} = p_{10} = U_0 \rho c \tag{12}$$

A "noise reduction" defined as the ratio of the sound pressure at the input to the sound pressure at the output is:

$$\frac{p_{21}}{p_{11}} = \frac{\cos \phi}{\sin \phi} = \frac{1}{\tan \phi} = \frac{1}{\frac{\omega m}{\rho c}} \tag{13}$$

A "noise reduction" defined as a ratio of incident sound pressure to transmitted sound pressure is:

$$\frac{p_{21}}{p_{+}} = 2 \cos \phi = 2 \sin \phi \ \frac{1}{\tan \phi} = 2 \sin \phi \ \frac{1}{\frac{\omega m}{\rho c}} \tag{14}$$

And finally, a "noise reduction" defined as a ratio of the sound pressures at position 2 before and after insertion of the mass is given by:

$$\frac{p_{21}}{p_{20}} = \frac{\cos \phi}{\cos (k\ell + \phi)} = \frac{\sin \phi}{\cos (k\ell + \phi)} \ \frac{1}{\frac{\omega m}{\rho c}} \tag{15}$$

If $\omega m/\rho c$ is large, as assumed initially, $\tan^{-1} \phi$ approaches 90° and $\sin \phi$ approaches unity. For this case, Eqs 13 and 14 differ only by a factor of 2, expressing the pressure doubling at the face of the massive barrier, where the incident pressure and the reflected pressure add to yield a pressure twice as great as the incident pressure. We might note that if the impedance at $x = \ell$ were small, the reflected pressure would be out of phase with the incident pressure and the difference between the incident pressure and the pressure at the input could be quite large. However, the input impedance of most noise control components will not differ very greatly from ρc and we may expect that, in general, the sound pressure measured at the input of a noise control element will not be very different from the incident pressure. It is perhaps worth pointing out that the square of Eq 14 is the expression for normal incidence transmission loss given in most texts for a massive wall, if $\omega m/2\rho c$ is much larger than unity.

The noise reduction quantity given in Eq 15 (which is usually called insertion loss) differs significantly from those in Eqs 13 and 14. In particular, the insertion loss depends not only on $\omega m/\rho c$ but also upon $k\ell$. The dependence on $k\ell$ indicates that this measure of acoustical effectiveness depends on the geometry in front of the mass, which in turn may be interpreted as indicating a dependence on the driving impedance of the source.

This example illustrates that the three definitions do not yield the same measure of noise reduction, even for a very simple acoustical system. There is only one case for which the three definitions yield the same result*; in all other cases, the noise reduction of a component depends upon the definition selected and on environmental factors (the source, load, and transmission impedances, etc.).

In the example above, the noise reductions could readily be calculated and compared with one another. For noise control elements in engine test facilities, the several noise reductions cannot be readily calculated because the several impedances are not known. Even if they were known, calculation of the several noise reductions would present an extremely difficult task. Each of the impedances, and hence the noise reduction, will depend upon frequency and the angle of incidence of the sound wave at the input. It would therefore be necessary to know beforehand the distribution of pressures as a function of angle of incidence. In the following paragraph

* The specific case is an acoustical system for which the source impedance and the input impedance of the noise control element are both ρc.

definitions of acoustical effectiveness which are appropriate to engine test cells are discussed. In Section III, the influence of environmental factors on noise reduction are pursued in more detail. The noise reduction quantities which are used throughout this text are also compared with other noise reduction quantities in an empirical manner.

B. Definitions of Acoustical Effectiveness for Aircraft Engine Test Facilities

Selection of an appropriate definition of acoustical effectiveness for aircraft engine test facilities must be made considering the procedures which can be used to measure acoustical effectiveness. As was suggested above, definitions related to sound energy or power are of limited value. Determination of sound power requires a knowledge of the phase angle between sound pressure and particle velocity, as well as the direction of the particle velocity, over the entire area of the input and output of an acoustical treatment. To date, no practical instrument has been devised for measuring true sound power in the field. It is necessary, therefore, to restrict the definitions to those relating to sound pressure.

Insertion loss, as defined above, cannot be used because of the practical difficulty of inserting and removing large noise control components in the field. This practical difficulty is unfortunate since the noise control engineer is usually concerned with an insertion loss measure of acoustical effectiveness.

The ratio of incident sound pressure to transmitted sound pressure must also be eliminated from consideration, on the grounds of inadequate instrumentation. Although certain correlation techniques might allow discrimination between incident and reflected waves, conventional measurement techniques do not. At present only the sum of the incident and reflected waves can be measured.

Thus, by a process of elimination, it is necessary to select the ratio of sound pressure on the input side to the sound pressure on the output side of the noise control element as a measure of acoustical effectiveness. Two such noise reductions are used in this report; one for impervious barriers such as walls, doors and windows, and one for acoustical treatments in air passages. For impervious barriers, the noise reduction, NR, is defined as[8]:

$$NR = SPL^*_1 - SPL_2, \qquad (16)$$

where

SPL_1 is the average SPL in the reverberant field on the source side

SPL_2 is the SPL near the barrier on the receiver side.

Where possible, the transmission loss (TL)** of the barrier, should be derived from the NR[8].

The noise reduction, L_{nr}, of an acoustical treatment in an air passage is defined as:

$$L_{nr} = (SPL_{1_{av}} + 10 \log_{10} A_1) - (SPL_{2_{av}} + 10 \log_{10} A_2) \qquad (17)$$

where

$SPL_{1_{av}}$ is found from the average value of the sound

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* $SPL = 20 \log_{10}(p/0.0002)$ db in which p is the sound pressure in microbar.

** Transmission loss equals $10 \log_{10} \frac{W_1}{W_2}$, in which W_1 is the acoustic power incident on the barrier, and W_2 is the acoustic power transmitted by the barrier.

pressure over the input area (A_1 sq ft) of the acoustical treatment.

$SPL_{2_{av}}$ is found from the average value of the sound pressure over the output area (A_2 sq ft).

To a rough approximation:

$$L_{nr} \approx PWL_1 - PWL_2 \tag{18}$$

in which

PWL_1* is the power level at the input,

PWL_2 is the power level at the output.

Equation 18 is only an approximation because the direction of velocity and the phase relations over the input and output areas are not known. Nevertheless, the area terms are retained in the definition L_{nr} for two reasons. The first reason is that gradual changes in the open area of an air passage may result in a change in SPL in the passage, without a loss of PWL. Inclusion of the area assures that such area changes are not identified as noise reductions.**

A second reason for using the area terms and employing the PWL concept in the definition is that this form of definition is readily extendable to acoustical treatments which have multiple inputs (e.g., a test cell with primary and secondary air inlets which have both common and individual treatments).

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* $PWL = 10 \log_{10} \frac{W}{10^{-13}}$ where W is the acoustic power in watts.

** An increase in area is frequently taken as a noise reduction quantity, and justifiably so in some cases. However, if an acoustical treatment changes area greatly from the "input" to the "output", the SPL will diminish, but the PWL will remain constant. The reduction in PWL is a more useful quantity than a reduction in SPL for design of test cells and suppressors.

The noise reduction, L_{nr}, of a multiple input system can be defined as the difference between the total PWL at the inputs to the total PWL at the outputs. The total PWL used is:

$$PWL_{total} = 10 \log_{10} [\text{antilog} \frac{SPL_1 + 10 \log A_1}{10} + \text{antilog} \frac{SPL_2 + 10 \log A_2}{10} + \ldots + \text{antilog} \frac{SPL_n + 10 \log_{10} A_n}{10}] \qquad (19)$$

It should be kept in mind that the PWL_{total} is not actually a power level, but merely a useful artifact for combining the inputs to the acoustical treatment. The acoustical behavior of multiple input systems is discussed in Section III and is not pursued further here. Some general limitations of the L_{nr} method are discussed below.

C. Limitations of the L_{nr} Definition of Acoustical Effectiveness

In order to use the L_{nr} noise reduction in the design of an engine test facility, one must know the SPL at the input to the acoustical treatment. The SPL at the input in the test facility cannot be obtained from the free field noise characteristics of the engine because the test facility may markedly change the noise characteristics of the engine. In addition, the acoustical treatment may also affect the noise characteristics of the engine. Thus it is necessary to determine the noise characteristics of engines in test facilities in order that the L_{nr} definition will be useful in the design of engine test facilities. The noise characteristics of engines in test facilities are discussed in Section V.

Another limitation of the L_{nr} definition is that the acoustical effects of certain noise control elements may be obscured and/or attributed to another noise control element. For example, consider a "straight-through" type of engine test cell and a "U" shaped test cell with identical acoustical treatments. The difference between the SPL at the output of the exhausts, for example, will be of the order of 15 db in the higher frequencies. It would seem reasonable, then, to assume that the difference in noise reductions (15 db) is attributable to the bend which is the only element not common to both test facilities. However, if L_{nr} measurements are carried out in both test cells, it will be found that the L_{nr} of the bend is only about 5 db. It is found, in addition, that the L_{nr} of the treatment following the bend in the "U" shaped cell is about 10 db greater than the L_{nr} of the same acoustical treatment in the straight-through cell. Thus measuring the L_{nr} of the bend alone does not determine the entire acoustical effect of the bend. To determine the total effect of a bend, it is necessary to consider both the L_{nr} of the bend and the change in the L_{nr} of another acoustical treatment caused by the bend.

The influence of the bends on the L_{nr} of an acoustical treatment following the bend is but one example of interactions of acoustical treatments with one another. These interactions are considered in more detail in Section IV.

SECTION III

SOURCES OF ERROR IN THE MEASUREMENT OF NOISE REDUCTION

In order to evaluate and extrapolate noise reduction data, one must have some understanding of the sources and magnitudes of the errors that result from experimental techniques. Two types of error are investigated in this section.

The first type of error is that caused by the random variations, about a mean value of noise reduction, that are obtained if a given type of noise reduction measurement is carried out several times. The sources of these variations are 1) instabilities in the data recording and reduction systems, 2) variations in source levels, and 3) variations of the noise level in the plane of the input or output of an acoustical treatment.

The second type of error is that caused by differences between measured values of noise reduction, as obtained with different experimental conditions and techniques. For example, if a jet engine operating at 100% of maximum revolution rate (rpm) is used as a noise source, the value of noise reduction obtained will not, in general, be the same as the value of noise reduction obtained if the jet engine operates at 55% rpm.

The sources of error of the first type are reviewed in paragraphs A through C below, and estimates of the magnitude of each error are obtained. In paragraphs D through G, three possible causes of the second type of error are investigated, in light of the information derived in paragraphs A through C. The three causes are differences in the measured value of noise reduction which result from: 1) different air flow rates (engine speed), 2) different noise sources (jet engine vs. explosive source), and 3) different measurement procedures

(L_{nr} method vs. EN-1). The entire analysis of errors is summarized in paragraph H.

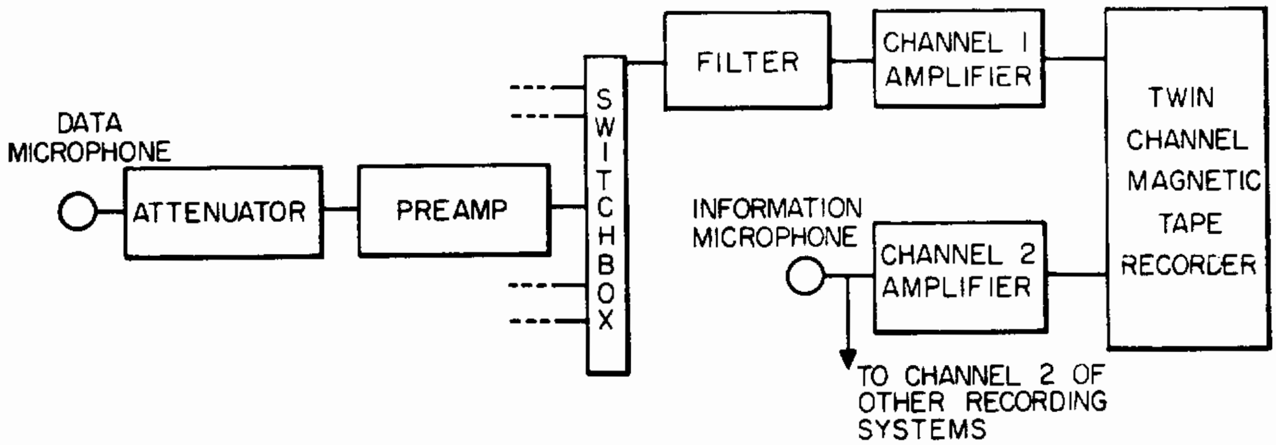
A. Measurement System

Errors caused by the data recording and reduction systems used to obtain a large portion of the data in this report are presented in parts 1 and 2 below. Although frequency analyzers are part of the data reduction system, a separate section is devoted to them because errors arising from the use of frequency analyzers are not solely related to the data reduction process.

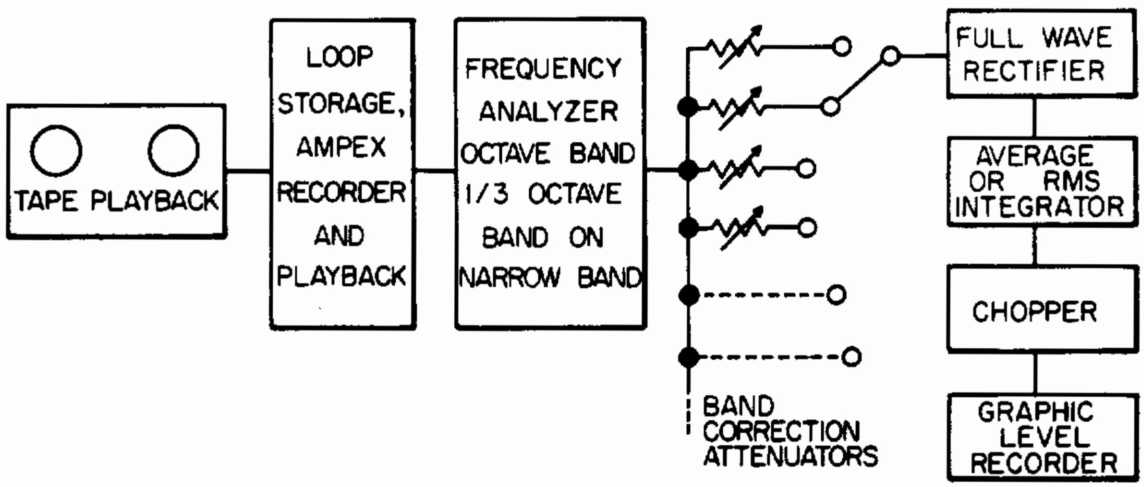
1. Data Recording System

Data recording equipment and techniques are described in References 9 and 10. The equipment used in the data recording system, outlined in Fig 2a, was to a large extent commercially available equipment which was modified for one or more of the following reasons: (a) to reduce temperature dependence of the sensitivity of the components; (b) to reduce harmonic distortion; (c) to reduce microphonics; (d) to improve frequency response and stability; (e) to increase signal-to-noise ratio.

The input to the data channel of the tape recorder was filtered, as needed, to assure an adequate signal-to-noise ratio over the entire frequency range of interest. For example, when the recorded noise sample had large low frequency components and small high frequency components, a filter that de-emphasized the low frequencies was used. The gain could then be increased enough to raise the high frequencies above the electrical background without overloading the low frequency signal.



a.) BLOCK DIAGRAM OF FIELD RECORDING SYSTEM.



b.) BLOCK DIAGRAM OF LABORATORY DATA REDUCTION SYSTEM.

FIG. 2 RECORDING AND DATA REDUCTION SYSTEMS.

The other channel of the twin-channel tape recorders was used to record pertinent information, such as the microphone number and position, the engine operating condition, attenuation settings of the recorder, time of day, etc. If, as was frequently the case, more than one recorder was used, the information channels of all recorders were connected to the same microphone.

The sources of error in the data recording system are discussed in detail in Reference 9. The major errors arise from: 1) the reciprocity calibration of the reference microphone, 2) the comparison calibration of the data microphones with the reference microphone, 3) the instability of the several components (with time and with temperature), and 4) the variations in SPL from the acoustic calibrators used in the field. Only variations about a mean value are of interest here. The absolute calibrations could be, for example, 10 db too high, with no error in the measured value of noise reduction.

The standard deviation of these errors is about 0.5 db for the system described above[9]. That is, if the same acoustic signal were recorded many times with different microphones, recorders, calibrators, etc. the distribution of the data would lie within $\pm$ 0.5 db of the mean value of all the data about 70% of the time, and within 1.0 db of the mean value about 95% of the time (assuming a Gaussian distribution of errors).

2. Data Reduction System

This system is described in detail in Reference 9. A block diagram of the data reduction system is shown in Fig 2b. The tape-recorded field data are first re-recorded on a tape loop. Each time the tape loop completes a cycle, the frequency analyzer, the attenuators, and the graphic level recorders step simultaneously. The "band correction attenuator" settings are determined from the frequency response characteristics of the microphones, recorders, filters, and components of the data reduction system. The 400 cps calibration signal provides a reference point that is used in conjunction with the band correction attenuators to obtain a plot of SPL vs. frequency.

Errors in the data reduction system are caused primarily by: 1) inaccuracies in attenuators, 2) a limited dynamic range of the integrator, and 3) the instability of the integrator between calibrations. The standard deviation of these errors is about 0.5 db for the system described above[9].

3. Frequency Analysis

The selection of an appropriate bandwidth for the measurement of various noise spectra has been discussed by many authors[11,12]. Selection of an appropriate bandwidth for the measurement of noise reduction presents different problems. The measured noise reduction of an acoustical treatment depends not only on its transfer function (i.e., noise reduction spectrum), but also on the spectrum of the noise input and the bandwidth of the frequency analyzer. If the noise spectrum and the noise reduction spectrum have constant slopes, it is possible to derive relations between the noise reduction in a frequency band, the noise reduction spectrum, the bandwidth, and the input spectrum. These relations are equally applicable to the calibration corrections that may be applied to microphones and other measurement equipment in an attempt to make their response independent of frequency. In general, the response cannot be corrected to be independent of frequency by use of single-number "band" correction factors.

For example, consider a component (a noise reduction element, a microphone, a tape recorder, etc.) whose transfer function decreases at a rate of 18 db/octave ($1/f^3$). For simplicity, and with no loss of generality, the transfer function is taken to be equal to unity at $f = 1$ and is therefore 1/8 at $f = 2$. The octave band transfer function, H, (a ratio of octave band sound pressure at the input to the octave band sound pressure at the output) can be calculated by the following expression:

$$H = \int_{f=1}^{f=2} g(f)\, 1/f^3\, df \Big/ \int_{f=1}^{f=2} g(f)\, df \qquad (20)$$

where g(f) is the spectrum level of the input.

If, for example, the input SPL increases at a rate of 18 db/octave on a spectrum level basis (21 db/octave in octave bands), then $g(f) = f^3$ and the octave band transfer function H is:

$$H = \frac{\int_1^2 df}{\int_1^2 f^3 df} = 4/15 \qquad (21)$$

or a 12 db reduction in SPL.

In the input decreases at 18 db/octave on a spectrum level basis, then $g(f) = 1/f^3$

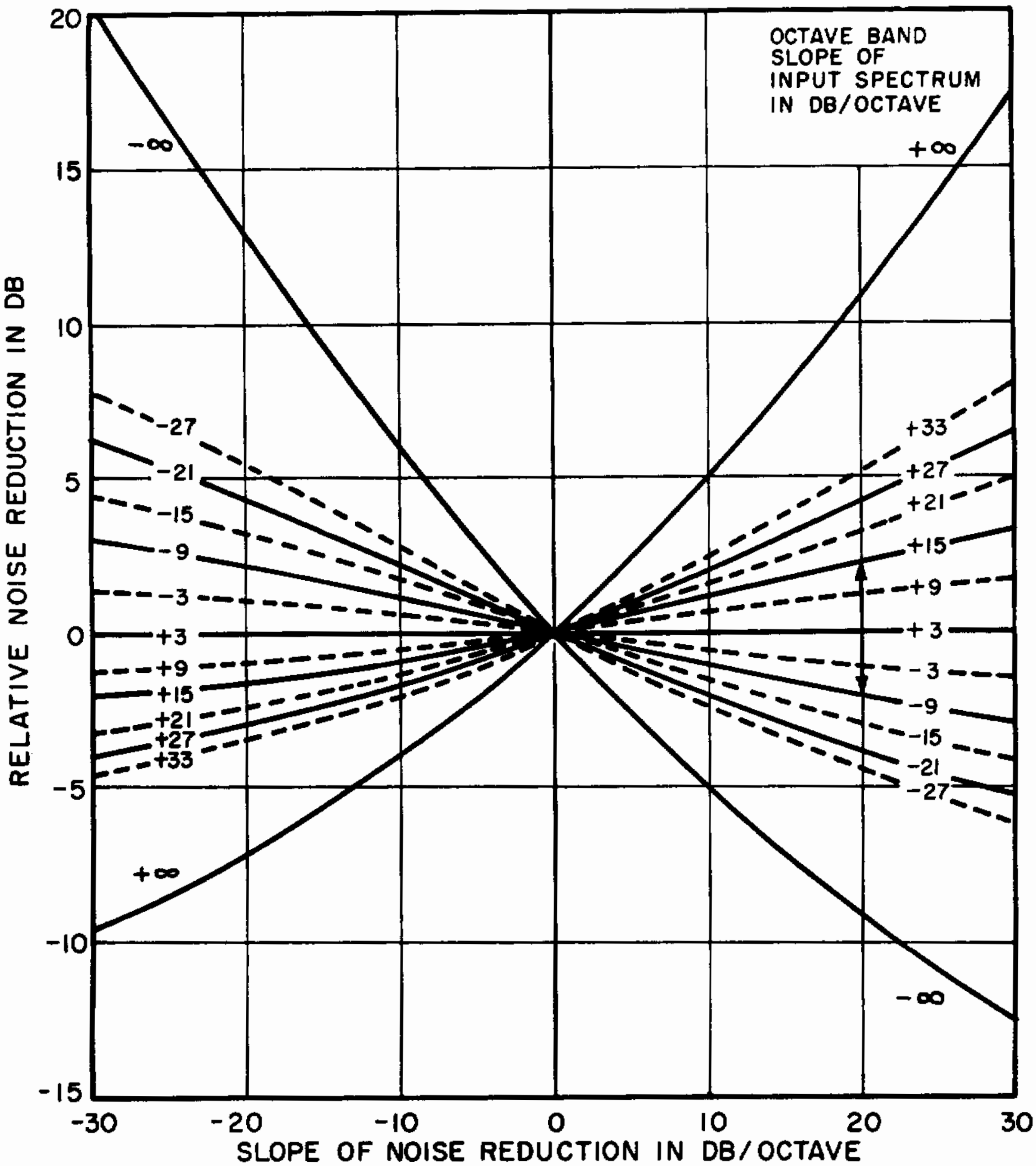


FIG. 3 RELATIVE OCTAVE BAND NOISE REDUCTION AS A FUNCTION OF THE SLOPE OF THE NOISE REDUCTION AND INPUT SPECTRA.

$$H = \frac{\int_1^2 1/f^3 \quad \cdot \quad 1/f^3 df}{\int_1^2 1/f^3 df} = 31/60 \qquad (22)$$

or a 6 db reduction in SPL.

As the slope of the input increases toward positive infinity, the noise reduction approaches the value at f_2 (18 db). As the slope decreases toward negative infinity, the noise reduction approaches the value at f_1 (0 db).

Obviously, the value of the octave band noise reduction may vary over a wide range as the input spectrum varies. Figure 3 has been derived by carrying out the calculations indicated by Eq 20, for a wide range of input and noise reduction slopes. This graph can be used to find the variation in octave band noise reductions with variations in input spectrum slope. (The reference level for the ordinate, relative noise reduction, is arbitrary and unimportant.) Certain conventions must be observed when using Fig 3. Noise spectrum slopes are given in terms of the octave band slopes, which are 3 db greater than the slopes on a spectrum level or "per-cycle" basis. Noise reduction is taken to be a positive quantity, and a noise reduction that increases with frequency is said to have a positive slope.

Figure 3 can be used to solve two problems. The first problem is to find a noise reduction for an arbitrary input spectrum from the noise reduction measured with a particular input spectrum. The second problem is to determine the octave band noise reduction for a given input spectrum when the "per-cycle" noise reduction is known. These problems can be solved if the noise reduction spectra and the input spectra have slopes that are reasonably constant over an octave band.

EXAMPLE 1:

Assume that the noise reduction of an acoustical treatment is measured with the use of an octave band filter and is found to be 18 db under the following conditions: the input spectrum slope is +15 db/octave; the slope of the noise reduction is 20 db/octave. It is required to find the octave band noise reduction of an input that has a slope of -9 db/octave.

From Fig 3, the relative noise reduction for an input slope of +15 db/octave and a noise reduction slope of 20 db/octave is +2 db. For an input spectrum with a slope of -9 db/octave, the relative noise reduction is -2 db. The difference in relative noise reduction is thus 4 db. The octave band noise reduction for a -9 db/octave input spectrum is 4 db less than that for a +15 db/octave input, or 14 db.

EXAMPLE 2:

Assume that the noise reduction of an acoustical treatment is given as a continuous function of frequency. The noise reduction is 10 db at 300 cps and increases at 25 db/octave to 35 db at 600 cps. It is required to find the noise reduction in the 300-600 cps band for a -9 db/octave input spectrum. This problem can be solved by remembering that the relative noise reduction at the lowest frequency in the octave band (300 cps) is obtained from the negative infinity curve.

The relative noise reduction for a -9 db/octave input spectrum is seen to be about 8.5 db greater than the noise reduction at 300 cps (read up from -∞ curve at A to -27 db/octave curve at B in Fig 3). Thus the noise reduction in the 300-600 cps band is (10 + 8.5) 18.5 db.* The same result could be obtained by observing that the relative attenuation at 600 cps is obtained from the positive infinity curve.

A chart similar to Fig 3 could also be derived for one-third octave band analysis. Calculations show, for example, that if the slope of the input spectrum is varied from -30 db/octave to +30 db/octave, the one-third octave band noise reduction will vary less than 1 db for any noise reduction slope in the range from +20 db/octave to -20 db/octave.

The standard deviation of errors arising from the use of a one-third octave band filter is estimated to be no more than 0.5 db for the range of input spectra and noise reduction spectra encountered in the data contained in this report.

B. Variations in Noise Source Levels

1. Jet Engine

The jet engine is a source of random noise. In order to analyze the noise data, it is assumed that the noise radiated from the engine is stationary** in time. If several measurements of jet noise are made with a short sample time,

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* Problems of the type illustrated by the second example can be more easily solved if Fig 3 is wrought in a slightly different form. See Fig 16 of Volume One of this report.

** Briefly, stationary implies that certain properties of the signal (mean value, rms value, etc.) are independent of the time the experiment is started. See Reference 13 for a discussion of this point.

a distribution of rms values is obtained. The variance of the rms values will depend on the amplitude probability function of the signal, the bandwidth, and on the length of the integration time. In general, the variance will decrease as the reciprocal of the integration time or bandwidth, and hence the standard deviation will decrease as the square root of the integration time or the bandwidth. Although the average value of the rectified sound pressure has generally been used rather than the rms value, the above considerations still apply. The variation in average values from 5-second samples (the shortest sample-time used for data presented herein) was analyzed for a recording of noise in a test section of an engine test cell[14/], with a jet engine operating at 100% of compressor revolution rate. From this recording, seventeen different samples, each 5 seconds long, were filtered in one-third octave bands of frequency and integrated.

The results of this analysis are given in Fig 4, which shows the absolute variation of the average SPL (over 5 seconds) of the seventeen samples[10/]. Fifty percent of the measured values fall within the shaded area. Ninety-five percent of the values fall below the upper solid line. If the distribution is normal, the standard deviation* averaged over all frequency bands is about 0.75 db (including the variations due to instabilities, drift, etc., of the entire data recording and reduction system). Since the standard deviation of errors in the data recording and reduction systems is of the order of 0.7 db, the standard deviation for the jet engine noise is negligible.

* The standard deviations, σ, reported here are the square root of the "best estimate of the variance, σ^2, of the population", which is slightly smaller than the best estimate of the standard deviation of the population. The resulting error is small (< 8%) for all sample sizes used in this section; see Reference 15.

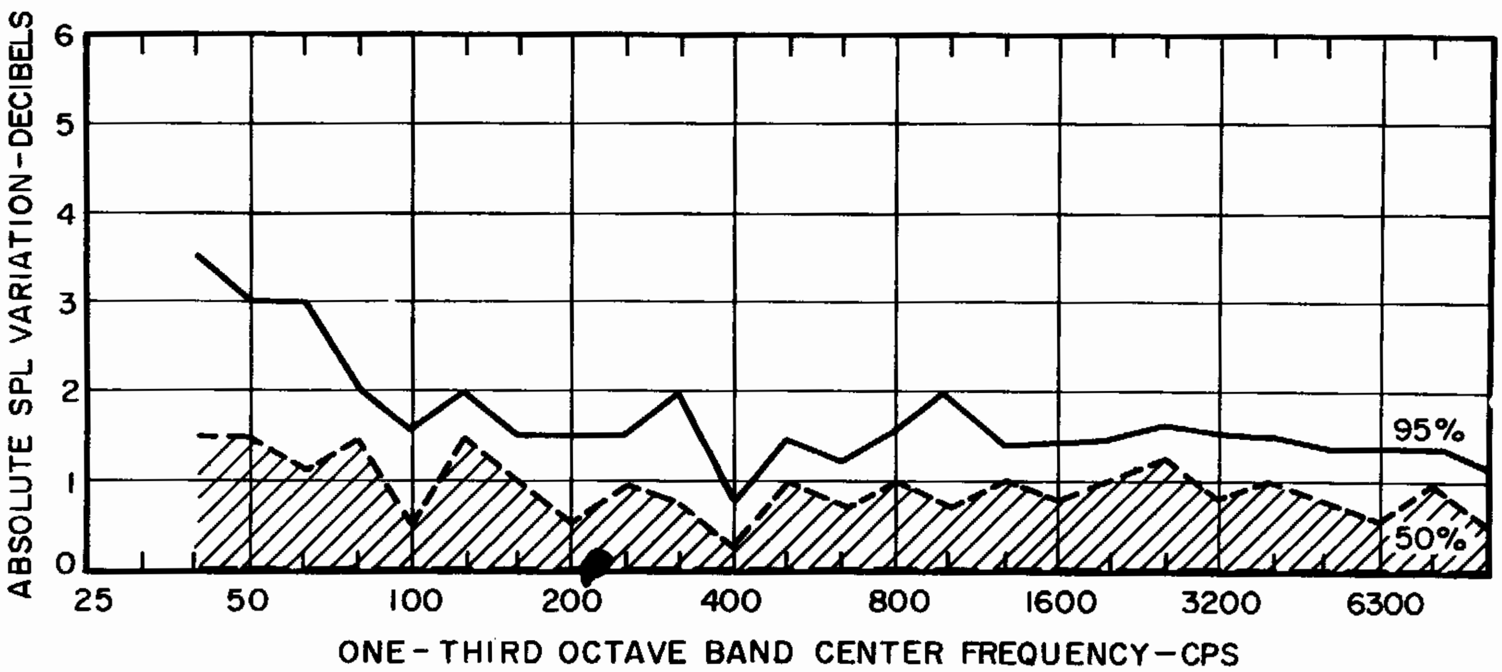


FIG. 4 THE ABSOLUTE VARIATION OF SOUND PRESSURE LEVEL AVERAGED OVER 5 SECONDS AS A FUNCTION OF FREQUENCY. SEVENTEEN DIFFERENT SAMPLES WERE TAKEN FROM A LONG-TIME RECORDING MEASURED IN THE TEST SECTION OF A CELL WITH THE ENGINE OPERATING AT 100 % OF MAXIMUM COMPRESSOR REVOLUTION RATE.

2. Explosive Noise Source

The explosive noise source (XNS) that was used to obtain most of the data contained in this report is a small cannon which fires blank 10-gauge shotgun shells[16]. Since the shells are not identical in composition, the SPL's are not identical each time a shot is fired. The absolute deviation from the mean value of sound pressure level measured for 18 shots is given in Fig 5[10]. The SPL's were all measured at a fixed position relative to the cannon in the test section of a jet engine test cell. The spread of data is larger than that for the 5-second samples of jet engines. If the distribution is normal, the standard deviation of the SPL distribution, averaged over frequency, is about 1 db*. Since the standard deviation of the errors in the measurement and data reduction systems is 0.7 db, the error introduced by the variation in the average values of SPL is about the same order of magnitude as the error from the data reduction system.

If "n" shots are averaged together, the standard deviation of the distribution of mean SPL for "n" shots will vary as $1/\sqrt{n}$. For example, if 4 shots (a typical number for the data used in this report) are averaged together, the standard deviation will be about 0.5 db. Stated in a more useful manner, this implies that the average value of 6 shots will be within 0.5 db of the mean value of a very large number of shots about 80% of the time, and within 1 db more than 95% of the time.

*The standard deviation at 60 cps is significantly higher. Sixty cycle signals, like the poor, are with us always.

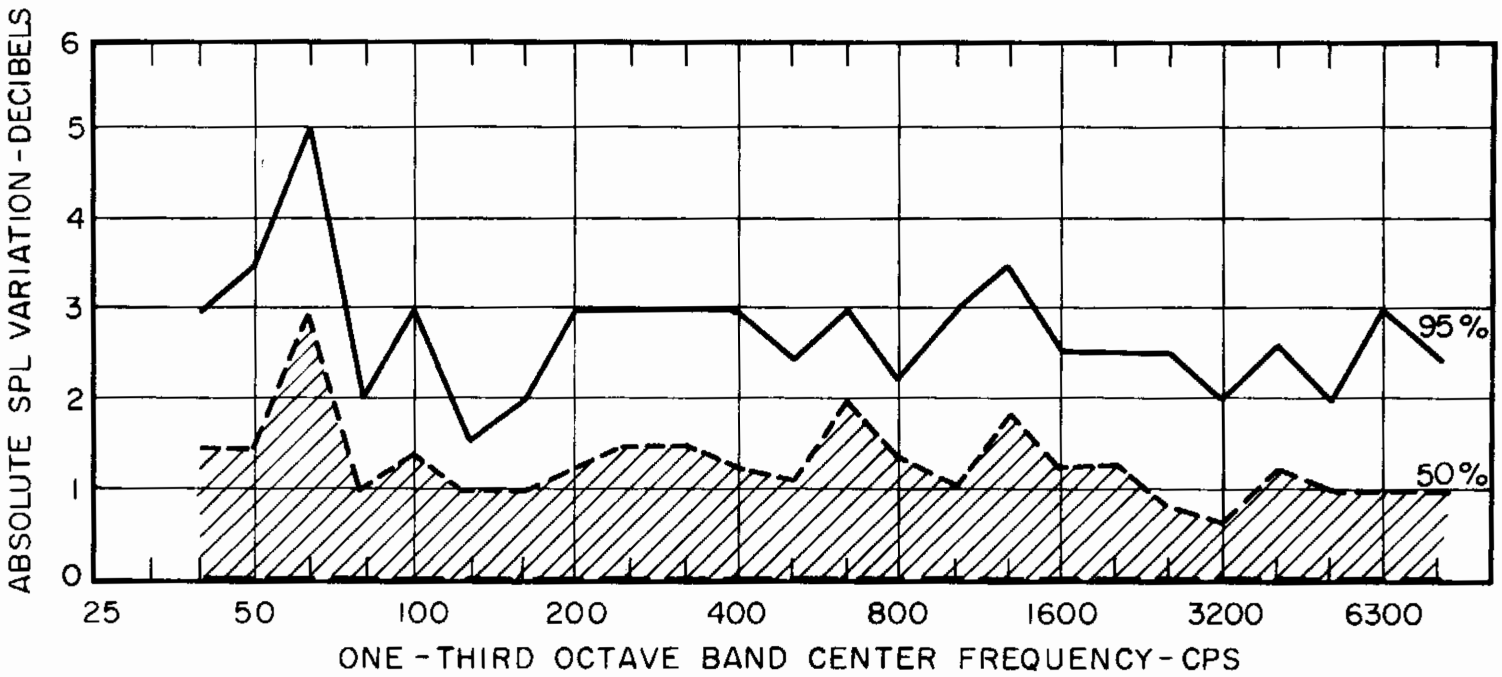


FIG. 5 ABSOLUTE VARIATION OF THE SPL INTEGRATED OVER 150 MILLISECONDS FOR 18 SAMPLES OF EXPLOSIVE NOISE SOURCE MEASUREMENTS.

C. Variations in Noise Levels in Space

1. The Distribution of SPL in Space

A measurement of SPL at a single microphone position in the input or output plane of an acoustical treatment will not serve to define uniquely the average SPL at the input or the output. However, a distribution of SPL's can be obtained by using several microphone positions in a grid at the face of an acoustical treatment. From this distribution, the space-average value of SPL and the standard deviation of SPL's around the space-average can be determined. The standard deviation can then be used to ascertain how many microphone positions will be required to obtain an average SPL that will be within X db of the "true"* space average, Y% of the time. To determine the variation of SPL around the average, it is assumed that there are no interaction effects between the variations in space, the variations in source levels, and the variations introduced by the data recording and reduction systems. If the distribution is normal, then the total variance, σ^2_{total} is:

$$\sigma^2_{total} = \sigma^2_{space} + \sigma^2_{source} + \sigma^2_{s} \qquad (23)$$

where: σ^2_{space} is the variance due to spatial variations of SPL,

σ^2_{source} is the variance due to the noise source, and

σ^2_{s} is the variance due to the measurement and data reduction system.

It will be shown below that σ^2_{total} is much greater than $(\sigma^2_{s} + \sigma^2_{source})$ and therefore σ^2_{total} is approximately equal to σ^2_{space}.

*The "true" space average refers to the value which would be obtained using very long samples obtained from a very large number of measurement positions.

FIGURE 6

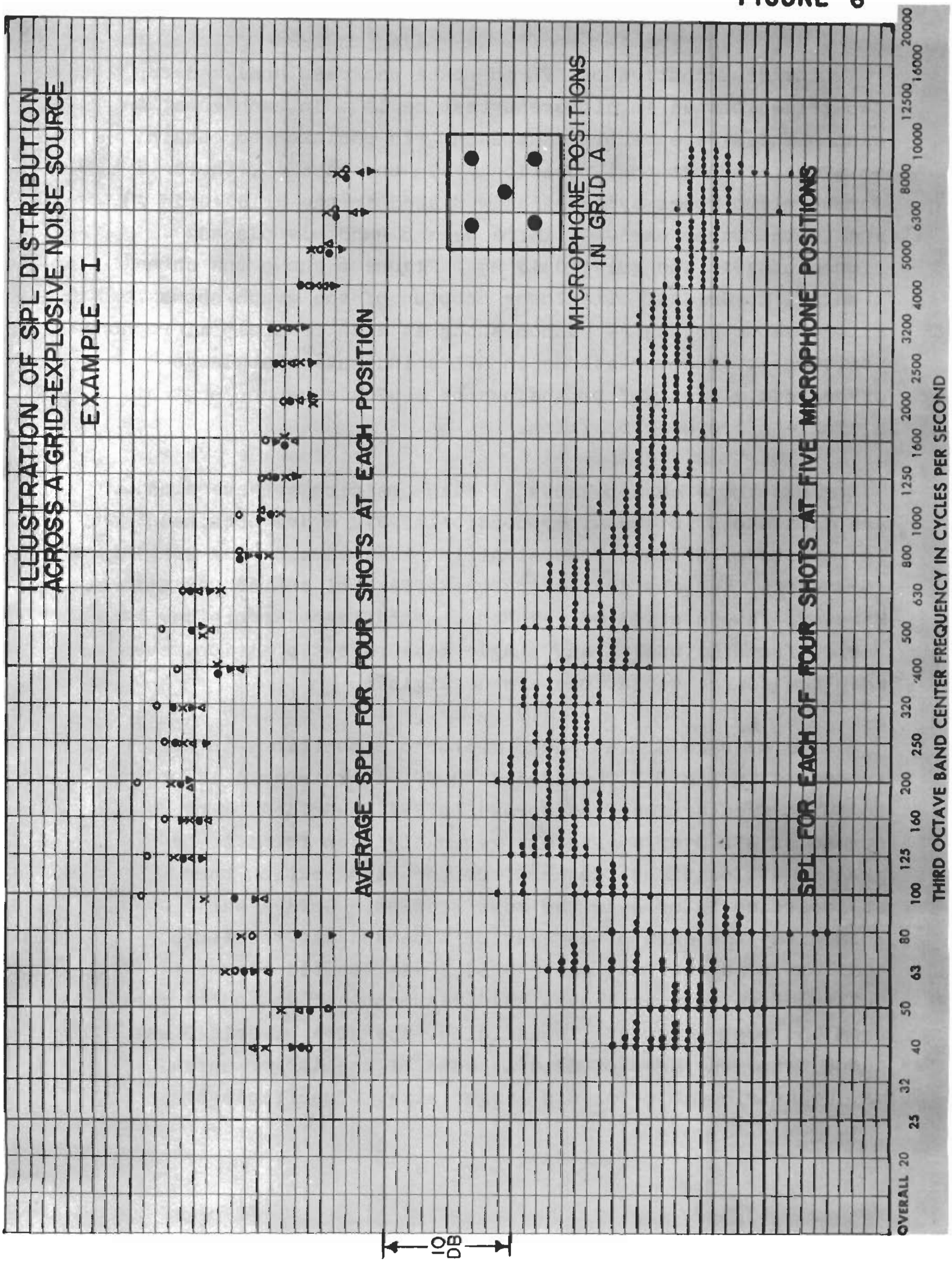


ILLUSTRATION OF SPL DISTRIBUTION
ACROSS A GRID-EXPLOSIVE NOISE SOURCE
EXAMPLE I
AVERAGE SPL FOR FOUR SHOTS AT EACH POSITION
MICROPHONE POSITIONS
IN GRID A
SPL FOR EACH OF FOUR SHOTS AT FIVE MICROPHONE POSITIONS
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
10 DB

The distribution of SPL in space has been analyzed for two grids located in different test cells. In each case, 4 cannon shots were recorded at each grid position to obtain an average SPL at that position. In one case[17], there were 5 microphones symmetrically placed in the grid (Grid A), and in the other case[18], there were 6 randomly placed microphones in the grid (Grid B). Figure 6 shows the data obtained at Grid A. The lower portion of the graph shows all 20 datum points (4 cannon shots at each of the 5 microphone positions). The variations in SPL's indicate that large errors could result from the use of a single cannon shot at a single microphone position.

When the 4 cannon shots at each position are averaged, the data shown in the upper portion of Fig 6 are obtained. The spread of SPL has been significantly reduced by averaging four shots, thereby decreasing the effect of source variations. However, it can still be seen that significant errors may result from the use of any single measurement position, even when the source variations are negligible.

The average SPL at the 6 microphone positions in Grid B are shown in Fig 7. The spread in SPL at this grid is somewhat greater than the spread in SPL at Grid A. The spread of SPL over a grid has generally been found to be greater at locations farther from the test section than at locations in or near the test section. Grid A was located at the exit of an eductor tube and was therefore near the test section, while Grid B was located near the outlet of an exhaust acoustical treatment and was quite far from the test section. In addition, the area of Grid B was about twice the area of Grid A; some of the larger spread may be attributable to this larger size. Insufficient

FIGURE 7

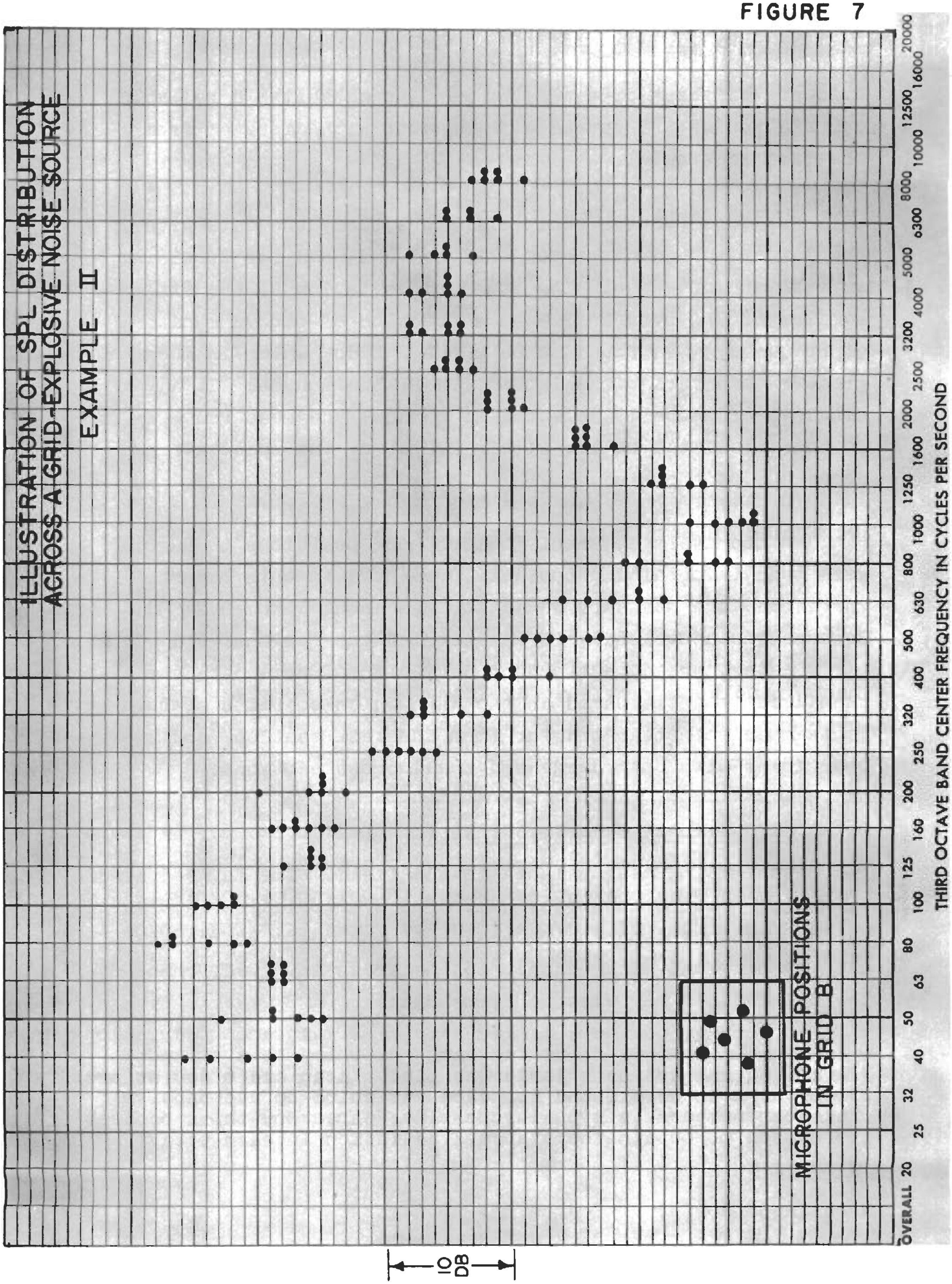


ILLUSTRATION OF SPL DISTRIBUTION
ACROSS A GRID-EXPLOSIVE NOISE SOURCE
EXAMPLE II
MICROPHONE POSITIONS
IN GRID B
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
10 DB

evidence has been obtained, as yet, to establish a reliable correlation of SPL spread with area or with distance from the test section.

For conservative engineering practice, the data with the larger spread have been analyzed and used for obtaining an estimate of error. The standard deviation of the six samples of SPL over Grid B as a function of octave bands of frequency* was found to be about 2.2 db, which is much greater than variations introduced by the source. If a single microphone is selected at random, the measured SPL is thus predicted to lie within 2.2 db of the space average about 70% of the time.

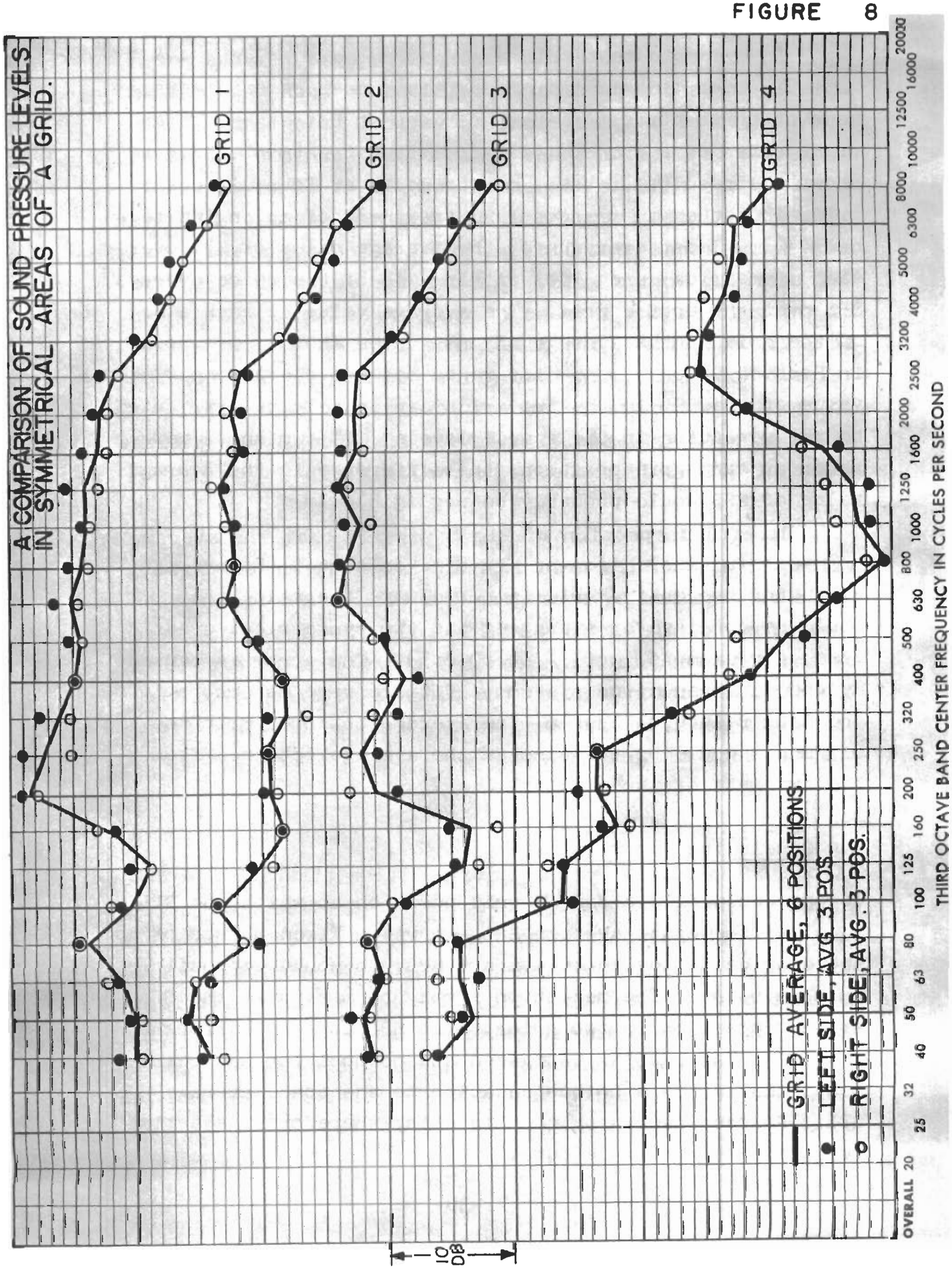
On the average, about 5 microphone positions in a grid were used in obtaining the data presented in this report. Therefore, the standard deviation was about 1 db ($2.2/\sqrt{5}$). Because the above analysis was carried out for the grid with the greater spread, a standard deviation of somewhat less than 1 db is anticipated in the data which contains measurements made both near and far from the test section, and over grids with both large and small cross sections.

2. Application of Symmetry Condition

When making measurements in the field, time may be saved by making measurements over half of a symmetrical area, rather than over the entire area. If the noise source is also symmetrical with respect to the areas involved, then the noise field may also be symmetrical.

*In an attempt to use a larger sample, a mean and 6 deviations from the mean were found for each one-third octave band. Eighteen numbers (six positions times three one-third octaves) were used to obtain the standard deviation in each octave band.

FIGURE 8


A COMPARISON OF SOUND PRESSURE LEVELS
IN SYMMETRICAL AREAS OF A GRID.
GRID 1
GRID 2
GRID 3
GRID 4
— GRID AVERAGE, 6 POSITIONS
● LEFT SIDE, AVG. 3 POS.
○ RIGHT SIDE, AVG. 3 POS.
10 DB
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND

In order to test this assumption, measurements were made at three corresponding positions on both the left and right sides of a symmetrical acoustical treatment. Figure 8 shows the results of these measurements at four different grids in the exhaust acoustical treatment of an engine test cell[19]. Almost everywhere, the average value of SPL over each symmetrical area lies within 1 db of the space average SPL over the entire grid. The average value at each side of the grid is the average of only three shots. As shown in Section B above, the difference between the average value of the SPL in the two grids can largely be attributed to the variation in the source levels. If two shots were taken at each grid position, the difference in the average value of SPL would probably be negligible.

Careful inspection of Fig 8 reveals that the average value for the entire grid is not always the value that would be obtained by averaging the two symmetrical areas (see, for example, grid 4 at 80 cps). The error is caused by the data reduction system. If there were no errors in the data reduction system, the average obtained from the six shots would be the same as the average obtained from the two sets of three shots.

3. Concluding Remarks

If one could find a single grid position at which the SPL equalled the space-average SPL, or was a fixed number of db above or below the space-average SPL, it would be possible to obtain the noise reduction simply by measuring the SPL at that position in two grids. The SPL at many different positions has been investigated and no position has been found that bears a unique relation to the space-average SPL. A space averaging technique is, therefore, essential to obtain reproducible data.

D. Total Error from Measurement System, Variations of Source Levels, and Variations of Noise Level in Space

1. Calculation of Total Error

The total variance of the distribution of all possible values of the space-average SPL in a grid is the sum of the variances of the several sources of error. The total variance, σ^2_{total}, is:

$$\sigma^2_{total} = \sigma^2_1 + \sigma^2_2 + \sigma^2_3 + \sigma^2_4 + \sigma^2_5 \qquad (24)$$

in which: σ^2_1 is the variance caused by the recording system, 0.25 db.

σ^2_2 is the variance caused by the data reduction system, 0.25 db.

σ^2_3 is the variance caused by 1/3 octave frequency analysis, 0.25 db.

σ^2_4 is the variance caused by the spatial distribution of SPL in a grid, 1.21 db, for four microphone positions.

σ^2_5 is the variance caused by the variation of source levels, approximately zero db.

For a measurement of space average SPL with an engine as a source, the total standard deviation, σ_{total}, will be about 1.5 db if 5-second samples are taken at each of four microphone positions. In the several surveys, longer samples at more positions were generally used, but the above estimate of σ_{total} will not be affected, since the variation in source level is negligible. If the explosive source is used and one shot is recorded at each of four microphone positions, the total standard deviation will also be about 1.5 db.

The standard deviations given above apply to the distributions of average SPL's in a grid. Noise reductions are obtained by subtracting the average SPL at the output grid from the average SPL at the input grid. The variance of the noise reduction is the sum of the variance of the average SPL in each grid. The variance of the output and input grids is assumed to be the same, so that the standard deviation is just $\sqrt{2}$ times the standard deviation of average SPL in a grid. The standard deviation of noise reduction values is, therefore, about 2.0 db.

In summary, if a noise reduction measurement is made using either a five-second sample of engine noise or one cannon shot, at each of four microphone positions in both the input and output grids, the value of noise reduction obtained will be within about 1.5 db of the true mean value about 70% of the time, and within 3 db over 95% of the time. The true mean value is the average value of noise reduction that would be obtained from a very large number of measurements at many different microphone positions.

2. Interpretation of Differences between Measured Noise Reduction Values

Suppose that the noise reduction, L_{nr}, of two identical acoustical treatments is measured in two identical test cells, a and b. The following steps are repeated many times: 1) The noise reduction in Cell a, L_{nra}, is measured; 2) the noise reduction in Cell b, L_{nrb}, is measured; and 3) L_{nra} is subtracted from L_{nrb}. The average value of $(L_{nra} - L_{nrb})$ will be zero, but the standard deviation of $(L_{nra} - L_{nrb})$ will be found to be $\sqrt{2}$ times the standard deviation of the L_{nr}'s[15]. Since the standard deviation for L_{nr} is about 2 db, the standard deviation of $(L_{nra} - L_{nrb})$ is almost 3 db if measured under conditions outlined in the previous paragraph. If the two acoustical treatments in

Cell a and Cell b are not identical, the mean value of $(L_{nra} - L_{nrb})$ is not zero, but the standard deviation is still 3 db.

If the standard deviation of the difference between two noise reduction measurements is greater than 3.0 db, then it must be concluded that another source of variation or randomness has entered one or both of the noise reduction measurements. In the following sections, the mean values and standard deviations of differences in noise reductions measured with different experimental techniques are investigated to determine the influence of various experimental techniques on the measured value of noise reduction.

E. The Influence of Air Flow on Noise Reduction

The noise reduction of acoustical treatments varies with the velocity of air flow. This variation has been experimentally investigated recently by Meyer, et. al[20]. In the evaluation program, it has not usually been possible to obtain measurements of air velocity through acoustical treatments. The variation of noise reduction with air flow has been investigated by an indirect method.

A change in noise reduction with air flow has been obtained by measurements during engine operation near idle condition and at military power. Near idle condition, the range of velocity in the intake treatments was estimated to be about 15 to 25 ft per second. At military power, the range in air velocity in the various test cells was about 40 to 60 ft per second. A comparison of these data, which is presented below, provides one measure of the effect of air flow on noise reduction.

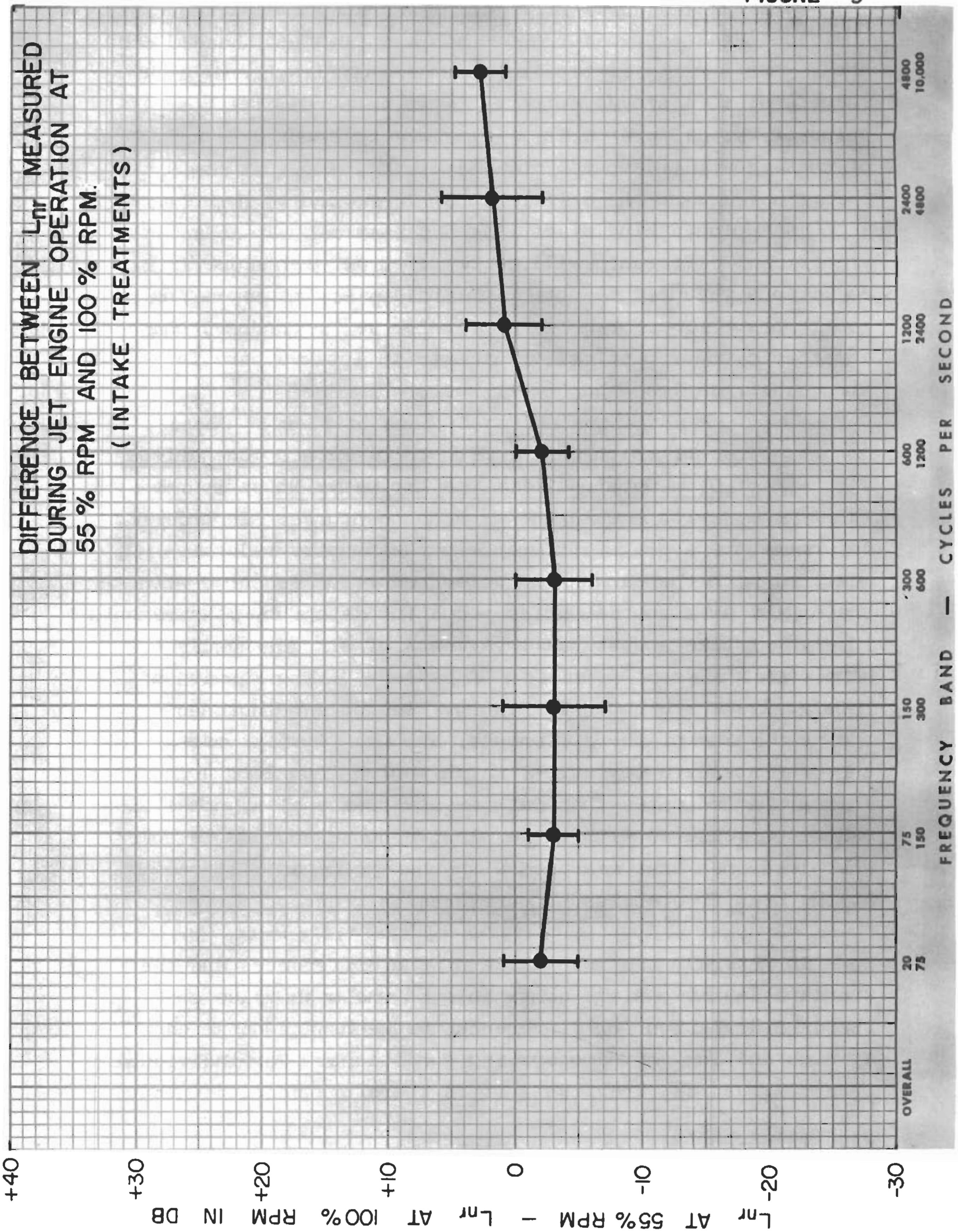
Another measure of the effect of air flow on noise reduction has been obtained by comparing measurements made at military power with measurements made with the explosive source. It is obvious that differences between the noise reductions obtained using the explosive source and those obtained during engine operation could be caused by factors other than air flow. It is initially assumed that the effects of other factors can be neglected and that the only difference between measurements with the explosive source and the engine is the change in air velocity.

1. Effects of Flow in Intake Treatments

a. Investigation of the Effects of Flow by Variation in Engine Speed. Four sets of noise reduction data were obtained using an engine at 55%, and at 100% of maximum compressor revolution rate[19, 21, 22, 23]. The data were all measured in intake acoustical treatments. It is estimated that the air velocities at 100% rpm were less than 60 ft/sec and those at 55% rpm less than 25 ft/sec.

In Fig 9, the average value of the difference and the standard deviation of the difference is given as a function of octave bands of frequency. As can be seen, the L_{nr} at 55% rpm is less than the L_{nr} at 100% rpm in the frequency range from 20 to 1200 cps and greater than the L_{nr} at 100% rpm in the frequency range from 1200 to 10,000 cps. The standard deviation varies from 2 to 4 db. The value of the standard deviation averaged over the eight octave bands is about 3 db. The small difference in the mean values indicates that the effects of flow are small at least over the range of velocities encountered.

FIGURE 9

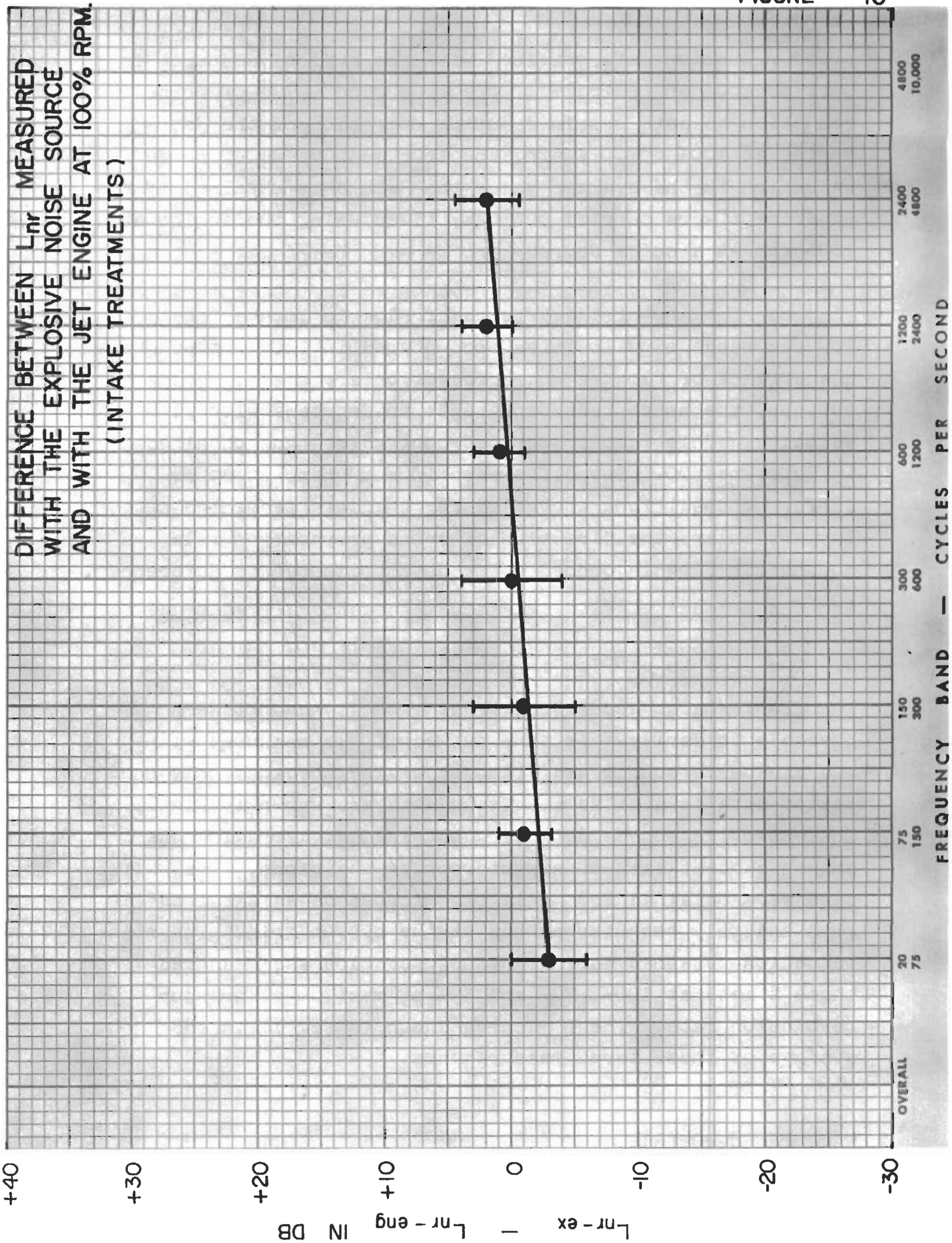

DIFFERENCE BETWEEN L_{nr} MEASURED
DURING JET ENGINE OPERATION AT
55 % RPM AND 100 % RPM.
(INTAKE TREATMENTS)
FREQUENCY BAND — CYCLES PER SECOND
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
+40
+30
+20
+10
0
-10
-20
-30
L_{nr} AT 55% RPM — L_{nr} AT 100% RPM IN DB

The intakes used for these measurements each contained about 3 or 4 noise control components (baffles, bends, etc). Therefore, if a single component were measured, the mean value of the difference in L_{nr}'s might be about 1/4 or 1/3 of that shown, or about 1 db. Since the sample is small, the mean value of the difference in L_{nr}'s can be neglected when measuring a single acoustical treatment. That is, no significant difference will be obtained between noise reductions measured at both high and low engine operating conditions. (The foregoing conclusion is, of course, only applicable for acoustical treatments in intakes, and for air velocities less than 60 ft/sec.)

b. Investigation of the Effects of Flow by Comparison of Data Obtained with the Explosive Noise Source and with the Engine as a Noise Source. In order to determine the average difference between noise reduction measured with the cannon and noise reduction measured with the engine as a source, noise reduction data from measurements in eight intake acoustical treatments of six test cells have been analyzed[14, 17, 19, 21, 22, 23]. The results of this analysis are shown in Fig 10. The bold points show the mean value of the difference, and the vertical bars show the standard deviation, which varied from 2 to 4 db. This graph indicates that, at the low frequencies, noise reduction measured with the explosive noise sources is somewhat less than that measured with the engine as a source. At the high frequencies, the reverse is true.

In general, the mean value of the differences and the standard deviations are comparable to the mean value and standard deviation for differences between data obtained at high and low engine operating conditions. For individual intake acoustical treatments it is therefore concluded that

FIGURE 10


DIFFERENCE BETWEEN Lnr MEASURED
WITH THE EXPLOSIVE NOISE SOURCE
AND WITH THE JET ENGINE AT 100% RPM.
(INTAKE TREATMENTS)
FREQUENCY BAND — CYCLES PER SECOND
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
+40
+30
+20
+10
0
-10
-20
-30
Lnr-ex — Lnr-eng IN DB

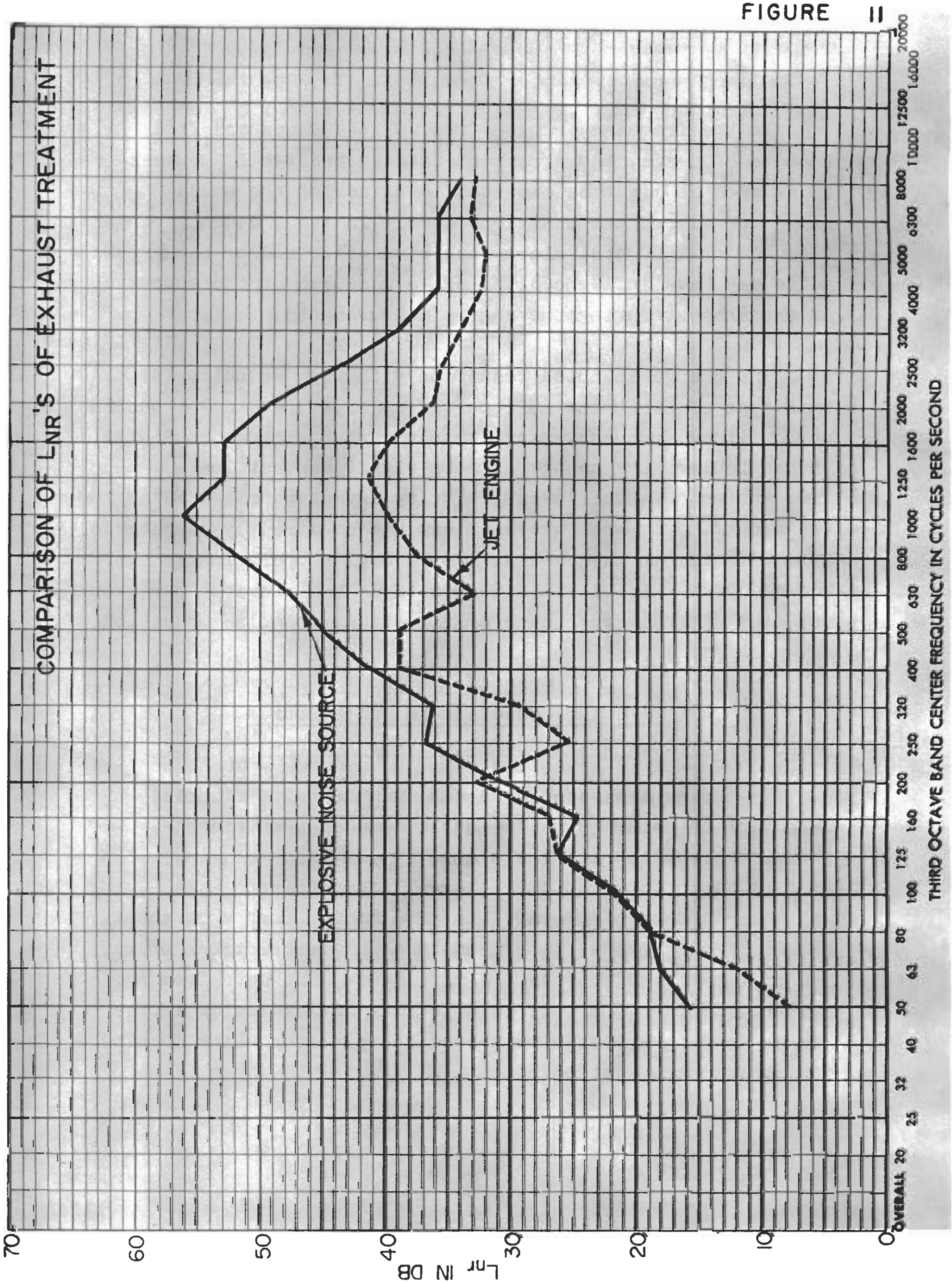
noise reduction data obtained with the explosive source is just as good an estimate of the true mean value of noise reduction as data obtained during operation of the jet engine at 100% rpm. Again, the true mean value of noise reduction is the average value that would be obtained if the experiment were carried out many times using the engine at 100% rpm as a noise source.

c. Summary of the Effects of Flow in Intake Treatments. For the range of velocities encountered in intake treatments, the effects of flow on noise reduction are small. Below 600 cps, the noise reduction tends to decrease with flow velocity. This result does not contradict that obtained in Reference 20. The difference is not unexpected, as different conditions prevailed for the data presented here. In particular, the direction of sound propagation relative to the air flow is opposite the data presented in Reference 20.

2. The Effects of Flow in Exhaust Acoustical Treatments

For exhaust acoustical treatments it has not been possible to derive relations between L_{nr}'s measured with the explosive source and with jet engine, because only one set of data is available. These data are shown in Fig 11. The explosive source data were obtained by averaging one cannon shot at each of six microphone positions. The engine data were obtained from only one microphone at the input grid and only two microphones at the output grid. The noise reduction obtained during engine operation was measured three times, at 100%, 70% and 55% of maximum compressor revolution rate. The L_{nr} curve shown in Fig 11 is the average of these three measurements.

FIGURE 11


COMPARISON OF LNR'S OF EXHAUST TREATMENT
EXPLOSIVE NOISE SOURCE
JET ENGINE
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
Lnr IN DB
0 10 20 30 40 50 60 70

In the frequency ranges from 20 to 400 cps and from 2500 to 8000 cps, the agreement between the two L_{nr} curves is as good as might be expected, considering the small number of microphone positions used. In the frequency range from 400 to 2500 cps, microphone wind noise and/or background noise may have influenced the SPL's measured at the output of the exhaust during the engine measurements.

The change in noise reduction with flow velocity for this type of treatment as reported by Meyer, et. al[20] occurs over a wider frequency range than the change shown in Fig 11. In particular, the change is more significant at low frequencies. As the lowest velocity investigated by Meyer was about 100 ft/sec, the data may not be directly comparable. (The velocity in the exhaust during engine operation is not known.) The possibilities suggested in the previous paragraph are more probably the cause of the change in noise reduction shown in Fig 11.

F. The Influence of the Noise Source on Noise Reduction Measurements

In the general case, one expects the value of noise reduction to depend upon the noise source. For example, the noise reduction of the simple system described in Section II would be quite different if the piston were characterized by a constant pressure rather than a constant velocity. In this section, some general restrictions on the use of substitute sources are presented. The data from the previous section are used to show that the explosive source can be used to approximate the noise reduction which obtains during engine operation.

1. General Limitations on the Use of a Substitute Source

In order that the measured noise reduction be independent of the source, the substitute noise source must generate a noise field similar in certain respects to the noise field of the jet engine. For example, the distribution of sound pressure in space must be approximately the same for both sources. The noise sources must therefore be located in the same position in the test cell. Because the source of noise from a jet engine is distributed in space, there may be no single appropriate position at which to locate the explosive noise source. However, for intake acoustical treatments, the jet engine noise source can usually be considered to be located at the upstream opening of the eductor tube. For this reason, most measurements were made with the explosive noise source positioned near the eductor tube opening.

Measurements of noise reduction of the exhaust acoustical treatments were also made with the cannon located near the eductor tube. If, however, the exhaust acoustical treatments are located near the jet engine, the effective location of the low frequency jet engine noise may lie within the acoustical treatment. In such a case, no SPL difference measurement of acoustical effectiveness will provide a useful indication of acoustical effectiveness of the exhaust treatment. Only an insertion loss measurement would be the useful way to measure acoustical effectiveness.

Certain noise reduction elements, such as exhaust diffusers, reduce noise levels by modifying the acoustic power radiated from the engine. Obviously, no artificial source can be used to measure these effects.

2. Special Limitations on the Use of Substitute Sources in Acoustical Treatments with Multiple Inputs

Certain acoustical treatments have multiple inputs to a common acoustical treatment. Consider, for example, the test cell shown in Fig 12. The primary and secondary air enter the air inlet and pass through a lined bend, a lined duct, and another lined bend. The primary or combustion air then enters the test section, but the secondary or cooling air passes around two more bends and through a lined duct before entering the test section.

Using the definition given in Section II, the L_{nr} of the total intake system is:

$$L_{nr} = PWL_{in} - PWL_{out} \qquad (25)$$

where PWL_{in} is the power level of the input and PWL_{out} is the power level at the output, as determined from Eq (19). PWL_{in} is the sum of the PWL at the primary air inlet, PWL_{pri}, and the PWL at the secondary air inlet, PWL_{sec} and is found by:

$$PWL_{in} = 10 \log_{10} [\text{antilog} \frac{PWL_{pri}}{10} + \text{antilog} \frac{PWL_{sec}}{10}] \qquad (26)$$

If the noise reduction for the primary air path is 10 db and for the secondary air inlet is 30 db, the PWL at the output is given by:

$$PWL_{out} = 10 \log_{10} [\text{antilog} \frac{PWL_{pri} - 10 \text{ db}}{10} + \text{antilog} \frac{PWL_{sec} - 30 \text{ db}}{10}] \qquad (27)$$

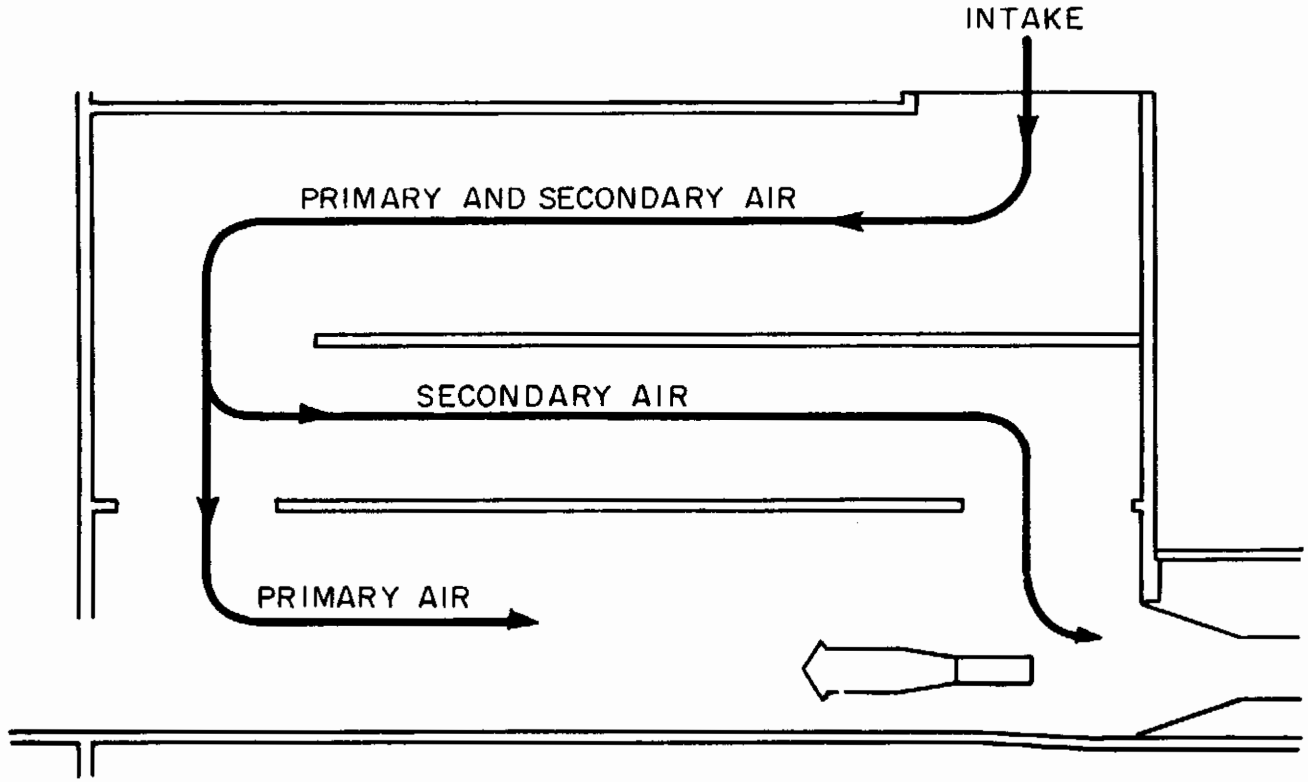


FIG. 12 ILLUSTRATION OF AN ACOUSTICAL TREATMENT WITH TWO "INPUTS."

In the simple case of a zero sound pressure at the primary air inlet, the L_{nr} is 30 db. In the case of zero sound pressure at the secondary air inlet, the L_{nr} is 10 db. In the case of finite sound pressures at both inputs, the value of L_{nr} will be greater than 10 db and less than 30 db. This can best be seen by substituting various combinations of power levels at the primary and secondary air inlets and carrying out the operations indicated in Eq 27. The table below shows, for typical input power levels, the resulting output power level and the total L_{nr} of the system.

	PWL_{pri}	PWL_{sec}	Total PWL_{in}	PWL_{out}	L_{nr}
1.	100	80	100	90 + 50 = 90	10
2.	100	90	100.5	90 + 60 = 90	10.5
3.	100	100	103	90 + 70 = 90	13
4.	100	110	110.5	90 + 80 = 90.5	20
5.	100	120	120	90 + 90 = 93	27
6.	100	130	130	90 + 100 = 100.5	29.5
7.	100	140	140	90 + 110 = 110	30

The L_{nr} is near its lower limit (10 db) only when PWL_{sec} is less than or about equal to PWL_{pri}. As the value of PWL_{sec} exceeds PWL_{pri}, the L_{nr} becomes greater than its lower limit (10 db). When the contribution at the output of the secondary air path is equal to or exceeds the contribution of the primary air path, the L_{nr} approaches its upper limit.

It is obvious from this simple example that the noise reduction of this intake system will depend not only on the noise reduction of the two air paths, but also on the relative magnitudes of the input power levels. Therefore, a substitute noise source can be used only if it can be demonstrated that relative values of the SPL's at the inputs will be the same for the substitute noise source as for the engine involved. Experience has shown that the above condition will not usually occur.

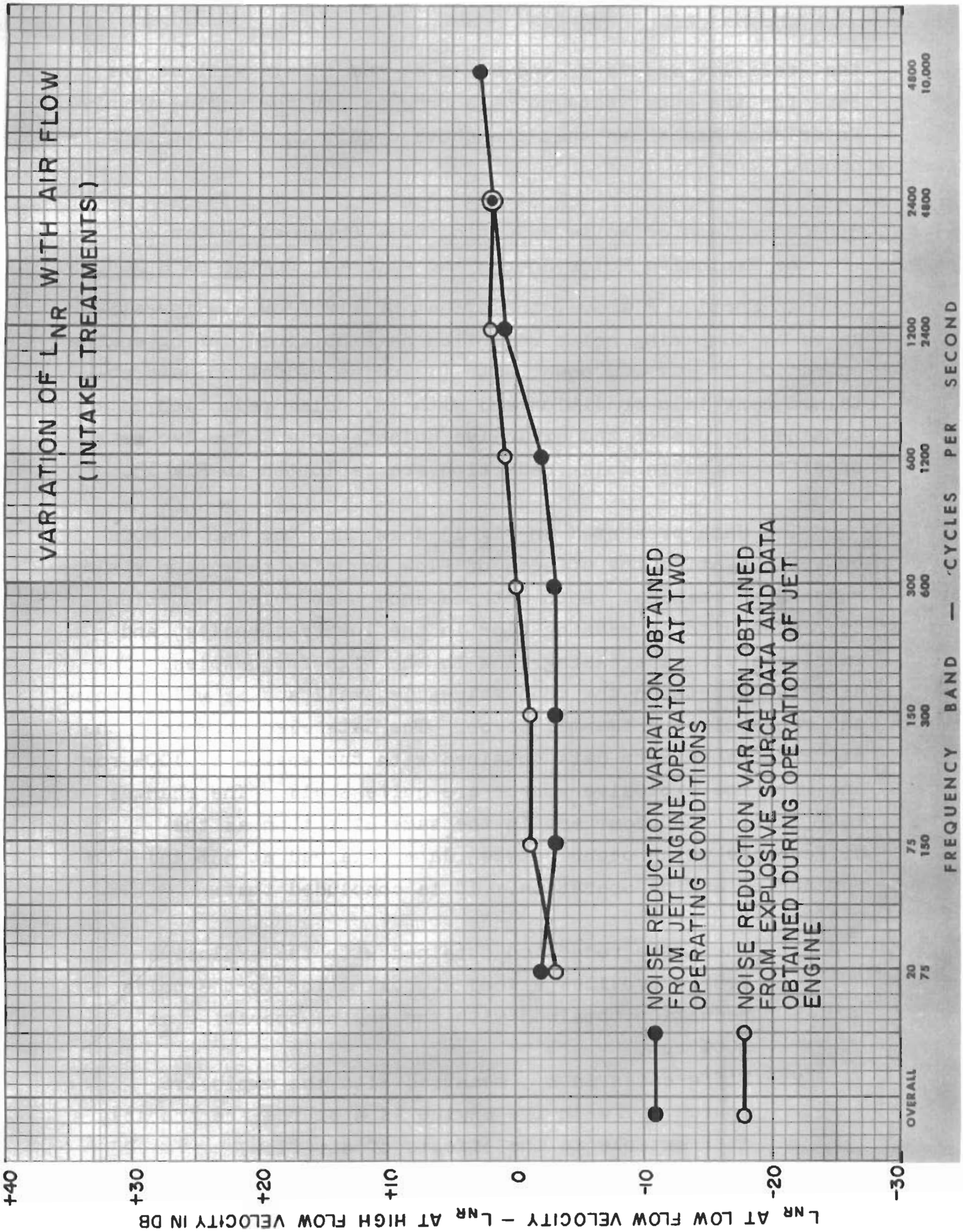
3. Comparison of Noise Reduction Data Obtained by Use of the Explosive Noise Source and with the Engine as a Source

In Section F above, the effect of air flow was investigated by comparing noise reduction data obtained during engine operation at maximum power with (1) noise reduction data obtained near idle condition and with (2) noise reduction data obtained by use of the explosive noise source. The two sets of differences in noise reduction are replotted in Fig 13.

The data presented in Fig 13 show that the two methods of investigating the effects of flow on noise reduction yield comparable results. Both methods show that noise reduction decreases with increasing flow in the low frequencies, and increases with increasing flow in the high frequencies. The difference between the mean values is small and is well within the range of experimental error. The data also show that the differences between data obtained with the engine at 100% rpm and with the explosive source are quite small. The changes in L_{nr} which do occur are in the same direction as the differences which occur between data obtained at 55% and 100% rpm during engine operation. It is concluded that:

1. The noise reduction obtained with the explosive source provides a good approximation to the noise reduction obtained with the engine operating at 100% rpm.
2. Differences in L_{nr} measured with the explosive source and with the engine as a source are probably attributable to the effects of air flow.

FIGURE 13


VARIATION OF LNR WITH AIR FLOW
(INTAKE TREATMENTS)
NOISE REDUCTION VARIATION OBTAINED FROM JET ENGINE OPERATION AT TWO OPERATING CONDITIONS
NOISE REDUCTION VARIATION OBTAINED FROM EXPLOSIVE SOURCE DATA AND DATA OBTAINED DURING OPERATION OF JET ENGINE
+40
+30
+20
+10
0
-10
-20
-30
LNR AT LOW FLOW VELOCITY — LNR AT HIGH FLOW VELOCITY IN DB
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
FREQUENCY BAND — CYCLES PER SECOND

G. The Influence of Measurement Procedures

In this section the noise reduction obtained by the L_{nr} method is compared with the noise reduction obtained by other measurement procedures.

1. Comparison of EN-1 Difference with L_{nr}

The EN-1 difference methods of evaluating acoustical treatments are presented in an Aircraft Industries Association report, "Uniform Practices for the Measurement of Aircraft Noise".[24] The EN-1 evaluations are presented for many configurations of engine test cells and ground run-up suppressors. The EN-1 evaluation prescribed for the exhaust acoustical treatment of jet engine test cells has been used frequently as a measure of acoustical effectiveness of the noise control components in engine test cells, and on occasion to describe the general acoustical effectiveness of test cells. The other EN-1 differences do not appear to be widely used.

In this section, the EN-1 difference method of evaluating exhaust acoustical treatments is reviewed in the light of data obtained from many evaluations of the acoustical performance of noise control components in jet engine test cells. It is found that:

1. EN-1 differences may either increase or decrease with engine operating condition.

2. With a fixed engine operating condition, different values of EN-1 differences may be obtained because the EN-1 microphone positions are not uniquely defined.

3. Within broad limits, the L_{nr} of an exhaust

acoustical treatment can be predicted from EN-1 measurements.

a. Definition of EN-1 Difference. The EN-1 difference measure of acoustical effectiveness is defined as the difference between the SPL at the engine EN-1 microphone and that at the exhaust EN-1 microphone.

The engine EN-1 microphone is located as follows[24]:

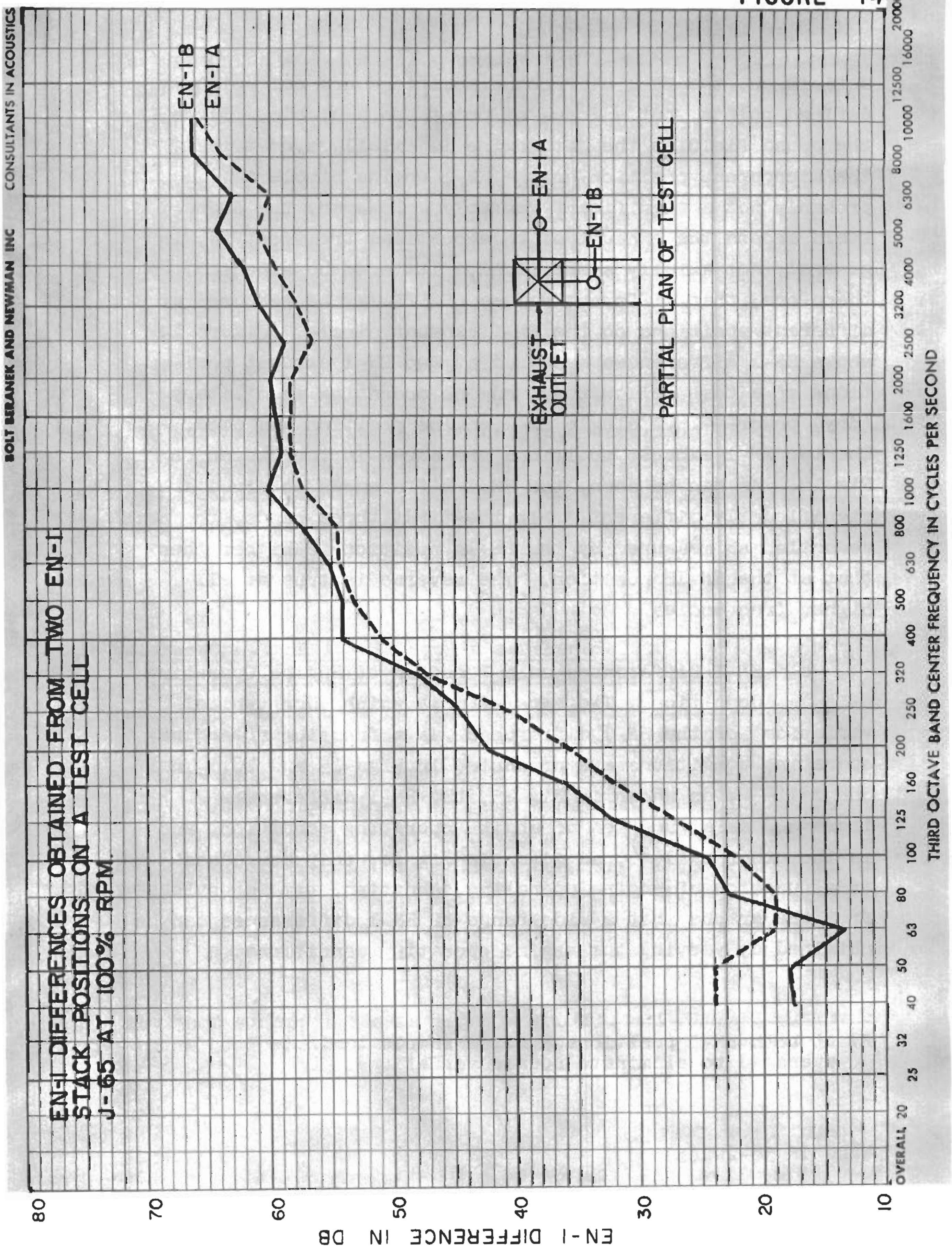
"The microphone should be located in a plane perpendicular to the engine axis and at a distance of two nozzle exit diameters aft from the rear of the engine and radially two nozzle exit diameters from the engine centerline. No measurement should be made at a distance of less than 3 ft from the nozzle center."

The exhaust EN-1 microphone position is located as follows[24]:

"The microphone should be located in a plane perpendicular to the axis of the soundproofing exit (referred to as the emitter) at a distance of one emitter diameter from the emitter plane and at a radius of one emitter diameter from the emitter centerline. Measurements should not be made at a distance less than 14 ft or more than 50 ft from the center of the emitter. (The 'emitter diameter' of an elliptical or rectangular opening shall be the minor dimension.)"

b. Variation of EN-1 Differences with Exhaust Microphone Position. As can be seen from the

FIGURE 14


BOLT BERANEK AND NEWMAN INC CONSULTANTS IN ACOUSTICS
EN-I DIFFERENCES OBTAINED FROM TWO EN-I
STACK POSITIONS ON A TEST CELL
J-65 AT 100% RPM.
EN-I B
EN-I A
EXHAUST
OUTLET
PARTIAL PLAN OF TEST CELL
EN-I DIFFERENCE IN DB
80
70
60
50
40
30
20
10
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND

definition, the EN-1 microphone positions are not uniquely defined points, but a locus of points on a circle. Since the sound field at the jet engine is axially symmetrical, the microphone position on the EN-1 engine circle is probably not a significant variable. However, the sound field around the exhaust of an acoustical treatment is, in general, not axially symmetrical. It is to be expected, therefore, that different values of SPL may be measured at different positions on the EN-1 exhaust circle. Figure 14 shows EN-1 differences measured at two EN-1 exhaust positions of a test cell with an engine operating at military (dry) power[19/]. A single EN-1 engine position was used. The two EN-1 exhaust positions are shown in the sketch on Fig 14. As can be seen, the EN-1 differences obtained are significantly different.* In the first three bands, the noise reduction as measured by the EN-1A position is higher than that at the EN-1B position. The reverse is true at all higher frequencies.

c. EN-1 Differences as a Function of Engine Operating Condition. Two sets of data which show typical variation of the EN-1 difference as a function of engine operating conditions are presented in Figs 15 and 16. In one cell[22/], as shown in Fig 15, the EN-1 differences increased as a function of engine operating condition, and in the other cell[19/], as given in Fig 16, they decreased as a function of engine operating condition. It is obvious from the graphs that a wide range of EN-1 differences can be obtained by varying the engine operating conditions.

* At both EN-1 positions a noise sample 1 minute long was averaged to eliminate source variations.

FIGURE 15

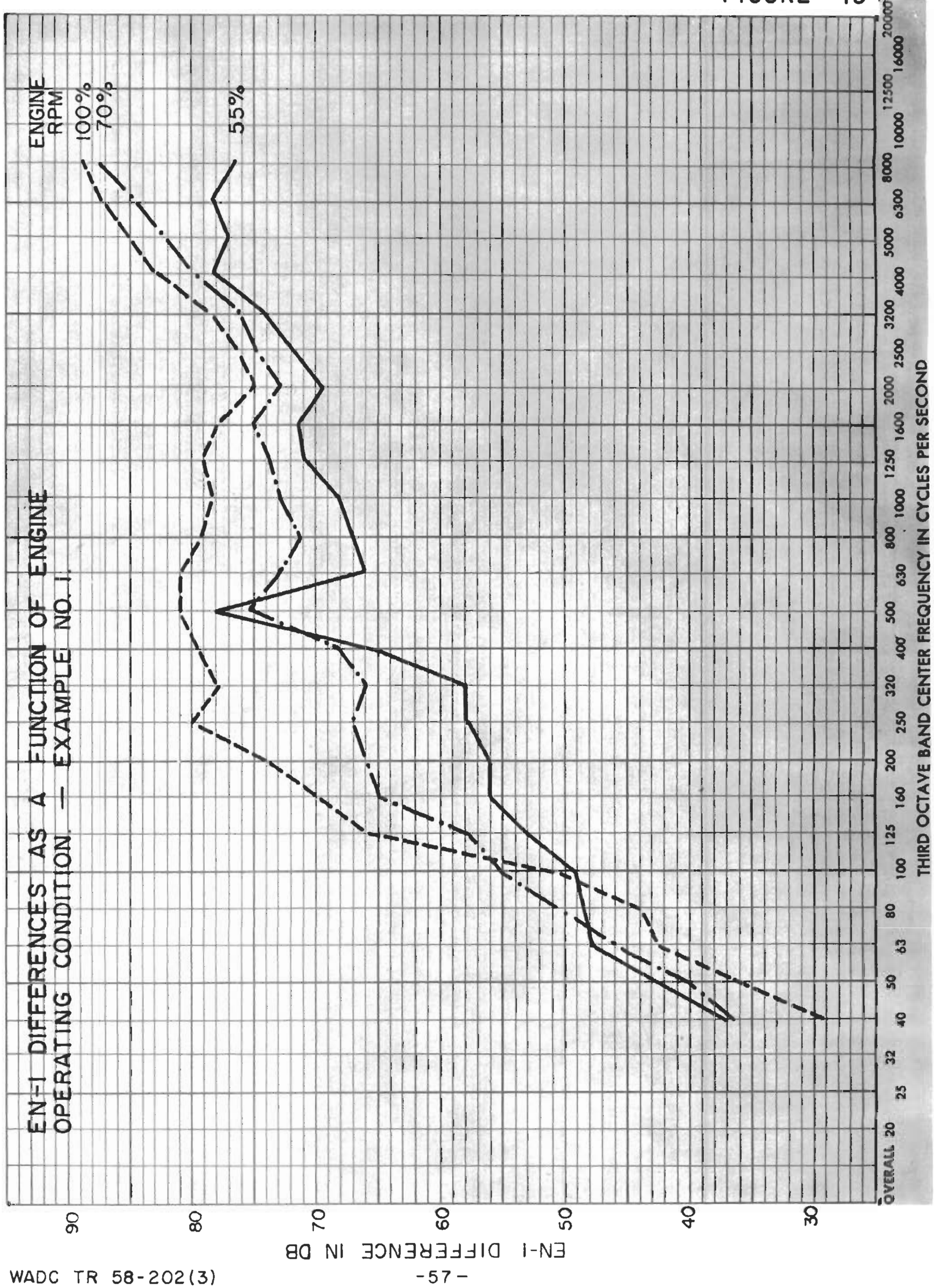
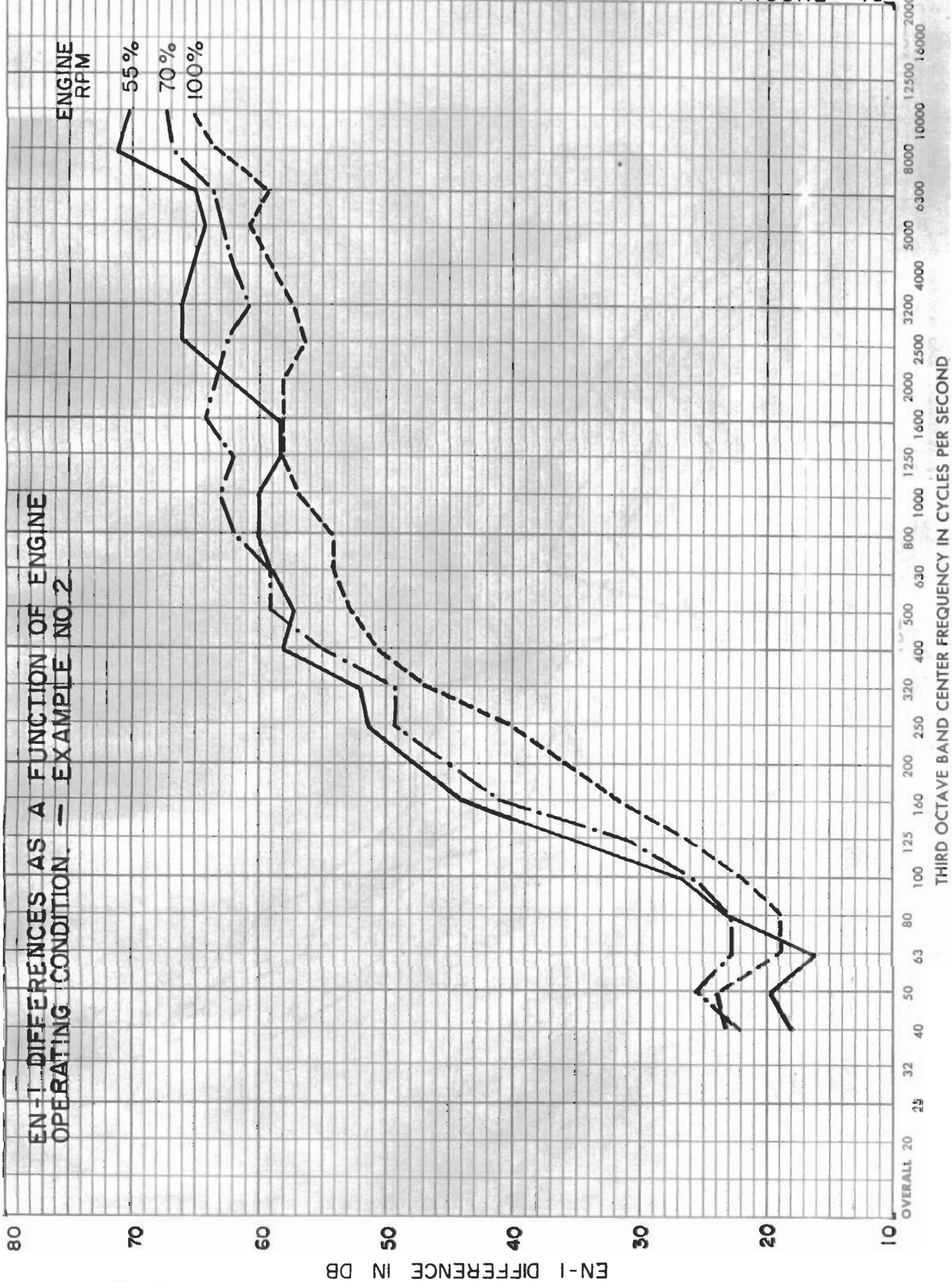

EN-I DIFFERENCES AS A FUNCTION OF ENGINE
OPERATING CONDITION — EXAMPLE NO. I
ENGINE
RPM
100%
70%
55%
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
EN-I DIFFERENCE IN DB
90
80
70
60
50
40
30

FIGURE 16



EN-I DIFFERENCES AS A FUNCTION OF ENGINE OPERATING CONDITION — EXAMPLE NO 2
ENGINE RPM
55%
70%
100%
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
80 70 60 50 40 30 20 10
EN-I DIFFERENCE IN DB

d. <u>Derivation of a Relationship between EN-1 Differences and L_{nr}.</u> Direct comparisons of EN-1* and L_{nr} measurements generally show EN-1 differences to be greater than L_{nr}'s, with the difference between the two increasing with frequency. In the following paragraphs the general form of a relation between EN-1 differences and L_{nr}'s is derived. This relation shows that the EN-1 difference depends on the L_{nr} of the acoustical treatments in the exhaust, the difference between the near field SPL and the PWL of the engine, the directivity of the exhaust outlet, and the size of the exhaust gas outlet. The constants in the derived relationship are evaluated by use of EN-1 differences and L_{nr}'s measured in seven jet engine test cells. This relationship can be used to obtain an approximation to the L_{nr} of the acoustic treatments** from the EN-1 measurements.

If it is assumed that there is a fixed relationship between the octave band SPL at the EN-1 engine position (SPL_{eng}) and the free field power level (PWL) of the engine, then:

$$SPL_{eng} = PWL - X \tag{28}$$

where X is derived empirically*** and is a function of frequency.

---

* All EN-1 data which have been compared with L_{nr} data in the section below were obtained at 100% of maximum compressor revolution rate (no afterburner).

** It should be pointed out that the EN-1 measurement technique and the L_{nr} derived from it, measures the total effectiveness of all of the component treatments in the exhaust system. The effects of bends will also be included in the measurements. Therefore, the EN-1 method is in no sense a measurement of the acoustical effectiveness of a single noise control component.

*** Although it is not relevant to this argument, the value of X has been found to be about 22 db for the difference between overall sound pressure and power level. However, X varies as a function of frequency since the spectrum at the EN-1 position is different from the far field power level spectrum.

The SPL at the exhaust stack EN-1 position, SPL_{exh}, may be written as follows:

$$SPL_{exh} = PWL - L_{nr} - 10 \log_{10} k4\pi \, (\sqrt{2}\, r)^2 - DI - Y \quad (29)$$

where

- k = a constant which lies between 0.5 and 1.0 depending upon whether the radiation from the exhaust stack can be considered hemispherical or spherical.
- r = the "emitter diameter" in ft as defined above.
- Y = a measure of the difference between the overall power level and the power level in an octave band, and is therefore a function of frequency.
- DI = the near field directivity index at the EN-1 exhaust stack position (see Section VI). If spherical divergence obtains between the exhaust stack and the EN-1 "sphere", then DI measures the ratio of the SPL at the EN-1 position to the average SPL over the sphere or hemisphere of radius $\sqrt{2}\, r$.

Subtracting one equation from the other:

$$SPL_{eng} - SPL_{exh} = L_{nr} + (Y-X) + DI + 10 \log_{10} k4\pi r^2 \quad (30)$$

If Y, X and DI are assumed to be constants, then Equation 30 can be written:

$$L_{nr} = \text{EN-1 difference} - A - 20 \log_{10} (r/10)^{*} \quad (31)$$

---

* For convenience the emitter diameter has been normalized to 10 ft. The last term in Equation 31 is zero for r = 10 ft.

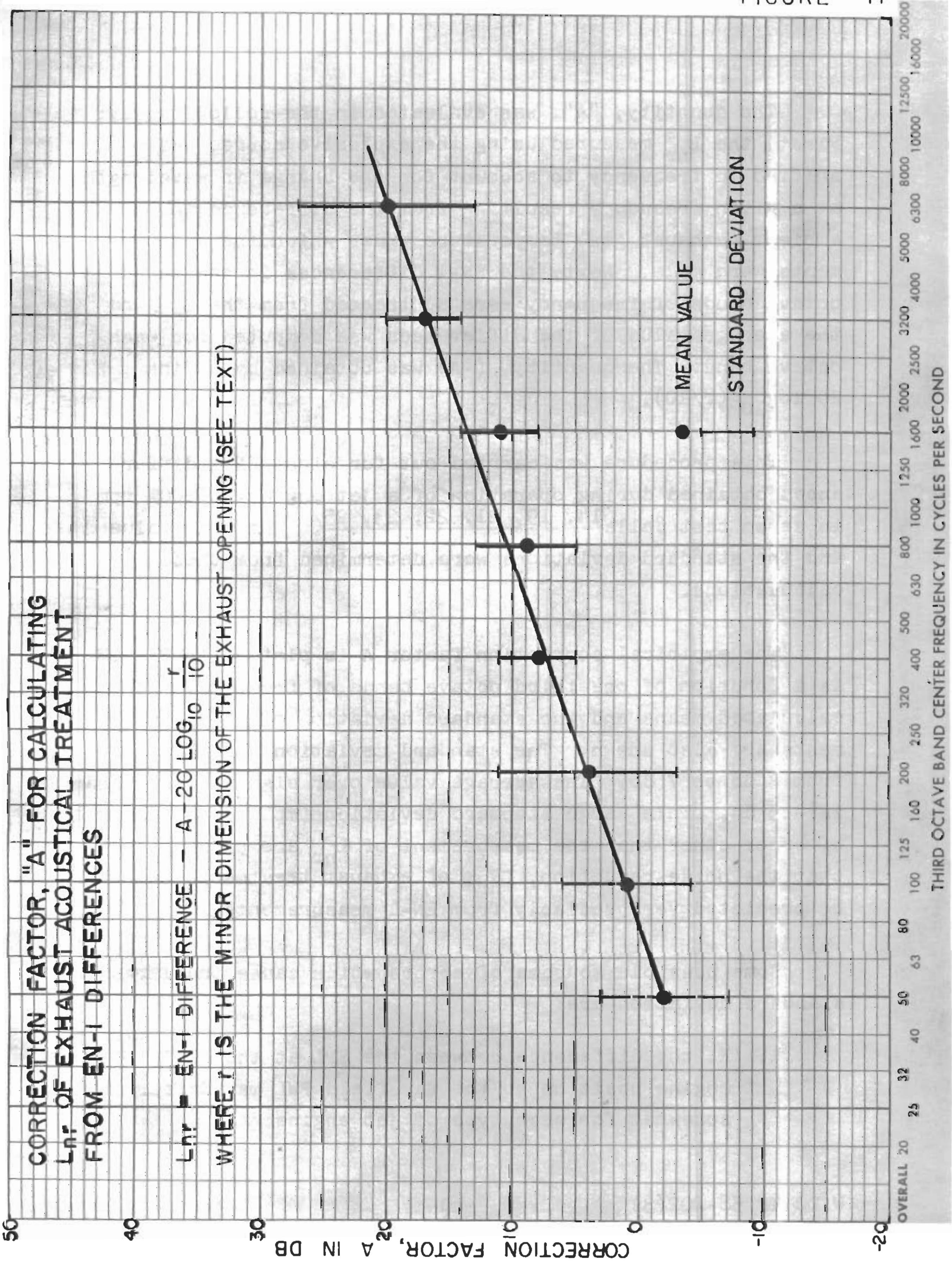
The quantity, "A", was evaluated in the following way: First, the L_{nr} measured using the explosive source was shifted in frequency to account for the change in wavelength at a given frequency resulting from the difference in the ambient (exhaust gas) temperature under which the measurements were made. Next, the EN-1 differences in one-third octave bands of frequency were subtracted from the L_{nr} and the average value of the differences was computed for each octave band. The quantity, (-A) was obtained by adding 20 $\log_{10}$ (r/10).

This procedure was carried out for eight EN-1 differences obtained during operation of a jet engine at 100% rpm in seven test cells[14, 19, 21, 22, 23, 25]. The mean value and the standard deviations were determined from these calculations.

The empirical correction factor A is plotted in Fig 17 as a function of one-third octave bands of frequency. The calculated means and the standard deviation of the measurements are also shown. The standard deviation for A varies from 3 db to 7 db. The average value over eight octave bands is 4.5 db. This large standard deviation implies that sources of error other than the measurement of L_{nr} are present and that the noise reduction, L_{nr}, of exhaust treatments cannot be predicted very reliably from EN-1 measurements.

Some possible sources of errors which cause the large standard deviation are:

1. X, the difference between the SPL at the EN-1 engine position to the far field PWL may differ somewhat for each type of jet engine.


CORRECTION FACTOR, "A" FOR CALCULATING
Lnr OF EXHAUST ACOUSTICAL TREATMENT
FROM EN-I DIFFERENCES
Lnr = EN-I DIFFERENCE – A–20 LOG10 r/10
WHERE r IS THE MINOR DIMENSION OF THE EXHAUST OPENING (SEE TEXT)
MEAN VALUE
STANDARD DEVIATION
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
CORRECTION FACTOR, A IN DB
50 40 30 20 10 0 -10 -20

2. DI may vary on the EN-1 exhaust "circle" of any test cell and may be different for each test cell.

3. Y, which measures spectrum shape, may be slightly different for each engine.

2. Comparison of Insertion Loss with L_{nr}

On one occasion, it was possible to obtain insertion loss measurements. Measurements were made in two engine test cells which were identical, except that one contained no acoustical treatment.* Sound pressure level measurements were made at the exhaust of the treated cell, at the exhaust of the untreated cell and at the input to the acoustical treatment in the exhaust of the treated cell. These data were used to obtain the insertion loss and the L_{nr} of the exhaust acoustical treatment. At the lower frequencies, the insertion loss noise reduction is about 10 db greater than the L_{nr} as is shown in Fig 18. At the high frequencies, there are only small differences between the two noise reduction curves. It is very improbable that the large (10 db) difference can be accounted for by random experimental error as a grid of nine microphone positions was used to obtain all average SPL's.

These measurements are consistent with the arguments presented in Section II, which show that insertion loss is generally not equal to L_{nr}.

H. Summary

The errors in the measurements of L_{nr} arise from (1) instability in the data recording and reduction system,

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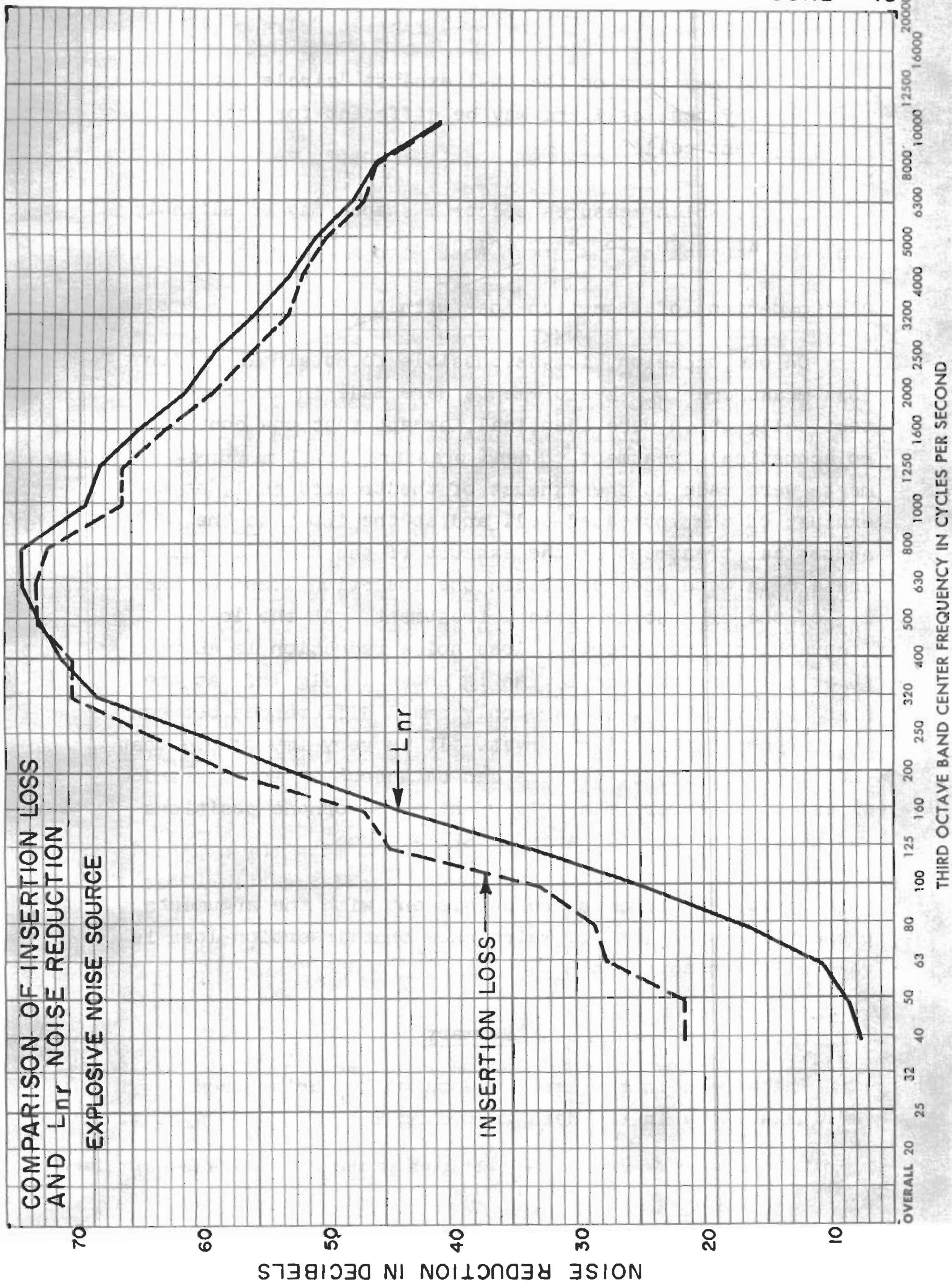
* Bolt Beranek and Newman Inc data obtained at the Gas Turbine Laboratories, Thompson Products, Painesville, Ohio.

FIGURE 18


COMPARISON OF INSERTION LOSS
AND Lnr NOISE REDUCTION
EXPLOSIVE NOISE SOURCE
Lnr
INSERTION LOSS
NOISE REDUCTION IN DECIBELS
70
60
50
40
30
20
10
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL
20
25
32
40
50
63
80
100
125
160
200
250
320
400
500
630
800
1000
1250
1600
2000
2500
3200
4000
5000
6300
8000
10000
12500
16000
20000

(2) variations in noise levels which result from the use of small samples to determine the average value of SPL at a point and, (3) variations in the space average value of SPL at the input and output of the acoustical treatments. For the data presented in this report, the standard deviation of noise reduction values is about 2.0 db.

Analysis of several sets of data indicate that the true mean value of the noise reduction of a single acoustical treatment can be approximated equally well using data obtained with (1) the explosive noise source, (2) a jet engine at 55% rpm and, (3) a jet engine at 100% rpm as noise sources.

EN-1 differences can be used to obtain an approximation to the noise reduction, L_{nr}, of an exhaust acoustical treatment. The large standard deviation of the difference between L_{nr} and EN-1 differences indicates that the prediction of L_{nr} from EN-1 differences in any one test cell may result in large errors.

SECTION IV

NOISE REDUCTION BY IMPERVIOUS BARRIERS

In the Air Force program, transmission loss data have been obtained for several types of wall structures. Some of these data are presented in this section. Data obtained for single-leaf structures indicate that the random incidence mass law[26] provides a reasonable estimate of transmission loss for the structures which have been encountered.

Data obtained for double wall structures are found to be about equal to the value predicted by normal incidence mass law for a single wall of the same total mass. The data obtained from the several surveys show conclusively that the increase in transmission loss predicted[26] for double wall structures is not obtained. The primary cause for the poor performance of double wall structures can usually be identified as various flanking paths.

Typical results obtained from several surveys for both double and single wall structures will be presented in the following paragraphs.

A. Calculation of Transmission Loss

Measurements of noise reduction were made during the acoustical surveys. The random incidence transmission loss has been calculated from the noise reduction measurements by use of the following formula[8]:

$$TL = NR + 10 \log_{10} \left(\frac{1}{4} + \frac{S_w}{R}\right) \qquad (32)$$

where NR = difference in SPL in decibels on the two sides of the wall, determined by measuring the SPL on the primary side with a microphone that is moved around in the reverberant sound field, and then subtracting the SPL measured with a microphone that is moved around in a region that is fairly near the surface on the secondary side.

TL = 10 times the logarithm to the base 10 of the ratio of the sound energy incident on the wall to the sound energy transmitted through the wall.

S_w = area of the transmitting wall either in square meters or in square feet.

R = room constant for the receiving room = $[S\bar{\alpha}/(1 - \bar{\alpha})]$, where S is the total area of the surfaces of the room on the secondary side and $\bar{\alpha}$ is the average absorption coefficient for the receiving room. S must have the same dimensions as S_w.

The transmission loss depends upon the angle of incidence of the sound waves in the source room. If the noise field in the source room is diffuse, the measured value of transmission loss should be approximately equal to the "random incidence mass law" value. In general, the source room for the measurements reported here was the test section of a jet engine test cell. The noise field in the test section is probably not diffuse. A discussion of the characteristics of the noise field in the test section is given below.

A section and elevation of a typical test section and control room are shown in Fig 19. The side walls, ceiling

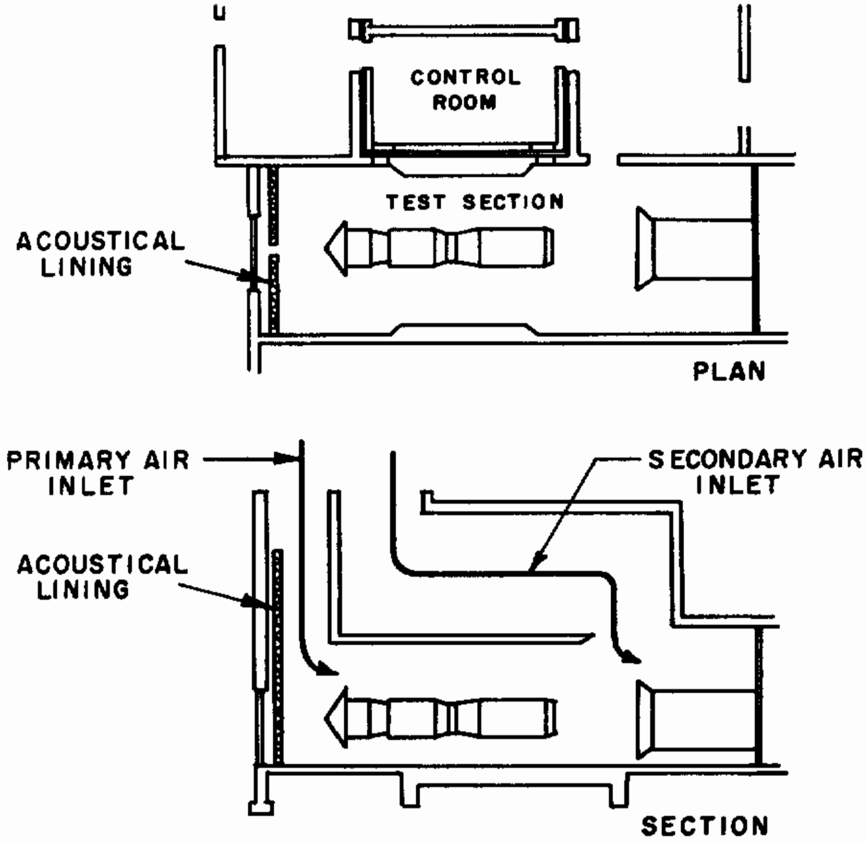


FIG. 19 PLAN AND SECTION AT THE TEST SECTION OF A TYPICAL TEST CELL.

and floors are usually concrete. The forward end of the test section is usually faced with several inches of glass fiber material, covered with a perforated metal. Thus, the sound waves that propagate in a direction normal to the side walls will interact with very little acoustical treatment. Furthermore, they will be at or near grazing incidence to the acoustical material with which they interact. For these two reasons, these waves will not be significantly damped.

On the other hand, the sound waves that are normal to the front and rear walls, or normal to the floor and ceiling, may be highly damped because of the presence of the acoustically treated end wall, the air inlets and the eductor tube. The distribution of acoustic energy impinging on the side walls, therefore, is not random. Instead, there will be more energy at or near normal incidence than at or near grazing incidence. The measured values of transmission loss should lie between the normal incidence mass law values and random incidence mass law values.

It should also be noted that the calculation of transmission loss from noise reduction data is particularly difficult for frequencies below about 100 cps. The noise field in the receiving room is not diffuse (an assumption on which Eq 32 is based), and in addition, the absorption coefficients of acoustical materials are not well known. Therefore, significant errors (of the order of 5 to 10 db) are possible in the calculation of transmission loss below 100 cps.

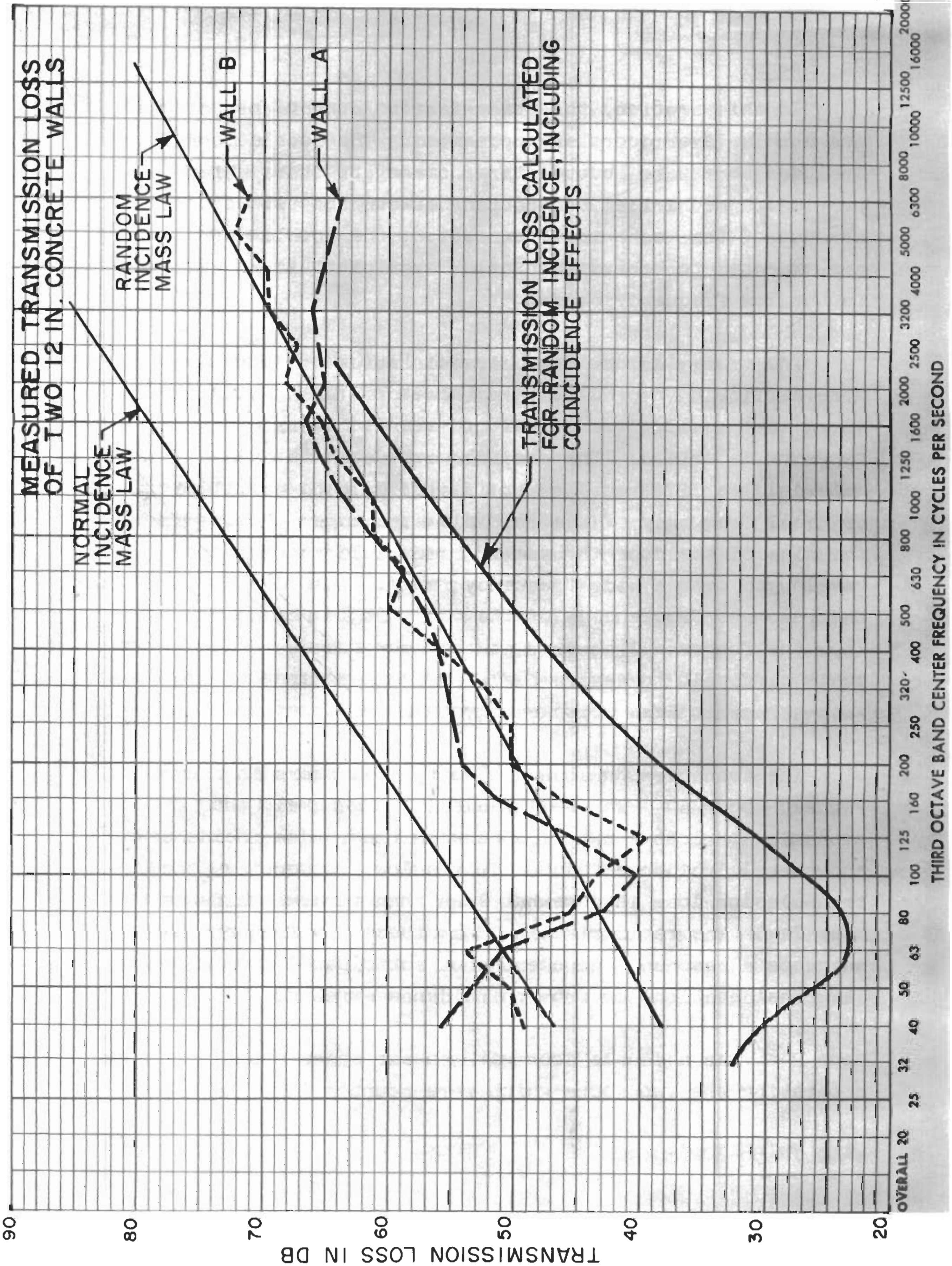
B. Single-Layer Partitions

1. Walls

Measurements of noise reduction were obtained for two single-layer walls which were not penetrated by doors or windows[19, 24]. Both walls separated the test section of one cell from the test section of an adjacent cell. The walls were 12 in. thick and roughly 15 ft high and 40 ft long. The transmission loss of both walls is plotted in Fig 20. The measured value of transmission loss is generally lower than normal incidence mass law would indicate, except in the 40 to 80 cps range. At about 80 to 100 cps, there is a distinct dip in the transmission loss, probably owing to wave coincidence, which can occur for these walls above 60 cps. Above this critical frequency[26, 27], the transmission loss is much lower than the normal incidence mass law value, and is slightly greater than would be predicted from the random incidence mass law. The transmission loss of both walls appears to be limited by flanking paths at about 65 to 70 db.

The transmission loss estimated for random incidence conditions including the effects of coincidence is also shown in Fig 20[27]. The calculated transmission loss is significantly lower than the measured value, and in addition, the calculated frequency for the coincidence dip is lower than the measured value by almost an octave. Both of these differences can be attributed to the "non-random" sound field in the test section, as discussed in Section A above. However, these data indicate that random incidence mass law will provide a realistic estimate of transmission loss for similar walls as used in the test sections of test cells.

FIGURE 20


MEASURED TRANSMISSION LOSS OF TWO 12 IN. CONCRETE WALLS
WALL B
WALL A
RANDOM INCIDENCE MASS LAW
NORMAL INCIDENCE MASS LAW
TRANSMISSION LOSS CALCULATED FOR RANDOM INCIDENCE, INCLUDING COINCIDENCE EFFECTS
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
TRANSMISSION LOSS IN DB
90 80 70 60 50 40 30 20

2. Doors

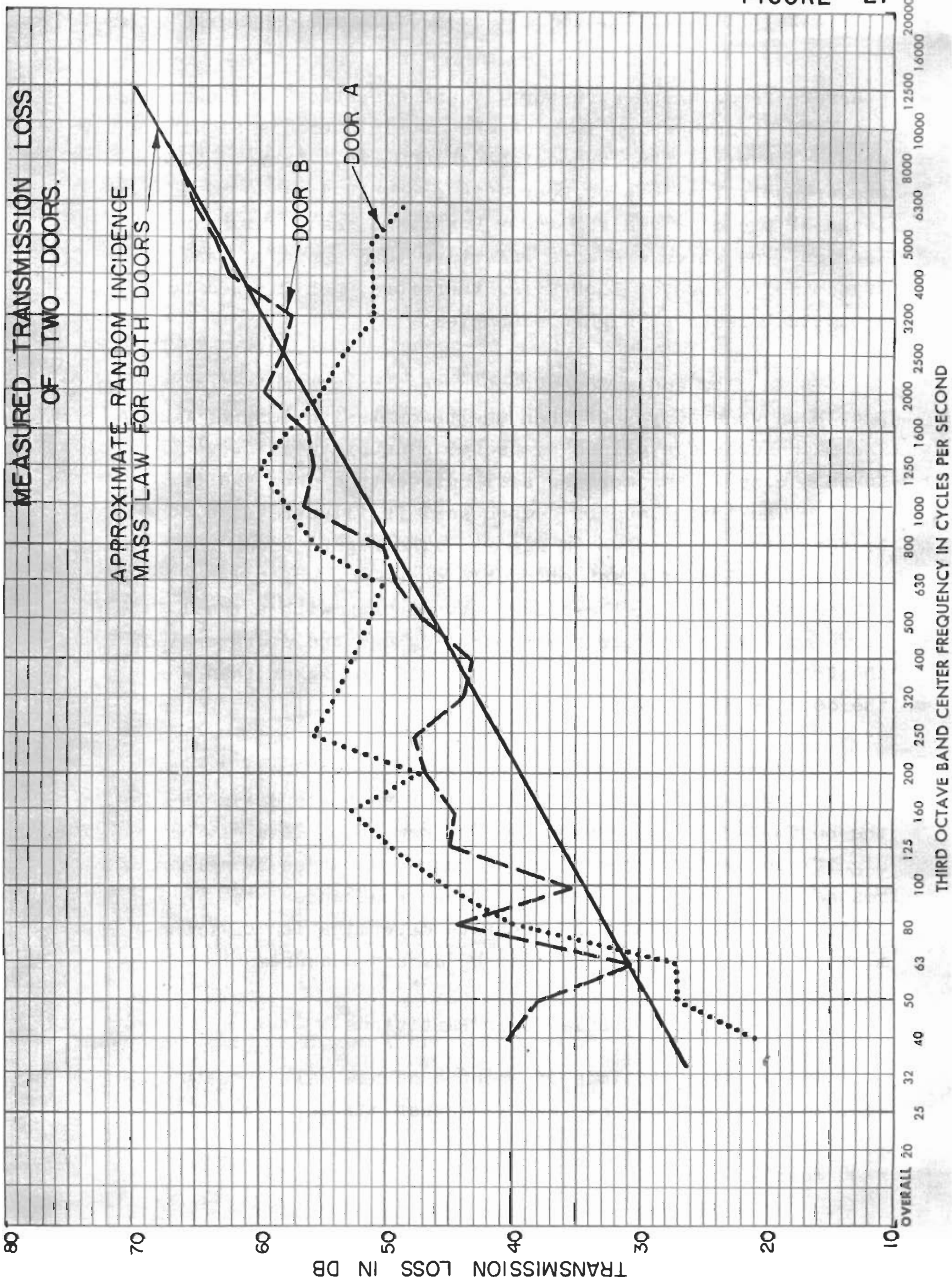
In this section, the transmission loss of both single and double layer doors are presented. The double doors included here were mounted in a common structure which provided a mechanical connection between the two doors. Because of the mechanical connection, the double doors are expected to behave as single layer partitions, at least in the low frequency bands.

The transmission loss characteristics of doors are more difficult to predict than those of single walls. The doors are usually constructed of several layers of different types of materials. The bending wavelength in each of the materials is different, so it is difficult to calculate the critical frequency of the multiple structure. Furthermore, the size of the door is usually comparable to the bending wavelength over a wide frequency range. In order to calculate the resonant frequencies of the door, it is necessary to know the forces and moments at the door frames. Furthermore, it is difficult to devise seals at the perimeter of the door which transmit less noise energy than the door itself.

The transmission loss of two typical doors is shown in Fig 21. Door A[28] was well sealed. In general, the transmission loss shown lies between the random incidence and normal incidence value. In the low frequencies, the transmission loss is somewhat less than random incidence mass law. The rapid rise in TL at these low frequencies suggests a resonant phenomenon not anticipated from mass law considerations or from coincidence effects.

Door B[22] was a double door with an air space about 2 ft deep separating the two doors which were mounted in a steel and

FIGURE 21


MEASURED TRANSMISSION LOSS OF TWO DOORS.
APPROXIMATE RANDOM INCIDENCE MASS LAW FOR BOTH DOORS
DOOR B
DOOR A
TRANSMISSION LOSS IN DB
80
70
60
50
40
30
20
10
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL
20
25
32
40
50
63
80
100
125
160
200
250
320
400
500
630
800
1000
1250
1600
2000
2500
3200
4000
5000
6300
8000
10000
12500
16000
20000

concrete frame which structurally and acoustically linked the two. The transmission loss of the double doors is significantly greater than the single door above 1600 cps. Above 1600 cps, the transmission loss of Door A is probably limited by acoustical leaks at the door seals. Door B has two seals which are in "series" and which are more effective than a single seal. The transmission loss of Door B is, therefore, not limited by acoustical leaks at high frequencies.

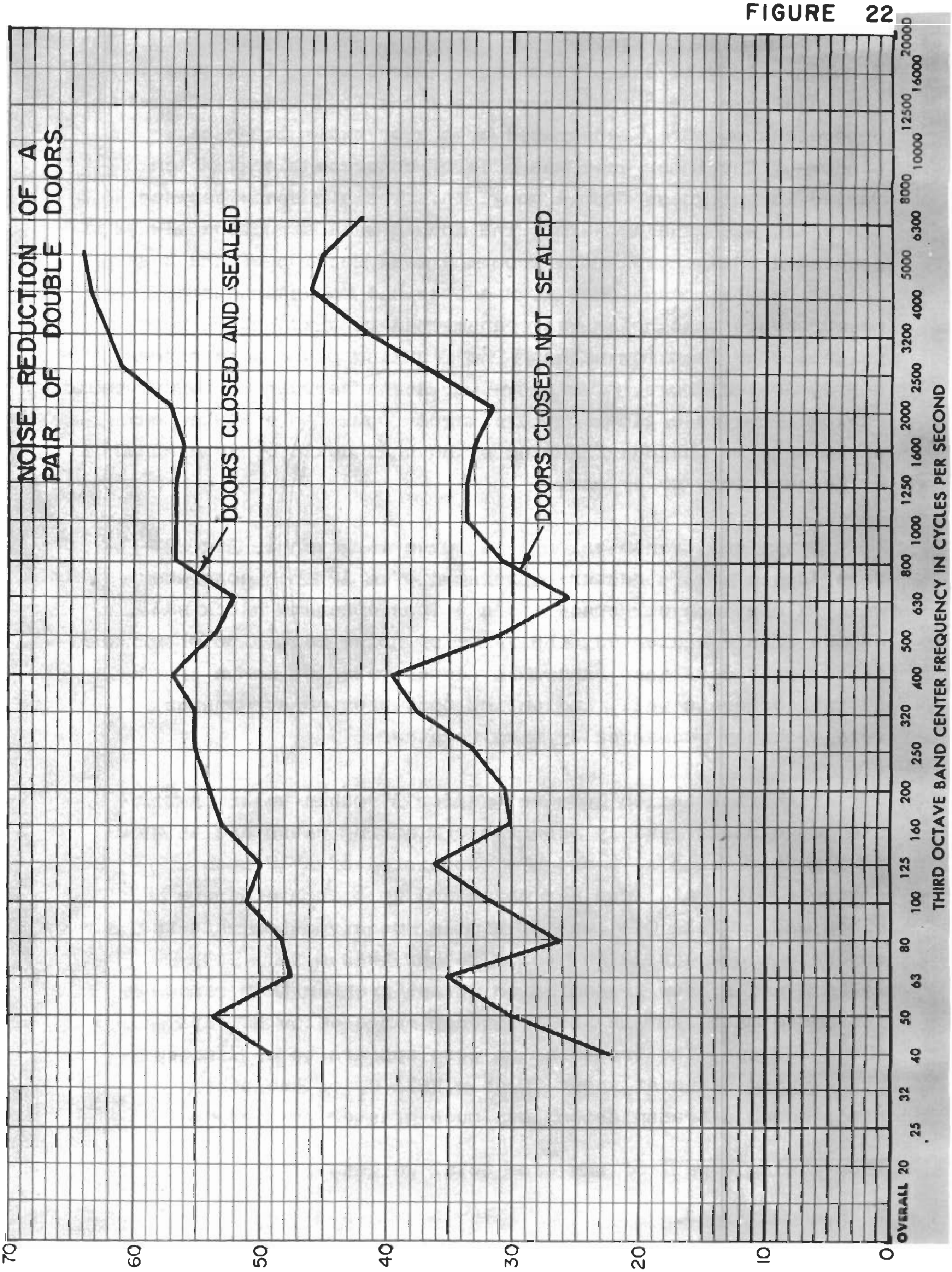
In the low and middle frequencies the transmission loss of Door B is generally less than the transmission loss of Door A. The main advantage gained by use of a double wall structure, in this case is an improved seal between the door and its frame. The importance of a good seal is illustrated by the transmission loss curve in Fig 22. The figure shows the noise reduction of a set of double doors[14], which were sealed by evacuating the space between them with a small air pump. With the air pump operating and the doors sealed, the transmission loss is about 15 to 25 db higher than when the door is not sealed.

In summary, the transmission loss of test cell doors may be severely limited by the lack of adequate seals at the perimeter. Furthermore, the transmission loss may be significantly decreased in some frequency ranges by mechanical resonances of the door. Because these resonances are difficult to calculate, an accurate determination of transmission loss should be obtained by direct measurement.

C. Double Layer Partitions

The noise reductions of about 8 double wall structures were measured during the several acoustical surveys. The

FIGURE 22


NOISE REDUCTION OF A
PAIR OF DOUBLE DOORS.
DOORS CLOSED AND SEALED
DOORS CLOSED, NOT SEALED
NR IN DB
70
60
50
40
30
20
10
0
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000

double walls were all of similar construction. In general, the wall near the noise source was a one ft thick reinforced concrete structure, penetrated by one or two multi-pane windows. The inner wall was made of concrete block, which varied in thickness from 4 to 12 in. The air space between the walls was usually 4 in. The double wall structure always separated the control room from the test section of the test cell. On most occasions, it was possible to ascertain by measurements that noise was transmitted to the control room primarily by flanking paths. The flanking paths were either poorly sealed doors, or windows, or ducts carrying instrumentation cables from the engine to the control room. In a few cases, there were no obvious flanking paths that could be determined by measurement, or by ear.

A typical transmission loss curve is given in Fig 23[19/]. This double wall structure consisted of a 12 inch concrete wall, a one inch air space and a 4 inch concrete block wall. As the figure indicates, the value of noise reduction obtained is somewhat less than random incidence mass law for a single 12 inch concrete wall, indicating that the two structures were probably mechanically tied together.

The transmission loss values for 3 double walls that did not exhibit readily observable flanking paths are shown in Fig 24. Walls A[22/] and B[23/] consisted of a 12 inch concrete wall, a 4 inch air space and an 8 inch concrete block wall. Wall C[17/] was of similar construction, but the concrete block was 12 in. thick rather than 8 in. Both walls A and B show transmission losses greater than random incidence mass law* in the frequency range below 800 cps. Above 800 cps, the transmission loss appears to be limited by flanking paths at about 70 db. Wall C, on the other hand, shows a transmission loss much greater than random

*For a single wall of the same total weight.

FIGURE 23

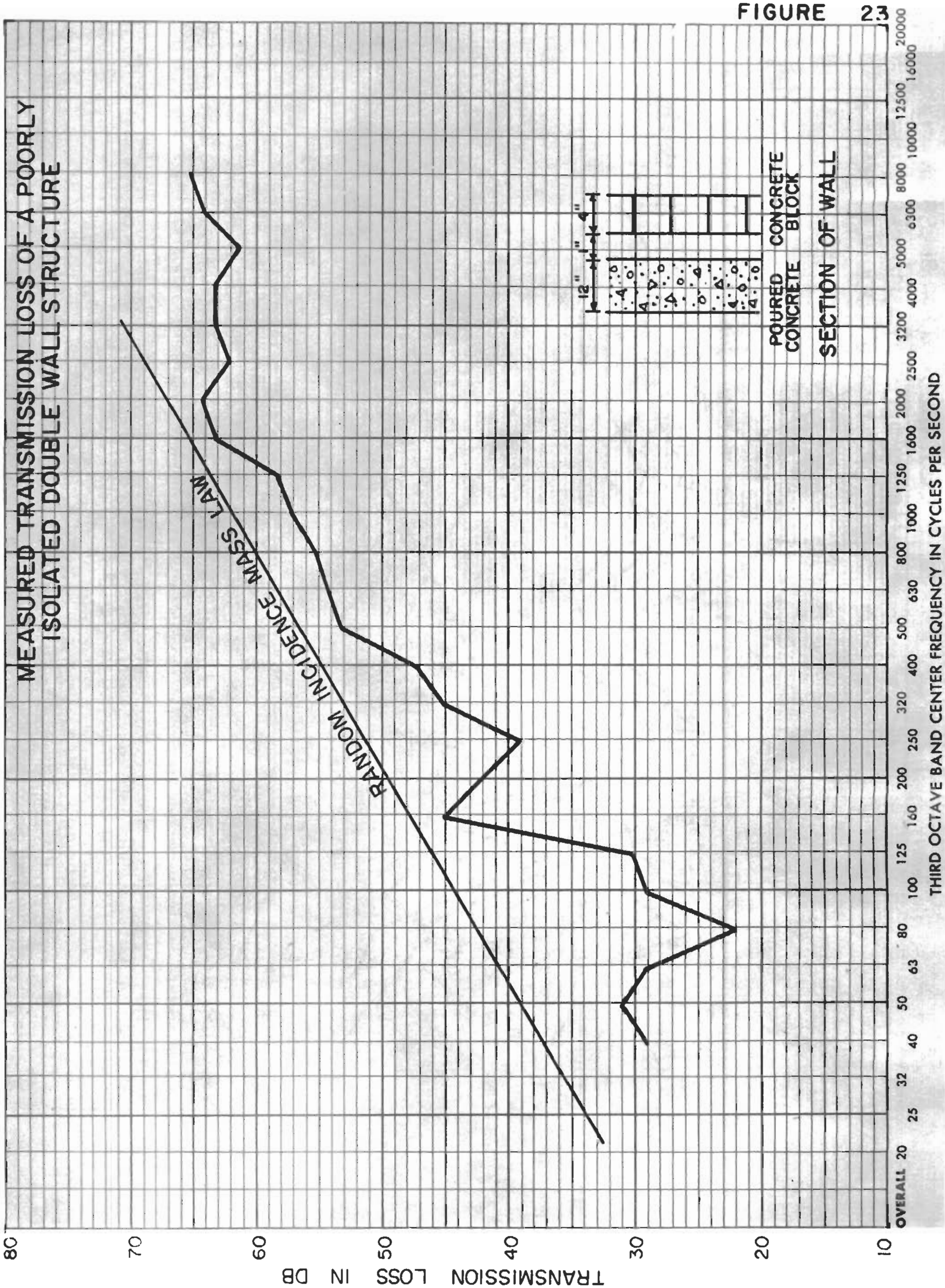
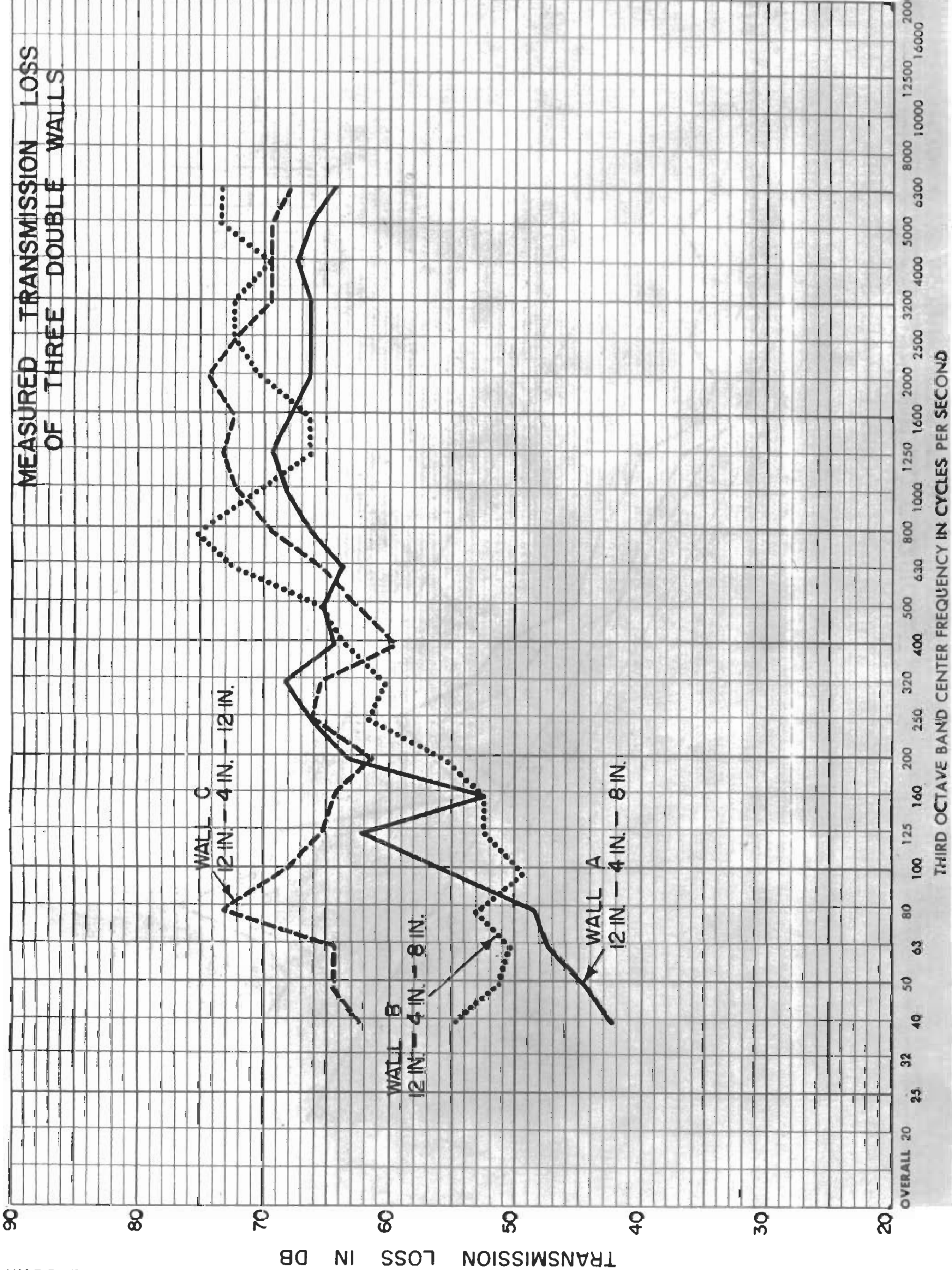

MEASURED TRANSMISSION LOSS OF A POORLY
ISOLATED DOUBLE WALL STRUCTURE
RANDOM INCIDENCE MASS LAW
12"
1"
4"
POURED CONCRETE
CONCRETE BLOCK
SECTION OF WALL
TRANSMISSION LOSS IN DB
80
70
60
50
40
30
20
10
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND

FIGURE 24


MEASURED TRANSMISSION LOSS
OF THREE DOUBLE WALLS
WALL A
12 IN. — 4 IN. — 8 IN.
WALL B
12 IN. — 4 IN. — 8 IN.
WALL C
12 IN. — 4 IN. — 12 IN.
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
TRANSMISSION LOSS IN DB
90 80 70 60 50 40 30 20

incidence mass law* in the lower frequencies. Wall C is also limited at about 65 to 70 db by flanking paths.

Three observations can be made about these data. First, conventional designs and construction techniques for double walls do not provide transmission losses greater than 70 db. Second, the upper limit on transmission loss is probably due to flanking paths. These paths include the ground which supports the double wall structure and further study of the transmission characteristics of soil is indicated. Third, the various theories presented to date for double wall structures do not yield realistic estimates of the transmission properties of double walls encountered in test cells. A theory must be developed which considers the stiffness of the walls as well as their mass.

It is not to be inferred from these remarks that it is not possible to build a double wall that has much greater transmission loss than a single wall of equivalent weight. Figure 25, for example, shows the measured transmission loss of a double wall structure which separated two broadcast studios[27/]. The wall consisted of two 4-1/2 in. thick brick walls which were plastered on one side. The air space between the walls was 12 in. There were no direct mechanical connections between the two studios. Both studios are supported on rubber mounts in order to minimize structural flanking paths.

The transmission loss for the double wall structure is considerably greater than the normal incidence mass law value for a single wall of the same total weight. At the

*For a single wall of the same total weight.

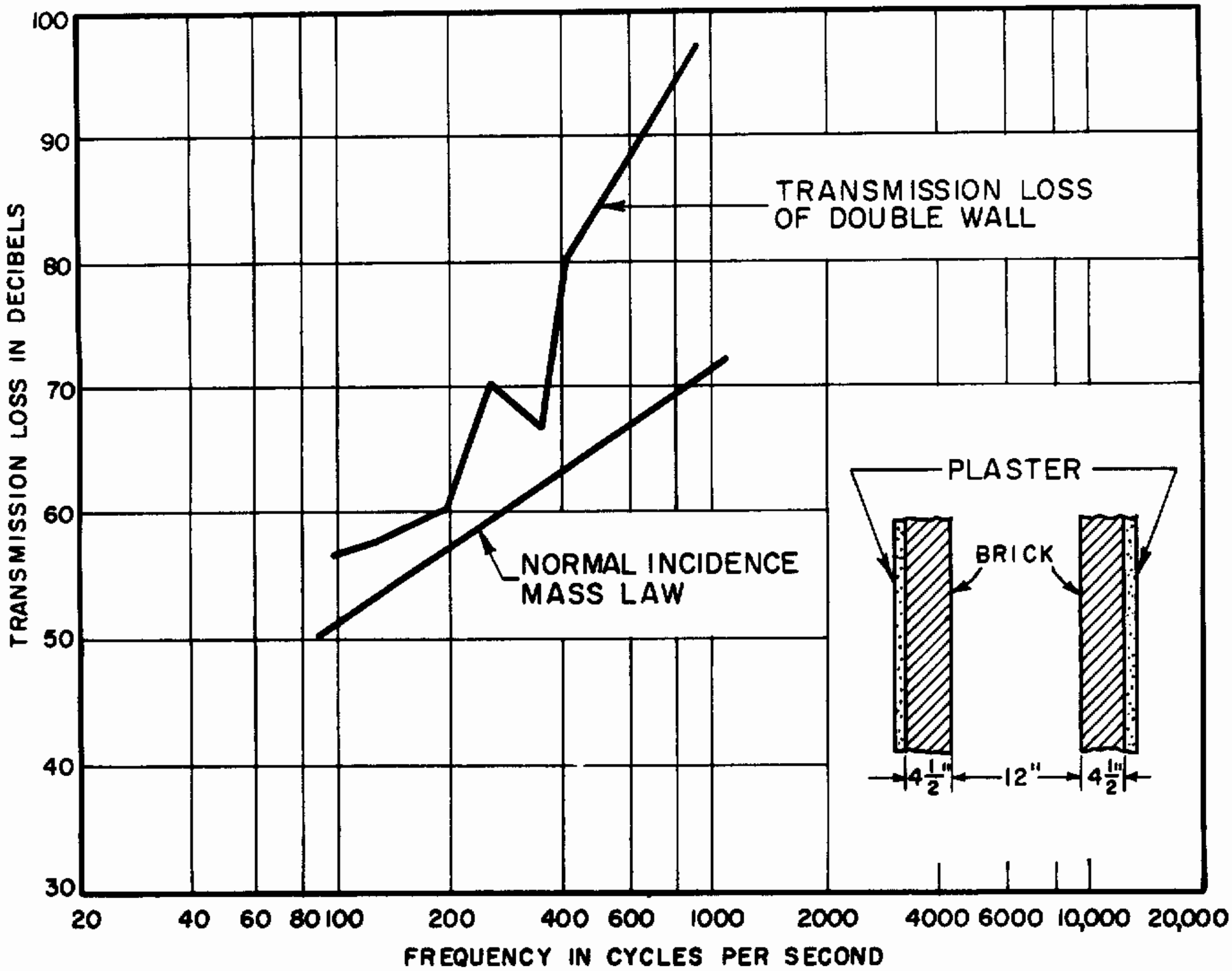


FIG. 25 TRANSMISSION LOSS FOR A DOUBLE WALL STRUCTURE.

highest frequency reported, about 900 cps, the noise reduction of the double wall is about 25 db greater than the normal incidence mass law value.

The primary problem in achieving high sound isolation with double walls is provision for adequate structural isolation between the two layers. Some typical details for good isolation are given in Section IV of Volume Two of this series.

SECTION V
NOISE REDUCTION OF NOISE CONTROL COMPONENTS FOR AIR PASSAGES

A. General Discussion

The reduction of noise through air passages is accomplished by three types of components: lined ducts, bends, and resonators. A lined duct has a perimeter which is covered with some form of absorptive acoustical material. For discussion purposes, parallel baffles can be considered as ducts with two sides lined and a rectangular cross-section. A bend may either be a change in direction of the air passage itself, or a change in direction of air flow caused by special design of a noise control component. Resonators, which are effective over a narrow frequency band, use combinations of acoustically massive and resilient components to effect noise reduction by reflection back towards the source. Under the USAF program of acoustical evaluations, measurements were made only for lined ducts, bends, and baffles. No resonator structures were encountered, and hence, no data are presented for resonators.

The noise reduction of lined ducts was discussed by Sabine[29], who showed that for low frequencies the total energy loss through a duct should be proportional to the ratio of the perimeter to the cross-sectional area, and to a small power of the absorption coefficient of the lining. Morse[7] derived the noise reduction for ducts by determining a solution for the wave equation which was applicable over a wider frequency range. Cremer[30] has presented Morse's results in a slightly different and more useful form, so that fewer calculations are required.

The results of Morse's and Cremer's work are applicable only to certain propagation conditions. For example, each of these theories deals only with the noise reduction of infinitely long systems. Neither considers the reflection of sound power back toward the source, caused by the change in impedance at the input to the duct. In addition, neither considers the possibility of the reflection of sound power back into the lined duct caused by the change in impedance at the outlet of the duct. The effect of the scattering of sound at the input to a lined duct was not considered by Morse and Cremer, but Young[31] has investigated this effect for one duct geometry.

Although Morse indicates that his results could be generalized to include the noise reduction for sound waves with a periodic distribution of pressure across the duct (higher order of modes), he presents results only for plane waves* and the first higher order mode. Cremer and Morse assume that the normal acoustic impedance of the duct wall is known and that the normal impedance is a point** function. It is usually necessary to measure the acoustic impedance of the duct lining, because small errors in the value of impedance may cause large errors in the estimation of noise reduction. Existing methods for calculating impedance of a typical lining encountered in test cells from the physical properties are not reliable enough to be used for predicting the impedance with the accuracy required.

*In an actual duct, a truly plane wave does not exist. Because energy flows into the boundary of the duct, the wave front is inclined toward the side walls.

**This assumption requires that there is no coupling between points of the absorbing material and, hence, no flexural wave propagation in the acoustical material. This assumption is generally accepted although, to the author's knowledge, experimental investigations of the validity of the assumption have begun only recently (in Göttingen).

Parallel baffles, which usually do not have a septum dividing them, present additional troubles. The effective normal impedance of such a baffle depends upon the sound pressure distribution on either side of the baffle. In turn, the pressure distribution depends on the impedance of the baffle. Therefore, a simultaneous solution to two or more wave equations may be required.

The usefulness of theories presented to date is therefore seriously limited because they relate to conditions that are not obtained in test cells. The differences between the results obtained from the present analysis and the theory are attributable to the effects of finite length, and to the differences between the attenuation for the first order mode and the attenuation for higher order modes.

1. A Qualitative Analysis of the Noise Reduction of Ducts and Baffles

Some of the differences between noise reduction for plane waves and noise reduction for higher order waves can be estimated quantitatively by considering how noise reduction may vary with angle of incidence at high frequencies (those for which the wavelength of sound is much smaller than the width of the duct or baffle). Consider, for example, the parallel baffle structure sketched in Fig 26.

The baffle is 2t in width, and has a thin heavy septum in the center. The on-center spacing of the baffles is D, the height is H and the length is L. The open spacing between the baffles, D', is equal to (D - 2t). A sound wave traveling parallel to plane B and entering at an oblique angle, θ, must travel a distance ($1/\cos\theta$) times farther

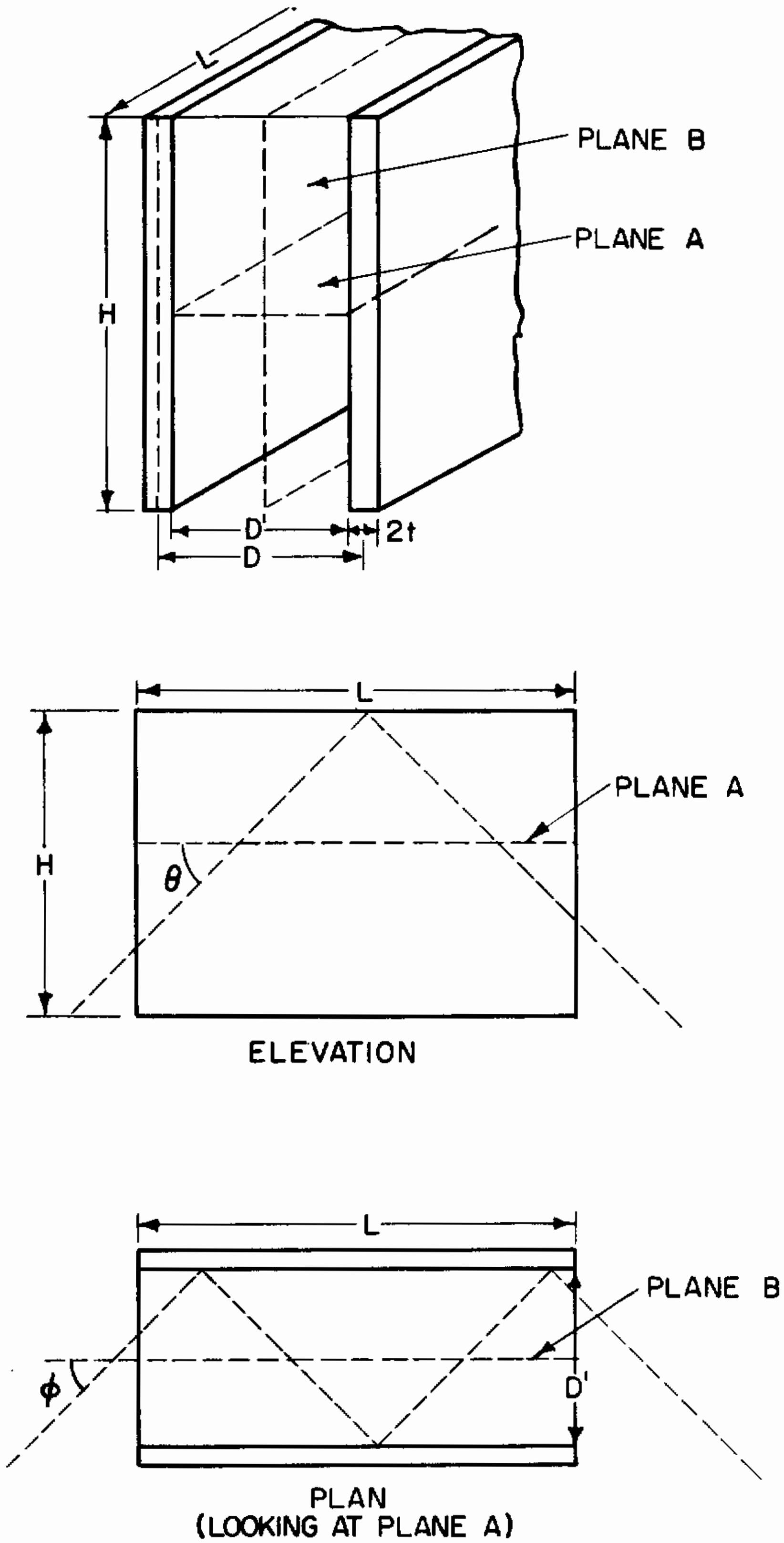


FIG. 26 GEOMETRY AND NOMENCLATURE FOR A PARALLEL BAFFLE STRUCTURE.

than the wave entering at normal incidence ($\theta = 0$). For example, if the noise reduction for a length, L, were 10 db, then the noise reduction for waves at 45° would be about 14 db. For waves at 60°, the noise reduction would be 20 db. As the angle of incidence approaches 90°, the noise reduction approaches infinity. If the distribution of the incident sound waves, $f(\theta)$, were known, it would be possible to obtain the noise reduction for any $f(\theta)$ as follows:

$$\text{Output/Input} = \int_{\theta = -90^{\circ}}^{\theta = +90^{\circ}} f(\theta) g(\theta) d\theta \Bigg/ \int_{\theta = -90^{\circ}}^{\theta = +90^{\circ}} f(\theta) d\theta \qquad (33)$$

where $g(\theta)$ is the ratio of the output to input for each angle θ, which equals $g(0^{\circ}) \cos \theta$,

$g(0^{\circ})$ is the ratio of sound pressure at the input to sound pressure at the output for normal incidence waves.

It is assumed here that any end effects resulting from reflection back toward the source are negligible or that they are small compared to $g(0^{\circ})$.

Now consider waves moving parallel to plane A of Fig 26. A sound wave entering at an oblique angle ϕ is reflected from one baffle and then the other until it leaves the baffle at the angle ϕ or $-\phi$. The number of reflections, n, is approximately:

$$n = \frac{L}{D'} \times \tan \phi \qquad (34)$$

The noise reduction of the baffle is $[1 - \alpha(\phi)]^n$, where n is $\frac{L}{D'} \tan \phi$. As ϕ approaches 90°, the noise reduction approaches infinity. As ϕ approaches 0°, the noise reduction approaches zero because the frequency is assumed to be high enough so that the duct is many wavelengths wide, and there is little interaction with the sound absorbing boundaries.

The noise reduction for waves at an angle of incidence ϕ is generally greater than the noise reduction for waves at an angle of incidence θ. Stated in another manner, for values of $\alpha(\theta)$ encountered in test cells, the noise reduction for waves with components perpendicular to the baffle is greater than the noise reduction for waves parallel to the baffle. Thus the noise field at the output of baffles will contain fewer higher order modes in a plane perpendicular to the baffle than in the plane parallel to the baffle.

In the low frequencies, a similar result is obtained because higher order mode propagation can occur only for frequencies at which the boundaries of the duct are separated by a distance greater than the wavelength of sound. At lower frequencies, where the wavelength is large, higher order modes decay even with no acoustical treatment on the perimeter. If H is greater than D', as is usually the case, higher order modes at any frequency will be propagated with less attenuation in plane A than in plane B.

In a test facility, the sound waves will not be parallel to either plane A or plane B. The noise reduction for these cases, however, will be similar in certain respects to either of the cases described above. In particular, the sound waves at any angle of incidence other than normal are expected to be attenuated more rapidly than the sound waves at normal incidence.

2. Some Implications of the Dependence of Noise Reduction on Angle of Incidence

Assume, for example, that the noise reduction for a parallel baffle structure is 4 db per foot for sound waves incident at 90° to 70° from normal, 3 db per foot for waves

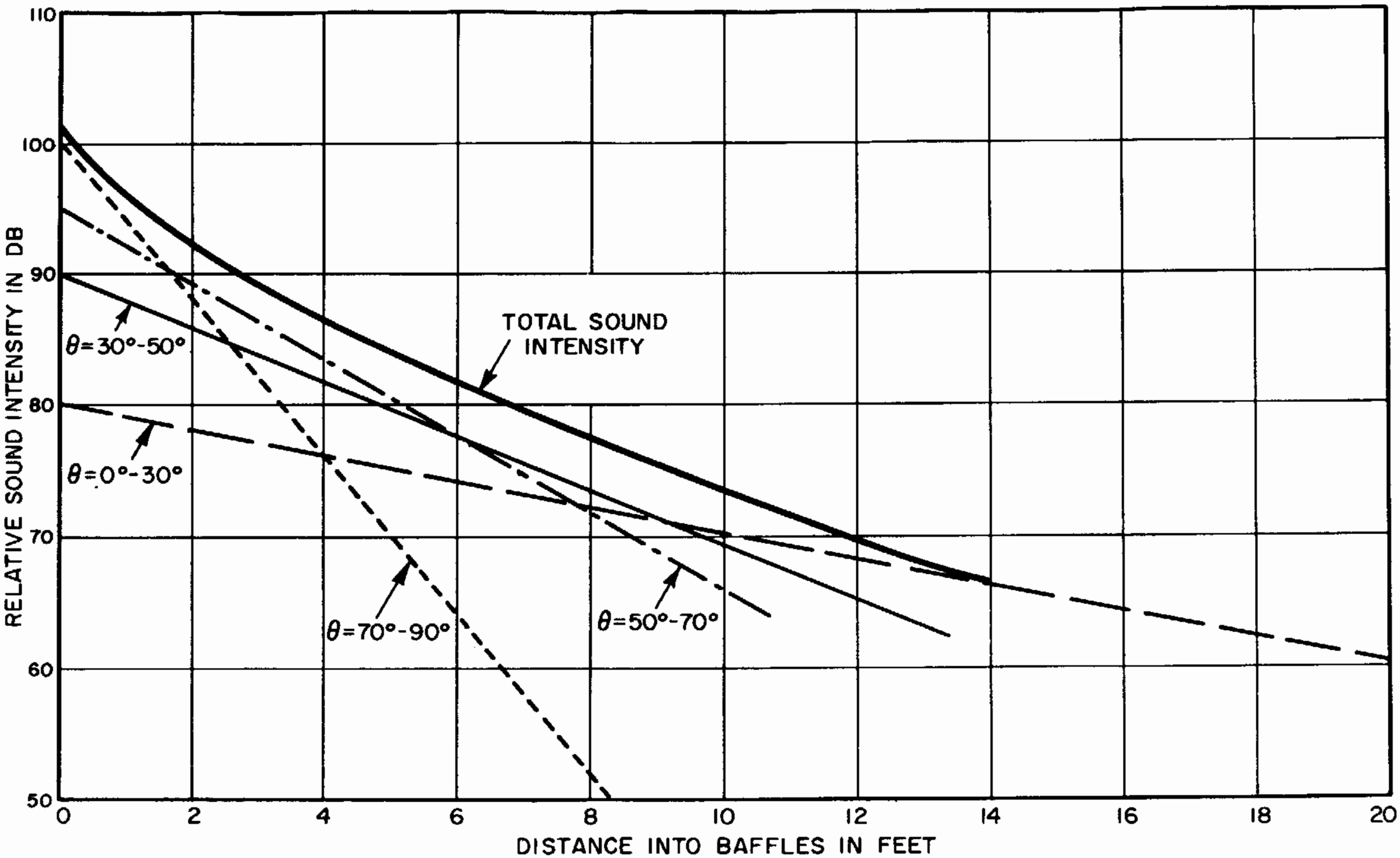


FIG. 27 SOUND INTENSITY AS A FUNCTION OF DISTANCE INTO A PARALLEL BAFFLE STRUCTURE

between 50° and 70°, 2 db per foot for waves between 30° and 50°, and one db per foot for waves from 0° to 30°. Furthermore, let the relative intensity level in each of these angular ranges be 100, 95, 90 and 80 db respectively, at the input. The total sound intensity at any point in this structure is obtained by a summation of the sound energy from 0° to 90°.

Figure 27 shows the sound energy in each angular range as a function of distance into the structure. At the input the total relative sound energy is about 102 db and the major contribution comes from the energy in the 70° to 90° region. At a distance of 10 ft into the structure, the level is about 83 db, with approximately equal contributions from the 30° to 50° and 0° to 30° angular regions.

The noise reduction per unit distance (the slope of the curve) varies considerably as a function of length into the treatment. In the first one foot of distance from the entrance, the noise reduction is about 6 db/ft. From 10 ft to 11 ft, the noise reduction is about 1-1/2 db/ft. At 20 ft, the noise reduction is 1 db/ft.

A value of noise reduction per ft obtained from a measurement from the input to the output is not the same as the slope of the curve. For example, a measurement of noise reduction of 10 ft of this treatment would show a noise reduction of 102-73 = 29 db or 2.9 db/ft which is almost twice the slope of the curve at 10 ft! Even at the distance of 20 ft, where the remaining energy is almost entirely in the 0 - 30° range, the measured noise reduction from 0 to 20 ft would be 2.1 db/ft rather than 1.0 db/ft, which is the slope of the curve at 20 ft.

By varying the relative sound energies in the several angular regions, and by varying the noise reduction values in each range, one can obtain different distances at which the noise reduction slope becomes constant. The values of noise reduction and relative sound energy for this example were selected to illustrate this point. However, the results are similar to those measured in the field, as demonstrated by the examples which follow.

3. Selected Examples from Field Measurements

In one acoustical survey under the Air Force program, the variation of SPL with distance into the treatment was investigated. The SPL was measured at several intervals in a square (8' x 8') lined duct. The acoustical lining consisted of about 6 in. of glass fiber blanket enclosed in perforated metal and backed with a two foot air space. The SPL's at the various measurement positions are shown in Fig 28. A single microphone position was located in the center of the duct at each 4 ft interval, and data were recorded during two shots from the explosive noise source. The SPL at each position is plotted relative to its value at the input. In the 300 to 600 cps band, the noise reduction in the first 4 ft is about 8 db or 2 db/ft. If this value were used for the noise reduction per foot, the total noise reduction for 20 ft of acoustical treatment would be 40 db. As can be seen, however, the total noise reduction in the 300-600 cps band is only about 25 db.

For longer or shorter treatments, from about 10 to perhaps 30 ft, the noise reduction in the 300-600 cps band could be expressed as $8 + 3/4\ \ell$ (the solid line on Fig 28). This formula would predict a noise reduction of 38 db for a 40 ft

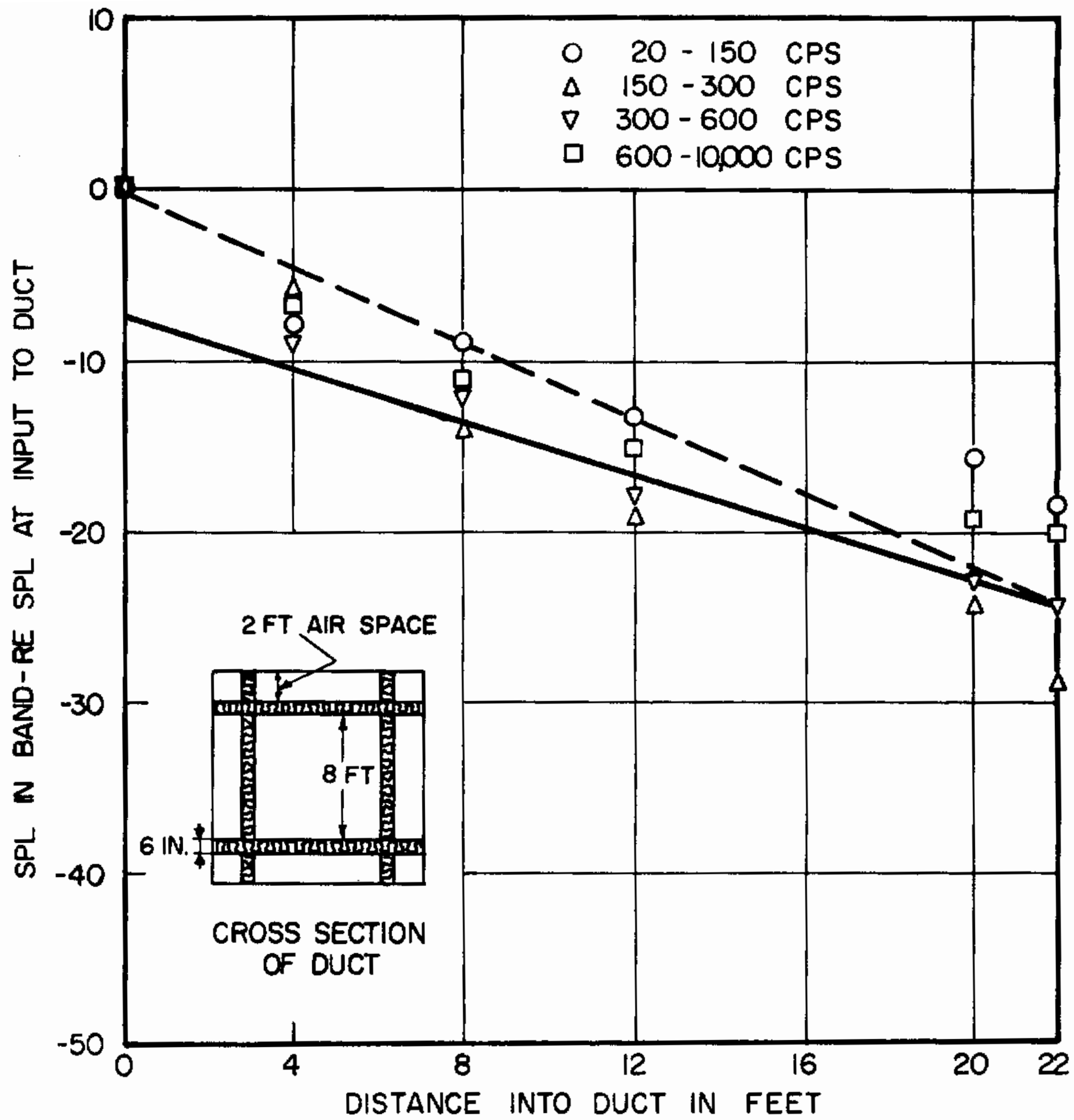


FIG. 28 FIELD MEASUREMENTS OF SOUND PRESSURE LEVELS IN A LINED DUCT.

treatment. If one measures only the SPL's at 0 ft and at 22 ft and concluded the noise reduction per foot for this treatment was 25/22 lb per foot, then the noise reduction of 40 ft of this treatment would be predicted to be 46 db. If the value of noise reduction in db per foot (with no added constant) is used to predict the noise reduction of a similar duct less than 20 ft long, the noise reduction will be always too low (see dashed line on Fig 28).

Data measured under this program of acoustical surveys are not the only data which clearly show that noise reduction per unit length varies with length. For example, an interesting article by Fitzroy[32/] in the Journal of the Acoustical Society of America in 1943 showed a similar result. Fitzroy made measurements with a pure tone at 2-1/2 ft intervals through 20 ft of several different types of acoustical treatments. His interpretation of the data is interesting. Having found that the noise reduction per unit distance is relatively large in the first few feet of the treatments, and then reaches a smaller constant value, he concludes "each cell (noise reduction component) reach(es) its maximum practical reduction beyond which the reduction is relatively small." This is another way of stating that the slope of the SPL vs. distance curve becomes constant.

Another interesting series of experiments was carried out under the Materiel Command of the Army Air Force in late 1943 and early 1944.[33/] Experiments somewhat different from those described for the 8 ft by 8 ft lined duct were carried out for parallel baffles about 3-1/2 in. thick and 16 in. on centers. The baffles consisted of three 6-ft sections which were installed, section by section, to obtain measurements of

6 ft, 12 ft and 18 ft baffles. A space averaging technique similar to that described in Section III was used for all measurements. Sixteen measurement positions were used in each grid. The noise reduction for each length was measured at six frequencies with a warble tone which varied ± 30 cps at frequencies below 1300 cps, and ± 90 cps at higher frequencies, In each test, six frequencies were used: 150, 300, 600, 1200, 2400 and 4,000 cps. Although the measurement equipment may not have been as stable as the system described in Chapter III, the error is probably smaller than the data obtained with the system described in Section III. The source (a loudspeaker) was more stable than an engine or an explosive source, and 16 microphone positions were used.

Figure 29 contains the data presented in Appendix 2 of Ref. 33. Clearly, the noise reduction per unit length is not a constant. For example, at 600 cps, for the first 6 ft of baffles the noise reduction is 22 db or 3.8 db per ft. From 6 ft to 12 ft the noise reduction is 14 db or 2.3 db per ft. From 12 to 18 ft the noise reduction is 13 db or 2.2 db per ft. This noise reduction for any length can be expressed as $(9 + 2.3\,\ell)$ db, at least in the range from 6 to 18 ft. This formula for noise reduction will more reliably predict the noise reduction for any length than would the product of ℓ and a db-per-ft value of noise reduction.

Hirschorn[34] also describes an experiment for which he encountered a variation of noise reduction per unit length with increasing length. His data show a slightly different variation of noise reduction with length than is shown in other experiments, but the results also bear out the arguments presented above.

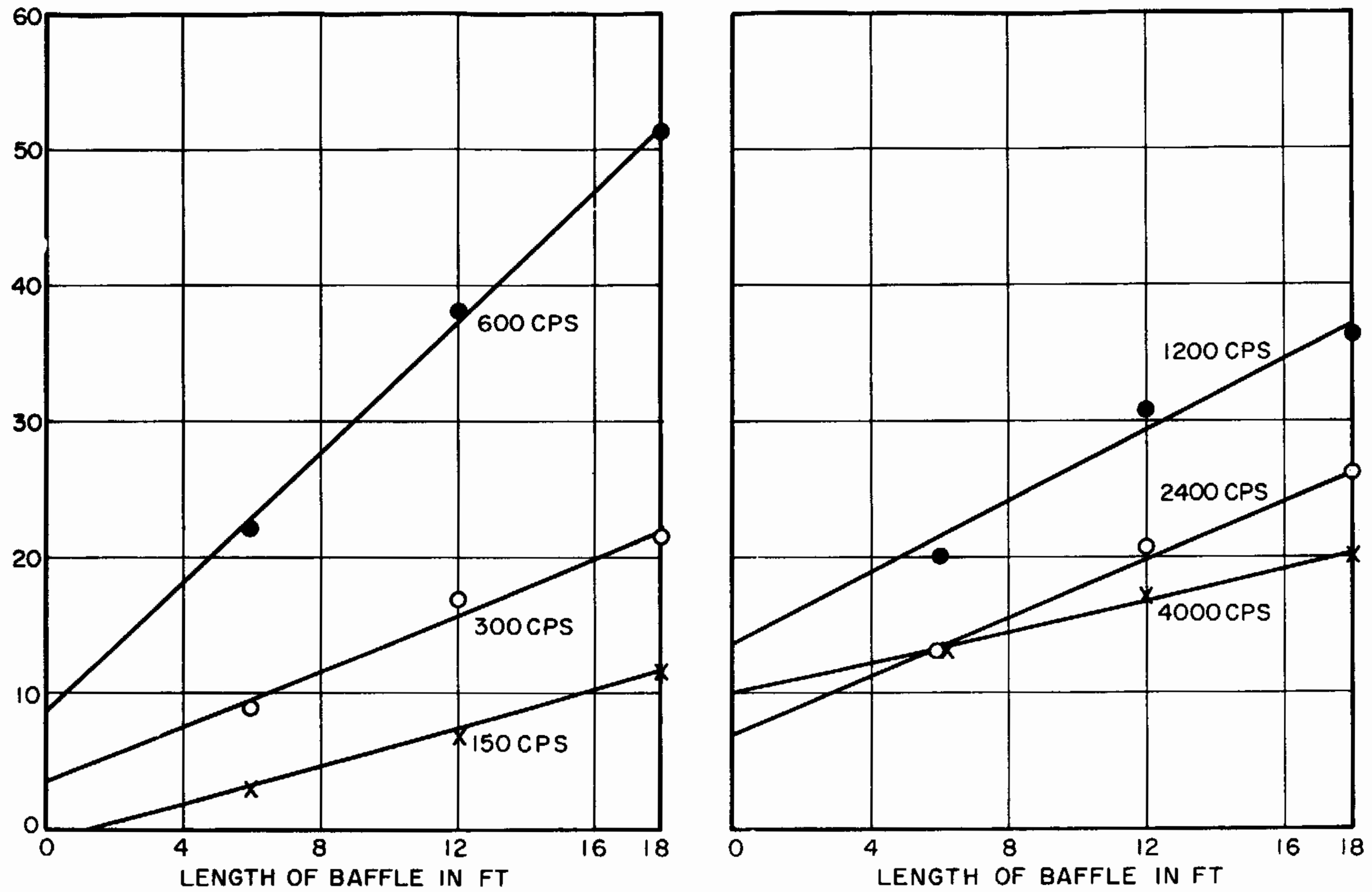


FIG. 29 NOISE REDUCTION VS LENGTH FOR $3\frac{1}{2}$ IN. THICK PARALLEL BAFFLES SPACED ON 16 IN. CENTERS

4. Summary

1. The noise reduction per unit distance does not adequately describe the noise reduction characteristics of baffles and ducts, except for a plane wave travelling parallel to the axis of the duct or baffle. For other inputs, with sound energy at several angles of incidence to the duct, the slope of an SPL vs. length curve varies with length. Hence, noise reduction per unit length is a function of length. The noise reduction per unit length approaches a constant value which should be the value obtained for plane waves. As a result, it is possible to express the noise reduction for a length ℓ, of the treatment as $(a + b\ \ell)$ db, where a is the intercept of the "linearized" SPL vs. ℓ curve at $\ell = o$, and b is the slope of the same curve. Obviously, there is no noise reduction for $\ell = o$ and there is a value of ℓ, below which the expression $(a + b\ \ell)$ is not valid. This value of "ℓ" can be determined from the SPL vs. distance curve.

2. As demonstrated by several examples, a noise reduction of the form $b\ell$ $(a = o)$, determined from measurements at the input and at the output of a finite length, ℓ_o, of the treatment will always yield a noise reduction which is too large for lengths of treatment greater than ℓ_o, and too low for lengths less than ℓ_o.

3. The values of a will depend both on the particular treatment involved and on the distribution of the incident sound energy as a function of angle of incidence. Since b should be equal to the noise reduction for plane waves parallel to the axis of the duct or baffle, b should not depend upon the input. Furthermore, b should approximate the value predicted by Morse and Cremer.

B. Noise Reduction by Thin Parallel Baffles

In the acoustical surveys carried out under the Air Force program, only two general classes of parallel baffles were encountered: (1) thin baffles, usually about 4 in. thick, spaced 12 in. on centers; and (2) thick baffles, between 2-1/2 and 4 ft thick, spaced 6 to 8 ft on centers. No baffles of intermediate dimensions were encountered. The noise reduction characteristics of the thin baffles are analyzed in this section. The noise reduction characteristics of thick baffles are analyzed in Section C.

1. Method of Analysis

The noise reduction characteristics of similar types of baffles, with similar types of noise input distribution, have been plotted as a function of length in order to find the constants a and b in the expression for noise reduction of a length ℓ:

$$L_{nr} = a + b\ \ell$$

Since the noise reduction depends upon the angle of incidence of the sound waves at the input to a treatment, the analysis is first carried out for those treatments with noise inputs which have similar noise distribution with respect to angle of incidence.

The estimation of the distribution of noise with respect to the angle of incidence is difficult. In general, one might expect that the input would be:

1. Random near the test section;
2. Somewhat biased just beyond a bend because of the higher order modes in certain directions;

3. Also somewhat biased if the input to the baffles is preceeded by other baffles which are oriented so that the planes of the two sets of baffles are parallel and;
4. Nearly plane if the input is preceeded by a long acoustical treatment, such as a duct which would suppress all the higher order modes.

Not enough data are available to investigate each class of input, (1) - (4) above, separately. However, only in case (4) should the noise input be nearly plane. This case is therefore not included in the initial analysis.

2. Analysis

The data included in the initial analysis were all derived from measurements of the noise reduction of similar parallel baffles. The parallel baffles included in this section were all 4 in. thick and spaced 12 in. on center. The baffles were covered with a perforated metal facing which enclosed about 4 in. of 4-1/4 lb/cu ft PF or TWF Fiberglas. The open area of the facings varied from about 20 to 35%.

The noise reduction, L_{nr}, for about ten sets of 4 in. thick parallel baffles spaced 12 in. on centers has been measured under the Air Force Program. Six of these baffles are included in a general class having more or less "random" inputs: two* 4 ft long,[23] and one 3,[19] one 6,[19] one 10[23] and one 12 ft long.[17]

* These data have been averaged to obtain a single L_{nr} for 4 ft baffle.

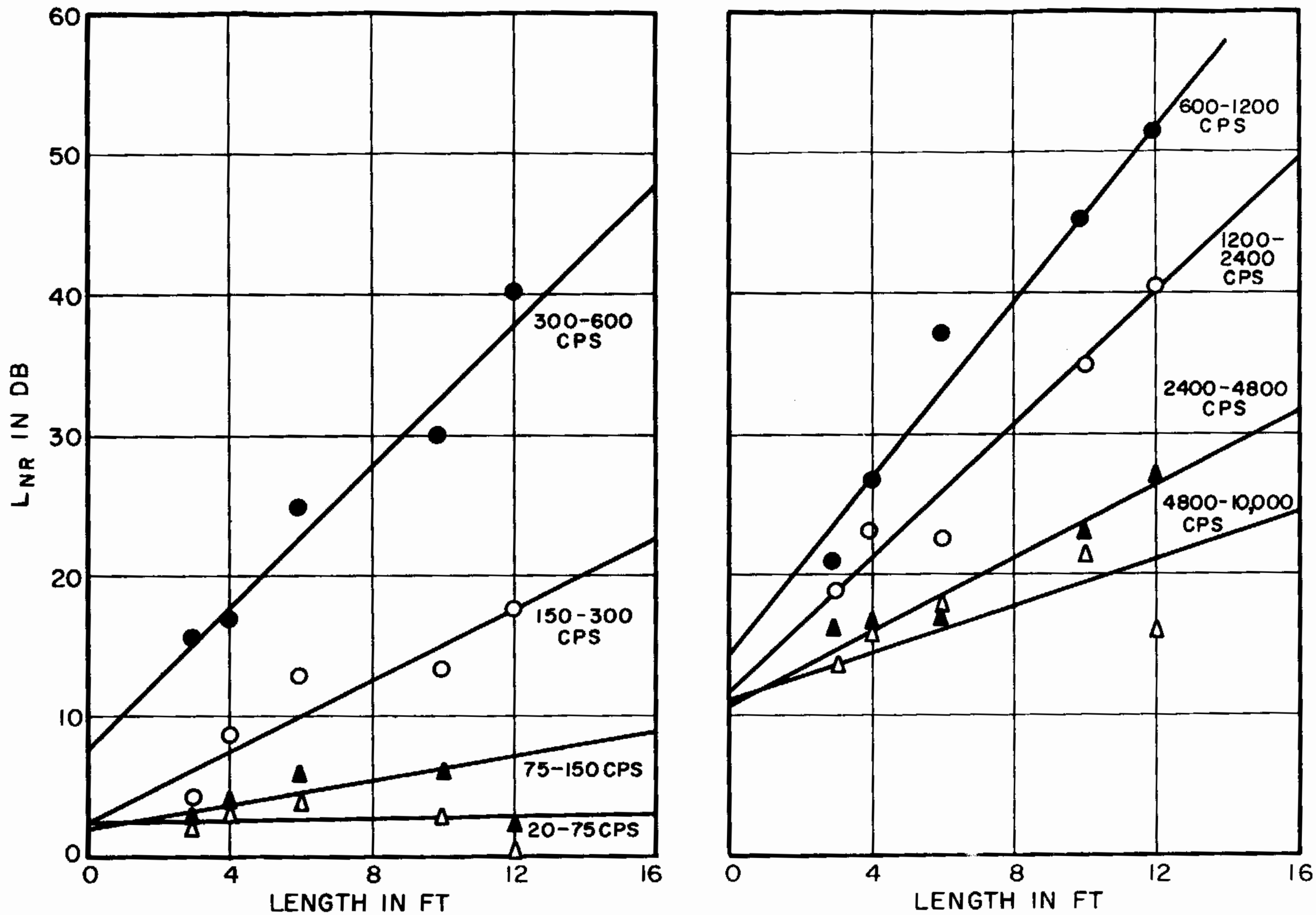


FIG. 30 NOISE REDUCTION AS A FUNCTION OF LENGTH FOR 4 IN. THICK BAFFLES SPACED ON 12 IN. CENTERS

To analyze these data, the one-third octave band data were averaged to obtain the octave band noise reduction for a +3 db/octave input slope (see Section III). The noise reductions for each octave band were then plotted as a function of the length of the baffle. These data are presented in Fig 30.

In the 20 to 75 cps band, the noise reduction appears to be nearly independent of length, and is on the average about 3 db. The 3 db noise reduction may result from reflection of energy towards the noise source because of the change in cross-section at the input and output of the baffles. Apparently, there is essentially no loss of energy as the wave propagates through the structure. The 3 db noise reduction appears to be significant; that is, it probably does not result from measurement error. The probability is less than 5% that the average value of six measurements of noise reduction would be 3 db if the true mean value were 0 db.

The data in the 75 to 150 cps band cannot be approximated very well with a straight line. Apparently, the noise reduction per unit length varies considerably with length in the range from 3 to 6 ft. The slope does not appear to approach a constant value until some length beyond 6 ft. The noise reduction in the range from 3 to 10 ft can be approximated by:

$$L_{nr} = 2 + 0.45\,\ell$$

where: ℓ is the length of the baffle in ft.

As frequency increases, the approximation of noise reduction with a straight line appears to become better, at least for lengths from 4 ft to 12 ft. The noise reduction for these

FIGURE 31

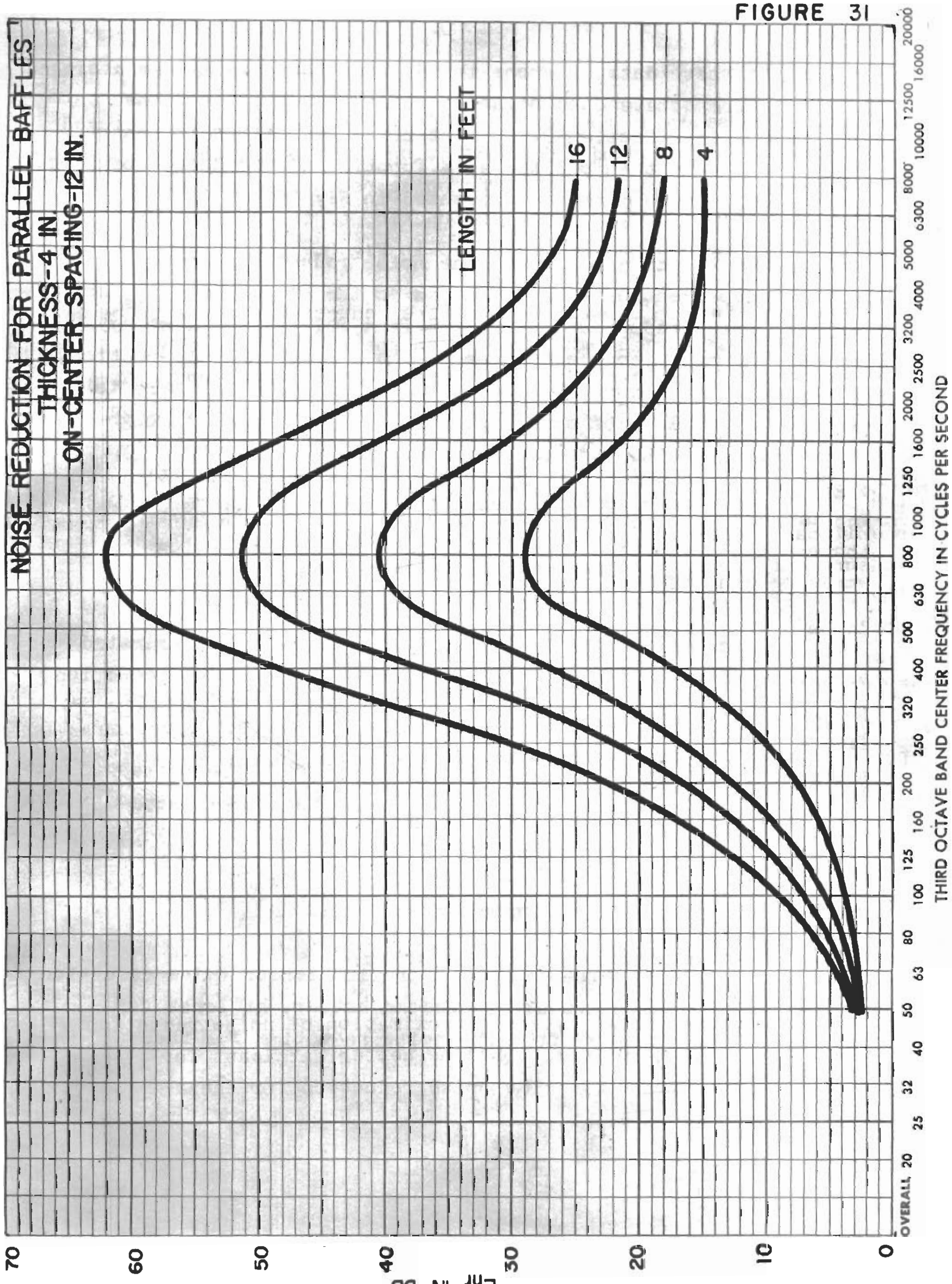
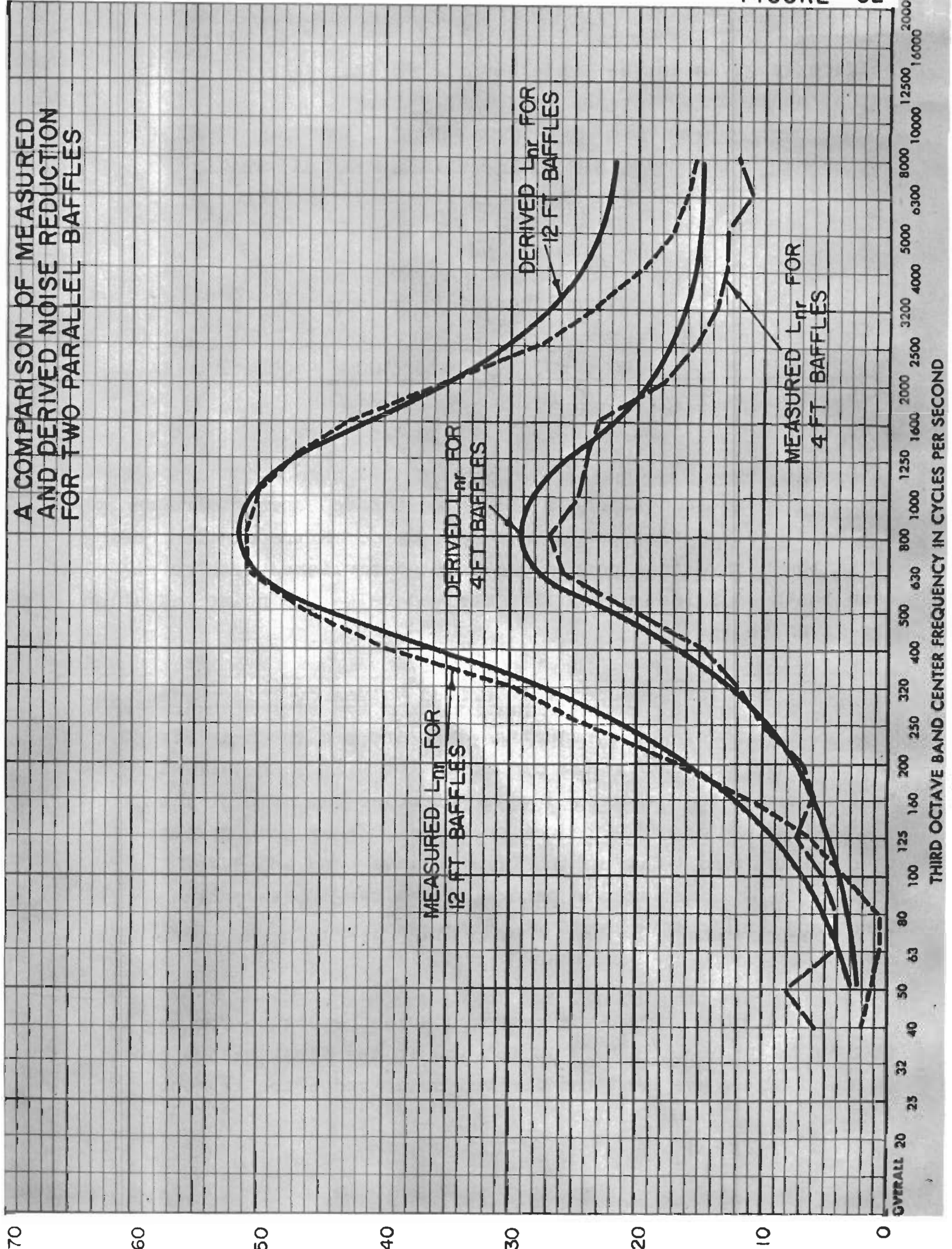

NOISE REDUCTION FOR PARALLEL BAFFLES
THICKNESS-4 IN.
ON-CENTER SPACING-12 IN.
LENGTH IN FEET
16
12
8
4
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL
20
25
32
40
50
63
80
100
125
160
200
250
320
400
500
630
800
1000
1250
1600
2000
2500
3200
4000
5000
6300
8000
10000
12500
16000
20000
0
10
20
30
40
50
60
70
Lnr IN DB

FIGURE 32



A COMPARISON OF MEASURED AND DERIVED NOISE REDUCTION FOR TWO PARALLEL BAFFLES
DERIVED L_{nr} FOR 12 FT BAFFLES
MEASURED L_{nr} FOR 12 FT BAFFLES
DERIVED L_{nr} FOR 4 FT BAFFLES
MEASURED L_{nr} FOR 4 FT BAFFLES
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
0 10 20 30 40 50 60 70
L_{nr} IN DB

baffles is given in the table below. The values given should not be applied for lengths less than 4 ft.

NOISE REDUCTION OF 4 IN. THICK PARALLEL BAFFLES, SPACED 12 IN. ON CENTERS FOR RANDOM INCIDENCE NOISE FIELD

$$L_{nr} = a + b\ell$$

Octave Bands of Frequency

L_{nr}	20 75	75 150	150 300	300 600	600 1200	1200 2400	2400 4800	4,800 10,000
a db	3	2	3	8	14	12	11	11
b db/ft	0	0.45	1.2	2.5	3.1	2.3	2.3	0.8

The noise reduction for one-third octave bands of frequency has been derived from the octave band data. Extrapolation between the octave band points has been accomplished with reference to the original data. The derived values of noise reduction are given in Fig 31.

3. Comparison of Measured and Derived Values of Noise Reduction

Four sets of noise reduction data are presented in Fig 32. The measured value of noise reduction for 4 ft long baffles is compared with the derived value of noise reduction. The measured value of noise reduction is as much as 5 db greater than the derived value in the first octave band and 4 db greater in the last octave band. In the range between the first and last octave bands the difference between the two curves is generally less than 3 db.

On the other hand, the measured noise reduction for the 12 ft baffles is about 3 to 5 db too low in the first octave band. In the last octave band the derived noise reduction is about

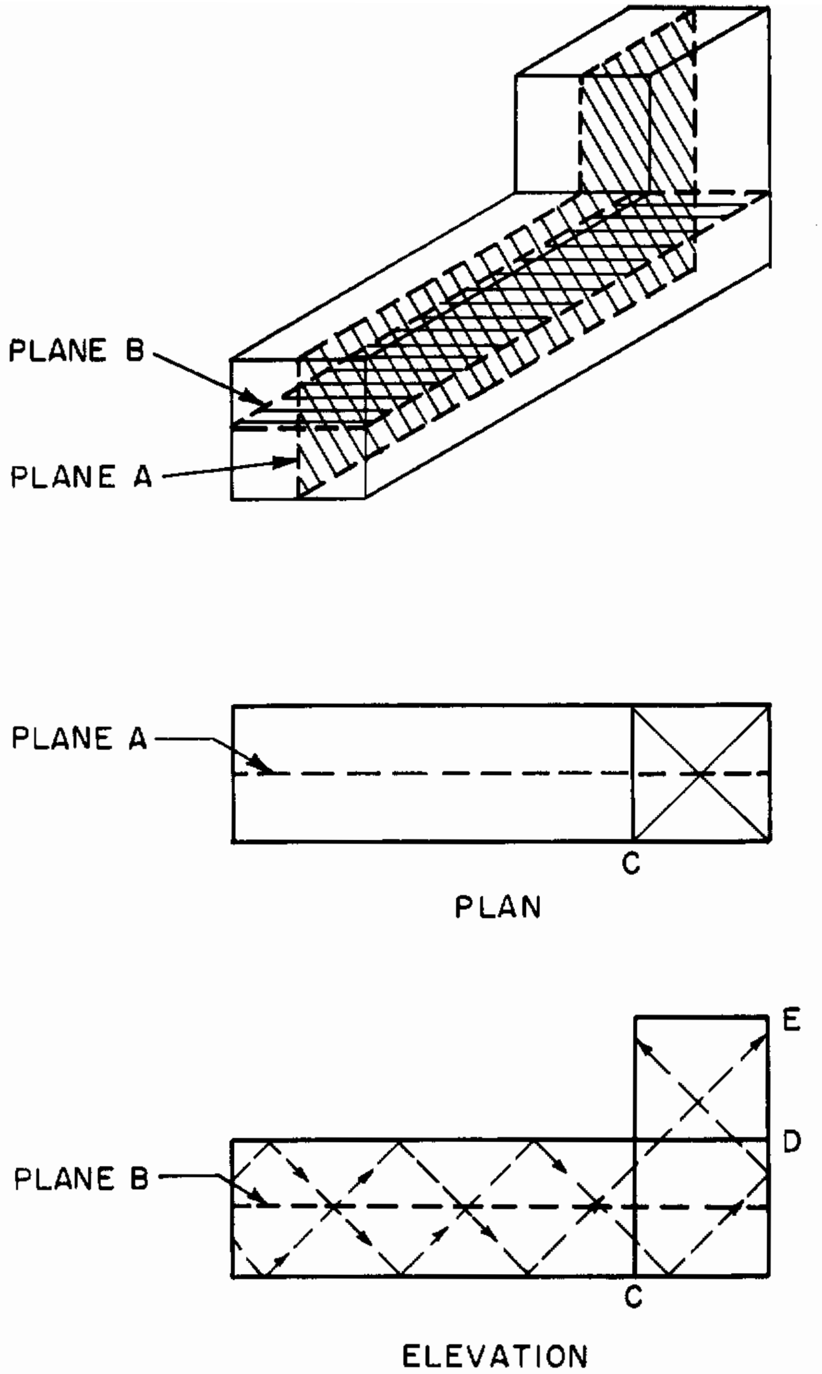


FIG 33 NOMENCLATURE AND A PLAN AND SECTION FOR A TYPICAL BEND

FIGURE 34

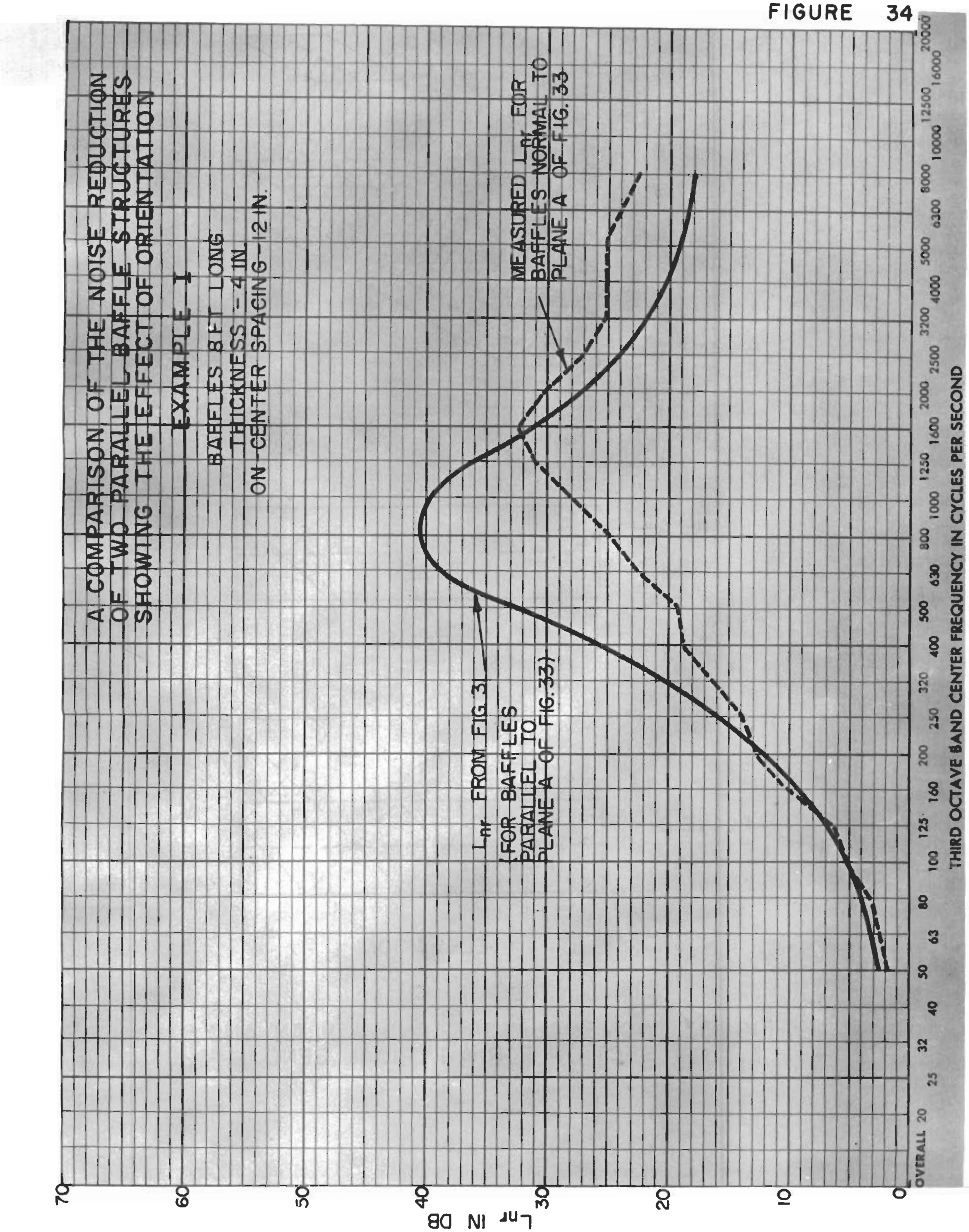
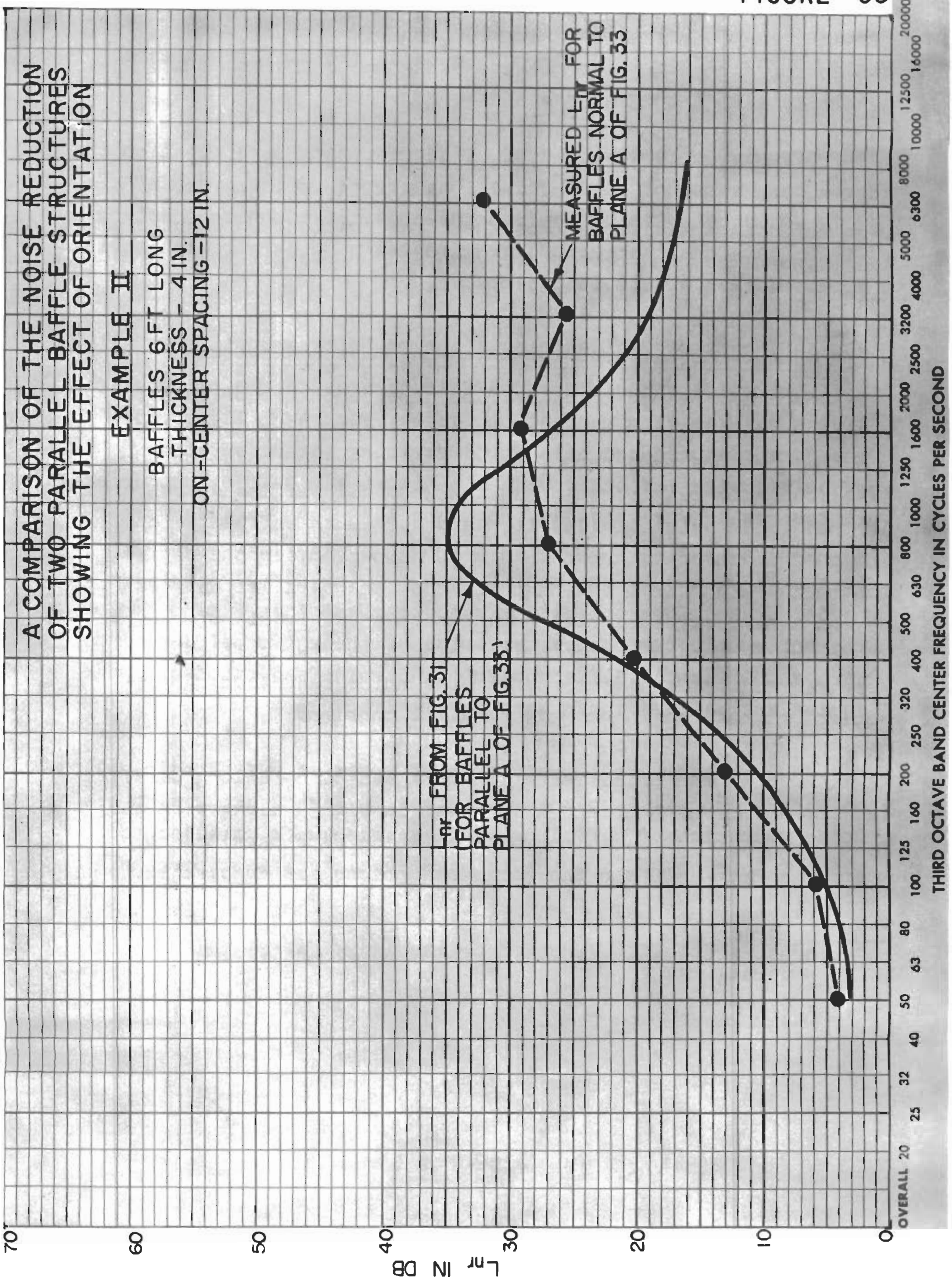

A COMPARISON OF THE NOISE REDUCTION
OF TWO PARALLEL BAFFLE STRUCTURES
SHOWING THE EFFECT OF ORIENTATION
EXAMPLE I
BAFFLES 8 FT LONG
THICKNESS - 4 IN.
ON CENTER SPACING - 12 IN.
MEASURED L_{nr} FOR
BAFFLES NORMAL TO
PLANE A OF FIG. 33
L_{nr} FROM FIG 31
(FOR BAFFLES
PARALLEL TO
PLANE A OF FIG. 33)
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
70 60 50 40 30 20 10 0
L_{nr} IN DB

FIGURE 35


A COMPARISON OF THE NOISE REDUCTION
OF TWO PARALLEL BAFFLE STRUCTURES
SHOWING THE EFFECT OF ORIENTATION
EXAMPLE II
BAFFLES 6FT LONG
THICKNESS - 4 IN.
ON-CENTER SPACING - 12 IN.
Lnr FROM FIG. 31
(FOR BAFFLES
PARALLEL TO
PLANE A OF FIG. 33)
MEASURED Lnr FOR
BAFFLES NORMAL TO
PLANE A OF FIG. 33
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
0 10 20 30 40 50 60 70
Lnr IN DB

4-6 db too high. In other octave bands the differences are generally less than 3 db.

If the derived values of noise reduction were equal to the true mean value of noise reduction, errors of the order of 2 db would be anticipated from the considerations in Chapter III. Therefore, the errors are generally no larger than those expected in a statistical sense.

4. Influence of Baffle Orientation on Noise Reduction

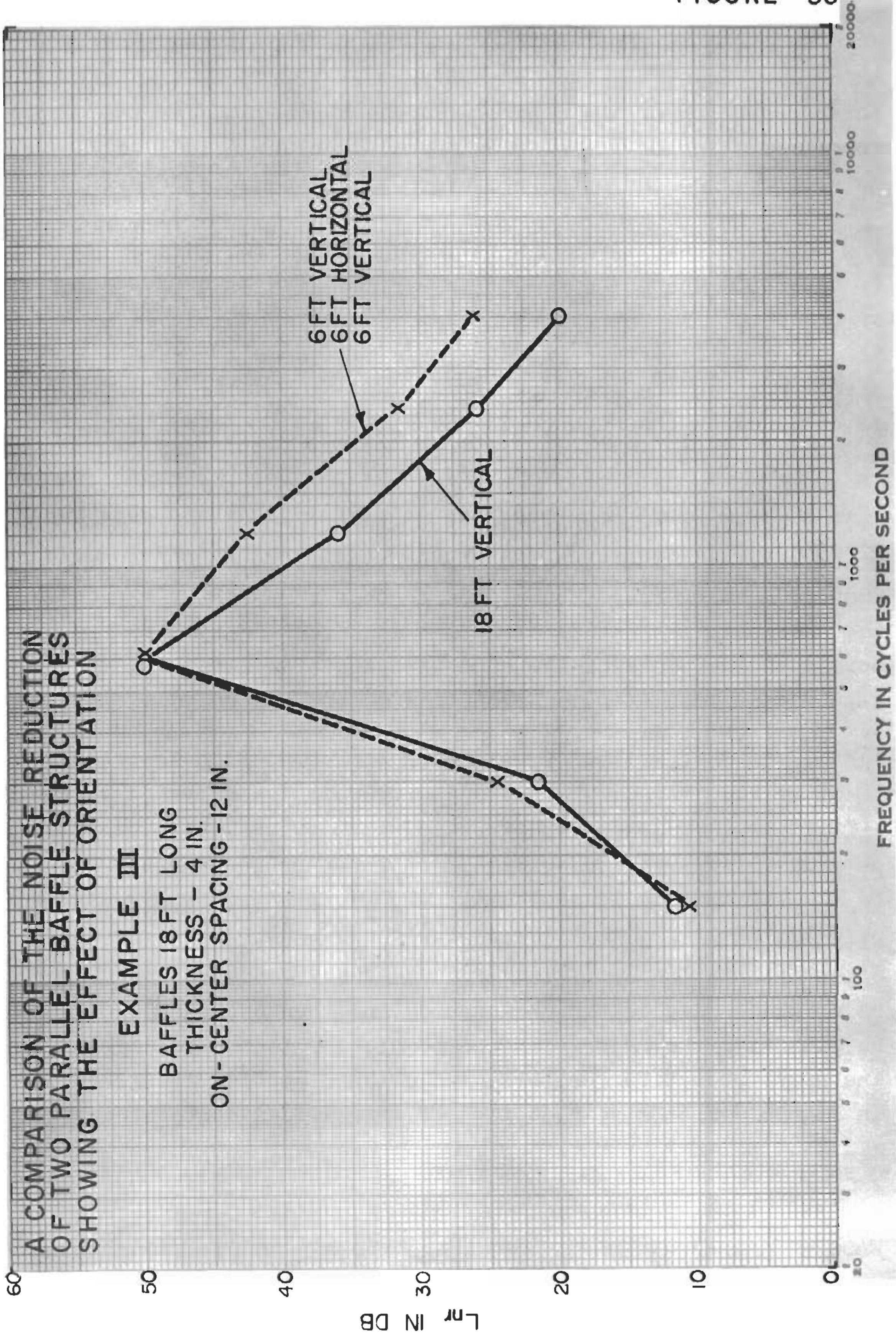
The noise reduction of baffles depends on the orientation of the baffles with respect to bends. Figure 33 illustrates the nomenclature used here for describing the orientation of baffle with respect to bends. Almost all baffles encountered in test cells were parallel to plane A.* Such baffles were included in the data shown in Fig 30 However, in one engine test cell[22] and in one wind tunnel,** the parallel baffles beyond a bend were oriented normal to plane A in Fig 33. The noise reduction characteristics of these two baffles were significantly different from the noise reduction of equivalent baffles which were parallel to plane A.

The measured noise reduction of a set of 8 ft long parallel baffles[22] normal to plane A is given in Fig 34 along with the noise reduction of 8 ft long baffles which were parallel to plane B. Below 200 cps the noise reductions are comparable.

* This orientation is dictated primarily by aerodynamic considerations.

** Private communication from W. J. Galloway. Data obtained from wind tunnel at Convair, San Diego.

FIGURE 36


A COMPARISON OF THE NOISE REDUCTION
OF TWO PARALLEL BAFFLE STRUCTURES
SHOWING THE EFFECT OF ORIENTATION
EXAMPLE III
BAFFLES 18 FT LONG
THICKNESS - 4 IN.
ON-CENTER SPACING - 12 IN.
6 FT VERTICAL
6 FT HORIZONTAL
6 FT VERTICAL
18 FT VERTICAL
FREQUENCY IN CYCLES PER SECOND
20
100
1000
10000
20000
L_{nr} IN DB
0
10
20
30
40
50
60

From 200 to 1600 cps the noise reduction of the baffles which are normal to plane A is significantly lower than the noise reduction of the other baffles. Above 1600 cps the noise reduction of the baffles normal to plane A is 3 to 7 db greater than the noise reduction of the other baffles.

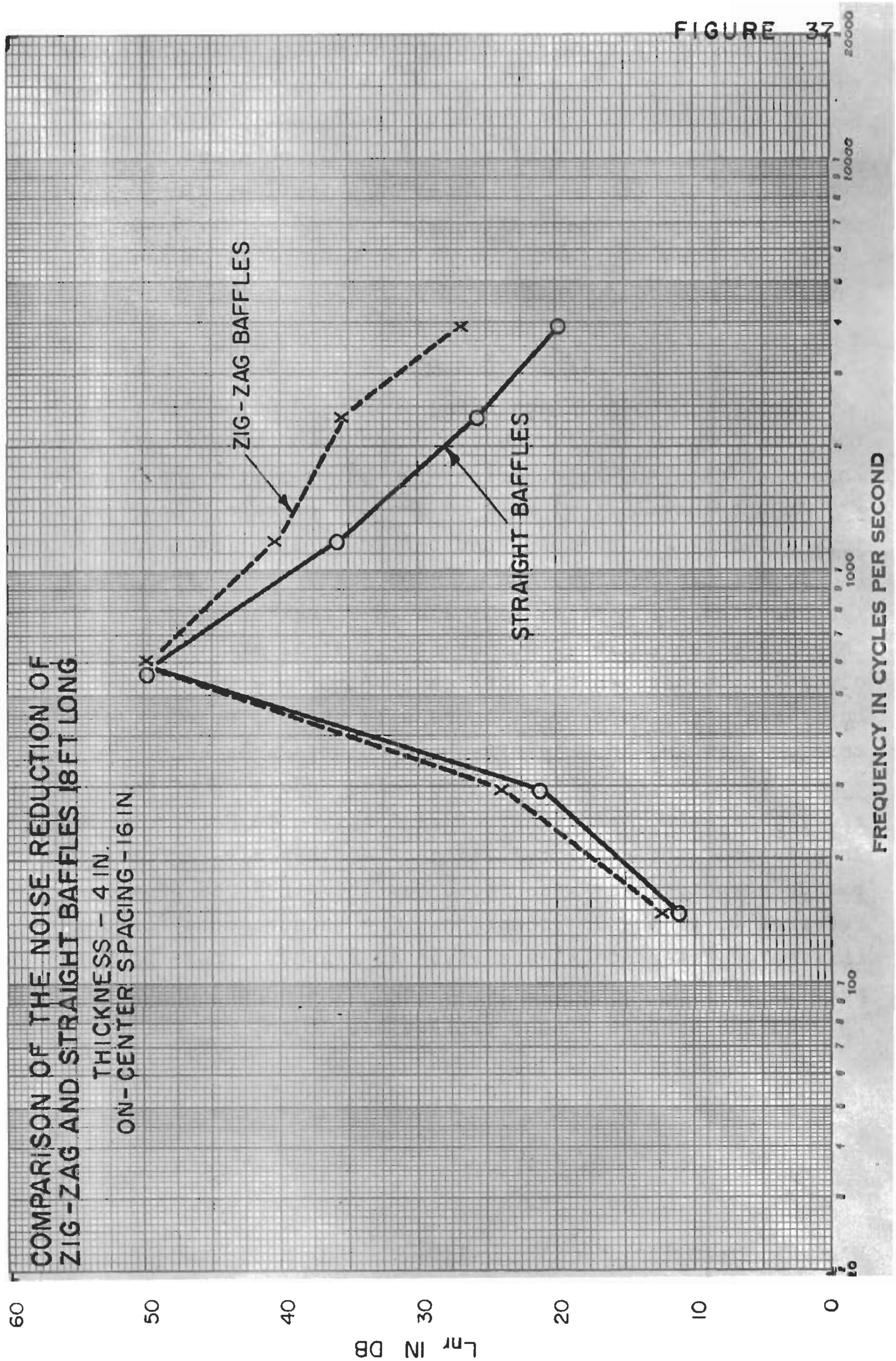
Similar results are shown in Fig 35 for two sets of 6 ft baffles. The noise reduction was measured in octave bands of frequency*; the values between the octave band center frequencies are interpolated. These data show the same differences as the previous data. Increased noise reduction is obtained in the high frequency region at the expense of a decreasing noise reduction in the mid frequency range.

No attempt has been made to generalize these data for other lengths. If increased high frequency attenuation is required, staggering the baffles so that they form a "zig-zag" air path is just as effective as orienting the baffles normal to the plane of the bend. In addition, there is no significant loss of noise reduction at the mid frequencies.

Another method of obtaining increased high frequency noise reduction, by varying the orientation of baffles, was investigated in Reference 33. The noise reduction of 18 ft of parallel baffles, oriented so that the major plane of the baffles was perpendicular to the ground, was compared with the noise reduction with another orientation. The other orientation consisted of 6 ft of baffles with their major plane parallel to the ground, and then 6 ft of baffles with the major plane perpendicular to the ground. The noise reduction for these two sets of baffles are given in Fig 36. The noise reduction of the baffles containing

* Private communication from W.J. Galloway. Data obtained from wind tunnel at Convair, San Diego.

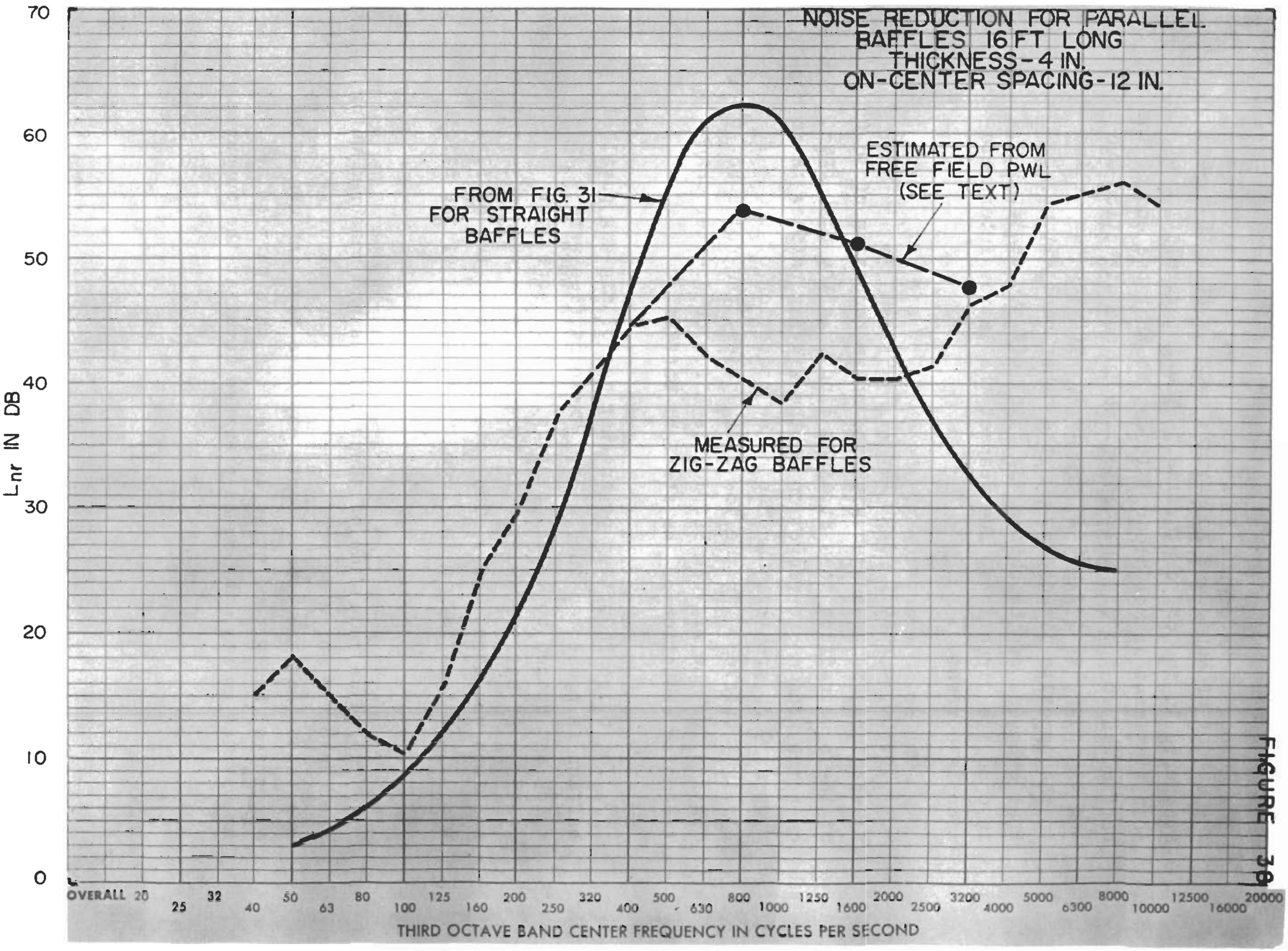
FIGURE 37


COMPARISON OF THE NOISE REDUCTION OF
ZIG-ZAG AND STRAIGHT BAFFLES 18 FT LONG
THICKNESS - 4 IN.
ON-CENTER SPACING - 16 IN.
ZIG-ZAG BAFFLES
STRAIGHT BAFFLES
FREQUENCY IN CYCLES PER SECOND
100
1000
10000
20000
60
50
40
30
20
10
0
L_{nr} IN DB

the horizontal section is about 5 db greater than the noise reduction of the baffles which were all vertical. The noise reductions below 600 cps are not significantly different. The increased noise reduction is probably obtained by suppression of higher order modes in a vertical plane. These modes are not completely attenuated by the vertical baffle structure. The reorientation of the baffles appears to suppress these components.

5. Noise Reduction of Zig-Zag Baffles

Another method of achieving increased noise reduction in the high frequencies (those for which the duct width is greater than a wavelength) is to orient the baffles at a small angle $\pm \theta$ to the longitudinal axis of the air passage. Near the test section, for example, the baffles will be at an angle $\div \theta$ to the center line, and at some distance, ℓ, from the test section the baffles will change direction so that they are at an angle $-\theta$ to the longitudinal axis. The included angle between the baffles is $(180^{\circ} - 2\theta)$. If the baffles are angled enough, there will be no line of sight through them and the high frequency sound waves cannot "beam" through the baffles. The noise reduction of one zig-zag parallel baffle structure is given in Reference 34. In addition, the noise reduction of another zig-zag parallel baffle structure was measured under the Air Force program referred to in the Introduction. The measurements for the baffles of Reference 33 are given in Fig 37. These baffles were 18 ft long and 3-1/2 in. thick. The normal distance between the centers of the baffles was 16 in.



NOISE REDUCTION FOR PARALLEL BAFFLES 16 FT LONG THICKNESS-4 IN. ON-CENTER SPACING-12 IN.
ESTIMATED FROM FREE FIELD PWL (SEE TEXT)
FROM FIG. 31 FOR STRAIGHT BAFFLES
MEASURED FOR ZIG-ZAG BAFFLES
L_{nr} IN DB
0
10
20
30
40
50
60
70
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND

FIGURE 38

The author of Reference 33 states "each (6 ft baffle was) staggered 16 in. off centerline." The interpretation of this statement is not clear. It may be that one end of the baffle was 16 in. off the centerline in one direction and the other end of the baffle was 16 in. off the centerline in the opposite direction. On the other hand, it may be that each end of the baffle was 8 in. off the centerline in opposite directions. We suspect the latter interpretation is correct.

The noise reduction of the 18 ft of zig-zag (three 6 ft sections) is presented with the noise reduction of 18 ft of the same baffles in a straight line. In the high frequencies, the noise reduction of the zig-zag baffles is as much as 9 db greater than the noise reduction of the straight baffles. In the low frequencies, the noise reduction for the zig-zag baffles is a few db greater than the noise reduction of the straight baffles.

The noise reduction of another set of zig-zag parallel baffles is presented in Fig 38. These baffles were 4 in. thick and spaced on 12 in. centers. There were four sections of baffles and the included angle between adjacent sections was about 152°. The total offset of the baffles was 12 in. that is, ± 6 in. from the mid-point of the baffle section.[21]

The noise reduction of these baffles was measured using the explosive noise source. The SPL at the output grid suggested that the measurements may have been influenced by acoustical background noise levels. In this particular acoustical survey, the sound pressure level at the output grid was also measured during engine operation. The free field power level and the measurements at the output grid* have been used to estimate the noise reduction in the frequency range from 800 to 2000 cycles. The noise reduction estimated from the free field PWL and the SPL during engine operation is also given in Fig 38.

The noise reduction of these baffles is generally greater than the noise reduction of 16 ft of straight parallel baffles except in the mid-frequency range where the noise reduction has been estimated from the free-field power level of the engine. In general, however, angling the baffles provides additional noise reduction, particularly in the high frequencies.

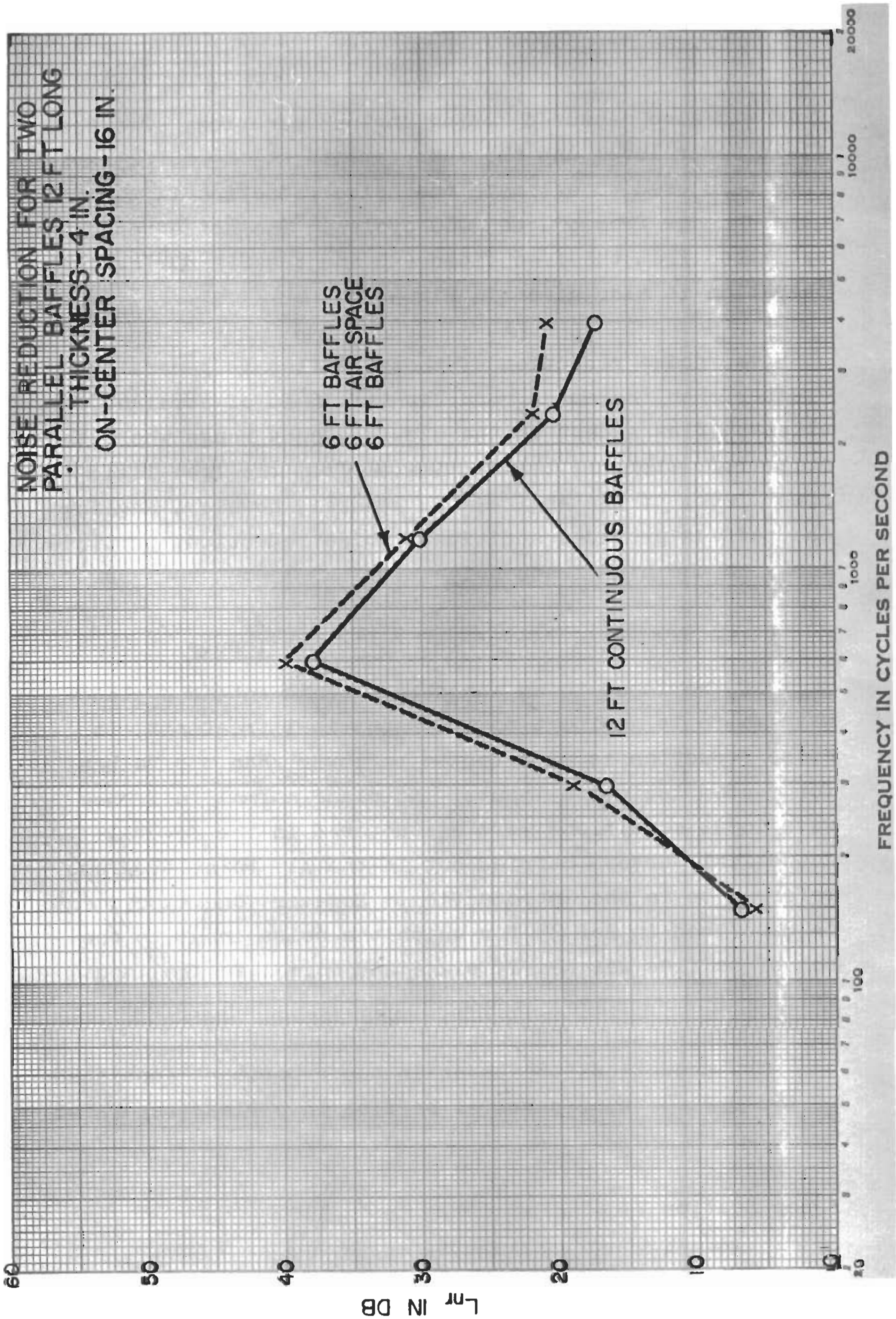
6. Noise Reduction for Normal Incidence Inputs

If the input to an acoustical treatment is a plane wave which is incident on the acoustical treatment, the noise reduction is expected to be lower than the values for random incidence inputs. No plane normal incidence wave input conditions were encountered in the several acoustical surveys carried out in the Air Force programs referred to in the Introduction. However, one experiment is described in Ref 33 for which the noise input to one set of baffles was nearly plane and normal to the input grid.

The acoustical treatment in the experiment consisted of two 6 ft sets of parallel baffles in series, separated

* See Section VI

FIGURE 39


NOISE REDUCTION FOR TWO
PARALLEL BAFFLES 12 FT LONG
THICKNESS-4 IN.
ON-CENTER SPACING-16 IN.
6 FT BAFFLES
6 FT AIR SPACE
6 FT BAFFLES
12 FT CONTINUOUS BAFFLES
FREQUENCY IN CYCLES PER SECOND
L_{nr} IN DB
0
10
20
30
40
50
60
20
100
1000
10000
20000

by an open air space 6 ft long. The major planes of the two sets of baffles were parallel to one another.

The noise input to the first 6 ft of baffles in the combination was random; however, the higher order modes were attenuated rapidly in the first 6 ft baffles so that the input to the second 6 ft of baffles was nearly plane and normal.

The noise reduction for this combination of baffles is given in Fig 39 along with the noise reduction for 12 ft of the same type of baffles, which consisted of two contiguous 6 ft sections. There is no significant difference between the noise reduction of these two baffle structures.

The noise reduction of the second 6 ft of baffles is just about equal to the difference between the noise reduction of 12 ft of continuous baffles and the noise reduction of 6 ft of continuous baffles. Therefore, the noise reduction for plane waves is given by the slope (from 6 ft to 12 ft in Fig 39) of the noise reduction curves as was suggested earlier. For plane waves, therefore, the noise reduction varies in direct proportion to length. In the general expression for noise reduction, $a + b\ell$, a is zero for normal, plane waves. If the input to the second 6 ft of baffles were random, the total noise reduction for the combination of baffles would be $2(a + 6b)$, a value which would be much greater than the noise reduction for 12 ft of contiguous baffles.

Thus, these data appear to indicate that the noise reduction for plane waves will be given by the slope of the noise reduction curves as obtained from measurements with <u>any</u> type noise field input, provided the measurements are made far enough into the baffle structure. All higher order modes in the baffles will decay faster than the lowest order mode and the noise reduction per unit distance will approach the value for the lowest mode.

7. <u>Comparison of Measured Noise Reduction with the Theory of Morse and Cremer</u>

The noise reduction measured for a random input should be greater than the noise reduction for a plane wave which is normal to the input grid. However, the noise reduction per unit length at some large distance into the treatment should approximate the noise reduction for the lowest order modes. Therefore, the slope of the noise reduction curves in Fig 30 should be approximately equal to the noise reduction predicted by Morse and Cremer.

Calculation of the impedance of the baffle structure is difficult because the parallel baffles did not contain a dividing septum. The energy flow into a baffle depends upon the sound pressure distribution on either side of the baffle. If the sound pressures on the two sides of the baffle are in phase, the particle velocity in the middle of the baffle is zero and the impedance is the same as if a septum were present. If the sound pressures are not in phase, the results of Morse and Cremer do not apply. The impedance of the 4 in. thick baffles with 12 in. on-center spacings has been calculated assuming the sound pressures on either side of the baffle to be in phase. Thus the 4 in. baffles are equivalent to a duct lined with two inches of acoustical material.

The noise reductions have been obtained from the impedance by use of Cremer's charts. The calculated noise reduction and the measured slope of the noise reduction curves of Fig 30 are given in the table below for several values of frequency.

PLANE WAVE NOISE REDUCTION OF 4 INCH THICK
PARALLEL BAFFLES SPACED 12 INCHES ON CENTERS

Frequency	Noise Reduction in db/ft Measured	Noise Reduction in db/ft Calculated
100 cps	0.4	0.075
410 cps	2.4	1.12
820 cps	3.1	2.85
1640 cps	2.4	4.5
6300 cps	0.8	1.42

The spectra of the calculated and measured noise reductions have the same general shape, but the maximum value for the measured noise reduction is about one octave lower than the maximum for the calculated noise reduction. Except for the two highest frequencies, the measured value of noise reduction is generally higher than the calculated value.

It is to be noted that the measured noise reductions are derived from octave band data. The noise reduction at any frequency may vary significantly from the octave band value, particularly if there is a narrow peak in the noise reduction spectrum. Thus, the significance of the difference in the measured and calculated peak values of noise reduction (3.1 and 4.5 db/ft respectively) is obscured.

C. Noise Reduction by Thick Parallel Baffles

The noise reduction, L_{nr}, of six sets of thick parallel baffles were measured under the Air Force program. The thickness of these baffles varied from 3 to 4 feet. The open spacing between them varied from about 2 to 5 feet. The materials used in the baffles were the same as those used in the thin baffles.

The noise reduction for these baffles has been plotted as a function of ℓ/D'* for each of the eight octave bands of frequency. In Figures 40 and 41, the noise reduction of any of the six baffles lies within 3 db or less of the straight lines fitted on each graph with one exception. The noise reduction for $\ell/D' = 3.0$ in the 20 to 75 cps octave band is significantly low. The noise reduction versus length curve is not evidently linear below $\ell/D' = 4$. Therefore, the values of noise reduction derived from these curves should not be used for ℓ/D' values much less than 4 in that band.

The range of values of D' for the data presented is from 2 to 5 feet. Over this range the noise reduction spectrum did not appear to vary significantly. For example, the frequency of the maximum noise reduction for D' = 5 ft and for D' = 2.5 ft was about the same.

Figure 42 shows the derived values of noise reduction for parallel baffles 3 ft thick on 6 ft centers based on the data in Figs 40 and 41. Two observations may be made about this data. First, the noise reduction above 1000 cps may be

* The data at ℓ/D' equals 3, 4, 4.25, 4.8, 5 and 7.2 are from references 19, 23, 19, 22, 18, 22, respectively.

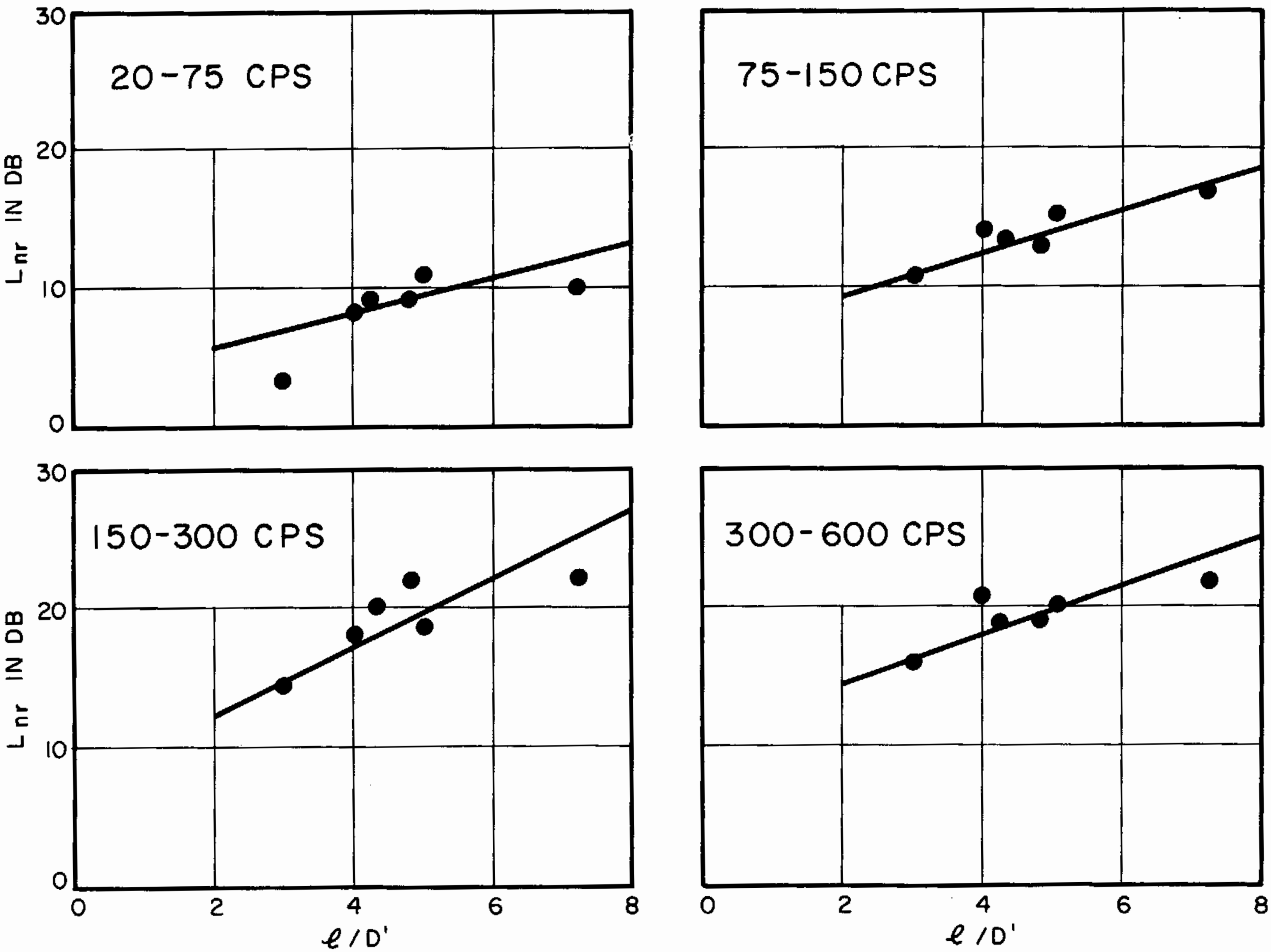


FIG. 40 NOISE REDUCTION VS ℓ / D' FOR THICK PARALLEL BAFFLES (SEE TEXT)

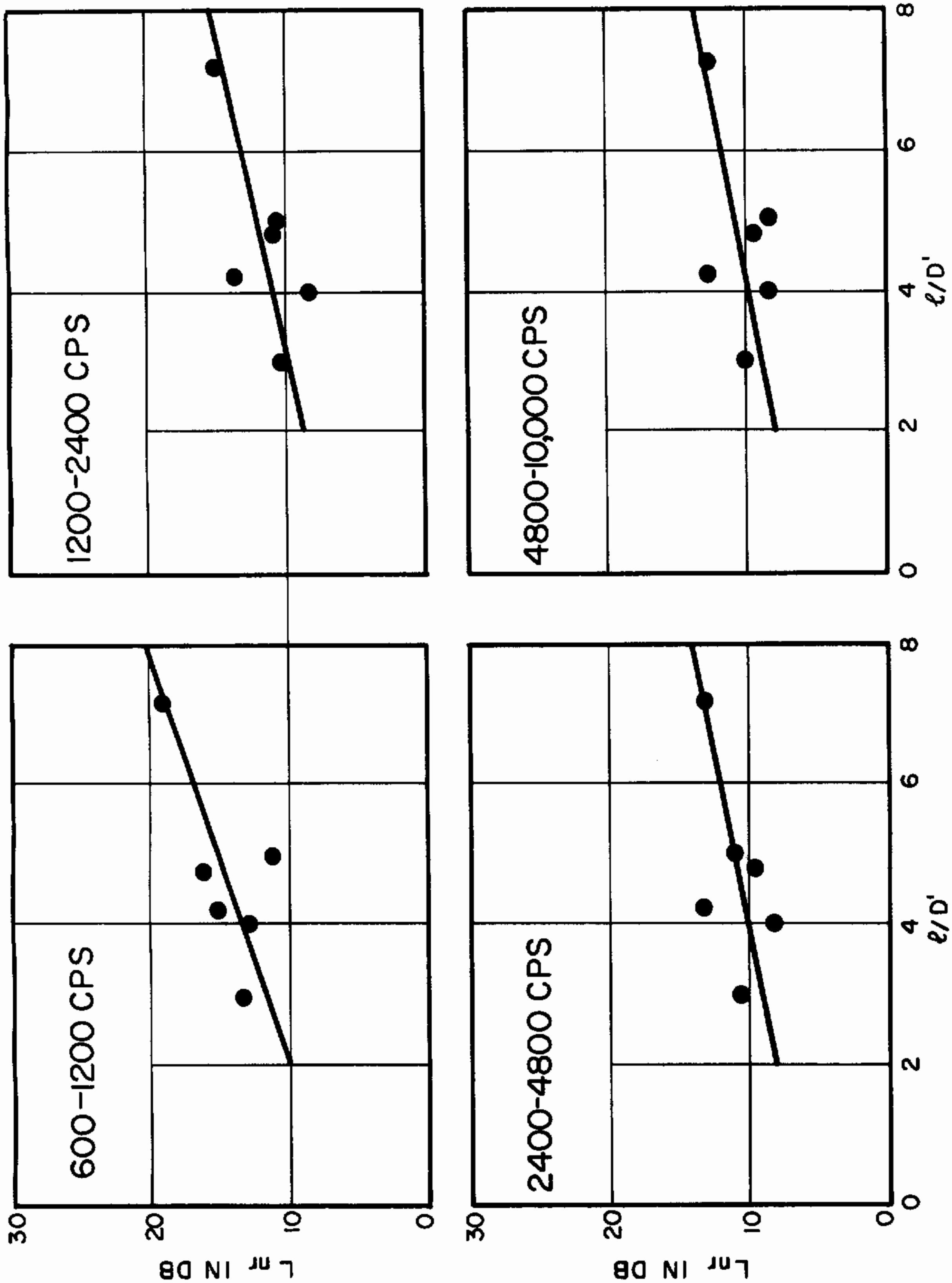
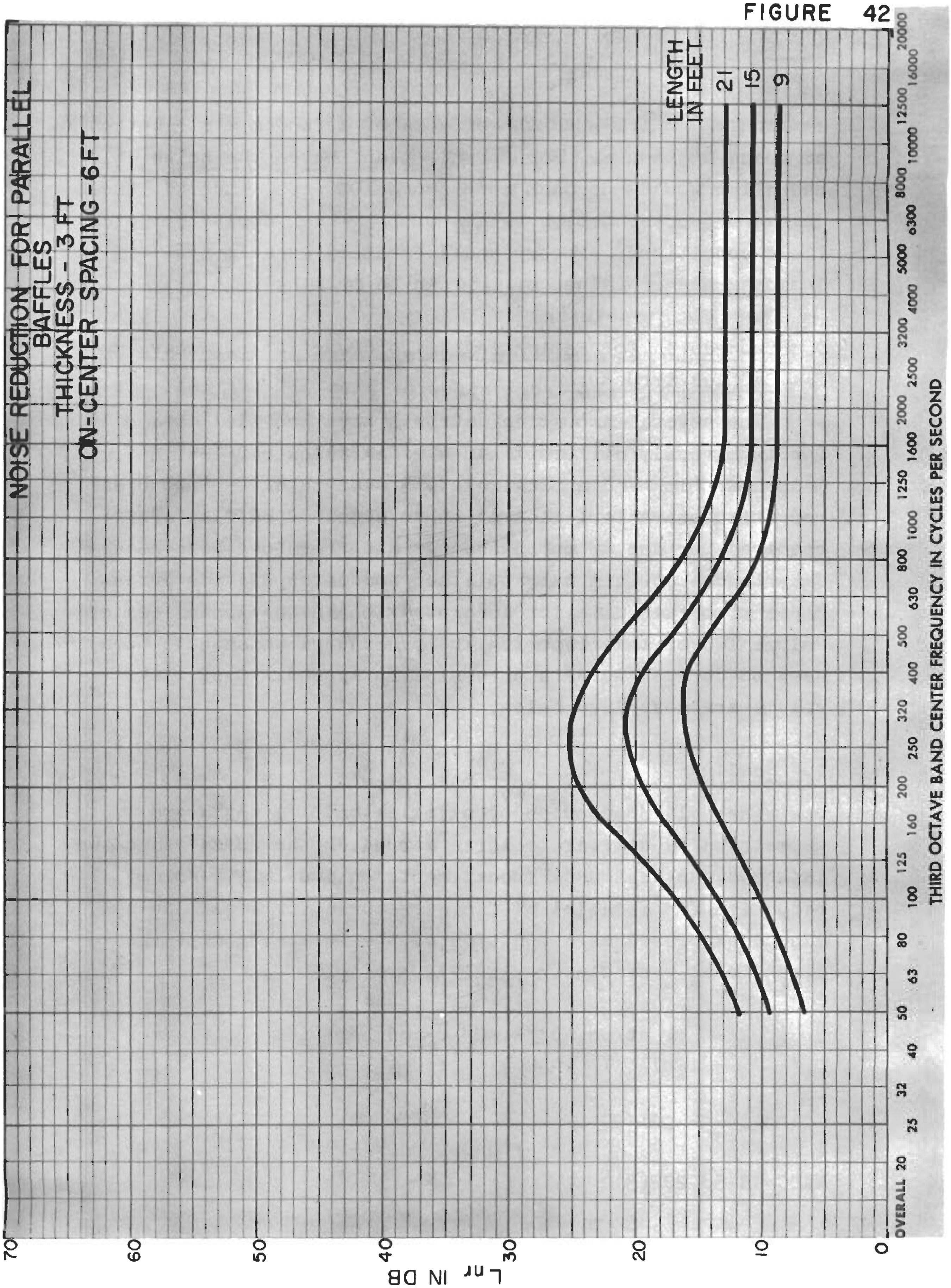


FIG. 41 NOISE REDUCTION VS ℓ/D' FOR THICK PARALLEL BAFFLES (SEE TEXT)

FIGURE 42


NOISE REDUCTION FOR PARALLEL BAFFLES
THICKNESS - 3 FT
ON-CENTER SPACING - 6 FT
LENGTH IN FEET
21
15
9
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
L_{nr} IN DB
0 10 20 30 40 50 60 70

attributed entirely to the effects of a random distribution of noise at inputs. For normal, plane waves the noise reduction would approach zero above 1000 cps. Second, these data are representative values of L_{nr} which apply to typical field conditions. Theory and laboratory experiments[37] indicate higher values of L_{nr} can be obtained.

D. Procedures for Estimating Noise Reduction of Other Baffles Structures

The several acoustical surveys have provided data only for two grossly different sizes of parallel baffles. (e.g., 4 inches thick and approximately 3 feet thick). The noise reduction characteristics of other sizes of baffles are required for design purposes. It has therefore been necessary to use certain extrapolation and interpolation procedures to obtain data for incorporation in the Appendices of Volume Two of this report. The procedures which have been used to derive the data presented in Appendix C of that volume are described below.

1. Scaling

The noise reduction of parallel baffles which are geometrically similar to those for which the noise reduction is known can be found by scaling techniques. The noise reduction for parallel baffle structures (or lined ducts) can be expressed as a function of the following non-dimensional variables:

$$rt/\rho c,\ D'/\lambda,\ H/\lambda,\ \text{and } L/D'$$

in which

r is the flow resistance per unit length of the acoustical material in the baffles,

t is the thickness of the baffles,

D' is the open width between the baffles,

λ is the wavelength of sound

H is the height of the baffles, and

L is the length of the baffles.

If each of these dimensionless variables are held constant, then the noise reduction will also remain constant. The variation of noise reduction with the height of the baffles is not generally included as a significant variable. For test cell structures, H does not vary greatly and the variation of H is neglected. It is noted here because higher order of modes in a vertical plane can propagate only for certain values of H/λ and the lowest frequency for which the end corrections are applicable depends both on D'/λ and on H/λ.

The noise reduction of a structure which is geometrically similar to one for which the noise reduction is known can be found by the following steps:

1) Scale all dimensions of the acoustical treatment for which the noise reduction is known to obtain the desired treatment that is geometrically similar.

For example, the noise reduction of 12' of baffles which are 4 in. thick and 12 in. on centers, could be obtained by multiplying all of the dimensions of 6' long, 2 in. thick baffles, 6 in. on center by 2. In this case, 2 is the <u>scale factor</u>.

2) Divide the frequency by the scale factor.

For the example being used, the peak noise reduction occurs at about 2000 cps for the 2 in. thick baffles. Thus, the peak noise reduction for the 4 in. thick baffles will occur at a frequency of 2,000/2 = 1000 cycles.

3) Divide the specific flow resistance (the flow resistance per unit length) of the acoustical lining material by the scale factor.

The total flow resistance is the product of the specific flow resistance and the thickness of the acoustical material. The thickness of the acoustical material is directly proportional to the scale factor and the specific flow resistance is inversely proportional to the scale factor. Thus, the total flow resistance is unchanged by the scaling procedure.

It is found from experience that the scaling procedure is only approximate. Therefore, it is desirable, where possible, to obtain the noise reduction of some unknown treatment by scaling down from a larger size and up from a smaller size.

2. Variation of Noise Reduction with Baffle Opening

In order to obtain a more complete set of data for Appendix C of Volume Two, it is necessary to use techniques other than scaling. It may be required, for example, to find the noise reduction of baffles which are 4 in. thick and 16 in. on centers from the noise reduction of baffles which are 4 in. thick and 12 in. on centers. Two such sets of baffles are not geometrically similar so that scaling techniques cannot be directly applied.

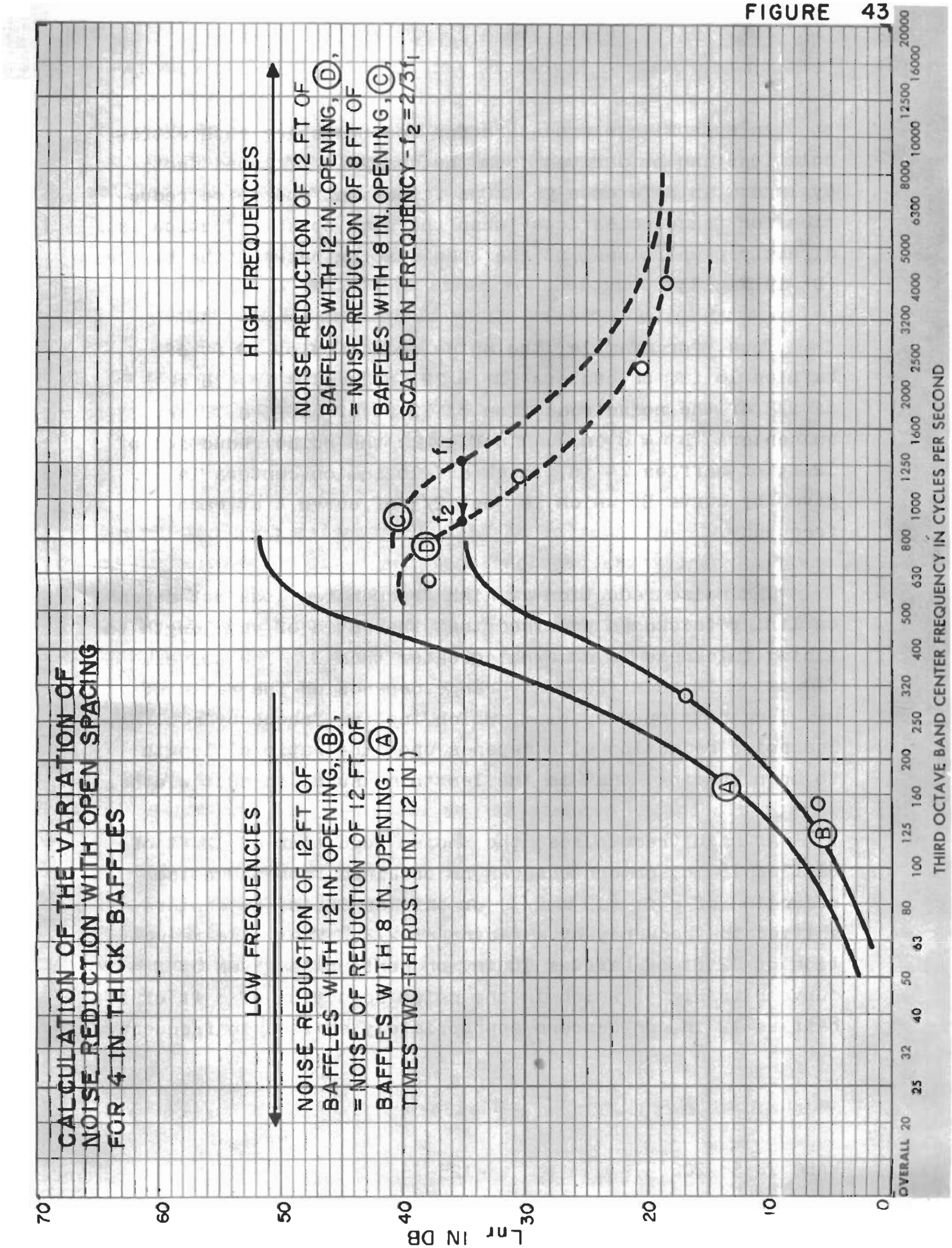
An approximate procedure for accomplishing such extrapolations can be derived from the analysis of lined ducts by Ingard in Reference 26. One finds that the noise reduction for frequencies lower than the peak noise reduction is directly proportional to the open spacing between the baffles (D' in Fig 33).

Thus, for example, the noise reduction of 12 ft of baffles 16 in. on centers (D' = 12 in.) and 4 in. thick is 2/3 of the noise reduction of 12 ft of baffles 12 in. on centers (D' = 8 in.). In Fig 42, the noise reduction of 12 ft of baffles, 4 in. thick and 16 in. on centers is given by curve B, which is just 2/3 of curve A at each frequency.

The noise reduction at high frequencies does not depend on lining thickness provided that the ratio of wavelength to lining thickness is somewhat greater than 1. The noise reduction in this frequency range depends on the ratio of wavelength to open spacing, which implies frequency scaling. The noise reduction also depends upon the ratio of length to open spacing, that is the length measured in duct widths.

At high frequencies, the noise reduction of 12 ft of baffles 16 in. on centers can be obtained from the noise reduction of 8 ft (12 x 2/3) of baffles 12 in. on centers shifted in frequency by a factor of 2/3. The noise reduction of 12 ft of baffles 16 in. on centers is given by curve D in Fig 43 which is the noise reduction of 8 ft of baffles 12 in. on centers appropriately shifted in frequency.

FIGURE 43


CALCULATION OF THE VARIATION OF NOISE REDUCTION WITH OPEN SPACING FOR 4 IN. THICK BAFFLES
LOW FREQUENCIES
HIGH FREQUENCIES
NOISE REDUCTION OF 12 FT OF BAFFLES WITH 12 IN. OPENING, B, = NOISE OF REDUCTION OF 12 FT OF BAFFLES WITH 8 IN. OPENING, A, TIMES TWO-THIRDS (8 IN./12 IN.)
NOISE REDUCTION OF 12 FT OF BAFFLES WITH 12 IN. OPENING, D, = NOISE REDUCTION OF 8 FT OF BAFFLES WITH 8 IN. OPENING, C, SCALED IN FREQUENCY - $f_2 = 2/3 f_1$
A
B
C
D
f_1
f_2
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
0 10 20 30 40 50 60 70
L_{nr} IN DB

As a test of the reliability of this method, the measured noise reduction for 12 ft of baffles 16 in. on centers (See Fig 29) is given by the open circles in Fig 43. As can be seen, the agreement is quite good. The author has also derived the noise reduction of 4 in. baffles 8 in. on centers by this procedure and has compared the results with recent data measured in England[35]. The results have been equally gratifying. However, it should be borne in mind that this procedure is approximate and the possibility of errors will increase with the range of extrapolation. It is not recommended, for example, that the noise reduction of baffles with 1 in. open spacing be derived from the noise reduction of baffles with 8 in. open spacing.

E. Noise Reduction by Lined Ducts

The noise reduction of six lined ducts was measured under the Air Force program. These lined ducts had nearly square openings which ranged from 6 ft x 6 ft to 10 ft x 10 ft. The lining of all but one of the ducts consisted of a Fiberglas blanket 4 to 6 in. thick, enclosed in perforated metal panels which were backed with an air space that varied in thickness from 1 to 2-1/2 ft. In one duct the Fiberglas was 2 ft thick and the air space was 1 ft deep. Three of the ducts[14(2), 23] had a single open area which filled the entire cross section of the cell. Three duct structures*[18, 21] consisted of four parallel ducts placed side by side in a square array. The ℓ/D' ratio for the ducts varied from 1.0 to 2.7.

* Two of the ducts were identical. The L_{nr} from these ducts have been averaged and are presented as a single point.

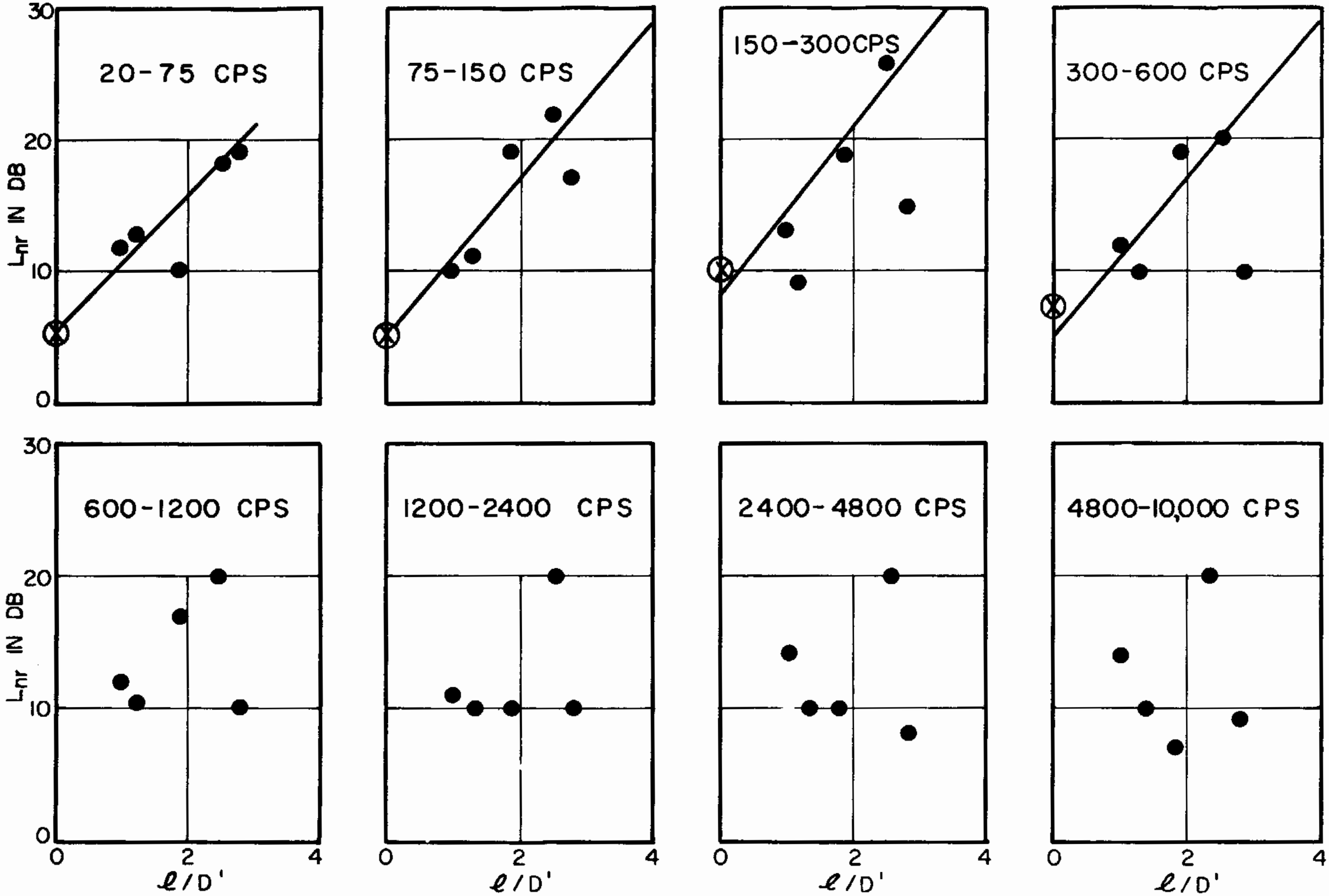


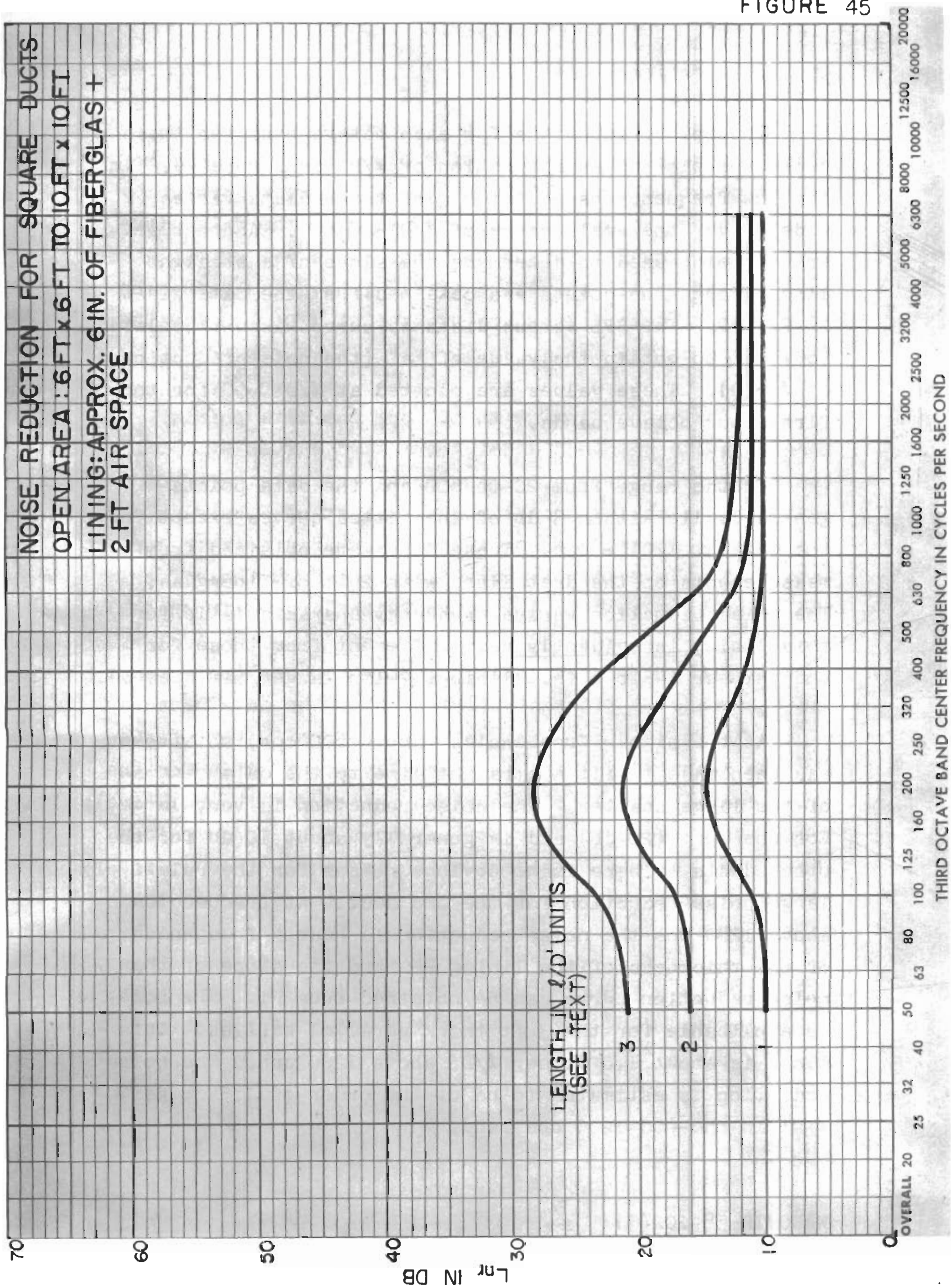
FIG. 44 NOISE REDUCTION VS ℓ/D' FOR FIVE DUCTS

The noise reductions for each octave band for these ducts are plotted as a function of ℓ/D' in Figure 44. In the low frequency range the L_{nr} of these ducts varies widely over the small range of ℓ/D' values. Another reference point used to determine the slope of a straight line through the points was obtained from the data given in Fig 42. The SPL versus distance curve was extrapolated to $\ell = 0$ to obtain the value of "a" (the noise reduction at $\ell = 0$). These values are plotted at $\ell/D' = 0$ for the first four octave bands.

In the range from 20 to 600 cps, the data points generally lie within 3 db of the fitted curves except for the data at $\ell/D' = 2.8$. These data were obtained from measurements on the duct lined with 2 ft of Fiberglas. The noise reduction versus frequency characteristic for this lining is evidently very different from those for the thinner (4 to 6 in.) linings (with larger air spaces).

At the higher frequencies (above 600 cps) no attempt has been made to fit a straight line to the data because of the large scatter. The noise reduction is very large for one duct (20 db) and is generally about 10 db for the other ducts. There is no obvious reason for the relatively large noise reduction for the one duct. At frequencies above 1200 cps the noise reduction is nearly independent of the thickness of the lining [26] and the slope of the noise reduction curve can be obtained from Fig 42 which is applicable for the 3 ft thick parallel baffles. This slope is about 1.0 db per ℓ/D' above 1200 cps. The noise reduction is estimated to be of the order of 1.5 db per ℓ/D' in the 600-1200 cps band.

FIGURE 45



NOISE REDUCTION FOR SQUARE DUCTS
OPEN AREA : 6 FT x 6 FT TO 10 FT x 10 FT
LINING: APPROX. 6 IN. OF FIBERGLAS + 2 FT AIR SPACE
LENGTH IN ℓ/D' UNITS (SEE TEXT)
3
2
1
70 60 50 40 30 20 10 0
L_{nr} IN DB
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND

The values of L_{nr} derived from Fig 44 and Fig 42 are given in Fig 45. These curves should be applied only for ducts with cross section in the range from about 6 ft x 6 ft to 10 ft x 10 ft. The lining for the ducts should be about 6 in. of Fiberglas (2-1/2 to 4-1/4 lb/ft^3 density) backed with an air space about 18 to 24 in. deep.

The noise reductions given in Fig 45 are applicable only for square ducts. The noise reduction for rectangular ducts can be derived from the noise reductions for parallel baffles of appropriate thickness and spacing.

F. Noise Reduction by Bends

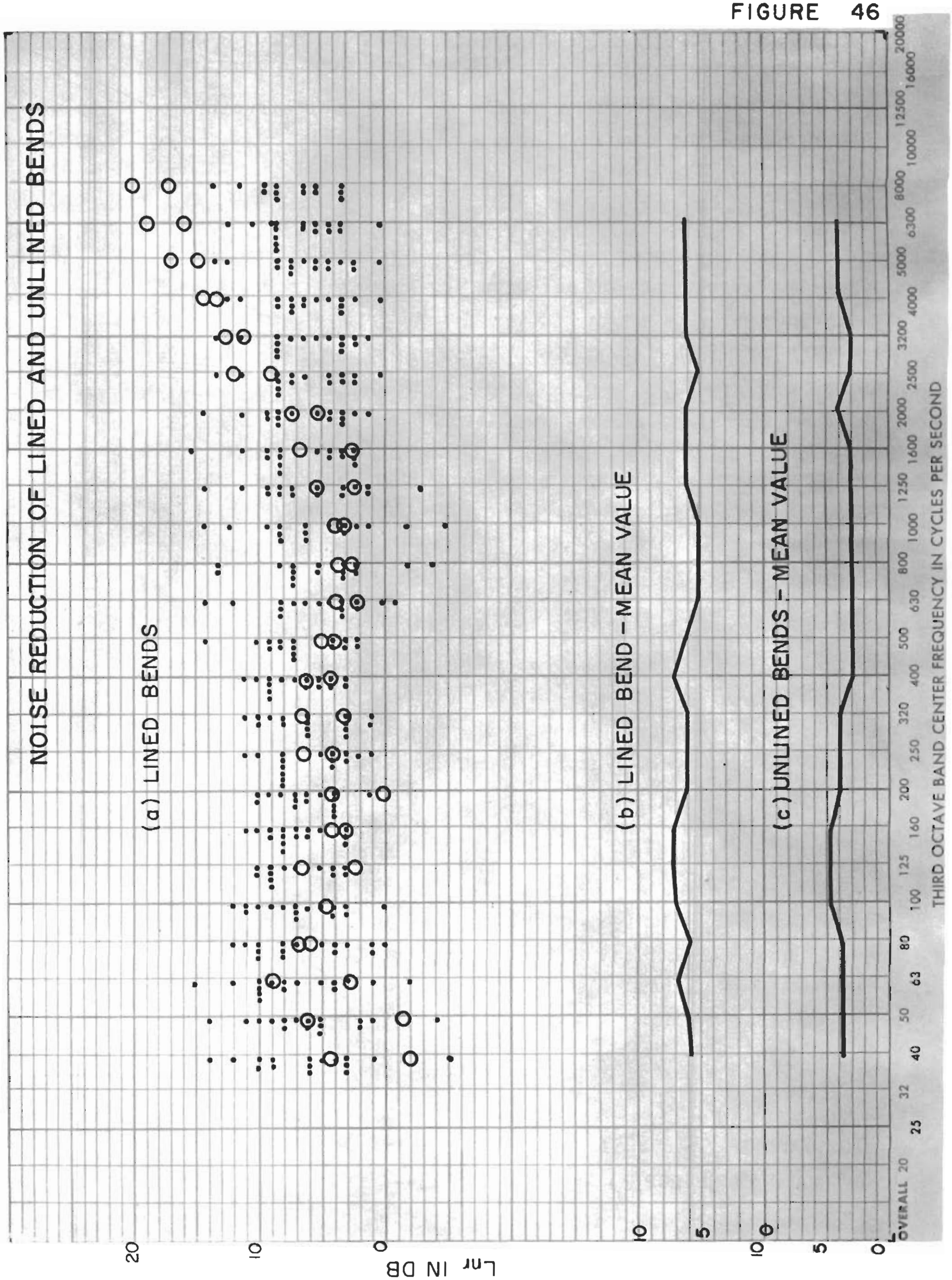
The noise reductions of 20 right angle bends were measured in the acoustical surveys under the Air Force program. Of these 16 were lined with acoustical material and four were unlined. The data which are presented subsequently are significantly different from those presented elsewhere in the literature[35, 36, 37]. The differences are attributable to the measurement techniques and to the difference between noise reduction for plane (first order) waves and for randomly incident waves.

In the three references cited measurements were made under controlled laboratory conditions which assured a plane wave in the "input" duct. In addition, the measurements were made by traverse techniques or by insertion loss techniques. Both of these techniques also measure effects of the bend on the noise field beyond a bend. These data do not include such effects. A bend scatters the sound energy into higher order modes so that the noise reduction of any treatment beyond a bend will be higher than for first order modes. The measurements of L_{nr} given here show only the difference in SPL at planes C and D in Figure 33.

The data obtained is summarized in Figure 46. The individual measurement points for lined bends are shown in Figure 46(a). The two open circles refer to bends which were preceeded by a baffle treatment which was parallel to plane B of Fig 33. For this particular geometry the noise reduction varies with frequency in a manner indicated by Lippert, Waters and King[35, 36, 37]. That is, the noise reduction increases rapidly for frequencies at which $\frac{\lambda}{4}$ is greater than the duct height. The duct height must be interpreted here to be the normal distance between the baffles preceeding the bend rather than the height of the duct containing the baffles. For this special case noise reduction characteristics follow the plane wave theory because the baffles suppress all higher order modes.

With the exception of this special geometry, the L_{nr} versus frequency characteristics of bends are essentially independent of frequency. The average values of noise reduction for lined and unlined bends are shown in Fig 46(b) and (c) respectively. As can be seen there is only a slight difference in the noise reduction of lined and unlined bends. Such a small difference in noise reduction (about 2 db) suggests that lining a bend in an engine test facility may not be an economical way to achieve noise reduction. However, the data do indicate that a lined bend will become much more effective in the

FIGURE 46


NOISE REDUCTION OF LINED AND UNLINED BENDS
(a) LINED BENDS
(b) LINED BEND - MEAN VALUE
(c) UNLINED BENDS - MEAN VALUE
THIRD OCTAVE BAND CENTER FREQUENCY IN CYCLES PER SECOND
OVERALL 20 25 32 40 50 63 80 100 125 160 200 250 320 400 500 630 800 1000 1250 1600 2000 2500 3200 4000 5000 6300 8000 10000 12500 16000 20000
20 10 0
10 5 0
10 5 0
L_{nr} IN DB

high frequencies if a baffle or long duct structure precedes the bend.

SECTION VI

INFORMATION REQUIRED FOR PREDICTING INSERTION LOSS FROM NOISE REDUCTION DATA

Prediction of insertion loss of a test facility requires knowledge of noise characteristics of engines operating in test facilities. The characteristics in free field are not sufficient, because they are modified by the enclosing facility. Relevant information on jet engines in test facilities is given in Paragraph A; on reciprocating engines in Paragraph B. Directivity characteristics of air intakes and exhausts, also needed for prediction of insertion loss, are presented in Paragraph C. Finally, some comments on measuring insertion loss are given in Paragraph D.

A. Noise Characteristics of Jet Engines in Test Facilities

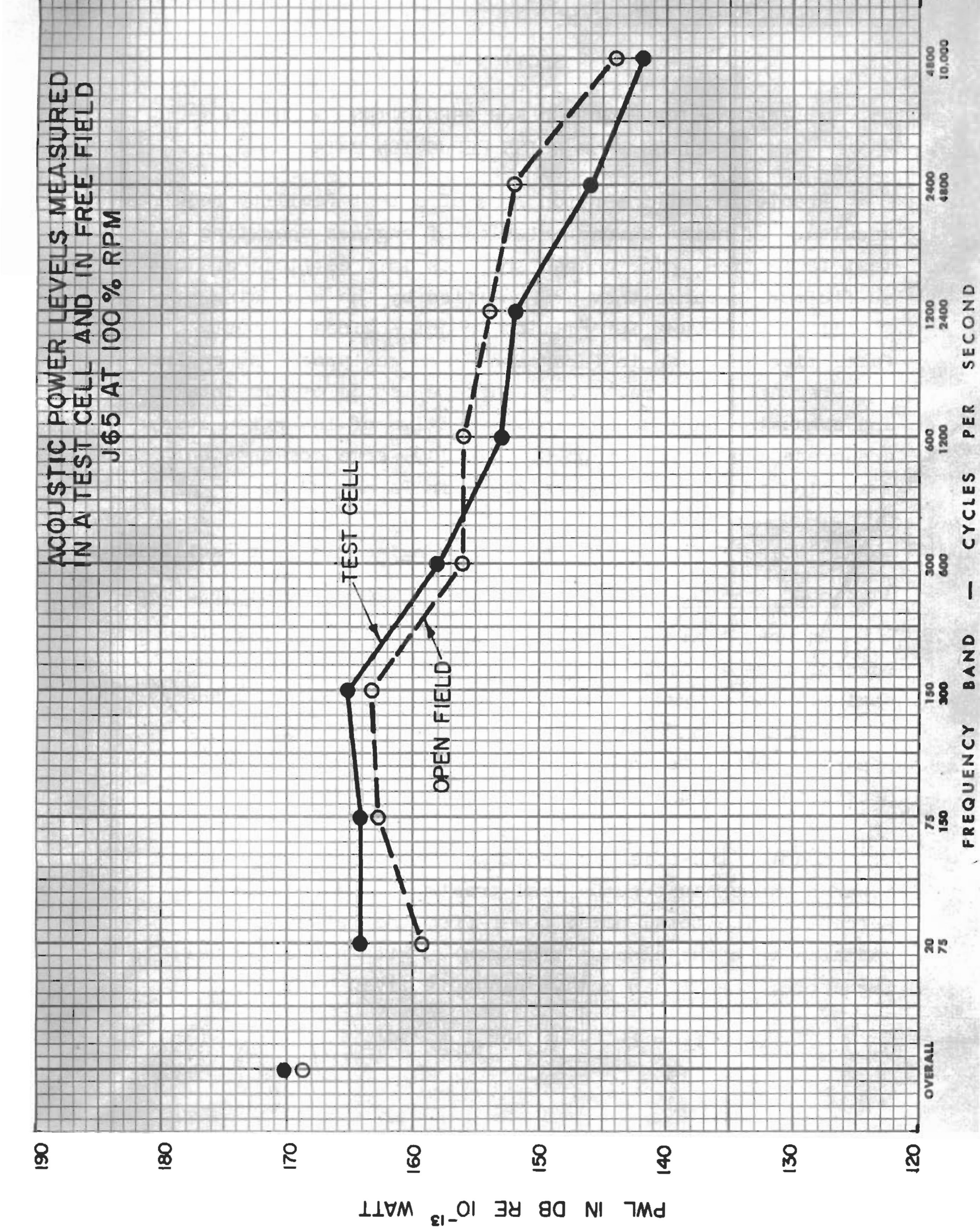
1. Sound Pressure Levels at the Exhaust Acoustical Treatment

In most test cells and ground run-up facilities, the eductor tube is located quite near the exhaust of the jet engine. The source of jet noise is distributed in space to the rear of the jet engine. The distance from the apparent source to the jet exhaust outlet increases with decreasing frequency.

Most of the noise is radiated into the eductor tube towards the exhaust acoustical treatment. More than 90 percent of the total noise power from the jet engine radiates into the exhaust treatment. Within one decibel, therefore, the SPL in octave bands is given by:

$$SPL = PWL - 10 \log_{10} A \qquad (35)$$

FIGURE 47


ACOUSTIC POWER LEVELS MEASURED
IN A TEST CELL AND IN FREE FIELD
J65 AT 100 % RPM
TEST CELL
OPEN FIELD
FREQUENCY BAND — CYCLES PER SECOND
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
190
180
170
160
150
140
130
120
PWL IN DB RE 10^{-13} WATT

where PWL is the open field power level of the engine in db re 10^{-13} watt (in octave bands),

A is the open area of the exhaust acoustical treatment in square feet.

This equation is used under the assumption that the free field power level of the engine is not changed by the eductor tube. This assumption has been checked by measuring the SPL at the input to one exhaust acoustical treatment;[19] the resulting power level is presented in Fig 47, along with the power level measured in the free field[38].

Since a single microphone position was used to determine the space-average SPL at the exhaust acoustical treatment, the differences between the free field power level and the power level in the test cell are subject to an uncertainty of the order of 4 decibels (see Section III, Para. D. Even allowing for this uncertainty, it appears that the eductor tube increases the power level slightly in the low frequencies, contrary to expectation.

In the higher frequencies, the PWL measured in the test cell is somewhat lower than the free field PWL. The microphone at the exhaust treatment of the test cell was located behind a blast deflector. The acoustic shielding provided by this deflector may account for the lower PWL in the test cell.

If a diffuser is attached to the engine in the test cell, the power level of the engine will be significantly decreased. It will be necessary then to measure the free field power level of the engine with the diffuser attached, in order to apply Eq 35.

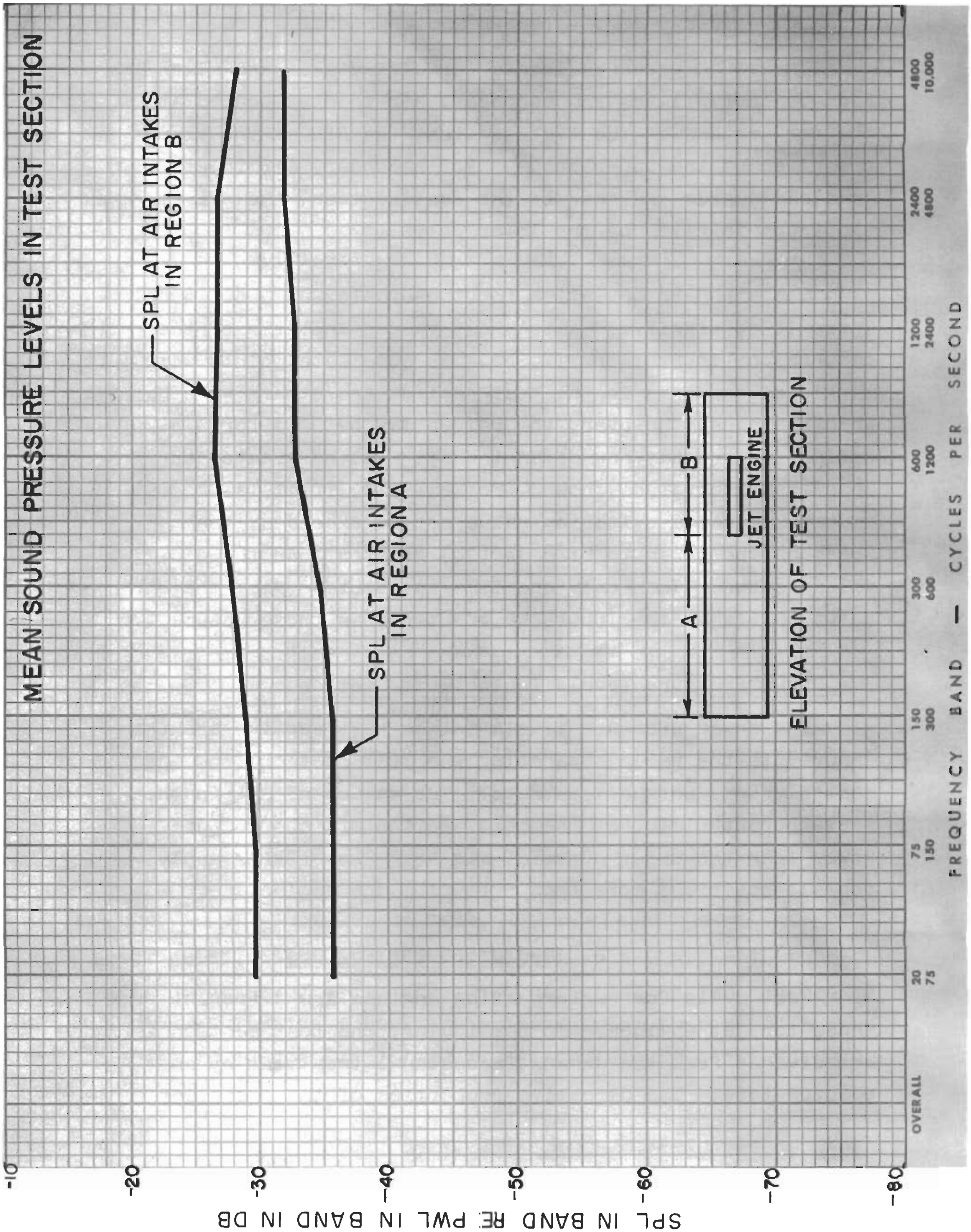
Although no measurements have been made of the SPL at the input to the exhaust acoustical treatment for ground run-up suppressors, the configuration of eductor tubes is similar to those in the test cells, and the considerations above should apply equally well to ground run-up suppressors.

2. Sound Pressure Level at Air Intakes

Measurements of SPL in eight different test cells were made at several locations in the test section including the primary and secondary air intake grids. These data indicate that the noise field in the test section is quite uniform except at the primary air inlet, which is usually located in the ceiling at the forward end of the test section. This position may be considered as being around a bend from the noise source.

A relation has been derived between the octave band SPL in the test section free field and the octave band PWL of the engine. For the range of areas involved (200-400 sq ft) the difference between SPL and the free field PWL does not appear to be dependent on area. The relation applies only for those test cells in which the engine exhaust is not isolated from the test section. One test cell[25] was encountered in which a "collar" was placed around the jet exhaust orifice. This collar, constructed of approximately 1/2 in. steel plate, effectively isolated the jet exhaust from the test section and the forward end of the engine. The SPL in test sections of this type must be obtained from detailed considerations of the structures involved. Omitting this exception, the values of SPL given in the paragraphs below apply to closely or loosely coupled engines and eductor tubes.

FIGURE 48


MEAN SOUND PRESSURE LEVELS IN TEST SECTION
SPL AT AIR INTAKES IN REGION B
SPL AT AIR INTAKES IN REGION A
A
B
JET ENGINE
ELEVATION OF TEST SECTION
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
FREQUENCY BAND — CYCLES PER SECOND
-10
-20
-30
-40
-50
-60
-70
-80
SPL IN BAND RE PWL IN BAND IN DB

The sound pressure levels for air intakes are given in Fig 48. For primary and secondary air inlets located at the forward end of the test section, the SPL at the inputs are given by the upper curve. For secondary air inlets located towards the rear of the test section, the SPL at the input is given by the lower curve.

The standard deviation for these data was about 4 db. There is no significance to these values in terms of the discussions in Section III, because some data points were single microphones in the test section and others were grid averages. However, the values may be used for estimating roughly how often the SPL will be greater or less than the values shown. For example, if it is important that a criterion value not be exceeded, the SPL's can be taken to be 8 db (2σ) greater than the values shown. Then only one time in about 40 will the actual SPL's at the inputs be greater than those assumed.

3. Sound Pressure Levels in the Test Section

The sound pressure levels in the test section are the "input" to the walls separating the test section from the control room, work spaces, etc. The values of SPL in the "reverberant field" of the test section are about 2 db lower than the upper curve of Fig 48. These SPL's apply for all positions more than 5 ft from the engine.

B. Reciprocating Engines

Test cells for reciprocating engines are different from those for jet engines. In particular, the test section is not divided from the exhaust acoustical treatment by an eductor tube. The directive properties of the propeller noise (dominant noise source) are such that about half of the acoustic energy flows toward the intake treatment and half flows toward the exhaust treatment. Thus anywhere more than about five feet from the propeller and the engine the SPL is approximately:

$$SPL = PWL + 10 \log_{10} A - 3 \text{ db} \tag{36}$$

where A is the cross sectional area of the test section.

C. Directivity

1. Definitions of Directivity Index

In the far radiation field of any noise source, the PWL level spectrum and the directivity pattern suffice to specify the source for the purpose of determining the sound distribution at all distances from the source, except for the disturbance introduced by the environment, such as air attenuation, refraction, reflection, etc.

The directivity in the far field is usually defined as the difference between the SPL at a point and the average SPL at a distance r, from the source:

$$DI(\phi, \theta) = SPL(r, \phi, \theta) - SPL_{av}(r) \tag{37}$$

where SPL (r, ϕ, θ) is the sound pressure level at a distance r, elevation θ, and an azimuth ϕ from the source

$SPL_{av}(r)$ is the average SPL at a distance r* from the source;

DI (ϕ, θ) is the directivity index (in db) at the angles ϕ and θ from the source.

Furthermore, the average sound pressure level can be related to the power level of the source, PWL, by:

$$SPL_{av}(r) = PWL - 10 \log_{10} A \tag{38}$$

where A (ft^2) is $4\pi r^2$ for spherical radiation and $2\pi r^2$ for hemispherical radiation from a source*.

By combining the two previous equations, DI can be expressed as:

$$DI(\phi, \theta) = SPL(r, \phi, \theta) + 10 \log_{10} A - PWL \tag{39}$$

$$= PWL(r, \phi, \theta) - PWL \tag{40}$$

Thus, DI (ϕ, θ) may be interpreted as the difference between 1) the power level that would be calculated if SPL (r, ϕ, θ) were assumed to be the average SPL and 2) the true power level of the source.

The EN-1 exhaust microphone is located in the near field of the exhaust gas outlet. In the near field, the directivity index may still be defined by Eq 37, but DI (ϕ, θ) becomes DI (r, ϕ, θ) as the difference between SPL (r, ϕ, Θ) and

* In electroacoustics DI is almost always defined in terms of the average SPL over an entire sphere enclosing the source. In aircraft noise control problems, it has been a common and perhaps unfortunate practice to take an average over a hemisphere. Care must be taken to assure which directivity is used.

$SPL_{av}(r)$ depends on r in the near field. Furthermore, in the near field, Eq 38 is no longer valid. Hence, a definition of the type given by Eq 37 is not useful for determining the PWL of the noise source.

As measurements of SPL at the EN-1 exhaust microphone position are usually made to determine the PWL of the exhaust gas outlet, the definition of DI at the EN-1 position shall be based on Eq 39:

$$DI_{EN-1} = SPL_{EN-1} + 10 \log_{10} \left[k4\pi \left(\sqrt{2}r \right)^2 \right] - PWL_{exhaust} \qquad (41)$$

where k is a number between 0.5 and 1.0,

r is the "emitter" diameter, e.g., the minor dimension of the exhaust,

$PWL_{exhaust}$ is the PWL of the exhaust gas outlet.

The number, k, would be 0.5 for hemispherical radiation and 1.0 for spherical radiation. For engine test cells, the exhaust system occupies a fraction of a sphere of radius $\sqrt{2}$ r, enclosing the exhaust outlet, and therefore k lies between 0.5 and 1.0. We have arbitrarily assumed k is 0.75.

Equation 37 is as valid a definition of DI in the near field as Eq 39. Equation 39 is arbitrarily chosen because it provides a way of interpreting DI in terms of PWL. Note, however, that in the near field Eqs 37 and 39 (or 41) are not generally equivalent because Eq 38 which relates them may not apply. Therefore, DI_{EN-1} as defined by Eq 41 cannot be determined by measurements of SPL over the EN-1 sphere alone. It is necessary to find the PWL of the exhaust gas outlet and them determine DI_{EN-1} from Eq 41.

2. Calculation of Directivity Index at the EN-1 Position

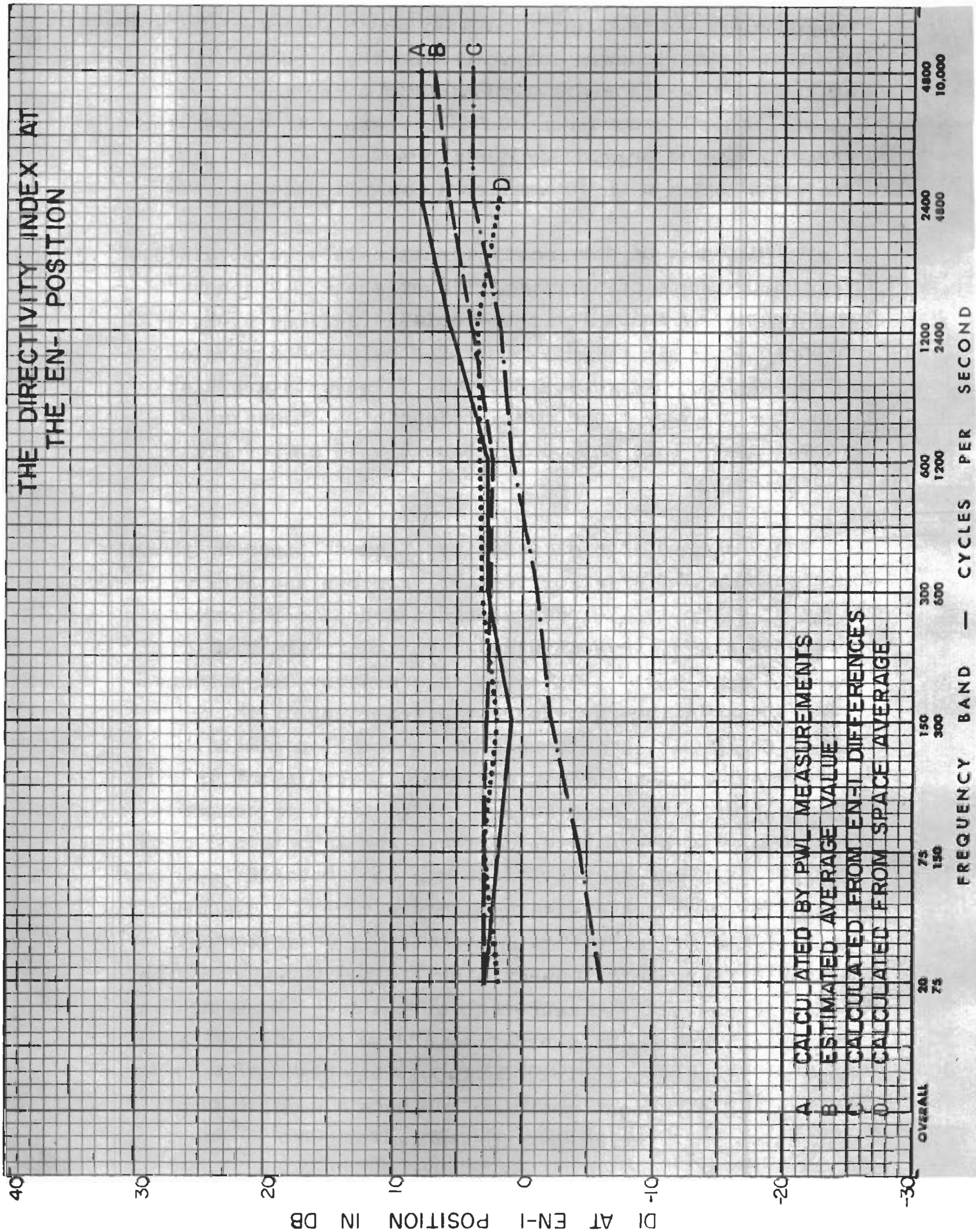
In three engine test cells[19, 21, 22], it was possible to determine the approximate power level of the exhaust gas outlet by measurements of SPL, which were made possible by special microphone holders and cables which were designed to withstand the high exhaust gas temperatures. On the average, two microphone positions were used. The average value of the directivity index at the EN-1 position has been found by application of Eq 41. These values of the directivity index are given by curve "A" in Fig 49. The directivity index is positive at all frequencies, being slightly greater at the high frequencies than at the low frequencies. The positive directivity index indicates that if the power level of the exhaust gas outlet were calculated using the SPL at the EN-1 position and the area of the EN-1 sphere, the value obtained would be greater than the actual power level of the exhaust stack.

Another method of evaluating the directivity index at the EN-1 position is suggested by Eq 30 and 31 in Section III. In that Section, a relation was derived between the EN-1 differences and the L_{nr} of the exhaust. Several quantities, including the directivity index at the EN-1 position, were expressed as the constant, A. Subsequently, the magnitude of A was determined from many sets of EN-1 measurements and L_{nr} measurements. Combining Eqs 30 and 31 yields:

$$DI_{EN-1} = A - (Y - X) - 32 \text{ db} \tag{42}$$

The values of Y and X were obtained from measurements of the SPL at the EN-1 engine position and measurement of the free field power level of three types of jet engines.

FIGURE 49


THE DIRECTIVITY INDEX AT
THE EN-I POSITION
A CALCULATED BY PWL MEASUREMENTS
B ESTIMATED AVERAGE VALUE
C CALCULATED FROM EN-I DIFFERENCES
D CALCULATED FROM SPACE AVERAGE
A
B
C
D
DI AT EN-I POSITION IN DB
40
30
20
10
0
-10
-20
-30
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
FREQUENCY BAND — CYCLES PER SECOND

Equation 42 and the quantities A, X and Y were used to evaluate the directivity index. The values of the directivity index at the EN-1 positions are given in Fig 49 by the curve labeled "C". These values of DI increase with increasing frequency as the previously derived values of DI, but they are about 4 or 5 db smaller in magnitude.

A third set of directivity index data were obtained from measurements of average SPL over the "EN-1 sphere"[14]. The directivity index for these data is defined by Eq 37 (which in the near field cannot be considered equivalent to the definition given by Eq 41.) These data are also given in Fig 49 by the curve labeled "D". The value of DI obtained by this method is about 4 to 5 db for all frequencies.

The values of the directivity index obtained by these three methods differ significantly from one another. The values of directivity index obtained from the first two procedures above are both based on the definition of DI for the near field. That is, they are derived by considering the relations between the power level of the exhaust gas outlet and the SPL at the EN-1 position. We believe that the first procedure of calculation is more reliable than the second. In the first procedure, the approximate power level of the stack is obtained by direct measurement of SPL. In the second procedure, the power level of the stack is obtained by: 1) the measurement of the SPL at the EN-1 position, 2) an estimation of the difference between the EN-1 SPL and the open field power level, 3) by measurement of the L_{nr} of the exhaust treatment, and 4) by estimation of the power level spectrum. Each of these steps, 1) through 4), may involve significant errors. The present estimate of the average value of directivity at the EN-1 position is, therefore, based primarily on the data obtained by the first procedure and is given by Curve B of Fig 49.

This average value is, at best, an estimate of the DI at the EN-1 position. The DI for any one cell will depend upon the position selected for the EN-1 measurement (the location of the microphone on the EN-1 circle) and the geometry of the exhaust gas outlet and its acoustical treatments.

3. Directivity Index for Air Inlets and Exhaust Gas Outlets

The intake and exhaust outlets of most engine test cells and ground run-up suppressors lie in a plane parallel to the ground. That is, the exhaust gas and intake air enter and leave the test facility in a direction perpendicular to the ground. Most of the noise radiated from these openings is, therefore, directed upwards. The significant directivity index of interest, DI (ϕ, θ), is the directivity index for positions in a plane parallel to the exhaust gas outlet or 90° to the direction of the air flow.

During the Air Force program of acoustical evaluations, the values of DI given in Reference 39 have been used for design balance studies (see Volume I). Where possible these values of DI have been compared with measured data and the agreement between the measurements and the value given in Reference 39 has been generally good. A few minor changes in the directivity indices have been made.

FIGURE 50

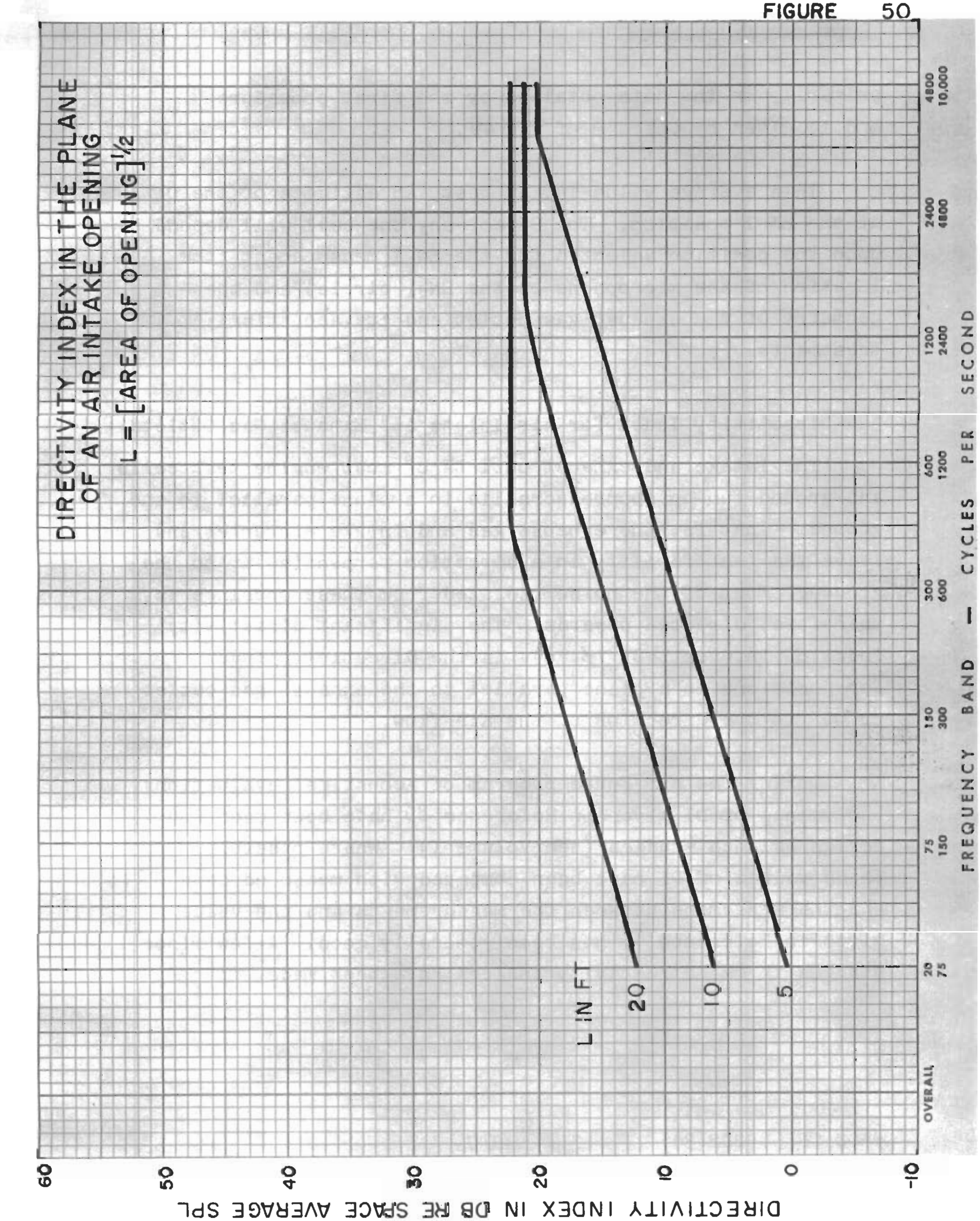
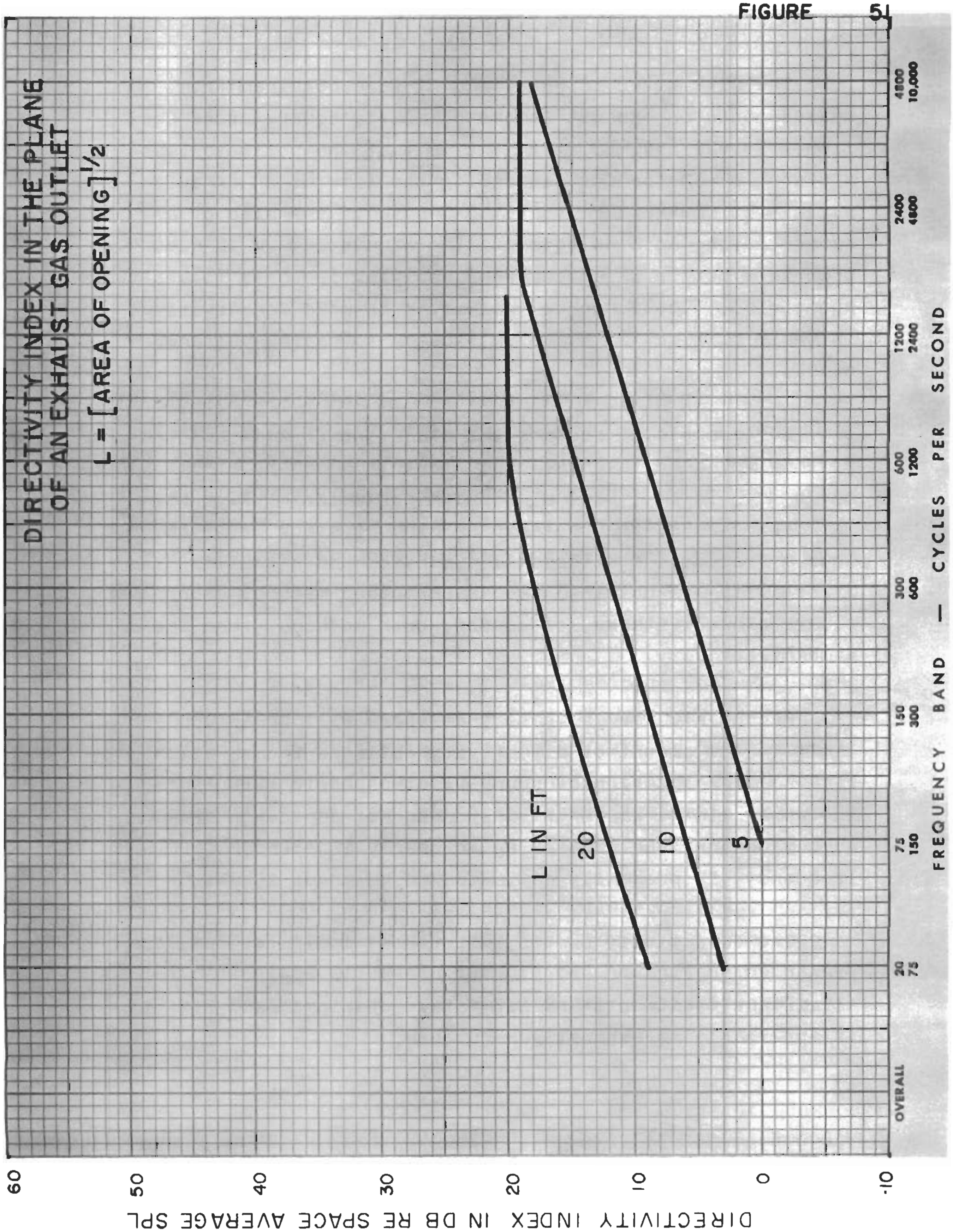

DIRECTIVITY INDEX IN THE PLANE OF AN AIR INTAKE OPENING
L = [AREA OF OPENING]^1/2
L IN FT
20
10
5
DIRECTIVITY INDEX IN DB RE SPACE AVERAGE SPL
60
50
40
30
20
10
0
-10
FREQUENCY BAND — CYCLES PER SECOND
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000

FIGURE 51


DIRECTIVITY INDEX IN THE PLANE OF AN EXHAUST GAS OUTLET
L = [AREA OF OPENING]$^{1/2}$
L IN FT
20
10
5
60
50
40
30
20
10
0
-10
DIRECTIVITY INDEX IN DB RE SPACE AVERAGE SPL
OVERALL
20 75
75 150
150 300
300 600
600 1200
1200 2400
2400 4800
4800 10,000
FREQUENCY BAND — CYCLES PER SECOND

The data given in Reference 41 were obtained from measurements on exhaust stacks of nearly square cross section. It was assumed at that time that the perimeter was a useful parameter for describing the size of an exhaust stack. It now appears that the length of the side of a square having an area equal to the exhaust stack in question is a better parameter when applying the data to arbitrary geometries. The suggested parameter is more conservative in that, smaller DI's are attributed to narrow rectangular stacks.

For example, a rectangular stack one ft wide and ten ft long would be taken as equivalent to a square stack having a 3.16 ft ($\sqrt{10}$) side and a perimeter of 12.6 ft. The perimeter of the rectangular exhaust stack is 22 ft. Thus a smaller characteristic dimension and a smaller directivity are obtained for the proposed method. For square exhaust stacks, of course, both methods yield the same directivity.

Recent studies indicate that there will be an upper limit to the amount of directivity obtained from a stack, because of scattering of sound by atmospheric turbulence. The upper limit, which probably does not depend on frequency because the scattering is nearly independent of frequency, is presently estimated to be about 20 db.

The directivity curves, with these modifications, are presented in Figs 50 and 51. Figure 5C is applicable to air intake treatments and air intake openings, and Fig 51 is applicable to exhaust gas outlets.

These directivity curves are average values of directivity in two senses. First, they are averaged over all azimuth angles around the outlet or inlet, and second, they represent average values for different types of cells. The value of the directivity index at any azimuth from a particular test facility may vary from the value shown.

In some test facilities, the plane of an exhaust gas outlet or an air inlet lies in a plane normal to the ground. That is, the air enters or leaves the test facility in a plane parallel to the ground. In other test facilities, the air inlets or exhausts lie in a plane horizontal to the ground, but a roof structure is placed above the inlet or exhaust so that the air is forced to enter or leave in a direction parallel to the ground. For either of these two cases, the <u>average</u> value of the directivity index may be taken to be 0 db.

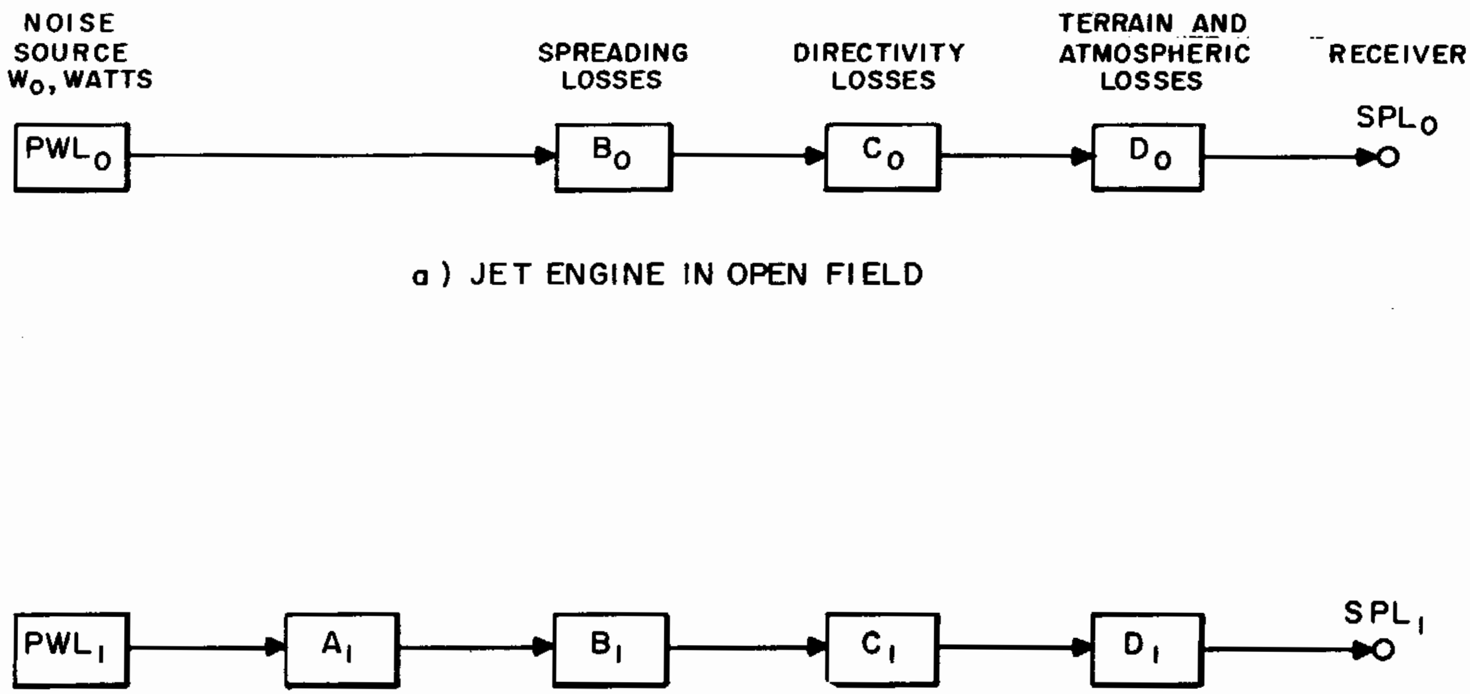
Reliable field measurements of directivity are seldom obtained because of the difficulty of measuring the total sound power radiated from an exhaust or intake stack, and because the distant field measurements can be complicated by contributions from several noise sources (the exhaust stack, intake stack, walls, doors, etc.). However, a few design balance studies (see Volume I) revealed that, in the three lowest octave bands, the exhaust outlet was the only significant noise source in some test cells[14, 19, 23, 25].

The difference between the measured SPL and the predicted SPL from the design balance study was used as an estimate of the accuracy of the directivity index. The mean "error" in the first three bands was about $\pm$ 1 db. The range of errors was from -3 to + 4 db. Further refinement of the directivity curves can be best obtained by carefully scaled model tests from which the effect of flow and gas temperature can be determined.

4. Measurement of the Insertion-Loss Noise-Reduction Provided by a Test Facility

The insertion-loss noise-reduction of the test facility is a useful measure of the acoustical performance of the facility as a whole. It is, by definition, the amount by which the sound pressure level is lowered at a particular point by "insertion" of the test facility.

The insertion loss is usually measured on a complete circle surrounding the test facility in order to find the acoustical effectiveness in all directions. There has developed recently the practice, of limited usefulness, of describing the performance of ground run-up suppressors by stating the noise reduction only at 45° from the jet stream axis. Noise problems may exist at any angle relative to the jet stream. Measurements at 45° alone do not tell enough about acoustical performance. It is not unusual to find a large (30 db or more) noise reduction at 45° from the jet stream, and a small or negative noise reduction at some other angle. If the measurements are to be generally useful, they must be made entirely around the test facility, not at one angle only.



b) JET ENGINE IN A TEST CELL

FIG. 52 A NOISE ENERGY FLOW DIAGRAM FOR AN ENGINE IN AN OPEN FIELD AND IN A TEST CELL.

A radius of 250 ft has been selected in order to obtain positions that would be far enough from the facility for the measured insertion-loss to be valid for greater distances also, but close enough so that atmospheric conditions would not unduly influence the measurements. The implications of this statement can be seen by considering the noise energy flow diagram shown in Fig 52.

In Fig 52 (a) the jet engine is represented as a noise source that radiates an acoustical power, W_o watts, or a power level, PWL_o decibels. The three blocks represent the several factors that attenuate the noise as it propagates from the engine to the measuring position. The SPL at the measuring position R ft from the engine can be written:

$$SPL_o = PWL_o - (B_o + C_o + D_o) \qquad (42)$$

The average SPL at a distance R from the receiver is $PWL_o - B_o$, where B_o measures the spherical divergence of sound from the source. The term, C_o, is a directivity correction which measures the amount by which the SPL at a particular distance, azimuth and elevation varies from the average SPL at that distance. The term D_o accounts for the attenuation of sound owing to atmospheric and terrain variables.

When the engine is placed in a test facility, the noise flow can be represented as shown in Fig 52 (b).

The SPL at the same position relative to the engine is now written:

$$SPL_1 = PWL_1 - (A_1 + B_1 + C_1 + D_1) \tag{43}$$

The terms in this equation are the same as those in Eq 42 with a new term, A_1, included. This term measures the net attenuation of the acoustical treatments in the test facility. If we now subtract Eq 43 from Eq 42, we obtain the insertion-loss-noise-reduction of the facility:

$$SPL_o - SPL_1 = (PWL_o - PWL_1) - A_1 - (B_o - B_1) - (C_o - C_1) - (D_o - D_1) \tag{44}$$

The first term on the right hand side of Eq 44 measures the change in PWL caused by the interaction of the test cell and the engine. This term is usually negligible in contemporary cells; certain design procedures, however, could make this term the most significant variable in test cell design.

The second term $(B_o - B_1)$ measures the change in SPL due to the change of the distance from the noise source to the receiver caused by "insertion" of the test facility. We call this term an <u>inverse square error</u>. If all other terms of Eq 44 were zero, the change in SPL would be, at most,

$$20 \log_{10} (R_1/R_o) = 20 \log_{10} [1 - (\ell/2R_o)] \tag{45}$$

where ℓ is the approximate length of the test facility,

R_o is the distance from the jet engine to the measurement point.

R_1 is the distance from the exhaust to the measurement point.

The above equation also assumes that almost all of the noise is radiated from the exhaust of the test cell. Another term 20 $\log_{10} \frac{R_o + (\ell/2)}{R_o}$, could be added to account for radiation from the intake. We shall consider only the exhaust term as this will give the greatest possible inverse square error.

The third term is the decrease in SPL due to the change in directivity caused by the test facility. In a well designed test cell the average value of $(C_o - C_1)$ is positive because a large fraction of the acoustic energy is directed upwards (i. e., away from a receiver on the ground).

The final term, $(D_o - D_1)$, measures the difference in atmospheric and terrain conditions for the two sets of measurements, i.e., before and after "insertion" of the test facility. It is usually impossible to make the two sets of measurements over identical terrain. In addition, the atmospheric effects are uncontrollable, and often they vary randomly with time, so that $D_o - D_1$ will not generally be zero.

Since the evaluation must be valid for the distant field, it is desirable to have the measurement position <u>far</u> from the engine or test facility where the "C" term is negligible. But to assure that the "D" terms are negligible, the measurement positions must be near the engine or test facility. A radius of 250 ft for the measuring circle has been selected as a compromise between these antithetical requirements.

The length of the test facility, "ℓ", will be less than 125 ft for most test cells and almost all ground run-up suppressors. Therefore, the inverse square error will be $20 \log_{10} \left[\frac{250 - (125/2)}{250} \right] = 3$ db at most, for a 250 ft measurement circle.

Recent studies[40] of sound propagation near the ground show that attenuation owing to terrain (typically 1 ft high dense ground cover) is negligible for distances less than 200 ft. At 250 ft, the effect of terrain is about 1 or 2 db; at 400 ft from the source, attenuations of 5 to 10 db are encountered. These studies also deal with sound propagation as influenced by wind velocity, wind velocity gradients and temperature gradients. Atmospheric effects will be negligible at all positions on a measurement circle of 250 ft radius, if the wind velocity at 20 to 40 ft above the ground is less than 5 knots. If a greater wind velocity is allowed, the circle must be made smaller to avoid wind effects. If the circle is made much larger, the allowable wind velocity must be lowered.

In summary, the radius of 250 ft is selected as an engineering compromise between the antithetical requirements of (1) a large distance which is needed to avoid near field effects and (2) a small distance which is needed to avoid atmospheric and terrain effects.

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