FEASIBILITY STUDY ON THE USE OF A SINGLE CORNER MICROPHONE POSITION IN THE DETERMINATION OF SOUND POWER LEVELS OF BROAD-BAND NOISE SOURCES IN A LARGE REVERBERATION ROOM

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LIST OF SYMBOLS

A	area
c	speed of sound
d	distance
dB	decibel
f	frequency
k_1, k_2, k_3	wave numbers
L_x, L_y, L_z	room dimensions
n_x, n_y, n_z	integers
P	pressure
S	surface area
T	time
V	volume, potential energy
x, y, z	axis coordinates
$\bar{\alpha}$	average sound absorption coefficient
ρ	density
δ	acoustic density change
∇^2	Laplacian operator
ψ	velocity potential

CHAPTER I

INTRODUCTION

In reverberation room testing the primary objective is to determine the sound power level of a sound source (17) (23). A sound source is defined as a device, machine, component, or subassembly which emits steady sound. The frequency spectrum of the sound is generally broad band in nature with or without discrete frequency and narrow band components. Typically, a sound source will occupy less than one percent of the volume of the test room. A test room is called reverberant when the room has a very small average sound absorption coefficient in the frequency bands of interest. This low sound absorption property allows the sound energy to be reflected off the walls of the chamber. Through multiple reflections the sound energy is theoretically distributed throughout the room. Sound energy cannot be directly measured but acoustic pressure is easily measured. The measurement of the acoustic pressure allows the calculation of the potential energy of the sound field by the expression,

$$V(x, y, z) = \bar{p}^2/2\rho c \tag{1.1}$$

where V is the potential energy, $\bar{p}^2$ is the mean-square acoustic pressure, ρ is the density of the air, and c is the speed of sound (9). Reverberation room testing techniques are primarily concerned with the determination of the mean-square acoustic pressure level for their determination of sound power levels.

The nature of the acoustic pressure in an ideal rectangular room is described mathematically as (4),

$$P = P_o \left[\cos\left(\frac{n_x}{L_x} x\right) \cos\left(\frac{n_y}{L_y} y\right) \cos\left(\frac{n_z}{L_z} z\right)\right] e^{j\omega t} \qquad (1.2)$$

where

P = acoustic pressure;

L_x, L_y, L_z = dimensions of the room;

n_x, n_y, n_z = non-negative independently variable integers;

x, y, z = coordinate axes; and

ω = circular frequency of vibration.

The acoustic pressure equation allows only certain pressure distributions to be present. The presence of the distribution is evident from Equation (1.2) at any fixed value of time t. The distribution can be visualized as a time varying lattice of maximum acoustic pressure points (3). The pressure lattice may be considered to be the superposition of many different modes of excitation. The modes or standing waves may be cataloged as follows (8):

Axial modes, one-dimensional waves moving between a set of parallel walls;

Tangential modes, two-dimensional waves moving along the walls of the walls;

Oblique modes, three-dimensional waves moving or "breathing" in and out.

Due to the boundary conditions imposed on the general wave equation in the classicial derivation of the acoustic pressure Equation (1.2), only certain modal frequencies can be excited in a given room. The frequencies of these modes are given by:

$$fn = \frac{c}{2} \left[\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2 \right]^{1/2} \tag{1.3}$$

This equation generates a series of "allowable" frequencies for any given set of room dimensions (8). The allowable frequencies will determine the range of frequency measurements which may be made and how well the acoustic pressure lattice is distributed. The distribution of the pressure lattice will depend on the number of modes excited. A simple approximate formula for the number of modes excited below a given frequency f_o is given by (8):

$$N \simeq \left(\frac{4\pi f_o^3 V}{3c^3}\right) + \left(\frac{\pi f_o^2 V}{4c^2}\right) + \left(\frac{f_o L}{8c}\right) \tag{1.4}$$

where

L = sum of the length of the room's walls;

V = volume of the room; and

A = total surface of the room.

For example, in the room used for experimentation in this study, 245 modes have frequencies below 125 Hz, 169,661 modal frequencies lie below 1250 Hz, and an astounding 164,618,706 modes lie below 12,500 Hz. The multitude of modes in the higher frequency ranges will increase the number of the acoustic pressure nodes in a given volume. The increase in the number of modes excited, coupled with a shorter wave length, allow the presence of an almost uniform energy distribution in the room. This result infers that the construction of very large reverberation rooms would aid in the determination of sound power levels for all frequency bands. There are, unfortunately, certain dissipative effects that offset the advantages of such large chambers. The upper limit on the size of a reverberation chamber is not based on modal excitation but on the dissipation of sound energy by the air present in the room (3) (4). A large ideal reverberation room would have a very diffuse sound field at high frequencies. Unfortunately, this is not the case due to dominant air absorption of sound in these upper frequency bands. The problem of determining a value for the average mean-square acoustic pressure in a reverberation room remains to be solved.

The traditional approach to determining the sound power level of a sound source is to sample the acoustic pressure lattice. This technique requires a time/space averaging system and has historically led to a number of problems. The primary difficulty is in knowing when the sound field has been adequately sampled. The sampling is to take place in a diffuse sound field. The radiation of sound by a sound source will define two regions in space: the near field and the diffuse field. In the near field the direct radiation of sound from the sound source dominates, and in the diffuse field the reflected, or reverberant, sound dominates. A large number of papers have been written on how these time/space average sound power level measurements are to be made (1) (5) (6) (12) (17) (20) (22) . These works deal with a variety of sampling techniques and room modifications to be used in these measurements. The work done to date with the diffuse field approach has been standardized. American National Standard S1.21-1972 is such a standard.

American National Standard S1.21-1972 employs the use of a traversing microphone or microphone array in the determination of sound power levels in a reverberation room. This standard stipulates that the microphone traverse or array must be in the diffuse sound field of the room. The requirements for compliance with the standard in regard to sound power measurements are given in summation below:

1. The shortest distance between the sound source and the microphone must be greater than

$$d = 0.008 \sqrt{V/T}$$

where V is the volume of the room in meters cubed, and T is the reverberation time in seconds.

2. The microphone traverse or array must not lie in a place within ten degrees of any room surface.

3. No point on the traverse, or microphone in the array, shall be closer to any room surface than one-half the wavelength of the lowest frequency of interest.

4. The microphone traverse or array must not intersect any area of air discharge or sound beaming of the equipment being tested.

5. The mean-squared sound pressure level average must have a corresponding standard deviation that lies within the tolerances given below for broad-band noise.

1/3 Octave Band Center Frequencies (Hertz)	Maximum Allowable Standard Deviation (Decibels)
100 - 160	1.5
200 - 630	1.0
800 - 2500	0.5
3150 - 10000	1.5

6. The mean-squared sound pressure level average must have a corresponding standard deviation that lies within the tolerances given below for discrete frequency tones.

1/3 Octave Band Center Frequencies (Hertz)	Maximum Allowable Standard Deviation (Decibels)
100 - 160	3.0
200 - 315	2.0
400 - 630	1.5
800 - 2500	1.0
3150 10000	2.0

7. The method of calculation for the determination of the standard deviation, sound power levels from the average sound pressure level, and the specifics for the running of these tests are outlined in the standard.

American National Standard S1.21-1972 outlines established procedures for reverberation room testing using the diffuse field technique.

This study searches for a new approach to the determination of the sound power level in a reverberant room. In the diffuse sound field the acoustic pressure lattice varies in time and space. This variation depends on the frequency content of the sound source. No point in the diffuse field has any special properties with respect to all of the modes in the room, thus requiring that some form of time/space averaging be undertaken. The need for space averaging could be eliminated if one point in the room was excited by every mode in the room. This point exists theoretically at the corners of the room. The special properties of the corner of the room may be seen by analyzing the acoustic pressure relationship (Equation (1.2)) with the coordinates for any corner inserted. The acoustic pressure is found to be a maxima. The corners of the room exhibit this pressure maxima for all modes of excitation making the corner unique in that aspect.

The presence of this unique point raises the possibility of using a single corner point for sound power level determinations. Ideally, a solid walled rectangular reverberation room and a single microphone would be the only apparatus needed to determine overall sound power levels. To comply with the diffuse field measurement standards, extensive modifications of basic solid walled reverberation rooms have been required. These modifications have included "selective" sound absorbing panels to detune room resonances and rotating diffusers to modify room geometry. A working definition of feasibility can be based on the elimination of these modifications.

Feasibility in this study is based on simplicity. If one can eliminate the use of such complex technical equipment as multiple microphone arrays, traversing microphones, large rotating diffusers, and selective sound absorption panels, without sacrificing precision in the sound power measurement, the technique will be called feasible.

In determining feasibility this thesis is more concerned with a proof of principle than in developing a practical measuring system. Therefore, only select representative tests have been made. This study is intended to determine the validity of a new approach to an old problem. If validity can be demonstrated, recommendations for further studies will be made.

CHAPTER II

EXPERIMENTAL PREMISE

Due to the lack of literature dealing directly with the concept of a corner measurement technique for the determination of sound power levels, it is necessary to draw on classical acoustic theory. This work is based on the premise that, for all modes of excitation, an acoustic pressure anti-node will exist in the corner of a rectangular reverberation room. This concept can be developed using classical energy principles. Either the eigenmode or the free-wave model of the sound field in a reverberent room may be used in this development (21). The potential energy function can be directly related to the acoustic pressure. Acoustic pressure will be measured experimentally; therefore, only the potential energy functions will be developed.

Consider a rectangular reverberation chamber with dimensions L_x, L_y, L_z, in the x, y, z directions, respectively. The simplified wave equation, in three dimensions, can be expressed as (4),

$$\frac{\partial^2 P}{\partial t^2} = c^2 \nabla^2 P \tag{2.1}$$

where

P = acoustic pressure;

t = time;

c = speed of sound; and

∇^2 = Laplacian operator.

If an axial mode is excited in a rectangular enclosure, an expression for the velocity potential is,

$$\psi(x, y, z, t) = \cos kx \cos\omega t \tag{2.2}$$

where $k = \omega/c$ is the wave number (9).

Acoustic pressure is related to the velocity potential by the equation:

$$P(x, y, z, t) = -\rho\left(\frac{\partial\psi}{\partial t}\right)\delta \tag{2.3}$$

where ρ is the density of the air, and δ is the acoustic density change.

The acoustic pressure of the axial mode becomes,

$$P(x, y, z, t) = \rho\omega \cos kx \sin\omega t \tag{2.4}$$

which yields a mean-square acoustic pressure of

$$\bar{P}^2(x, y, z) = \frac{1}{2}\rho^2 \omega^2 \cos^2 kx \tag{2.5}$$

Recalling that the potential energy density (V) is directly related to the mean-square acoustic pressure ($\bar{P}^2$) by the expression (1.1),

$$V(x, y, z) = \frac{\bar{P}^2}{2\rho c}$$

the potential energy density for an axial mode is,

$$V(x, y, z) = \frac{1}{4}\rho k^2 \cos^2 kx. \tag{2.6}$$

Next, consider a tangential mode in the above manner. The velocity potential for a tangential mode, from classical theory is,

$$\psi(x, y, z, t) = \cos k_1 x \cos k_2 y \cos\omega t \tag{2.7}$$

with x and y representing the two space coordinates. Equivalent expressions can be generated, using any pair of space coordinates, without affecting the general result. In the tangential mode expressions $k_1 = \frac{n_x \pi}{L_x}$, $k_2 = \frac{n_y \pi}{L_y}$, n_x and n_y are integers, and $k^2 = k_1^2 + k_2^2$. Developing the potential energy function from the pressure expressions,

$$P(x, y, z, t) = \rho\omega \cos k_1 x \cos k_2 y \sin\omega t \tag{2.8}$$

$$\bar{P}^2(x, y, z) = \frac{1}{2} \rho^2 \omega^2 \cos^2 k_1 x \cos^2 k_2 y \tag{2.9}$$

yielding a potential energy density of,

$$V(x, y, z) = \frac{1}{4} \rho k^2 \cos^2 k_1 x \cos^2 k_2 y \tag{2.10}$$

The final mode type to be considered is the three-dimensional oblique mode. The velocity potential for an oblique mode is given by,

$$\psi(x, y, z) = \cos k_1 \quad \cos k_2 x \cos k_3 y \cos\omega t \tag{2.11}$$

where $k_z = \frac{n_2 \pi}{L_z}$, n_z is an integer, and $k^2 = k_1^2 + k_2^2 + k_3^2$. As in the axial and tangential modes, the acoustic pressure relationship will generate the potential energy density. The relations,

$$P(x, y, z, t) = \rho\omega \cos k_1 x \cos k_2 y \cos k_3 z \sin\omega t \tag{2.12}$$

$$\bar{P}^2(x, y, z) = \frac{1}{2} \rho^2 \omega^2 \cos^2 k_1 x \cos^2 k_2 y \cos^2 k_3 z \tag{2.13}$$

yield a potential energy density,

$$V(x, y, z) = \frac{1}{4} \rho k^2 \cos^2 k_1 x \cos^2 k_2 y \cos^2 k_3 z \tag{2.14}$$

Kinetic energy densities for each of the modes may also be generated from the velocity potential function.

The average value of the potential energy density function in space may be expressed as,

$$<V(x, y, z)> = <\cos^2 k_1 x> <\cos^2 k_2 y> <\cos^2 k_3 z> \quad (2.15)$$

In order to investigate the nature of the acoustic pressure in the corner of the room, it will be necessary to make use of a well-known mathematical result. The average value of the cosine squared is one-half,

$$<\cos^2 kx> = \frac{1}{2} \quad (2.16)$$

Each of the individual modes may now be evaluated at different points in space.

The premise for this paper can be established by the following evaluation, which demonstrates the unique character of the corner of the room. The idea is to locate the maxima of the potential energy functions and show that they coincide. The point of coincidence will be an acoustic pressure anti-node for all modes. The existence of a point of coincidence will enable the detection of all the modes excited in the room from this one point.

Consider the axial mode with its potential energy function (Equation (2.6)),

$$V(x, y, z) = \cos^2 kx$$

In the interior of the room the spatial average is

$$<V(x, y, z)> = \cos^2 kx = \frac{1}{2} \quad (2.17)$$

The spatial average along a wall may be one-half or one, depending on the orientation of the wave:

$$<V(x, y, z)> = \frac{1}{2} \quad (2.18)$$

when the wave is traveling parallel to the wall or,

$$\langle V(x,\ y,\ z)\rangle = 1 \tag{2.19}$$

when the wave is incident upon the wall. The spatial average along any edge of the room is seen to be one or one-half from the same principles. However, at any corner of the room the spatial average must have the value of unity. The unity value is enforced mathematically by the cosine function's argument (kx) becoming an integral multiple of Pi. This is a result of the definition of k: $\frac{n\pi}{L}$. At the corner,

$$\langle V(x,\ y,\ z)\rangle = 1 \tag{2.20}$$

for all axial modes.

Now consider a tangential mode's potential energy function (Equation (2.10)),

$$\langle V(x,\ y,\ z)\rangle = \frac{1}{4}\,\rho k^2 \cos^2 k_1 x \cos^2 k_2 y$$

In the interior of the room,

$$\langle V(x,\ y,\ z)\rangle = (\tfrac{1}{2})(\tfrac{1}{2}) = \frac{1}{4} \tag{2.21}$$

The potential energy function at the wall is either,

$$\langle V(x,\ y,\ z)\rangle = (\tfrac{1}{2})(\tfrac{1}{2}) = \frac{1}{4} \tag{2.22}$$

or,

$$\langle V(x,\ y,\ z)\rangle = (\tfrac{1}{2})(1) = \frac{1}{2} \tag{2.23}$$

once again depending on orientation. In the corner of the room the potential energy function has the value of unity,

$$\langle V(x,\ y,\ z)\rangle = (1)(1) = 1 \tag{2.24}$$

The final mode type to be considered is the three-dimensional oblique mode. The potential energy function for the oblique mode is (Equation (2.14)),

$$V(x, y, z) = \frac{1}{4}\rho k^2 \cos^2 k_1 x \cos^2 k_2 y \cos^2 k_3 z$$

Spatial averages for the potential energies are as follows:

In the diffuse field,

$$\langle V(x, y, z)\rangle = (\tfrac{1}{2})(\tfrac{1}{2})(\tfrac{1}{2}) = \frac{1}{8} \tag{2.25}$$

Near a wall,

$$\langle V(x, y, z)\rangle = (1)(\tfrac{1}{2})(\tfrac{1}{2}) = \frac{1}{4} \tag{2.26}$$

Near an edge of the room,

$$\langle V(x, y, z)\rangle = (1)(1)(\tfrac{1}{2}) = \frac{1}{2} \tag{2.27}$$

In the corner,

$$\langle V(x, y, z)\rangle = (1)(1)(1) = 1 \tag{2.28}$$

A brief mapping of the maxima of the potential energy functions for the various modes will be used to summarize this classical development (Figure 1). The axial modes have potential energy maxima along the walls of the room. The tangential modes have potential energy maxima along the edges of the room, and the oblique modes have potential energy maxima in the corners of the room. For every mode excited, the corner of the room is an acoustic pressure maxima. In classical acoustic theory the energies of any number of independently excited modes are additive. These two classical results seem to indicate that a corner microphone technique may be used for the determination of sound power levels in a rectangular room.

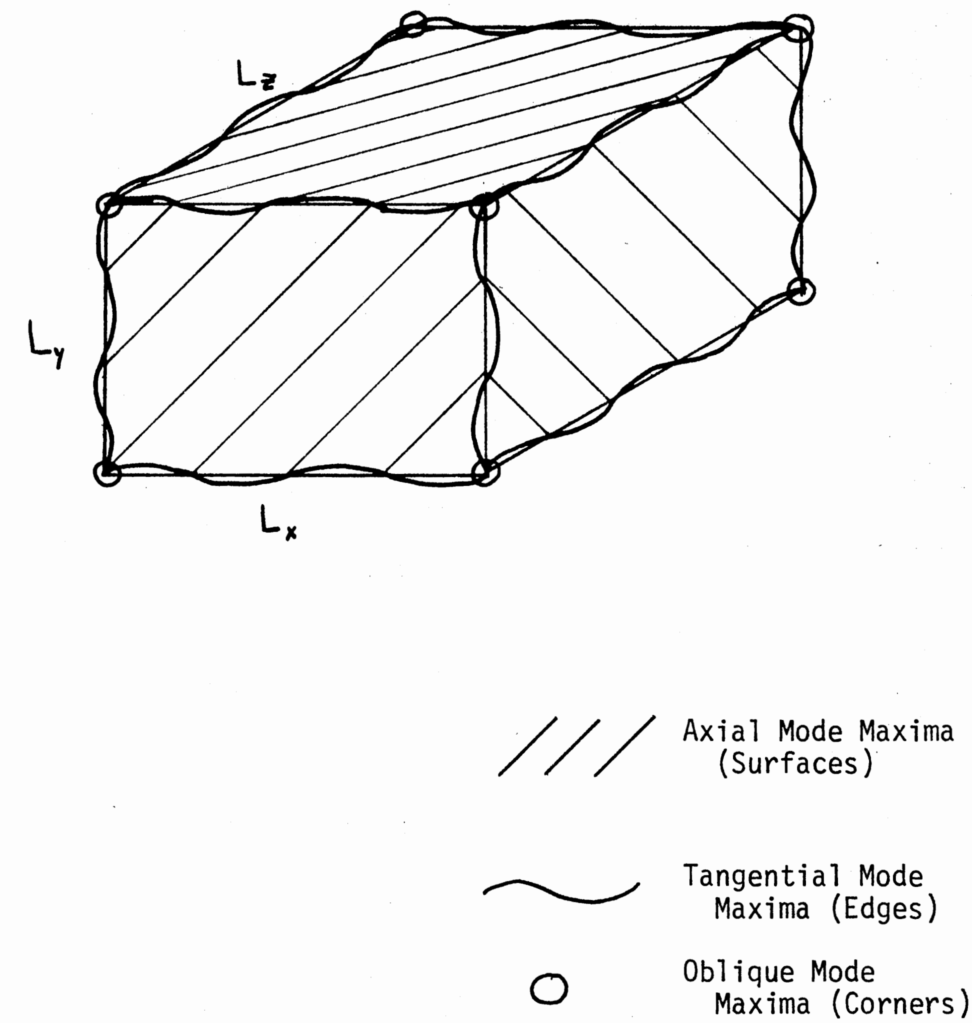

Figure 1. Potential Energy Maxima in a Rectangular Reverberation Room

The development used to support the following experiments is strictly classical. The position and nature of the sound source, as well as wall and air absorption effects, have not been considered. The theoretical model has recognizable potential difficulties, by being based on a loss less system in a perfectly rectangular room. Only experimentation can determine the extent of these model limitations.

CHAPTER III

EXPERIMENTAL INVESTIGATIONS

Procedure

A comparison of the accepted standard diffuse field measurement techniques with the proposed corner measurement technique must be made in order to show feasibility. Comparison tests were run using broad-band noise sources. A discrete frequency sound source was used to survey the sound field of the reverberation room before the broad-band noise tests were run. The survey tests were run using a small loudspeaker, with a microphone placed in the corner of the room. The loudspeaker was driven at various frequencies with a constant, root-mean squared, voltage input. The sound source was placed at a number of points on the floor of the room so that an estimate of the importance of position in regard to acoustic pressure in the corner of the room could be made. These points were located a distance of two meters or more from each other. None of the positions were within two meters of any wall nor within three meters of any major room irregularity. The two-meter distance was chosen to ensure that the sample points would be uncorrelated. A separation between microphones of more than one-half a wave length in a diffuse sound field has been shown to be uncorrelated (7). Using the principle of reciprocity sound sources separated by more than one-half a wave length should also be uncorrelated.

The primary feasibility demonstration of the corner measurement technique employed the use of borad-band noise sources. Broad-band noise sources were chosen because of the simplicity with which their acoustic pressure level may be measured in the diffuse field. Two different types of tests were run with the broad-band noise sources. The first was a comparison of the average sound pressure levels of two ILG reference fans. The standard diffuse field testing technqiue and the corner measurement technique were both used. This test was run to see the correspondence between the two techniques.

The second test investigated the effect of the sound source on the correspondence between the techniques. In these tests a loudspeaker, driven by a broad-band noise generator, was used as an alternate sound source. The algebraic difference between the corner and diffuse field measurements of the average sound power level was calculated as a correction factor. The correction factors obtained for the alternate sound source and the ILG fans were compared. Eight different one-third octave bands were used in these broad-band noise measurement technique comparisons.

In all of the tests which were run the corner microphone was lying on the floor of the reverberation room at an angle of 45 degrees from the vertical walls of the chamber. The microphone was equipped with a random incidence corrector which was placed against the walls of the chamber. A graphic level recorder was used to collect data for the determination of the average value and standard deviation of the acoustic sound pressure level. The microphone signals were filtered into approximate one-third octave bands by a frequency analyzer. The basic procedures of American National Standard S1.21-1972 were followed in the diffuse field measurements. A six-position microphone array, using a movable microphone boom, met the time/space standard deviations set by the testing standard. The six positions were sampled independently and the ensemble averaged. The noise sources were located at the same point in the room in all of the broad-band noise tests. This point was located in the central region of the floor of the enclosure.

Experimental Apparatus

The primary piece of equipment used in reverberation room testing is the reverberation room itself. The reverberation room used in these tests is located in Stillwater, Oklahoma, under the direction of the School of Mechanical and Aerospace Engineering of Oklahoma State University. The room is unusually large with a volume of approximately 810 cubic meters (11.5 x 9.7 x 7.3 meters) which causes air absorption to be a major concern at high frequencies. The ceiling of the room is irregular. Beams and slanted panels provide reflection and dispersion effects that are not easily evaluated. Another major irregularity of the room is a wooden staircase, which will also produce unknown reflection and diffusion effects. A one-third octave band analysis of the reverberation times and sound absorption coefficients for this room may be found in the Appendix. The effects of the room's irregularities will limit the confidence of any conclusions drawn in the extreme frequency ranges. The low frequency ranges have few excited modes. This results in a non-uniform sound field. The room's irregularities will compound the problems associated with this non-uniform sound field. Measurements in the high frequency ranges will be largely affected by reflection and beaming effects and by air absorption.

The Oklahoma State University Reverberation Room has been used in previous investigations in which the testing procedures outlined by the American National Standard S1.21-1972 were applied. The most extensive of these was written by David Ross during the summer of 1973 (14) (15). These previous tests showed the reverberation room to be well within the qualification requirements for the testing of broad-band sound sources. The broad-band noise source which was used in the qualification testing of the reverberation room was a standard ILG reference fan. The room did not meet the criteria for measuring discrete frequency type sound sources. The broad-band and discrete frequency tests were run using both six- and twelve-microphone arrays and ensemble averaging. Use of more than six microphone positions yielded no significant improvement in the accuracy of the broad-band sound power level measurements. Advancement was made toward the qualification of the Oklahoma State University Reverberation Room in regard to discrete frequency sound sources by Mark Sanborn in 1977 (16). In the later tests the microphone array used by Ross was replaced by a traversing microphone system. The room once again failed to meet the requirements for discrete frequency sound sources in two of the four different one-third octave bands tested, but the error level had been reduced from that obtained by Ross. The broad-band qualifications were met and surpassed by the traversing microphone technique. This reverberation room has historically met all of the standards and qualification requirements necessary for the testing of broad-band noise sources using the simplest of instrumentation systems, but to date has had only limited success in meeting the criteria for testing sources with discrete frequency components. The relative ease with which an accurate broad-band measurement may be obtained and an inability to meet the standards necessary to do meaningful discrete frequency testing have limited this investigation to broad-band noise sources. The previous works indicate that a six-microphone position array, along with ensemble averaging of the sound pressure level, will be quite accurate for the purposes of this study.

The measuring system is comprised of a microphone with a random incidence corrector, a preamplifier, a frequency analyzer, and a graphic level recorder. This measurement system is quite adequate for feasibility tests; however, the filter characteristic of the frequency analyzer is not standard. This generates some discrepancy between what this thesis calls a one-third octave band and what is defined as a standard one-third octave band. A standard one-third octave band has a bandwidth of approximately 23 percent of the center frequency. The closest approximation to a one-third octave band available on the frequency analyzer which was used had a bandwidth of only 21 percent of the center frequency. The skirt shape of the analyzer's filter differs from that of the standard one-third octave filter. These discrepancies are unimportant in a feasibility study, but would have to be standardized for a day-to-day use in the noise industry.

Results

The average sound pressure level was found to be both frequency and position dependent in the discrete frequency survey tests. There was as much as a fifteen-decibel difference in the sound pressure level with changes in the source position. All of the sound source positions tested were in the diffuse sound field of the reverberant chamber. The sound source, a loudspeaker, had a constant power input and was driven with a fixed set of frequencies. Deviations in the corner sound pressure level were found to be frequency dependent. A wide range of acoustic sound pressure levels was generated in the corner measurements whenever any two source positions, with the same set of input frequencies, were tested. No simple relationship was found between the sound pressure level measurements and the sound source position in the room. These results are not surprising when considering the acoustic pressure lattice concept of a reverberation room. The lattice idea is based on the concept of acoustic pressure nodes and anti-nodes. Acoustic pressure nodes exist throughout the room, so that any position in the room has the possibility of lying at one or more of these nodes. It is impossible to excite a standing wave in a room at a nodal point. A standing wave will be driven to its fullest extent at an anti-node. A mode driven at an anti-node could exhibit unproportional dominance in the corner of the room. While a mode that is being driven at a nodal point could be absent from the frequency spectrum being measured in the corner of the room. This observation of extreme position dependence of the sound pressure level, as measured in the corner of the room, immediately limits the feasibility of the corner technique. If a corner measurement technique for the determination of sound power levels is to be used, the location of the sound source will be limited to a small region of the reverberation chamber.

The comparison tests, using two ILG reference fans for sound sources, demonstrated a fixed relationship between the diffuse field measurement and the corner measurement techniques. Both of the ILG reference fans were tested using a standard diffuse field technique and the corner measurement technique. The fans were tested separately at the same position and orientation on the floor of the reverberation room. The diffuse measurements surpassed the criteria set by American National Standard S1.21-1972 with regard to allowable time/space standard deviation. The two ILG reference fans generated a stable broad-band noise spectrum (Table I). The fans are essentially identical and display similar noise spectrums when measured with standard techniques. The similarity in the diffuse field measurement is repeated in the corner measurement. Sound sources located at the same floor position, generating similar noise spectrums, with the same directional characteristics, appear to generate like sound pressure levels in the corner of the room. Simply stated, like sources generate like noise spectrums in both the diffuse field and corner measurement techniques. The corner measurement exhibits the same basic noise spectrum for both fans and the diffuse field measurements exhibit the same basic noise spectrum for both fans. However, the corner and diffuse field noise spectrums differ. This difference is found primarily in the higher frequency ranges, where substantial air absorption is present. In the lowest one-third octave band investigated, the corner measurement of the sound pressure level was found to be larger than the diffuse field measurement. This result could be due to the pressure amplification properties of the corner, or to a directionality characteristic of the sound source. The attenuation of the high frequency signals in the corner measurement can be assumed to be a result of the air absorption effects. The large volume of the test room and the room's irregularities lead to questionable conclusions in the higher frequency ranges. The room's obstructions will produce irregular scattering effects in the higher frequency ranges. This scattering may be dispersing the sound field in such a way as to shadow the corner acoustically.

TABLE I

ILG REFERENCE FAN TEST DATA

One-Third Octave Band (Hertz)	Sound Pressure Level (dB) Diffuse Field*	Corner Measurement
	Fan I	
100	76.6	79.5
200	76.1	76.0
400	76.0	75.5
800	75.8	74.4
1600	75.7	73.5
2500	75.7	71.1
5000	75.4	69.5
10000	75.3	68.8
	Fan II	
100	76.5	79.2
200	76.1	76.2
400	77.0	76.1
800	77.2	74.5
1600	76.5	70.6
2500	75.0	70.6
5000	76.0	70.5
10000	74.5	69.0

*The diffuse field measurement was made with a six-position microphone array in accordance with American National Standard S1.21-1972 (2).

The tests have shown the feasibility of a quick test of like broad-band noise sources using a comparison procedure. This kind of test could be used in quality control applications where an approximate output noise spectrum is known for a type of noise source and a particular unit needs to be tested. The test runs have used only broad-band noise sources; therefore, no conclusions can be drawn for sound sources with discrete frequency or narrow band components.

The diffuse field to corner measurement relationship was shown to be dependent on the sound source in the alternate sound source tests. A loudspeaker, driven by a random noise generator, was substituted for the ILG reference fan in these tests. The diffuse field and corner measurement tests were run in the same manner as the earlier tests. The time/space standard deviations of the sound pressure level was found to be on the order of one decibel. This standard deviation is small enough to show the basic trends in the two measurements and was approximately constant throughout the frequency bands tested. The deviation can be assumed to be a characteristic of the noise source itself and not one of the room or instrumentation system (see Table II).

A relative correction factor was used to compare the two types of sources, the ILG reference fans and the loudspeaker. The correction factor used was the algebraic difference between the diffuse field sound pressure level and the corner measurement of the sound pressure level. The data obtained from the two ILG reference fans was averaged to generate a single correction factor for the purpose of comparison. The reduced data (Table III) show that there is very little correspondence between the two correction factors. Large differences are found in the higher frequency bands. The lower frequency ranges show a relatively

TABLE II

LOUDSPEAKER BROAD-BAND NOISE TEST DATA

One-Third Octave Band (Hertz)	Sound Pressure Level (dB)	
	Diffuse Field*	Corner Measurement
100	79.2	76.6
200	72.4	69.5
400	66.7	65.7
800	62.8	64.2
1600	62.3	63.8
2500	61.5	62.7
5000	61.6	61.4
10000	62.0	61.5

*The diffuse field measurement was made with a six-position microphone array in accordance with American National Standard S1.21-1972 (2).

TABLE III

CORNER MEASUREMENT CORRECTION FACTORS

One-Third Octave Band (Hertz)	Correction Factors (dB)*	
	Loudspeaker	ILG Fan
100	2.6	-2.8
200	2.9	0.0
400	1.0	1.0
800	-1.4	1.4
1600	-1.5	2.2
2500	-1.2	4.5
5000	0.2	5.7
10000	0.5	6.0

*The correction factor is the algebraic difference between the diffuse field sound pressure level and the corner sound pressure level measurements.

close (3 dB) correspondence between the two measurements. This is an acceptable error band for the approximate determination of sound power levels. Discrepancies above the 1600 Hertz one-third octave band are too large for even approximate sound power measurements. Air absorption becomes very important in these frequency ranges. Directional properties of the sound source may also become significant at these frequencies. A loudspeaker is a highly directional sound source. These factors raise serious questions as to the applicability of a corner measurement technique for high frequency measurements in a room with dominant air absorption characteristics. The application of a corner measurement technique is severely limited by the apparent source dependence of the correction factor which is needed to relate the corner technique to the established diffuse field techniques.

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. A corner measurement technique for the determination of sound power levels is feasible for broad-band noise sources if:
    a. The sound sources tested have the same basic noise spectrum and directional properties.
    b. The corner measurement has been calibrated against a standard diffuse field technique.
    c. Only one sound source position and orientation is used.
2. Approximate sound power level measurements may be made for broad-band noise sources in large reverberation rooms using a corner measurement technique if:
    a. An uncertainty of three decibels is acceptable.
    b. The testing is limited to the frequency bands in which air absorption does not dominate.
    c. A single source position is used.
    d. The correction factor for that source position has been calibration with a known source.

Recommendations

This study, using its definition of feasibility, has identified only one possible application for the corner measurement technique. This application is in testing large numbers of products in a quality control operation. A program to develop a "practical" single microphone data acquisition system for rectangular reverberation rooms would be in order for this case.

The instrumentation necessary for a practical system would consist of a corner microphone (pointing into the center of the room), a preamplifier, a spectrum shaper, and a spectrum analyzer. This system and a standardized system would have to be available to generate an accurate correction factor for the spectrum shaper. The qualification tests for both broad-band noise and discrete frequency sound sources, as found in American National Standard S1.21-1972, should be made for both instrumentation systems. If the correction factors for the discrete frequency tests and the broad-band noise tests correspond, then a practical system could be established. The chief advantages of the corner microphone technique are its simplicity and a lack of physical obstructions in the test room. These warrant the time and expense of further investigation.

A practical system could possibly lead to other applications such as:

1. Progressive improvement tests, where a product is modified to reduce its noise level and is tested a multitude of times.

2. Improvement on the rough estimate application (conclusion 2) by rotating the sound source to offset the directionality problems.

3. Testing of air-moving devices such as fans and air conditioning units. This technique would allow normal operation and air flow of such units while making noise measurements. The measuring position is outside the normal region of air discharge.

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APPENDIX

TABLE IV

OKLAHOMA STATE UNIVERSITY REVERBERATION ROOM CHARACTERISTICS

One-Third Octave Band	Reverberation Time (Seconds)	Average Sound Absorption Coefficient ($\bar{\alpha}$)
100	15.5	0.015
125	14.1	0.016
160	14.7	0.016
200	15.6	0.015
250	12.7	0.018
315	12.3	0.019
400	11.0	0.021
500	10.6	0.022
630	10.3	0.023
800	9.5	0.025
1000	9.3	0.025
1250	8.2	0.029
1600	6.8	0.034
2000	6.2	*
2500	5.3	
3150	4.2	
4000	3.6	
5000	3.0	
6300	2.5	
8000	2.1	
10000	2.0	

*The dominance of air absorption at frequencies above this point prevents calculation of the sound absorption coefficient beyond this point (see Reference (14)).

VITA 2

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Master of Science

Thesis: FEASIBILITY STUDY ON THE USE OF A SINGLE CORNER MICROPHONE POSITION IN THE DETERMINATION OF SOUND POWER LEVELS OF BROAD-BAND NOISE SOURCES IN A LARGE REVERBERATION ROOM

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