ANECHOIC CHAMBER DESIGN AND

ACOUSTICAL ANALYSIS OF ROOM 1-051

by

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ABSTRACT

An anechoic chamber was needed to isolate sensitive optical equipment mounted on an optical table in room 1-051. The design involved developing methods to attenuate sound generated in the room, sound transmitted through the walls, and vibrations transmitted through the floor. An analysis of the present conditions was done to determine the design objectives and to provide a basis for comparison with the treated chamber.

Secondary walls insulated with fiberglas will provide attenuation of the sound transmitted through the walls. The ceiling will be lowered to ten feet and a fan installed to provide air flow and equal heating. All of the inner surfaces except the floor will be coated with a 4" layer of Sonex acoustical foam. A double door system leading into the hall should provide enough insulation to maintain the chambers acoustic integrity. The total cost of the project is estimated to be $5,000.

The experimentation showed little correlation between sound pressure and mechanical vibration of the test stand. It appears that sound pressure greater than 70 dB are necessary to produce vibrations large enough to be measured with our equipment. The optics table isolators provided by the manufacturer appear to successfully eliminate all transmission of vibrations from the floor.

Thesis Advisor: Alex Slocum

Title : Assistant Professor of Mechanical Engineering

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I would like to thank Alex Slocum for his guidance, friendship, and patience on not just one, but two seperate projects. A special thanks goes to Jeff Morse at Eckel Industries, Inc. who explained what it all really meant, without trying to sell his product.

The brothers of SAE deserve mention for providing the atmosphere that made it all bearable. Thanks, Nancy, for proof reading so carefully. And most of all, Mom and Dad for supporting me all the way through and letting me make my own mistakes.

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CHAPTER 1

INTRODUCTION

Echo-free chambers are used for three general applications. 1) They are used when experimentors want to determine the sound output of machinery such as gas turbines or compressors, or 2) to determine the effects of direct noise on some object. For example tests have been run to examine jet engine noise and determine its effect on the hull of an airplane. 3) Anechoic chambers are also used to isolate delicate instrumentation from outside sources. The latter was the purpose of designing an anechoic chamber in room 1-051 at M.I.T.

An optics table was purchased for the development of a laser interferometer based measurement system. The equipment is highly sensitive and can be upset by vibrations with amplitudes as small as 0.1 micron. Such vibrations could be either airborn or transmitted from the floor to the table. The project consisted of designing a suitable anechoic treatment for the walls of the chamber and evaluating the performance of the pnuematic legs that were inteded to provide isolation from the floor.

This thesis will first introduce sound and vibration analysis and detail the measurement method. Then it will discuss the constraints and the details of the design. Lastly, it will examine the analysis and the importance of the results. Unfortunately, the construction could not be completed due to financial considerations, but the plans will be carried out next fall. This means, however, that no data could be assimilated for the treated chamber and no quantatative analysis of the design could be made at this time. Such an analysis will be done when the room is completed in the fall.

CHAPTER 2

HISTORICAL PERSPECTIVE

Anechoic, or echo-free, chambers operate on the principle that the power of reflected sound waves could be reduced by lining all of the reflective surfaces with acoustically absorbant material. Ideally, the reflected waves could be reduced to the point where their effect on the systems and instrumentation in the chamber was negligible. The design of echo free chambers began during the second world war and has reached the point where noise reduction levels of 99.9 % are possible.

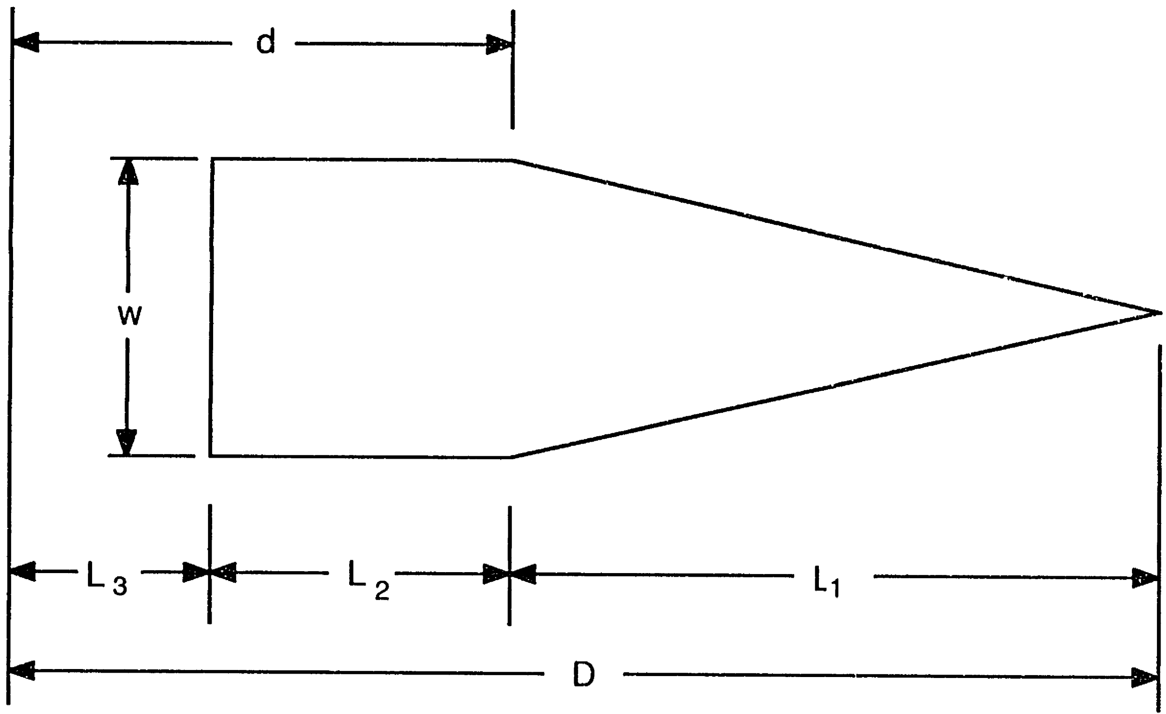
The first anechoic chamber was designed in 1936 by E.H. Bedell. It was lined with $18\frac{1}{2}$" of layered muslin and flannel [1]. Another chamber built a few years later in Berlin used pyramidal muslin bags stuffed with rock wool to a density of about 12 lbs./ft^3 [1]. Though these chambers reduced the reflected noise, the first real effort to optimize the design was not made until 1946 when L.L. Beranek and H.P. Sleeper, Jr. published a paper on the subject [1].

The investigation of anechoic chambers in the U.S. began during World War II when the National Defense Research committee sponsored a program in acoustical research at Harvard University's Cruft Laboratory. The program was designed to study acoustic phenomenon and develop acoustic instrumentation. The conditions of the study required the ability to differentiate between source waves and reflected waves, therefore the design of the chambers needed to be optimized [1].

The investigators were faced with two immediate questions: what material and in what shape? Beranek and Sleeper examined five different structural shapes and over a dozen different materials [1]. The testing was performed using the impedence tube method which had been detailed by H.J. Sabine in 1942 [2]. A noise source was placed at one end of a 15 foot long steel tube and the structure under examination at the other. A sound wave propogated down the tube was reflected back by the structure. By examining the standing wave pattern, the percentage of sound reflection could be calculated [3,4]. The results of their experimentation show that the linear wedge structure had the lowest percentage of sound pressure reflected. The exponential tapered pyramidal structure was nearly as effective, and could possibly be improved with more research, but the cost of design and construction made the further study of this configuration impractical at the time. The material chosen for the main body of the wedges was PF Fiberglas with a density of 2.5 lbs./ft^3 [1].

Beranek and Sleeper continued their investigation of the linear wedge structure to determine the optimum dimensions and construction method. By adopting the sound absorbtion coefficient, α, they set a standard which the industry has followed ever since. Their experimentation revealed that the critical variables in the wedge design were the base depth, d, and the resistence to flow, R_f, of the material (See Fig. 1). The angle of incidence also proved important and could be varied by adjusting the taper length. They tested fiberglas wedges with densities between 2.5 and 9 lbs/ft^3 and showed that the lower the density, the more effective the material [1]. The effectiveness of a material was determined by its *cutoff frequency*, which Beranek defines in the following excerpt:

> "The cutoff frequency is defined as that frequency at which the pressure reflection rises to 10 percent of the pressure in a normally incident sound wave. This corresponds to the frequency at which the absorption of sound energy drops to 99



D = Total Depth
L1= Taper Length
L2= Base Length
L3=Airspace Length
d = Base Depth
W = Base Width

Figure 1. Dimensions of Linear Wedge Structure.

percent or at which there is a sound reduction of 20 dB for a single reflection.[1]"

The determination of the cutoff frquency has been refined to a simple test that can be quickly preformed in an anechoic chamber to determine the value for the entire room, not just a single wedge of material. A sound source is placed in one corner of the room and measurements are taken on a line from the source to the center of the room. The cutoff frequency is the lowest frequency for which there is a 6 dB reduction in the sound pressure level with each doubling of the distance from the source. Thus the sound level is 6 dB lower at 4' than at 2', and 12 dB lower at 8' than at 2'. The dimensions have been standardized and it has been determined that the depth of treatment, D, must be one quarter of the wavelength of the cutoff frequency [5]. This relationship is shown in Figure 2.

The results of this first in-depth study have been used widely by the industry and have become the basis on which several national standards have been established [3,10]. Along with the work done by B.G. Watters in 1958 on the design of wedges [6], and H.J. Sabine on the impedence tube method [2], Beranek and Sleeper's report has raised acoustic engineering from the level of guesswork to that of a real science in a span of ten years.

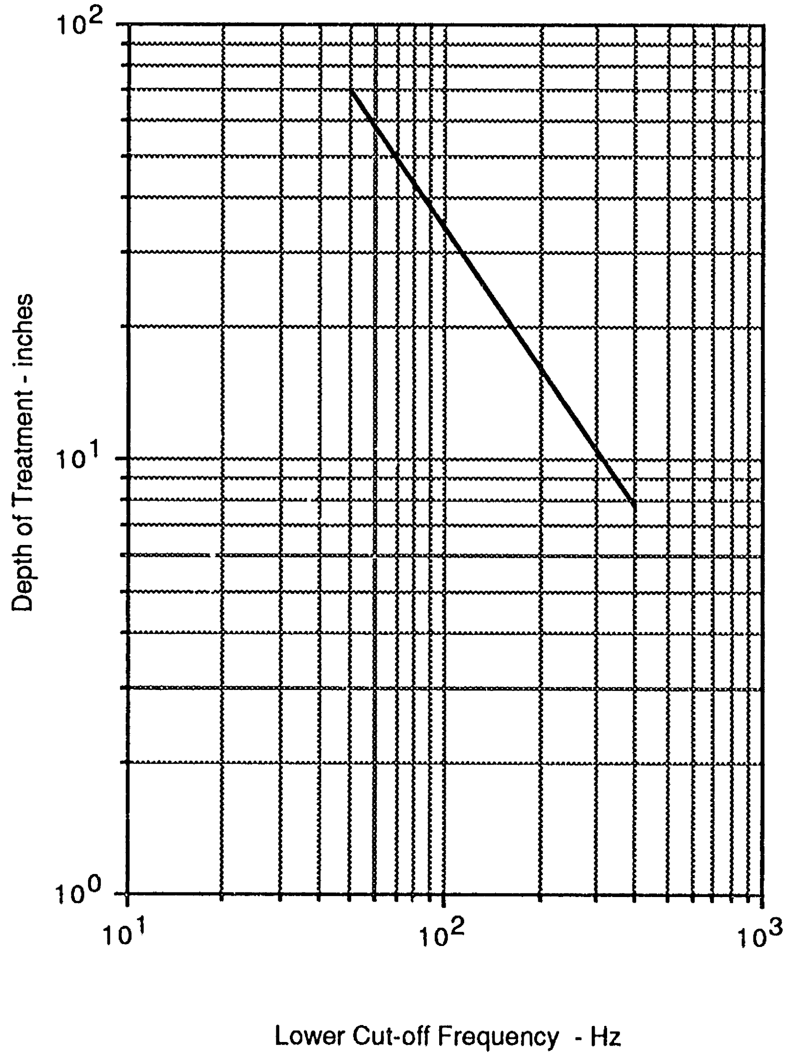


Figure 2. Cutoff Frequency as a function of Treatment Depth.

CHAPTER 3

THE NATURE OF SOUND AND VIBRATION

Although the basic theories of vibration have been understood for a long time, the science of controlling, or attenuating, these waves has only recently been studied and implemented in the engineering world. Because of the degenerating effect excessive noise can have on human hearing, sound waves are the most common form of vibration that needs to be controlled. The range of applications extend from noise analysis at airports and along highways to sound booths in the music industry. In conditions where there is large machinery it is desirable to eliminate a much wider range of vibrations and to avoid resonance. This condition can result in exaggerated vibrations in the object which can quickly lead to fatigue or failure and sacrifice the accuracy of delicate instrumentation. In order to accurately analyze a situation, we must examine not only the natural frequency of the object, but the excitation waves also. Thus it is imperative that we have a working understanding of vibrational waves before we attempt to control them.

3.1 VIBRATIONAL WAVES

In any real life situation, a vibrational wave will be made up of several wave forms of different magnitudes and different orientations. We will begin our study by defining those aspects that are common to all waves and then proceed to those specific to sound waves.

For analytical purposes we divide vibrations into three categories of wave forms, each of which can be accurately described with just two variables: amplitude and frequency. The *amplitude*, A, is the magnitude of particle motion in a given direction. In the case of *compression waves*, the magnitude is exerted along the axis of wave propogation. *Longitudinal* *waves*, on the other hand, have a magnitude perpendicular to the plane of motion, while *transverse waves* have a magnitude perpendicular to the direction of propogation but in the plane of motion (See Fig. 3). The *frequency*, f, is the number of times a periodic wave goes through one complete cycle in a given time period. It is usually measured in cycles per second, or Hertz (Hz).

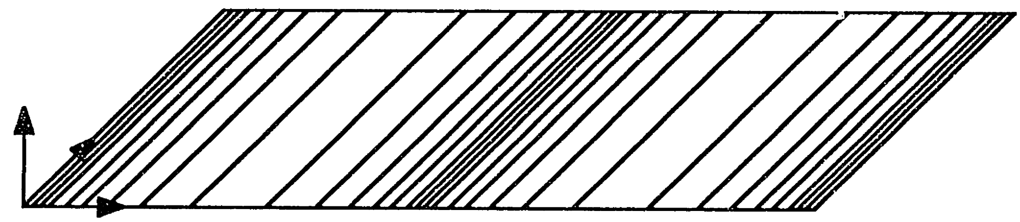
Most waves that we deal with are periodic, which means they proceed in a series of identical cycles. Only these waves can induce a sustained response from an object over any significant length of time. Aperiodic, or random, waves are less common and can produce only random fluctuations, not sustained vibrations. The period, T, of a wave and its wave length, λ, can quickly be calculated from the frequency using the relationship

$$\lambda = \upsilon T = \upsilon / f \qquad (1)$$

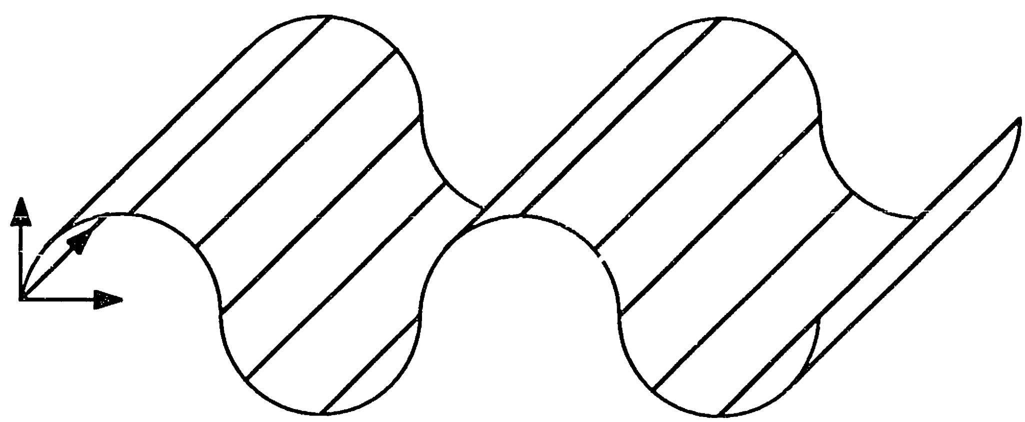
where υ is the velocity of the wave.

3.2 SOUND

By definition sound waves must have frequencies within the normal range of hearing for a healthy adult, which is 20 to 16,000 Hz. The velocity of a sound wave is given by C, which is characteristic of the medium in which it is traveling. The simplest, single frequency, periodic sound wave is heard by the human ear as a pure tone. Most other sounds can be modeled as a summation of harmonic waves whose frequency is an integer multiple of some base frequency. Fourier analysis is the process by which engineers can subdivide a give periodic wave into its sinusoidal, harmonic parts. The noise made by compressors, motors and fans are periodic while dishwashers and automobile engines emit aperiodic sounds due to their fluctuating rotational



(a)



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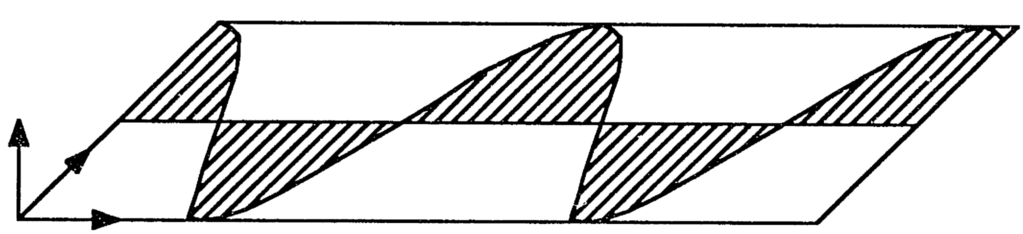

(c)

Figure 3. Wave forms. Wave Direction is Left to Right for a) Compression Wave, b) Longitudinal Wave, and c) Transverse Wave.

speeds [7]. The latter must be modeled using an infinite series of infinitessimal pure tones with distinct magnitudes.

Sound is most often modeled as a one dimensional compression, or progressive, wave [8]. Its speed is a function of the temperature and the medium through which it travels. The speed of sound in air at standard atmospheric pressure is given as:

$$C = 20.05\ (R)^{1/2} \quad \text{ft/sec} \tag{2}$$

where R is the temperature in degrees Rankine [9].

During the development of sound measuring equipment a confusing number of conventions have developed. Engineers in various fields have adopted their own conventions while the public uses another. For the sake of clarity we will attempt to stick to the most widely used convention throughout the report, but often reference data is based on one of the other systems. In order to avoid the confusion bound to arise, we will define the basic measurement systems now and describe their relationship to one another.

Most acoustic problems require the measurement of the force that the sound wave exerts on an object, be it a mechanical instrument or the human ear. The sound pressure is a function of the wave magnitude at a given time. Since the magnitude can be modeled as a sinusiodal function, or a summation of several such functions, the pressure, P, can be stated as a function of time, t:

$$P(t) = P_R \,\mathrm{Sin}\, [\ 2\,\Pi\, f\, t\ ] \tag{3}$$

Where P_R is the reference pressure and f is the frequency. In analysis, however, we generally only refer to the *root mean squared pressure* , P_{RMS}, which is simply the square root of the sum of the squares of the pressures. Algebraically;

$$P_{RMS} = \{P_a^2 + P_b^2 + \ldots + P_n^2\}^{1/2} \tag{4}$$

The *sound intensity*, I, is a measure of the sound energy that flows through a given area in a unit time. It can be calculated using the equation

$$I = P^2_{RMS}/\rho\, C \tag{5}$$

Where ρ is the equilibrium gas density.

The *power*, W, of a source can be determined by integrating the sound intensity over the surface area, S, upon which the wave acts:

$$W = \int I_S \, dS \tag{6}$$

Often it is neccessary to determine the sound power output of a single source. The American National Standard Institute provides a detailed testing method for determining the sound power levels of noise sources using anechoic chambers. The method involves taking ten sound intensity measurements in a sphere around the source and calculating the total power from the average sound intensity [10].

Many experiments call for the measurement of the sound pressure on some object due to a noise source of known power. Because of the wide range of sound levels in acoustic work (from about 10^{-10} to 10^{8} Watts) it is convenient to measure sound as a logarithmic ratio between the point under investigation and some reference point, W_0. Because the ratio is dimensionless, the log function gives us a answer in decibels (dB). Both the sound power and the sound pressure are measured on logarithmic scales.

The *sound power level* is given by

$$L_W = 10 \log_{10} W/W_o \quad \text{Re } W_o \tag{7}$$

The international standard reference power, W_0, is 1 picowatt (12^{-12} watts) [11].

Microphones are generally used to measure the *sound pressure level* at a point in space.

$$L_P = 10 \log_{10} (P^2/P^2_{ref}) = 20 \log(P/P_{ref}) \tag{8}$$

The convention for the reference pressure level is 20 micropascals. A similar relationship exists for the sound intensity level, however for most practical purposes the pressure level and the intensity level can be considered equal [5]. Thus the sound pressure level can be related to the sound power level using:

$$L_W = L_p + 10 \log S \qquad \text{dB re } 10^{-12} \text{ watt} \tag{9}$$

where S is the surface area over which the pressure is uniform.

Logarithmic ratios cannot be added or subtracted directly. Rather they must be converted back to their relative pressures and then combined. When we compute the new level we find that a doubling of the pressure only increases the sound level by 6 dB. Multiplying the pressure by ten produces an increase in the pressure level of 20 dB. Thus, if we wanted to reduce the sound pressure level in our anechoic chamber by 95%, we would need a reduction of:

$$20 \log (.05/1) = -26.02 \text{ dB} \tag{10}$$

Many acoustic instruments are further adjusted by what is called weighting. There are three different weighting scales: A, B, and C. *A-weighted* sound is adjusted to account for the fact that the human ear is not equally sensitive to all frequencies. For example a 1000 Hz tone at 50 dB and a 125 Hz tone at 66 dB sound equally loud to the ear. The A-weighted adjustments are listed in Table 1. Though usually designated by dB(A), this scale has been so widely adopted by acoustic engineers that this designation is often left off in their literature. The B and C scales were created for even more specific uses, but are rarely, if ever used anymore.

Table 1: Relative frequency Response of a Sound Level Meter with A-Weighting to Sounds Arriving at Random Incidence.

Frequency (Hz)	A-weighting (dB)
50	-30.2
63	-26.2
80	-22.5
100	-19.1
125	**-16.1**
160	-13.4
200	-10.9
250	-8.6
315	-6.6
400	-4.8
500	**-3.2**
630	-1.9
800	-0.8
1000	**0.0**
1250	0.6
1600	1.0
2000	1.2
2500	**1.3**
3150	1.2
4000	1.0
5000	**0.5**
6300	-0.1
8000	-1.1
10,000	**-2.5**

CHAPTER 4

THEORIES OF SOUND AND VIBRATION CONTROL

In analyzing vibration problems, be they sound or other waves, engineers almost always approach the problem from a systems point of view. The vibrational waves are the input and the objects they strike are the system parts. The reaction of the object is the system response. Objects react with a combination of the three possible responses: reflection, absorbtion, and transmission (See Fig. 4).

In the early days of acoustics, objects were rated by their reflection coefficient. However, it soon became easier to refer to an object's absorbtion coefficient. Whatever energy was not reflected must have been absorbed, and it seemed a more fitting measure of a material's accoustic properties. In dealing with an object's acoustical absorbtion, we refer to its absorption coefficient, α, to determine the percentage of energy absorbed or reflected by a material. Note that the absorbtion coefficeint is a function of the frequency of the input sound, and is generally lower for lower frequencies. The absorbtion coefficient is measured in Sabins/ft^2, or the number of Sabins absorbed per unit area. One Sabin is the equivalent absorbtion of one square foot of perfectly absorbant material.

The total absorbtion of the room can be calculated by multiplying the absorbtion coefficient by the surface area, S, of each surface of the room. For example, we can determine the total absorbtion of our 10.6' x 13.2' x 11.9' chamber before treatment. The calculations are shown in Table 2.The total absorbtion before treatment can be calculated simply by adding the absorbtion values for each surface: the floor, the ceiling and the four walls. This results in an absorbtion of 41.2 Sabins in the chamber [12].

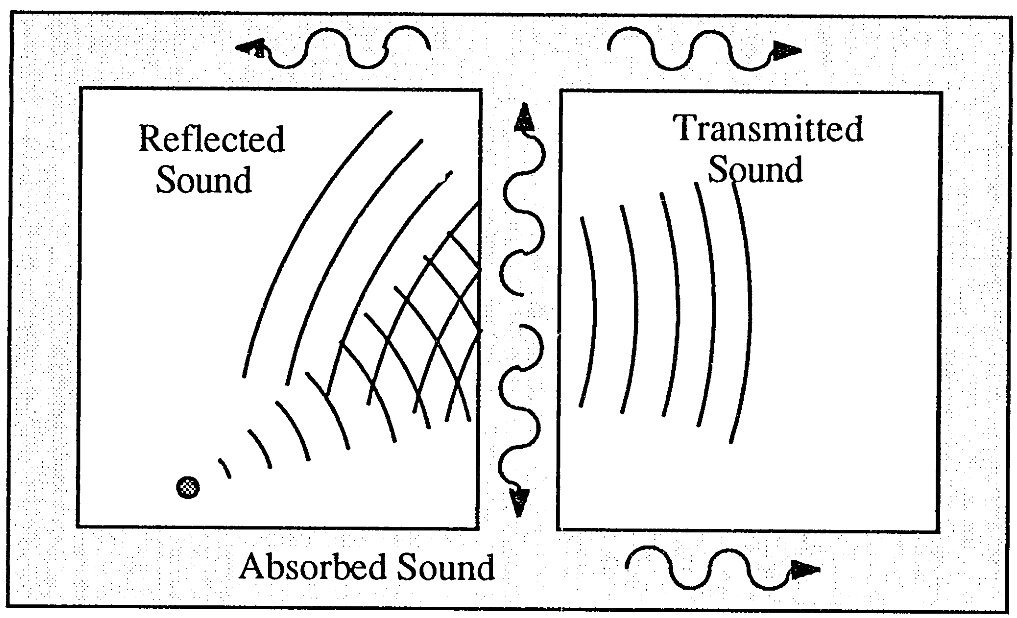

Figure 4. Sound Transmission Paths of Noise Emiited by a Source. Shows Reflected, Absorbed and Transmitted waves.

Table 2: Calculation of Sound Absorbtion of 1-050 Before Treatment.

Surface	α(sabins/ft^2)	S(ft^2)	A
Concrete Floor	0.02	141	2.8
Ceiling Plaster	0.03	141	4.2
Wall Plaster	0.06	570	34.2

Transmitted vibrations become a problem if we are working in an environment in which loud noises or strong vibrations are present due to uncontrollable activities such as trucks traveling on a nearby highway or the slamming of a door in a distant part of the building. These waves can be attenuated by reducing the power transmitted at each point where the wave must change mediums and insulating the room with sound damping material. The actual modeling of these systems is beyond the scope of this work, but we can apply several rules-of-thumb to our own design. From a simple systems point of view, the output can be greatly reduced by adding an appropriate mass-spring-damper subsystem. Essentially, energy is used in the excitation of each subsystem so that less is transmitted. If we are careful to avoid the natural frequencies of the system, this can be a very effective damping method.

Building secondary walls is a common method of adding such a subsystem to an anechoic chamber. The walls are supported by flexible mounts with a small air space between them and the original walls. The secondary wall, usually made out of gypsum board or likewise dense material, acts as the excitable mass and the air space as a pnuematic damper.

The effect is that the vibrations of the gypsum board tend to be much smaller than those of the excited wall.

Transmitted waves can also enter a chamber through the floor. Machinery in one part of the building can upset delicate instrumentation in another through the transmission of its vibration waves in the floor. Thus it becomes necessary to isolate either the machinery or the instrumentation, or both. Machinery vibrations are often attenuated by placing isolators underneath the feet which dampen the vibrations transmitted to the floor. Most often these isolators are made of Neoprene rubber or stiff metal springs, or a combination of both.

The methods of insulating the instrumentation are much more complicated. Occasionally it is done by isolating the entire slab of floor. This slab is mounted to the structural supports using stiff mechanical isolators, or by sinking its supports directly into the ground so that it does not touch the rest of the building. A more common method is to build a floating floor in which a structural slab is mounted on some resilient layer on top of the original floor.

CHAPTER 5

CHAMBER DESIGN

The chamber which we designed is not a true anechoic chamber. By strict definition, an anechoic chamber has absorbant treatment on all surfaces, including the floor. A room with the floor exposed, like ours, is called a hemi-anechoic chamber. Whereas an anechoic chamber mimics the conditions of sound high above the surface of the earth, a hemi-anechoic chamber mimics conditions on the surface of the earth with no reflective surfaces nearby.

The design of the actual chamber was constrained by many variables. The overwhelming consideration in the 10.6' x 13.3' chamber was space. Once the optics table was installed the governing dimension became the width. The workspace in this direction was reduced to 75.7 inches which had to accomodate two walkways and two wall treatments. Assuming that a person needed two feet on either side of the table as a minimum workspace, the maximum combined treatment thickness was only 27.7 inches, or 13.8 inches on each wall. This thickness was further reduced when the decision was made to install a 36 inch wide instrumentation cabinet in the south east corner. This reduced the possible wall thickness to 8 inches on each side.

The size constraint limited the maximum treatment thickness to 8 inches on the east and west walls, and therefore a minimum cutoff frequency of 400 Hz. It was stressed by members of the industry that the system was only as good as the weakest point, however it made sense to us that if some parts could be made more effective, they ought to be. Thus, since we had more room on the north and south walls, we added a thicker treatment.

Cost was the second most important constraint. The retail price for a custom made anechoic chamber is $100 a wedge. With three wedges per 4 square feet, it would cost nearly $54,000 to treat the entire chamber, which was not practical given the current financial situation. It would be possible to construct crude wedges on our own at a much cheaper cost, however, this would require a great deal of time and would necessitate cutting fiberglass boards, the dust of which may be cancer causing.

A much cheaper and less dangerous solution was found. Illbruck, in Minneapolis, produces an open-celled foam product called Sonex which is

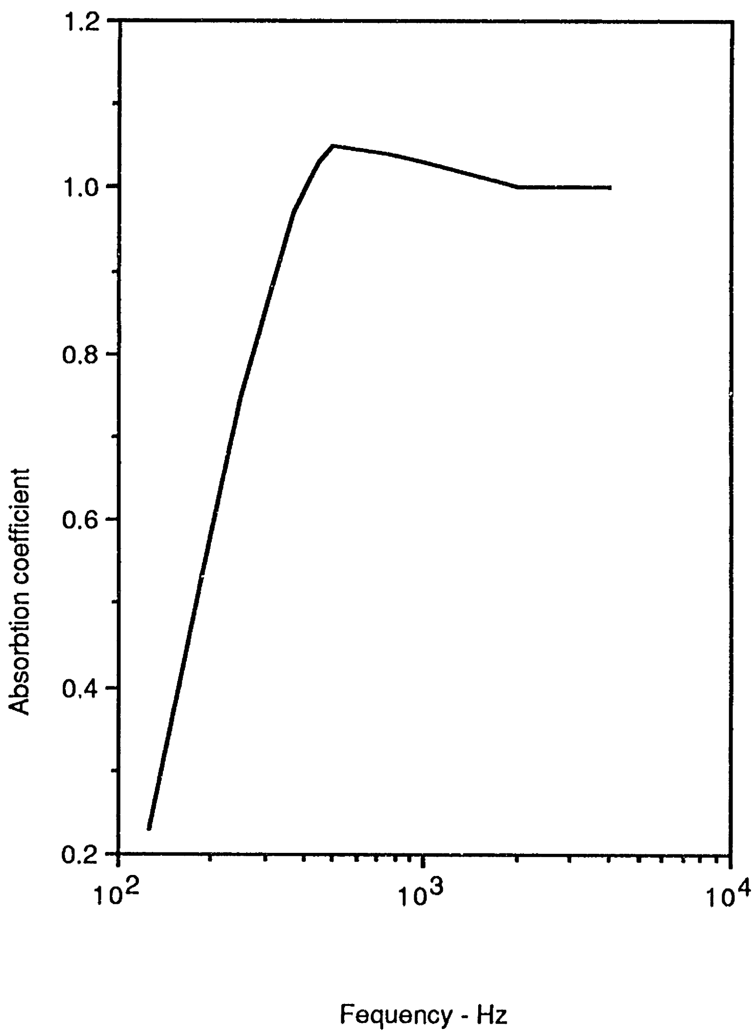

Figure 5. Sound Absorbtion Coefficient for
4" Sonex Across the Frequency Range

marketed as an inexpensive means of achieving high sound reduction [13]. The absorbtion coefficients for 4" thick Sonex are given in Figure 5. Recalling the method for determining absorbtion, we can calculate the total absorbtion of the chamber after treatment. The Noise Reduction Coefficient (NRC) for 4" Sonex across the frequency band is 0.96. Since it will cover 711 ft^2, the Sonex will provide a total of 682.6 Sabins of absorbtion. Added to the absorbtion of the floor, the chamber will have a total absorbtion of 685.4 Sabins as compared to 41.2 before treatment.

The fact that we were using only four inches of anechoic foam left us another four inches to install a secondary wall. Gypsum board is a suitably dense material for the secondary wall and would provide the necessary mass. By placing four inches of low density fiberglas foam between the primary and secondary walls, we could provide mechanical damping, and acoustic and thermal insulation. It would also prevent the propogation of sound waves parrallel to the wall in the air space. The overhead view of the chamber is shown in Figure 6 and the cross-sectional view in Figure 7. For safety reasons, we decided to extend the secondary wall only to eight feet, the height of a standard 4x8 sheet. The gypsum board should be $^1/_4$" thick and be attached as loosely as possible in order to minimize the vibrations transferred from the primary wall. One inch wide C-channels spaced every 12 inches along the top and bottom edges ought to provide enough support without too much transmission. The inset in Figure 7 shows a more detailed view of the attachment method.

The ceiling should be lowered to ten feet by constructing a frame with sixteen inch centers made out of two-by-fours (See Fig. 8). A support structure can easily be made by bolting two-by-fours into the wall joists. Then, ten foot long cross beams can be nailed to the support boards. The

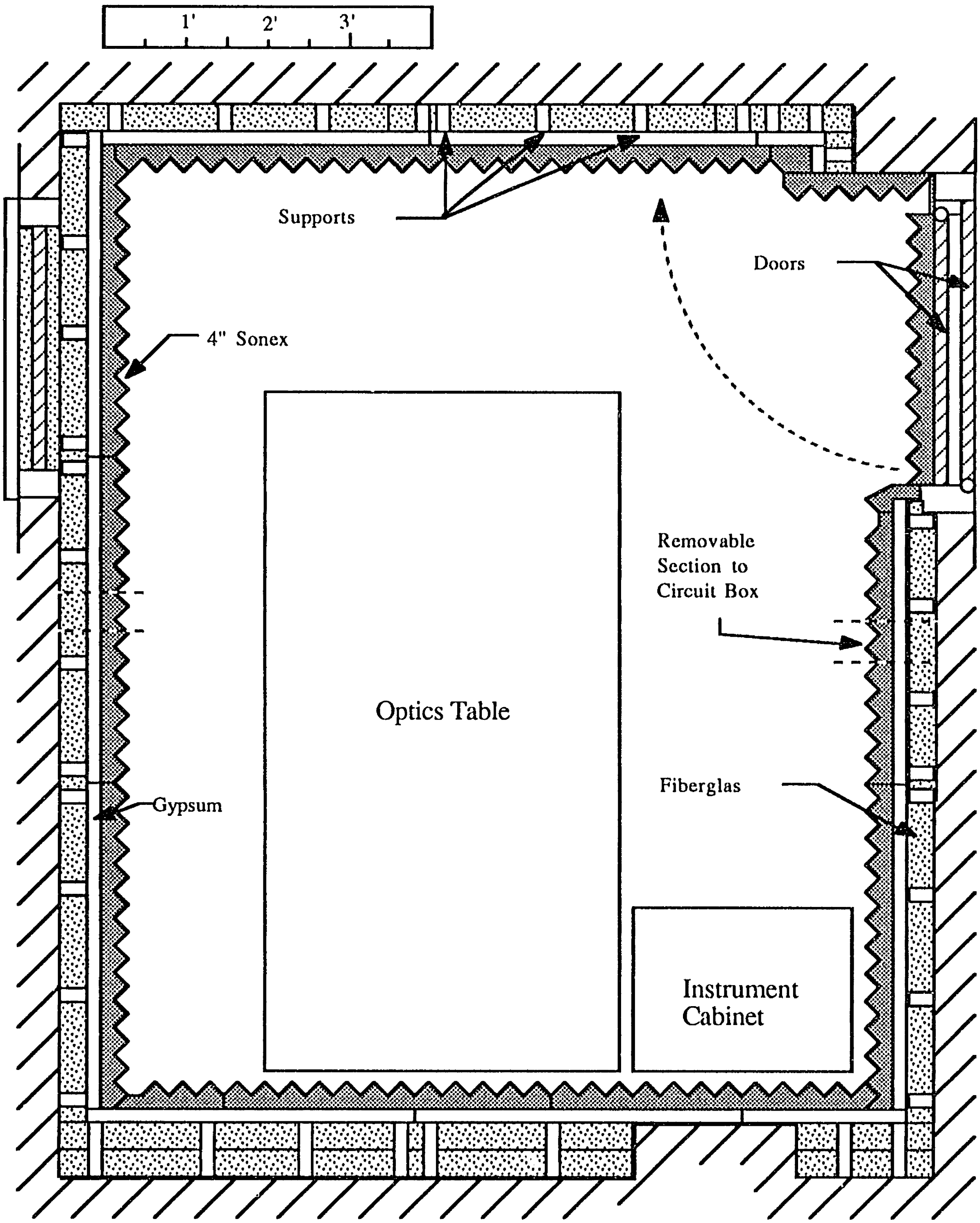

Figure 6. Overhead view of the anechoic treatment design for room 1-051.

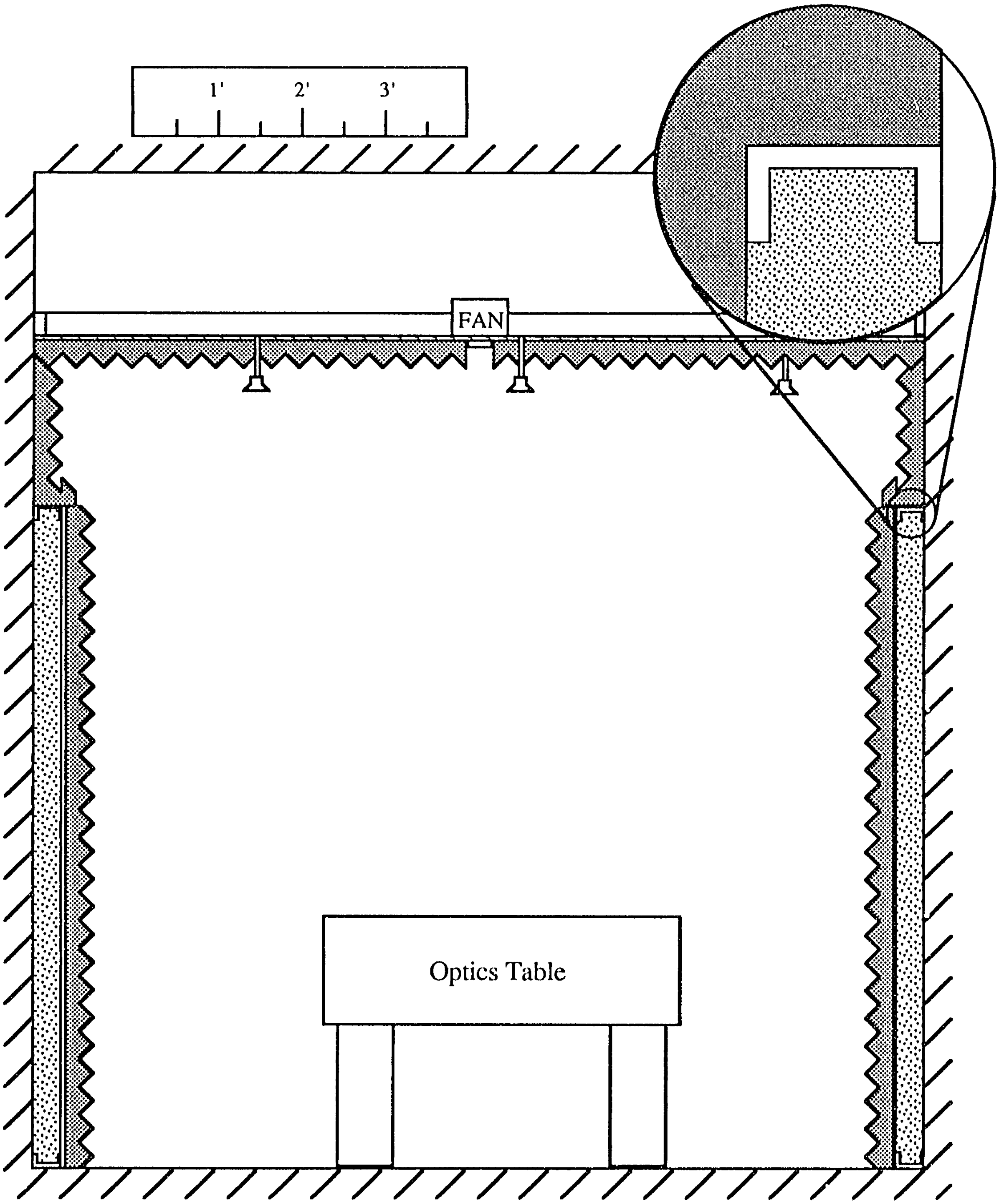

Figure 7. Cross-Sectional View of Anechoic Treatment Design.

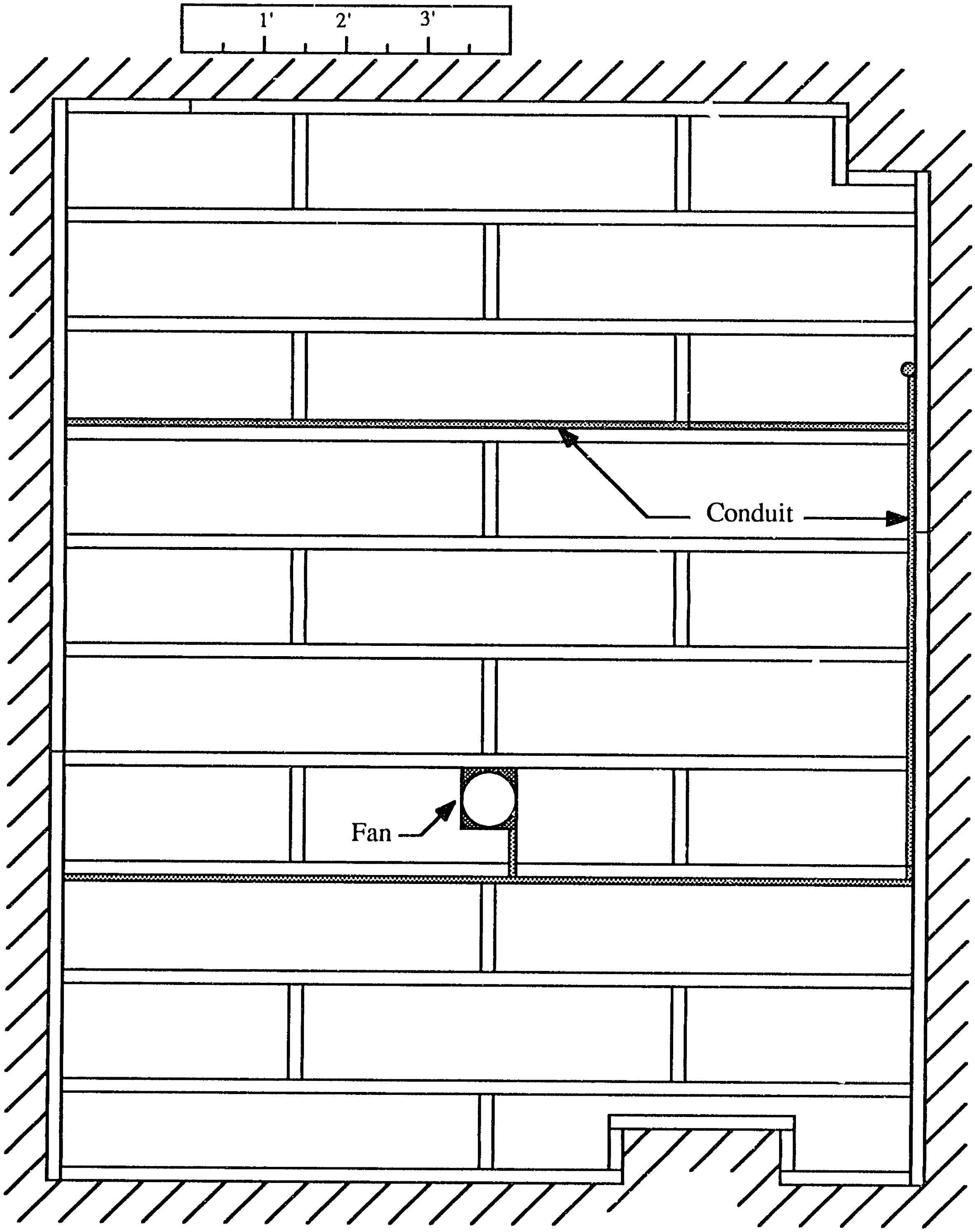

Figure 8. Ceiling Frame made of 2x4's. Shows possible Conduit Orientation and fan location.

gypsum board should be mounted to the underside of the cross beams, and Fiberglas sheets placed above the new ceiling. The Sonex can then be glued to the underside of the construction. Along the corners the Sonex should be meshed so that there are no air gaps and none of the supporting structures can be seen.

The flourescent lights should be replaced with flood lights and rewired so that the conduits are mounted to the cross beams. The lights can be supported on six inch hangers so that the bulbs hang approximatley one inch below the wedge tips. Whenever possible, the hangers should penetrate the valleys in the Sonex material. A fan should be mounted in the ceiling to provide air flow and equalize the temperature throughout the room. The overriding consideration when choosing a fan is that it be quiet.

The doors are acoustically the weakest portion of the room [5]. The west door will be nailed shut and boarded over on the outside so that it cannot be opened. A layer of fiberglas placed between the plywood and the door will provide extra insulation and damping. On the inside an extra layer of fiberglass can be installed to insure that there is no airspace in which vibrations might develop.

The east door should be reversed so that the original door opens into the hall. Because glass and gratings provide little acoustic impedence, they should be removed from the door or covered with a layer of gypsum and fiberglas. A secondary door will be built which will swing in and be acoustically insulated. It should be built out of two by fours and gypsum board, filled with fiberglas, and have a layer of Sonex applied to the chamber side. It is extremely important that the Sonex on the door meshes with that on the wall, or they will be destroyed when the door opens. To insure this, mount the sheet on the wall and then mesh another sheet onto it so that the back of the second sheet faces the room. Open the door and mark the exact location of the Sonex so that it can be mounted in the same position. It may even be possible to put the glue on the door and press it against the back of the meshed Sonex. In order to provide enough room for the door to open it may be necessary to install a spacer to move the hinges farther from the north wall (See Fig. 9).

It is crucial that the air gap between the door and the floor be as small as possible to prevent sound leakage. Some method should be devised to have both doors close tightly. Perhaps a lining can be placed around the frame that is compressed when the door is closed. This lining must extend across the floor also.

The vent leading into the room should be removed and if the duct cannot be boarded over, it should be lined with fiberglas to attenuate any sounds coming through the pipes. Any holes in the walls should be filled with fiberglas and then caulked. There are two circuit boxes in the chamber and these should be made accessible by cutting out a square of the treatment which can be removed by hand (See Fig. 6). The location of the boxes should be signalled by red paint or some other easily noticable method. Baffles will need to be built between the electronic instruments and the optical equipment. These baffles can be plywood coated on both sides with standard acoustic foam.

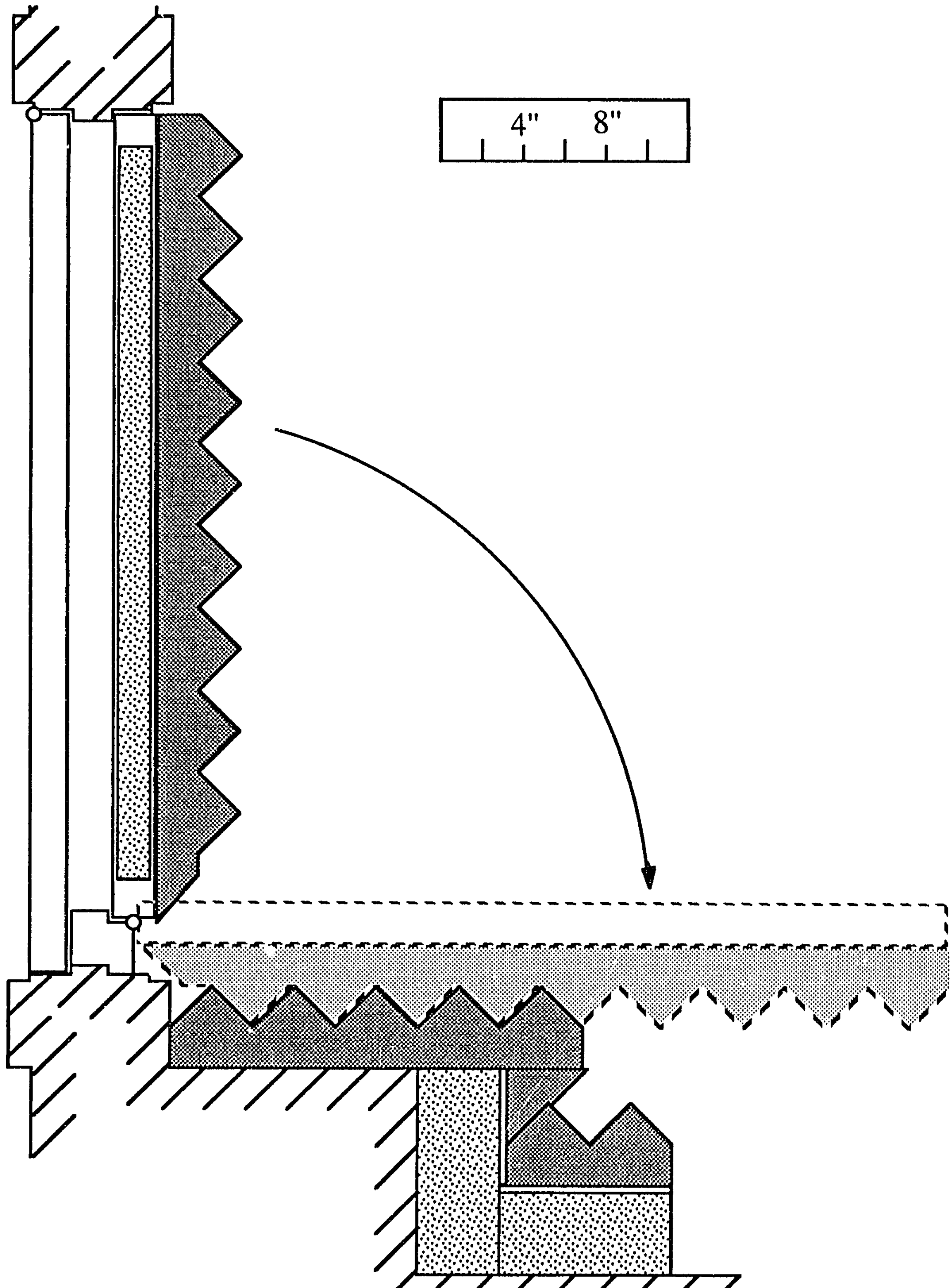

Figure 9. Detail of East Door Design.

CHAPTER 6

EXPERIMENTAL RESULTS

In order to measure the effectiveness of the anechoic treatment, and of the Melles-Griot optics table isolation system, we ran several experiments which can be compared to data taken when the room is finished. The object of the first series of experiments was to establish a correlation between the sound pressure level and the mechanical vibrations of the instrument support equipment. The second series of tests was run to determine the relationship between floor vibrations and table responses.

The measurements were taken using a condensor microphone made by B&K Instruments, Inc. with a 0.52" diameter cartridge. Mechanical vibrations were measured with a servo accelerometer (Sundstrand Data Control, model 300A1). Both were mounted on instrumentation stands provided by the table manufacturer to support optical equipment. The data was recorded and analyzed using a Hewlett Packard dynamic signal analyzer (model 3562A).

During the first series of tests the HP was in Sine Swept mode in which it averaged ten data points at each frequency. A random noise source was pointed in various directions and readings were taken. Figure 10 shows the sound pressure levels when the noise source was off. This data can be assumed to be the average background noise transmitted into the room from the environment. Note that the peak at 60 Hz is due largely to the electrical noise given off by the flourescent lights. Above 350 Hz the pressure levels are negligible.

The pressure levels created when the source is pointed directly at the microphone at a distance of 6" are shown in Figure 11. In this case there are

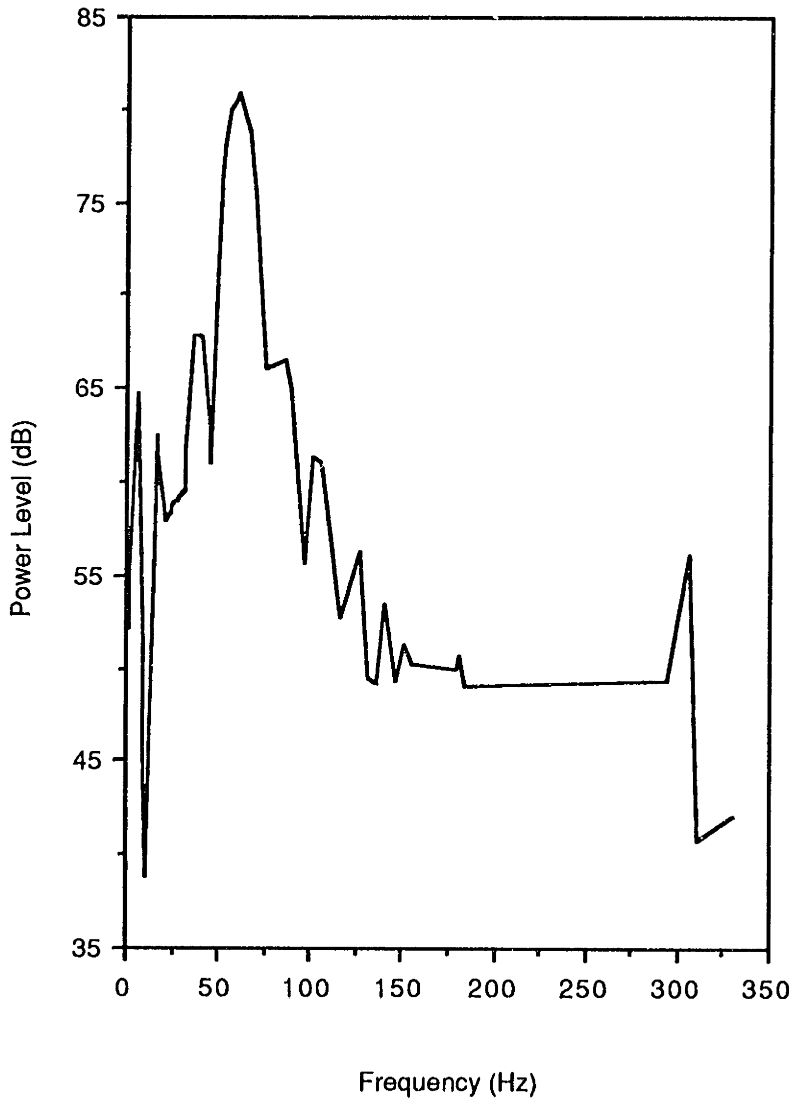

Figure 10. Background Noise in Room 1-051.

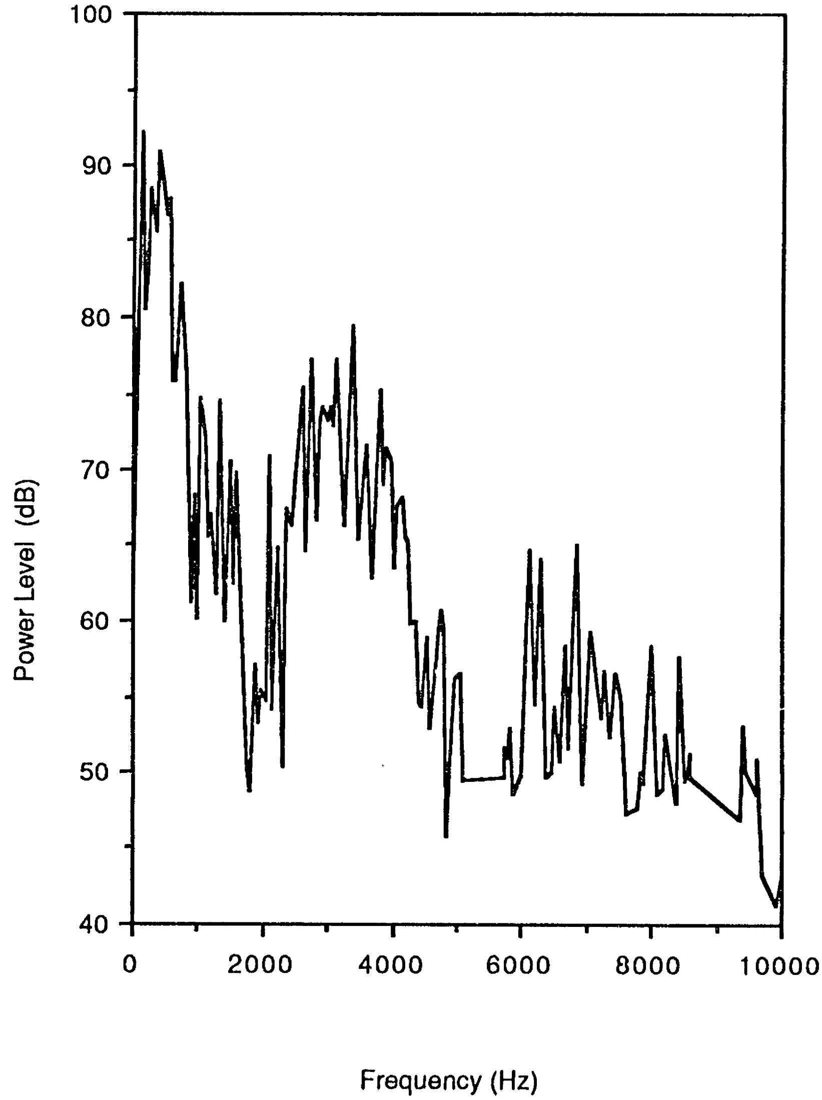

Figure 11. Sound Levels of Direct Noise

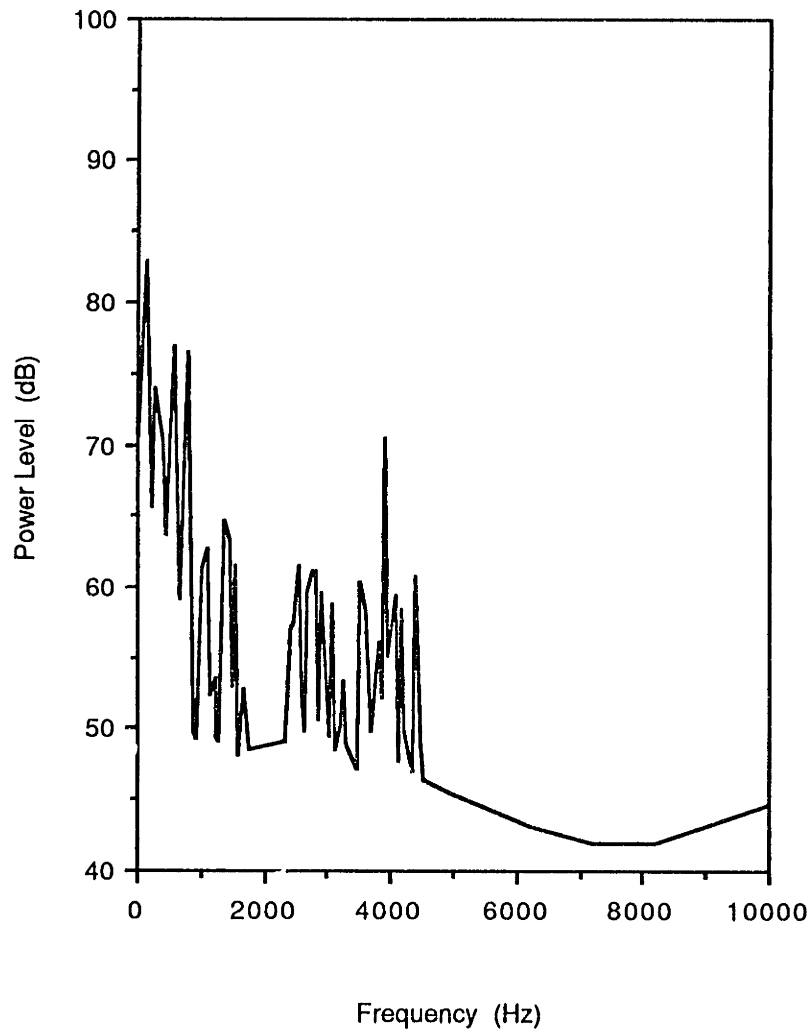

Figure 12. Sound Levels of Reflected Noise

pressure peaks up to 1 kHz, the largest once again being at 60 Hz with a magnitude of 92.2 dB. There is a wide band of peaks between 2 kHz and 4 kHz with an average pressure level of 75 dB and another between 6 kHz and 7.5 kHz with a average pressure level of 58 dB.

Figure 12 shows the pressure levels when the source is pointed directly at the wall so that only reflected noise reaches the microphone. The average pressure levels are about 10dB lower than those of direct noise. This corresponds to a pressure ratio of

$$P_{dir}/P_{reflect} = 10^{(.5)} = 3.16$$

It is interesting to note that frequencies over 5 kHz were not reflected off of the walls, leading us to believe that the walls have an appreciable amount of attenuation at high frequencies. The accelerometer was recording data under the same conditions. There was no apparent difference in the vibrations whether or not the noise source was on, thus no correlation could be made between the the noise levels and the vibration of the instrument stand. Apparently it will take pressure levels much greater than 70 dB to induce vibrations in the instrument stand large enough to be measured by our intruments.

The second series of tests consisted of pounding the floor near the leg of the table with a sledge hammer and attempting to correlate the resulting table vibrations to those of the floor. The H.P. was in time-capture mode in which it records the peaks at each frequency over a ten second period. Although measurable vibrations occured in the floor as a result of the impulse, there appeared to be no sympathetic vibrations in the table. This led us to conclude that the pneumatic isolators were effectivley reducing transmitted vibrations [14]. The data provided by the manufacturer on the isolators is shown in Figure 13. However, the isolators only dampen vertical

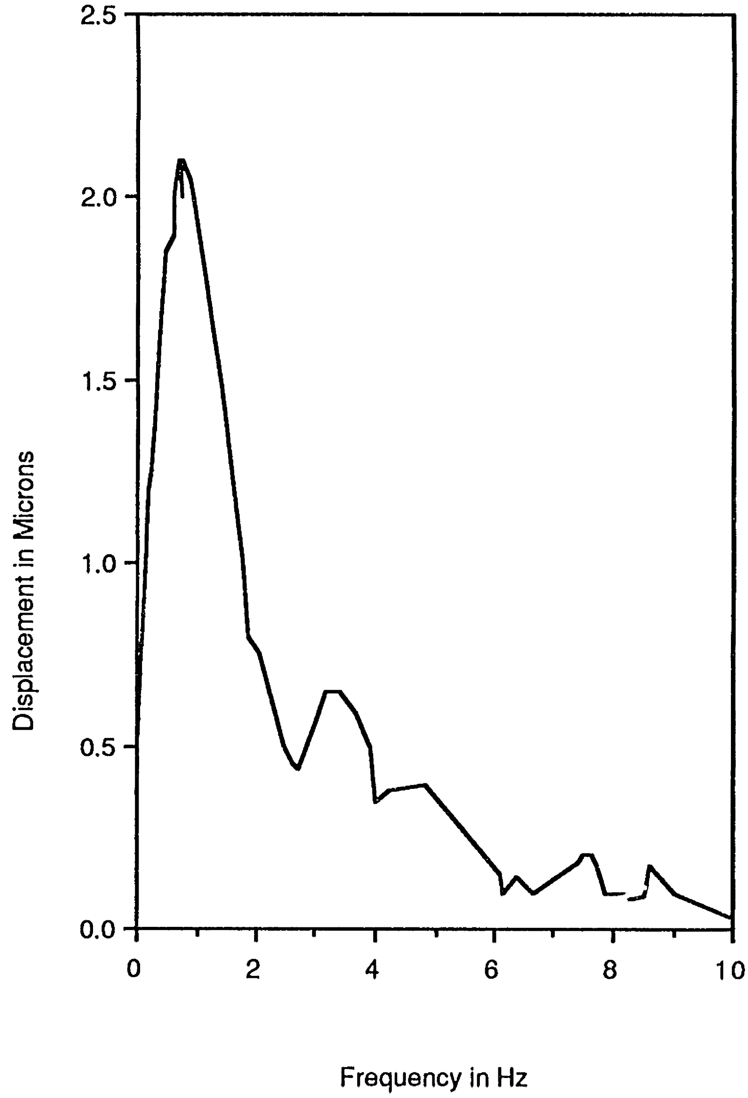

Figure 13. Transmissibilty Curve of a
Melles Griot Isolator

motion of the legs, and the table may be subject to compression waves in the floor which would put a buckling moment on the table. Therefore, we recommend placing some mechanical isolators, such as rubber, under the legs which will provide damping in shear and prevent the transmission of horizontal motions

CHAPTER 7

CONCLUSIONS

The proposed design attempts to attenuate all three possible souces of vibration in a very small area. Internal sounds are minimized by coating the walls with Sonex acoustical foam which will absorb nearly 95% of the sound energy. Sound transmitted through the walls is reduced by the addition of a secondary wall made of gypsum and mounted to the primary walls by four one-inch brackets spaced every 12". The space between the two walls is filled with 2.5 lbs/ft^2 density fiberglas which serves as acoustic damping and thermal isolation. It also prevents sound waves from propagating parallel to the wall in the space. Vibrations traveling in the floor are effectively minimized by the pnuematic isolators , but further isoaltion will be provided by adding mechanical dampers under the legs. These dampers must be effective in shear to provide attenuation of the compression waves propagating in the floor.

The experimentation showed the conditions of the chamber before treatment, but few useful conclusions can be drawn without comparing them to the post-treatment data. The reflected sound pressure in the room now is only about one third of that of direct noise. We were able to calculate the expected total absorbtion of the room to be 685.4 Sabins.

The experiments ought to be repeated for the chamber after treatment to determine the ratio of direct to refelcted sound and to establish the cutoff frequency. Only in this way can the actual effectiveness of the chamber be ascertained, and the equipment be properly designed.

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APPENDIX

PARTS LIST

12	Cartons of 4" Silver Coated Sonex w/ 4 sheets per carton
10	Adhesive Cartridges
1	Applicator
23	4'x8' Sheets of 3# Fiberglas, 4" thick
20	4'x8' Sheets of Gypsum Board, 1/4" thick
3	4'x8' Sheets of Plywood, 3/8" thick
16	Two-by-Fours, 12' long
100	4" C-Channels, 1" thick