CHARACTERIZATION OF THE CENTER FOR ACOUSTICS AND VIBRATION (CAV) REVERBERATION CHAMBER

A Thesis in
Acoustics
by
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Abstract

The most frequently used measure of the acoustical quality of a room is reverberation time. Reverberation times are computed following various standards including ASTM C423, in which the decay of sound is measured. Experimental modal analysis is a procedure typically used on plates and shells in which loss factors, modal masses, resonance frequencies, and mode shapes defining the object are extracted using measured frequency response functions (FRFs). These methods may be applied to reverberant rooms to measure low frequency reverberation times using loss factors.

Results of the characterization of the reverberation chamber in Penn State’s Center for Acoustics and Vibration (CAV) are presented. Reverberation times, levels of background noise, and five ideal measurement locations that give results matching those of the overall chamber are among the topics discussed. The results of a modal analysis experiment performed in the reverberation chamber are also presented. A modal overlap factor (MOF) of 1 is found to be below 125 Hz, indicating that the chamber is reasonably diffuse at and above that level. While the Schroeder frequency is found to be 378 Hz. The extracted loss factors are related to the reverberation times and compared to the results of the ASTM tests.
The results compare favorably, with the reverberation times from the modal study being slightly lower.
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<th>Definition</th>
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<tbody>
<tr>
<td>$\alpha$</td>
<td>Absorption coefficient,</td>
</tr>
<tr>
<td>$\bar{\alpha}$</td>
<td>Average absorption coefficient of an acoustic enclosure,</td>
</tr>
<tr>
<td>$\bar{\alpha}'$</td>
<td>Average absorption coefficient calculated for Millington and Sette reverberation equation,</td>
</tr>
<tr>
<td>$\alpha_i$</td>
<td>Absorption coefficient of the $i$th surface in an acoustic enclosure,</td>
</tr>
<tr>
<td>$\alpha_n$</td>
<td>Absorption coefficient computed from loss factors,</td>
</tr>
<tr>
<td>$\delta_n$</td>
<td>Modal damping factor (1/s),</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Loss factor (1/s)</td>
</tr>
<tr>
<td>$\rho_m$</td>
<td>Density of air (kg/m$^3$),</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Wavelength (m),</td>
</tr>
<tr>
<td>$\Psi$</td>
<td>Eigenfunction describing pressure at a point in a room.</td>
</tr>
<tr>
<td>$\Psi_n$</td>
<td>Normalized mode shape,</td>
</tr>
<tr>
<td>$\tau_r$</td>
<td>Average time between reflections (s),</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Angular frequency (1/s),</td>
</tr>
<tr>
<td>$\omega_0$</td>
<td>Angular resonant frequency (1/s),</td>
</tr>
<tr>
<td>$\omega_i$</td>
<td>Angular frequency below the resonance which has magnitude 3dB lower than the peak (1/s),</td>
</tr>
<tr>
<td>$\omega_n$</td>
<td>Angular resonant frequency (1/s),</td>
</tr>
</tbody>
</table>
\( \omega_u \) Angular frequency above the resonance which has magnitude 3dB lower than the peak (1/s),

A Complex constant,

\( A \) Total area of absorption in an acoustic space (Sabines),

\( A_{\text{panel}} \) Total absorption of a sample (Sabines),

\( A_1 \) Total absorption of the empty reverberation chamber (Sabines),

\( A_2 \) Total absorption of the reverberation chamber with sample in place (Sabines),

a Acceleration (m/s²),

\( a_0 \) Arbitrary constant,

\( a_1 \) Arbitrary constant,

B Atmospheric Pressure (Pa),

\( B_0 \) 101300 (Pa),

C Arbitrary constant,

c speed of sound (m/s),

D Energy lost per cycle (W/s),

d Decay rate (dB/s),

\( d_{\text{air}} \) Decay rate caused by air (dB/s),

\( d_{\text{avg}} \) Average decay rate (dB/s),

\( d_i \) ith decay rate (dB/s),

\( d_{\text{stdev}} \) Standard deviation of the decay rates (dB/s),

\( E_r \) Energy density inside an acoustic enclosure (J/m³),

\( E(t) \) Reduction in sound energy,

F Force (N),

f Frequency (Hz),

xiii
$f_{n_x n_y n_z}$ Eigenfrequency associated with an eigenvalue (Hz),

$f_s$ Schroeder frequency (Hz),

$K_n$ Constant related to whether a mode is axial, tangential, or oblique,

$k$ Stiffness or spring constant (N/m),

$k_{n_x n_y n_z}$ Eigenvalues of a system (1/m),

$k_x$ Wavenumber is the x direction of an acoustic enclosure (1/m),

$L$ Perimeter of a room (m),

$L$ Length of a pipe (m),

$L_m$ Mean free path length between reflections (m),

$\bar{L}_p$ Mean pressure of the reverberation chamber (dB),

$L_w$ Sound power level of a source (dB),

$L_{x,y,z}$ Length of corresponding dimension (m),

$MOF$ Modal overlap factor,

$m$ Absorption cause by air (1/s),

$m$ mass of an object (kg),

$m_n$ modal mass (kg),

$n$ number of reflections,

$n(\omega)$ Modal density of an acoustic volume,

$p$ Pressure in a pipe (Pa),

$p(r)$ Pressure at distance $r$ from the source (Pa),

$p_x$ Change in ambient pressure (Pa),

$Q$ Quality factor

$Q$ Volume velocity ($m^3/s$),

$R$ Sound power reflection coefficient,
$R_m$ Mechanical resistance constant (kg/s),

$r$ Distance from source to receiver (m),

$r_c$ Critical distance (m),

$r_r$ Rate of reflections in an acoustic space (1/s),

$r_r'$ Position of a receiver,

$r_s$ Position of a source,

$S$ Total surface area of an acoustic enclosure (m$^2$),

$S_i$ Surface area of the $i$th surface in an acoustic enclosure (m$^2$),

$SWR$ Standing wave ratio,

$T$ Temperature (°C),

$T$ Period of vibration (s),

$T_{60}$ Reverberation time (s),

$t$ time (s),

$u_x$ Fluid particle velocity in the x-direction (m/s),

$V$ Volume of an acoustic space (m$^3$),

$W$ Power being put into an acoustic enclosure (W),

$W_T$ Total energy in a system (W),

$x$ Complex displacement (m),

$x$ displacement from equilibrium position (m),
Acknowledgments

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Chapter 1

Introduction

The Center for Acoustic and Vibration (CAV) located at the Pennsylvania State University contains a sound transmission loss test facility. This facility is composed of a reverberation chamber and a hemi-anechoic chamber, which share one common wall. A window in the common wall allows for panels to be mounted, allowing for sound transmission tests. The overall goal of the work in this thesis is to update the characterization of the reverberation room using the most current and modern methods. The task of characterizing the chamber was taken on so that other experimental work may take place in the facility. This thesis focuses on the characterization of the reverberation chamber in the facility, including a modal analysis approach, typically used in the analysis of structures.

1.1 Typical Reverberation Rooms and Usage

Reverberation rooms have several uses and purposes in the field of acoustics research. They allow for several different tests such as sound absorption, acoustic scattering, sound power measurements, and tests of loudspeaker and microphone performance. All these tests have the requirement of a diffuse sound field, which is one of the greatest utilities of reverberation rooms. Most of the tests involve the
measure of a reverberation time, which is the time it takes sound to decay to one millionth its initial intensity or by a level of 60 decibels.

1.1.1 General Design and Features

The goal of design in most reverberation chambers is a long reverberation time, low background noise level, and a diffuse sound field. Reverberation time is directly related to the volume of the reverberation room while inversely related to the absorption in the room. Thus, a room with a long reverberation time would have a huge volume with very little absorption. However, a problem with this general design is encountered with the presence of air absorption. With increasing volume comes increasing absorption from the humidity in the air, which becomes problematic at high frequencies. Consequently, limiting the volume of the room is not advantageous in creating a diffuse sound field.

A diffuse field is defined as a sound field in which reflected sound waves of random incidence from all directions combine in an enclosure to give equal energy density at any location in the space. Creating a smaller acoustic enclosure introduces fundamental room modes at higher frequencies, making the room less diffuse in the low frequency regime. Typically, a room is considered truly diffuse above the Schroeder frequency, a frequency at which there are three modes within a given mode’s half power bandwidth [1]. There is no truly good way to eliminate these low order modes. However, acoustician Bolt came up with ideal room ratios that evenly space the modes [2]. By using Bolt’s room ratios the effects of degenerate modes, modes that have the same resonance frequency, become inconsequential as they will rarely occur with these geometries. This is advantageous as room modes will not stack on top of each other, thereby, creating areas of non-diffused energy density.

A commonly used method to break up room modes is to negate typical rectangular room geometry. By slanting parallel walls between 5° and 15° flutter echo,
a strange ringing or hissing caused by parallel walls can be avoided [3]. Another method used by acoustician Botsford involves adding cylindrical irregularities to a room’s rectangular walls [4]. The above methods are great design principles to follow when designing and constructing a reverberation chamber, however, these strange construction practices are not always permissible. Often to achieve a diffuse field within a reverberation room, the use of hanging, wall, or a rotating vane diffusers are used; in many cases all of them. Hanging diffusers are made of many types of semirigid, nonabsorbent materials, and are simply hung at various angles from the ceiling. Wall diffusers are also nonabsorbent and made up of various shapes and sizes in order to break up the flat surface of the wall. Xiang Duanqi’s paper on design of reverberation chambers uses spherical reinforced concrete diffusers to aid in the creation of a diffuse field [3]. Rotating vane diffusers are also often used as they diffuse sound by continually changing the angle at which sound is being reflected. Tichy and Baade cite three different effects caused by rotating vane diffusers that aid in the diffusion of sound [5].

A low noise floor is the last requirement of a reverberation room. This requirement helps ensure that an adequate signal to noise ratio be met during all tests. This is achieved by using massive, thick, heavy walls, floors, and ceilings, which keep unwanted noise out of the testing space. Many times a room within a room construction method is used, in which there are two chambers, one within the other, both of which have extremely thick walls. Also, in many cases the entire reverberation room is built on a separate foundation from the rest of the building; this isolates the room from any vibrations that may infiltrate from sources in the main building. Duanqi illustrates other measures to achieve a low noise floor in his paper on design of reverberation rooms [3].
1.1.2 Facilities at Penn State

The reverberation chamber at Penn State’s CAV facility is of rectangular geometry. The dimensions of the space are $5.69 \times 6.71 \times 3.47$ meters, yielding a total volume of $132.48 \text{ m}^3$, and total surface area of $162.42 \text{ m}^2$. Figures 1.1 and 1.2 create a good visual of the space, with the rectangles on the closest wall in Figure 1.2 being the entry door and transmission loss window. The walls of the room are constructed of concrete cinder blocks that have been painted several times. The floor in the chamber is a slab of concrete that has been sealed to make it as smooth and nonporous as possible. The material of the ceiling is smooth metal deck. All of the materials used are highly reflective and none absorb sound well.

![Figure 1.1. Photo of the Reverberation Chamber in the CAV facilities at Penn State.](image)

The chamber is equipped with an extremely heavy soundproof door, which is used for entry. The door is also highly reflective and has rubber seals to ensure
a complete seal when the door is completely closed. On the wall adjacent to the hemi anechoic chamber is a $1 \times 1$ meter opening, where panels may be mounted. Transmission loss tests can then be conducted on the panels mounted in this window. One simply excites the reverberant room with a sound source then measures the sound that successfully travels through the mounted panel into the neighboring hemi anechoic chamber.

Also, on the floor of the chamber is an isolated $1.5 \times 4.3$ meter area. This area is floated on soft porous rubber and not connected to any other portion of the chamber. The chamber also contains a ceiling mounted turntable that has the ability to have a frame that contains a microphone array mounted to it. When the turntable is operating one can build up a map of pressure in a hemisphere around a source, by using different heights. The turntable may also be used with a rotating vane diffuser.
1.1.3 Tests and Usage of Reverberation Rooms

Reverberation chambers are frequently used to measure the absorption coefficients ($\alpha$) of a material. The absorption coefficient is defined as the ratio of absorbed energy to incident energy. For small absorption coefficients, values of $\alpha \leq .1$, very little sound energy is lost and most of the initial wave front is reflected back. With a value of 1, the absorption coefficient is indicative of complete absorption and no sound energy is reflected back. Absorption coefficients are a way to acoustically classify building materials, and are of great advantage for modeling rooms in the field of architectural acoustics.

The absorption coefficient is measured using a change in reverberation time or decay rate of sound. Using this method the reverberation time of the room is measured with and without the sample in place. Because reverberation time is a function of absorption, and absorption is directly related to the absorption coefficient, a value for the absorption coefficient can be extracted from an equation relating reverberation time to the absorption coefficient of the sample. Standards ISO 354 and ASTM C423 were written to focus on the correct procedures to follow when measuring absorption coefficients in a reverberation chamber [6, 7]. These standards contain specific requirements that must be met in order to obtain valid results. Most of the requirements involve meeting certain levels of diffusivity in the reverberation chamber, and give different conditions to follow when mounting a sample in the chamber.

Another frequent test in reverberation rooms is the measure of scattering coefficients of samples. When sound is incident on a flat surface at an incident angle from the normal, it is reflected at an angle that is identical to the incident angle. The scattering coefficient is a measure of energy that doesn’t reflect back at the correct reflected angle. For a flat surface all the reflected energy will be at the reflected angle. However, when a surface isn’t completely flat some energy will be reflected at different angles, which is what the scattering coefficient measures.
The scattering coefficient is another measure of materials that is of great use to architectural acousticians.

Standard ISO 17497 contains all of the details in how to measure scattering coefficients in reverberation rooms [8]. This standard procedure involves measuring the reflected energy at the angle of reflection of a large circular sample on a turntable. Impulse response data is then collected at different rotated angles of the turntable. The scattering coefficient can then be pulled out of this data using equations. The standard has strict rules on humidity and temperature as small changes in these values can greatly effect the results of the test.

Reverberation chambers are also used to make sound power measurements. The goal of this type of measurement is to know the sound power of some noise source in the room. Some methods used to measure sound power are done in an anechoic space. However, due to the high cost of these spaces, a method was developed for reverberation rooms, as they are much cheaper to construct. Standard ISO 3471 contains requirements regarding procedures to follow when making sound power measurements [9]. The standard contains a grueling qualification procedure that must be met in order to get valid results with other noise sources. Once qualified, results can be found by finding the mean square pressure at known locations and applying a correction factor for the room.

1.2 Decay of Modes

In any structure or acoustic volume are resonances. The resonance frequencies are typically determined by the system’s physical properties and boundary conditions. In a room, or acoustic volume, resonance frequencies are defined by walls (boundary conditions) and the speed of sound in air (physical property), and are called modes of an acoustic enclosure. The resonance frequencies are well spaced in the low frequency regime, while much closer at higher frequencies.
Modes are best excited near their frequency of resonance. Also, once excited they tend to stay excited for some time and don’t stop vibrating or oscillating immediately after the excitation stops. The rate at which each mode decays is similar to the way that sound decays, and the two can be related mathematically.

1.3 Brief History of Reverberation Time

Wallace Clement Sabine is considered the founder of reverberation time. In 1895 while he was a young physics professor at Harvard University, he was given the task to find a remedy for a lecture hall in the newly constructed Fogg Art Museum.[10] The lecture hall in Fogg was modeled after the best sounding lecture hall on campus - Sanders Theatre, however once completed the Fogg lecture hall was the worst sounding lecture hall on campus. It was estimated that when a person stopped talking in Fogg lecture hall their voice could be heard for an additional five and a half seconds. With human speech an additional 15 words could have been spoken in that time, making speech recognition extremely difficult in the lecture hall. Sabine had no prior experience or background with sound, and took a very broad approach to the problem [11].

Sabine set off with a series of experiments in three different lecture halls to find out what makes some halls acoustically better than others. He and a group of assistants spent each night for the next several years experimenting in different lecture halls. Using an organ pipe source, they measured the time for sound to decay to an inaudible level using their ears and stop watches. They often moved numerous set cushions from Sander Theatre to Fogg lecture hall and measured decay rates with varying numbers of cushions. In some cases they moved up to 242 seat cushions.

After several years of experiments Sabine was able to find a relation between the absorption in the room, the size of the room, and the acoustic quality he defined
as reverberation time. Reverberation time is a measure in seconds of the amount of time it take for sound to become inaudible. Sabine continued studying other rooms and began classifying good sounding rooms by their reverberation times, a practice that is still followed to this day.

Soon after Sabine’s findings he helped to design the Symphony Hall in Boston. This hall is still considered today to be one of the best sounding halls for classical music concerts.

1.4 Scope of Thesis

Chapter 2 of the thesis will provide more background information needed to understand the results of the experiments performed in the reverberation chamber. Chapter 3 will focus on the experimental setup used to perform all the different experiments, including two detailed studies to choose some of the parameters used in the experiments. It will then describe both the reverberation time experiments and the modal analysis experiment. Chapter 4 looks at the results of all the different experiments preformed. The reverberation times computed in two different ways are looked at as well as mode shapes extracted from the modal analysis tests. The loss factors extracted during the modal analysis tests are then compared to absorption coefficients and reverberation times found in the reverberation chamber. The final chapter summarizes the results of the modal approach and weighs on the pros and cons of this approach. It then provides some possible topics for further research.
Chapter 2

Theory

2.1 Mathematical History of Reverberation Time

The first formula to find reverberation time was found experimentally by Sabine. Since then many others including Eyring, Millington, Sette, and Fitzroy have added to his original findings, adding twists or looking at the theory a different way. This first section looks at the mathematical history of reverberation time.

Reverberation time is the most frequently used measure of an acoustical space or enclosure. It is the time required for acoustic energy to drop to one millionth its initial value or by 60 dB.

Before beginning the mathematical description of reverberation time, a specific frequency must first be defined. The Schroeder frequency:

$$f_s = 0.329 c^{3/2} \sqrt{\frac{T_{60}}{V}}$$

(2.1)

defines a point in frequency space where one moves from individual well separated modal resonances to overlapping normal modes; where $c$ is the speed of sound [m/s], $V$ is the volume of the room [m$^3$], and $T_{60}$ is the reverberation time [s]
[1, 12]. The Schroeder frequency isn’t the exact location where the transition occurs but is a good estimate. The Schroeder frequency can also be defined as the location at which approximately three modes fall within a given mode’s half power bandwidth. At and above this frequency the sound field is considered diffuse, while below this frequency the sound field is more modal.

Typically when looking at reverberation time and other parameters that define acoustic spaces, only acoustic energy is considered. Individual sound pressure and particle velocity are not analyzed, thus all relevant phase information is lost. The Schroeder frequency also loosely defines this point in frequency space when it is fair to ignore all phase information. Above the Schroeder frequency it is fair to look at only acoustic energy while below the frequency not including phase information may bias measurements of reverberation time or other acoustic room measures.

The original reverberation equation experimentally derived by Sabine is:

$$ T_{60} = 0.164 \cdot \frac{V}{A}, $$

where $A$ is the total area of absorption [m$^2$], and the coefficient 0.164 is a value first found empirically thru Sabine’s experiments. Later a mathematical formula was derived for the coefficient, related to the speed of sound $c$ [m/s]:

$$ \text{coefficient} = \frac{4 \ln(10^6)}{c}. $$

Due to the dependance on the speed of sound the coefficient varies in literature from a value of 0.161 – 0.164. The derivation of this equation is to follow.

The validation of Sabine’s reverberation equation comes from assuming uniform energy density throughout the acoustic space. This is another way of saying that the sound field within the room is diffuse. Of course this is never completely true as complete diffusion implies no net energy flow. In real acoustic enclosures the absorption at walls tends to attract an energy flow away from an acoustic source.
Define the rate at which sound is being absorbed in an acoustic enclosure as 
\[ AE_r c / 4, \] where \( E_r \) is the energy density \([J/m^3]\). The rate at which sound is being absorbed in addition to the rate that sound increases in the enclosure’s volume, \( V dE_r / dt \), must be equivalent to the power being put into the acoustic enclosure \( W \) with units \([W]\). This yields a governing first order differential equation to describe sound growth or decay in an enclosure:

\[
V \frac{dE_r}{dt} + \frac{Ac}{4} E_r = W. \tag{2.4}
\]

If the room is assumed to be in steady state with a constant value of energy density, and the sound power source is suddenly turned off at time \( t = 0 \), the solution to equation 2.4 is given by:

\[
E_r = \frac{4W}{Ac} e^{-\frac{Ac}{4V} t}. \tag{2.5}
\]

By taking equation 2.5 and assuming a one millionth drop in energy:

\[
\frac{1}{10^6} = e^{-\frac{Ac}{4V} T_{60}}, \tag{2.6}
\]

and solving equation 2.6 for \( T_{60} \):

\[
T_{60} = -\frac{4 \ln(10^6)}{c} \frac{V}{A}. \tag{2.7}
\]

When using a common value for speed of sound at room temperature, \( c = 343 \), the coefficient becomes 0.161. It turns out that Sabine’s empirical value of 0.164 matches well with the actual derived value. A value of 0.161 will be used from here as opposed to 0.164 for the remainder of the paper.

Absorption in a room is generally caused by the materials of the ceiling, walls, and floor. By defining the average absorptivity as:

\[
\bar{\alpha} = A / S \tag{2.8}
\]
where $S$ is the surface area of the acoustic space [m$^2$], a modification to Sabine’s reverberation equation 2.2 can be made.

$$T_{60} = \frac{.161V}{S\bar{\alpha}}$$

(2.9)

Often a goal is to predict a reverberation time given absorptivity or absorption coefficients of surfaces in a room, not to find absorptivity given a reverberation time. By simply adding up the surfaces with known absorption coefficients and taking an average, the average absorptivity can be defined:

$$\bar{\alpha} = \frac{1}{S} \sum_{i=1}^{S} S_i\alpha_i,$$

(2.10)

where $\alpha_i$ is the absorption coefficient of the surface area $S_i$

So far, the only acoustic losses come from interaction with surfaces. There are, however, also acoustic losses due to sound’s interaction with air itself. Assuming that the air absorption causes a similar exponential decay, the modified solution to the differential equation contains an extra factor $m$ [1/s] in the decay:

$$E_r = \frac{4W}{Ac}e^{-[(Ac/4V)+m]t}.$$

(2.11)

Again assuming that reverberation time is the time it takes for sound to decay to one millionth its initial level, an expression for reverberation time including air absorption can be derived:

$$T_{60} = \frac{.161V}{S\bar{\alpha} + 4mV}.$$

(2.12)

Here, $m$ is absorption cause by air. For relative humidity $h$ in a rage of 20% - 70% the expression for the absorption by air $m$ is given by:

$$m = 5.5 \times 10^{-4} \left(\frac{50}{h}\right) \left(\frac{f}{1000}\right)^{1.7},$$

(2.13)
where absorption by air is frequency dependent. Absorption caused by air can become a significant form of energy loss in rooms with large volumes. On the other hand the absorption caused by air can usually be ignored in very small rooms.

Kuttruff points out that the Sabine equation fails for cases of high absorption [13]. In the case when all surfaces in a room are completely absorptive ($\bar{\alpha} = 1$) Sabine’s equation still yields a finite reverberation time. This is obviously nonphysical as a room that contains surfaces that don’t reflect energy can’t reverberate.

Eyring also noted the basic error in Sabine’s reverberation equation. He claimed that his equation is essentially a “live” room formula [14]. Eyring set off on his own path to characterize reverberation times based on mean free path between reflections. It can be shown that the mean free path length between reflections in a rectangular room is $L_m = 4V/S$ [m] [15]. An average time between reflections can then be found using the speed of sound:

$$\tau_r = \frac{4V}{cS} [s]. \quad (2.14)$$

The rate of reflections $r_r = cS/4V$ [1/s] is simply the inverse of the time between reflections $\tau_r$.

During each reflection the energy of the sound wave is reduced by a factor of $(1 - \bar{\alpha})$. After $n$ reflections the energy is reduced to a level of $(1 - \bar{\alpha})^n$. Relating the number of reflections $n$ to the rate of reflections $r_r$ and time $t$, the reduction in sound energy is:

$$E(t) = (1 - \bar{\alpha})^{r_rt}. \quad (2.15)$$

By the definition of reverberation time, setting equation 2.15 equal to $10^{-6}$:

$$T_{60} = \frac{\ln 10^{-6}}{r_r \ln (1 - \bar{\alpha})} = \frac{4 \ln 10^{-6} V}{cS \ln (1 - \bar{\alpha})} = \frac{.161V}{-S \ln (1 - \bar{\alpha})}. \quad (2.16)$$
where air absorption is ignored. By adding a factor of $4mV$ to the denominator, air absorption can be accounted for. For cases when $\bar{\alpha}$ is small $\ln (1 - \bar{\alpha}) \approx -\bar{\alpha}$ and the Eyring reverberation equation becomes the Sabine reverberation equation. Eyring’s reverberation equation solves the problem introduced earlier of a completely absorbent room with $\bar{\alpha} = 1$. Eyring’s reverberation equation does, however, assume that the absorption in the acoustic enclosure is uniformly distributed.

Acousticians Millington and Sette also derived a reverberation formula based on similar concepts used by Eyring [16, 17]. They, however, calculate the absorption coefficient in a different way:

$$\bar{\alpha}' = -\frac{1}{S} \sum_{i=1}^{S} S_i \ln (1 - \alpha_i).$$

The average isn’t of the absorption coefficient but of the natural logarithm of $(1 - \alpha_i)$. When plugging absorption in Equation 2.17 into a general reverberation equation, the final reverberation time is given by:

$$T_{60} = \frac{.161V}{-\sum_{i=1}^{S} S_i \ln (1 - \alpha_i)},$$

where air absorption can be accounted for with additional factor of $4mV$ in the denominator. Sette claimed that the reverberation formula derived wasn’t a general formula, but a useful formula in the case when: “any ray of sound, after repeated reflection will have struck any one surface in proportion to the ratio of the area of that surface to the total room surface [17].” This is the basic assumption used throughout the entire derivation. This is different than the assumption used in Eyring’s derivation in which each surface receives a proportional share of the total energy in the room at each reflection.

The reverberation equation was then left alone until 1959 when Fitzroy attacked the validity of the derivations made by others before [18]. He cited cases of erratic
reverberation times when absorption wasn’t uniformly distributed throughout the acoustic enclosure. In a case presented in a book by Dr. Knudsen and Dr. Harris a room was given a highly absorptive ceiling, yet relatively reflective side walls and floor [19]. The reverberation time found using formulas was a little over half a second while the measured time for the space was around two seconds. This motivated Fitzroy, as engineers can’t tolerate such large variations when working in the field. Fitzroy speculated that the sound field may settle into patterns of oscillation along an acoustic enclosure’s three major axes. This lead Fitzroy to a reverberation equation that borrowed from the ideas of Eyring, but scaled each decay time in the direction with the surface area in that direction. The final equation Fitzroy derived is:

\[
T_{60} = \left(\frac{x}{S}\right) \left[\frac{.161V}{-S \ln (1 - \bar{\alpha}_x)}\right] + \left(\frac{y}{S}\right) \left[\frac{.161V}{-S \ln (1 - \bar{\alpha}_y)}\right] + \left(\frac{z}{S}\right) \left[\frac{.161V}{-S \ln (1 - \bar{\alpha}_z)}\right],
\]

where \(x, y, z\) is the total surface area in a given direction, \(\bar{\alpha}_i\) is the absorption coefficient in the corresponding direction, and air absorption is ignored.

When using the Fitzroy reverberation equation the problems presented with nonuniform distribution of absorption were fixed. Before even publishing the paper Fitzroy experimented with the equation making many case studies for over eight years [18]. Even with the remarkable results using equation 2.19, it is rarely used in architectural acoustics, where favor is given to Sabine and Eyring. This seems incorrect as often in auditoriums and concert halls most absorption comes from seats and people in the audience; a case in which the absorption most definitely is not uniformly distributed.
2.2 Response of Rooms

Low frequency response of rooms is quantified by the boundaries and shape of a room. In a typical rectangular hard walled room, such as a reverberation chamber, the normal component of velocity must vanish at each boundary. From the Euler equation, the slope of the pressure normal to the boundary must also vanish:

\[
\frac{\partial u_x(0)}{\partial t} = -\frac{1}{\rho_m} \left( \frac{\partial p_x}{\partial x} \right)_{x=0},
\]

where \(u_x\) is the fluid velocity in the x direction, \(\rho_m\) is the density of the fluid, and \(p_x\) is the change in ambient pressure in the x direction. Using the Helmholtz equation:

\[
\frac{d^2 p_x}{dx^2} + k_x^2 p_x = 0,
\]

and the defined boundary conditions for pressure yields a solution of:

\[
p_x(x) = A_0 \cos(k_x x),
\]

with

\[
k_x = \frac{n_x \pi}{L_x},
\]

where \(n_x\) is an index, and \(L_x\) is the length of the x- dimension. Similar results are found for the y and z coordinate system.

Because rectangular coordinates are separable, the \(k\)'s in all coordinates can be combined to yield the eigenvalues of the system:

\[
k_{n_x,n_y,n_z} = \pi \left[ \left( \frac{n_x}{L_x} \right)^2 + \left( \frac{n_y}{L_y} \right)^2 + \left( \frac{n_z}{L_z} \right)^2 \right]^{1/2}.
\]

The eigenfunctions associated with the eigenvalues are found by multiplication by
cosine functions:

$$\Psi(x, y, z) = C \cos \left( \frac{n_x \pi x}{L_x} \right) \cos \left( \frac{n_y \pi y}{L_y} \right) \cos \left( \frac{n_z \pi z}{L_z} \right),$$  \hspace{1cm} (2.25)

where C is an arbitrary constant. The formula describes normal modes of a room. Anytime the value of a cosine becomes 0, the entire response vanishes. This creates nodal planes or planes of no pressure in the x, y, and z direction. The indices \(n_x\), \(n_y\), and \(n_z\) indicate the number of nodal planes in the x, y, and z direction.

The eigenfrequency associated with each eigenvalue is found by:

$$f_{n_x n_y n_z} = \frac{c}{2\pi} k_{n_x n_y n_z} = \frac{c}{2} \left[ \left( \frac{n_x}{L_x} \right)^2 + \left( \frac{n_y}{L_y} \right)^2 + \left( \frac{n_z}{L_z} \right)^2 \right]^{1/2}.$$  \hspace{1cm} (2.26)

The total response of the room is defined by the eigenfunction of the room, the source frequency content, and the location of the source and receiver. In general the low frequency regime is dominated by resonance peaks, while at higher frequencies resonances are so close that they combine to give a flat response. The low frequency modes of a room can be quantified using modal analysis, where well spaced resonances are modeled as simple harmonic oscillators.

### 2.3 Overview of Harmonic Motion

Modal response functions, both dynamic and acoustic, can be modeled as summations of harmonic oscillator responses. Experimental modal analysis analyzes transfer functions and models them with harmonic oscillators. To understand what the modal analysis software is doing, a review of the mathematics behind a harmonic oscillator is needed.
2.3.1 Math of a Simple Harmonic Oscillator

A harmonic oscillator is any system that when displaced from equilibrium, experiences a restoring force proportional to the displacement. This restoring force is known as Hooke’s Law:

\[ F = -kx, \]

where \( F \) is force [N], \( k \) is stiffness or spring constant [N/m], and \( x \) is displacement [m]. The negative sign in equation 2.27 indicates that the force is opposed to the motion, and is thus a restoring force. By letting the only force in the system be defined by Hooke’s law the system becomes known as a simple harmonic oscillator and it undergoes sinusoidal motion about its equilibrium position. An easy way to picture simple harmonic motion is a mass spring system, shown in Figure 2.1, in which a mass is attached to a spring which in turn is fixed to a wall. When the mass is displaced in one direction a restoring force pulls it back to equilibrium, but over shoots the equilibrium and begins a displacement in the opposite direction; the process then repeats again and again.

![Figure 2.1. Schematic of a simple harmonic oscillator](image)

Relating Hooke’s law to Newton’s second law:

\[ F = ma, \]
where $m$ is the mass of an object [kg] and $a$ is its acceleration [m/s$^2$], the following relationship is found:

$$-kx = ma = m\frac{d^2x}{dt^2}. \tag{2.29}$$

Rearranging some terms and setting the expression equal to zero, a more useful second order homogeneous differential equation is found:

$$m\frac{d^2x}{dt^2} + kx = 0. \tag{2.30}$$

By defining the angular resonance frequency as $\omega_0 = \sqrt{k/m}$ [1/s], Equation 2.30 can be modified:

$$\frac{d^2x}{dt^2} + \omega_0^2x = 0. \tag{2.31}$$

Equation 2.31 is the most general equation of motion that describes a simple harmonic oscillator which has a well known solution. The solution can be expressed in terms of exponentials, sines and cosines, or series solutions. The complex exponential and sines and cosines solution are shown in Equation 2.32, where $C$, $a_0$, and $a_1$ are all constants determined by the initials conditions.

$$x = Ce^{-i\omega t}$$

$$x = a_0 \cos(\omega t) + ia_1 \sin(\omega t) \tag{2.32}$$

### 2.3.2 Math of a Harmonic Oscillator with Damping and Applied Force

So far a harmonic oscillator with only one force has been looked at; a restoring force due to a spring or stiffness defined by Hooke’s Law. In real world systems the only force on the oscillator isn’t defined by Hooke’s Law; in real life there is friction or drag on the oscillator that eventually brings the oscillator’s motion to a stop.
By assuming that damping on the oscillator is proportional to the velocity at which the oscillator is traveling a new force can be added to the system. The new added force will model the oscillator as it decays back to its equilibrium position. A new schematic of an oscillator with damping is shown in Figure 2.2; now a dashpot will eventually bring the oscillation to a stop. Also displayed in the schematic are relations of force caused by the dashpot to the velocity $u$.

![Figure 2.2. Schematic of a damped harmonic oscillator](image)

The dissipative force is defined mathematically by:

$$F_r = -R_m \frac{dx}{dt}, \quad (2.33)$$

where $R_m$ is the mechanical resistance constant [kg/s]. Simply relating all the forces as before the differential equation governing the motion of the oscillator is:

$$m \frac{d^2x}{dt^2} + R_m \frac{dx}{dt} + kx = 0. \quad (2.34)$$

The solution to this equation can be found using complex exponentials. The solution won’t be discussed now but can be found in Morse and Kinsler et al. [20, 21]

Thus far only two forces are acting on the system, a restoring force and a damping force. When the oscillator is driven by an external force $F$ [N], the differential equation governing the motion is no longer homogeneous:
Assuming a time harmonic complex driving force:

\[ m \frac{d^2x}{dt^2} + R_m \frac{dx}{dt} + kx = F(t). \quad (2.35) \]

Due to the fact that the driving force is complex the solution to equation 2.36 gives a complex displacement, where the real part is the actual physical displacement. Since the displacement is time harmonic just as the drive force is: \( x = Ae^{-i\omega t} \), where \( A \) is complex. Plugging in the assumed complex solution to equation 2.36 yields the following:

\[ (-A\omega^2m - iA\omega R_m + Ak)e^{-i\omega t} = Fe^{-i\omega t}. \quad (2.37) \]

Using the relation: \( x = Ae^{-i\omega t} \), and solving for the complex displacement:

\[ x = \frac{1}{-i\omega R_m - i(\omega m - k/w)} Fe^{-i\omega t}. \quad (2.38) \]

### 2.3.3 Loss Factor and \( Q \) of oscillators

In structural acoustics losses are typically modeled to be proportional to displacement not to velocity as was derived above. This isn’t an issue as it just changes the math a little bit. Modeling the system with losses proportional to displacement yields a new differential equation:

\[ m \frac{d^2x}{dt^2} + (1 + i\eta)kx = Fe^{-i\omega t}. \quad (2.39) \]
where \( \eta \) is the loss factor [1/s]. A loss factor is the fraction of energy lost per radian, of the oscillating system, defined mathematically as:

\[
\eta = \frac{D}{2\pi W_T},
\]

where \( D \) is the energy lost per cycle [W/s], and \( W_T \) is the total energy [W]. By a similar process used in the viscous damping the complex displacement can be shown to be:

\[
x = \frac{F e^{-i\omega t}}{[-\omega^2 m + k(1 + i\eta)]}
\]

The loss factor of a system can be related to the quality factor, \( Q \), of an oscillating system. The quality factor measures the sharpness of a resonant peak. It is defined mathematically by:

\[
Q = \frac{\omega_0}{\omega_u - \omega_l},
\]

where \( \omega_u \) and \( \omega_l \) are the angular frequencies above and below the resonant peak \( \omega_0 \) at which the power of the oscillator has dropped to half its peak value. Often the \( Q \) of an oscillator is measured using Equations 2.42, by extraction of ‘3 dB down’ points from a plot of the resonant peak.

The relationship between loss factor and \( Q \) is:

\[
Q = \frac{1}{\eta}.
\]

Thus a system with a high \( Q \) has a sharper resonant peak and smaller loss factor, meaning energy is dissipated at a slower pace, while a low \( Q \) system has a broad resonant peak and higher loss factor.
2.4 Experimental Modal Analysis Theory

Experimental Modal Analysis is the process of finding properties of structures, such as resonance frequencies, mode shapes, modal masses, and loss factors. This is typically done for multi-dimensional objects such as plates and cylindrical shells.

Experimental frequency response functions (FRFs) are created between a applied inputs and reference responses at multiple locations. For structures the FRF between a displacement $x$ at position $r_r$ and the force $F$ at position $r_s$ is given by:

$$\frac{x_n(r_r, \omega)}{F(r_s, \omega)} = \frac{1}{m_n} \frac{\Psi_n(r_r) \Psi_n(r_s)}{-\omega^2 + \omega_n^2 + i \eta_n \omega_n \omega},$$

where $m_n$ is the modal mass, $\omega_n$ is the angular resonance frequency, and $\Psi_n$ is the normalized mode shape [22].

In a structural experiment the force and displacement between several locations are known as they are measured by a force hammer or shaker and accelerometers. Using standard modal analysis techniques the desired resonance frequencies, modal masses, mode shapes, and loss factors are extracted for each mode. A Complex Mode Indicator Function (CMIF) technique can be used to separate modes, find resonance frequencies and mode shapes [23]. A Rational Fraction Polynomial scheme can then used to refine damping and resonance frequency estimates [22, 24].

The same procedure can be applied to an acoustics enclosure. In this case the applied input comes from a speaker sound source (volume velocity) and the reference response locations are measured with microphones (Pascals). The FRF in this case would be a units of Pa·s/m³. The same modal extraction techniques can be employed to find the resonance frequencies, mode shapes, loss factors, and modal masses for the acoustic volume.
Experimental Procedures and Pre-studies

3.1 Experimental Setup

The experimental setup is designed to make two types of measurements, decay times and extraction of modal parameters. The decay experiments are found by using a source-on source-off method at several locations. The modal experiment finds FRFs between several source and receiver locations, then uses standard modal extraction techniques to find the desired modal parameters.

3.1.1 Experimental Hardware

The sound source used in all the experiments is a Brüel & Kjær Type 4292 OmniPower Sound Source. The source’s frequency response is nearly flat, and the source attempts to radiate sound as a monopole point source, with its twelve 5” speakers arranged equally around the enclosure. A picture of the source is shown in Figure 3.1. The DC impedance of the source is 5.30 Ω. Calibration curves for the source are given in Appendix C.2.

The amplifier used to power the Brüel & Kjær Type 4292 OmniPower Sound
Source is a Crown XTi2000. The amplifier is rated at 475 Watts per channel at 8 ohms. The Crown source amplifies all frequencies equally and has a nominal 32.9 dB gain [25]. An experiment was done on the amplifier to find performance parameters that the manufacture’s specifications didn’t provide. The experiment found the gain at each knob click through the range of the 20 clicks. This was done using an Agilent 33220A Arbitrary Waveform Generator to feed the amplifier and measuring the voltage across a 5Ω resistor. The results show that the amplifier’s gain is very linear through the range of steps and the total gain of the amplifier is 38.8 dB. More information on the calibration can be found in Appendix C. A picture of the Crown XTi2000 amplifier can be found in Figure 3.2.

![Picture of Crown XTi2000 amplifier](image)

**Figure 3.2.** Crown amplifier used to power the Brüel & Kjær Source.

The microphones used are two phase matched type TEF04 microphones manufactured by Goldline. The two serial numbers of the microphones are: sn# 601300 and sn# 601302. The microphones are omni directional, with a 1/2” diameter condenser capsule. Both microphones have a sensitivity of 7.5 mV/Pa, and require 12 - 48 V of phantom power to operate. The frequency response of both microphones is nearly flat; the frequency response of microphone sn# 601302 is given in Appendix C.3. The output connector of the microphones is an XLR 3 pin male adapter. A picture of the two microphones used in all tests is found in

![Picture of two microphones](image)

**Figure 3.1.** Brüel & Kjær Type 4292 OmniPower Sound Source.
The microphone preamps used in all tests are housed in an EASERA GATEWAY, made by Renkus-Heinz. The EASERA GATEWAY is a modified PreSonus FireBox which has modified frequency response and gain settings with AD/DA converters specifically made for use with software programs EASERA and EASERA PRO [26]. A picture of the EASERA GATEWAY is shown in Figure 3.4. The preamps on the GATEWAY accept XLR 3 pin male adapters, and have phantom power of 48 V, which is required by the Goldline microphones used in the tests. In the experiments the GATEWAY is used in two different ways; the first is a standalone mode, in which only the microphone preamps are used. The second is full operation in which the microphone preamps and the AD/DA conversion are used.

Additional experiments were also performed on the GATEWAY to determine the gain of the microphone preamps when used in standalone mode. The experiments were done in a similar manner as the tests on the Crown XTi2000 amplifier. An Agilent 33220A Arbitrary Waveform Generator fed the preamp and then the output voltage was recorded. This process was repeated for all 40 steps of the preamp knob. The results of the test found that the preamp had a total of 59.8 dB gain, although the gain wasn’t linear through the gain steps on the preamp knob. The preamp appears to have three different trends, or three different gain stages to raise the voltage of the microphone signal. This isn’t a problem as the gain at each preamp step is known, and the gain between steps can easily be compared if calibration is to be altered. The calibrated curve for the preamps is given in Appendix C.

In order to protect the B&K source from being driven beyond its limits a Fluke 45 Dual Display Multimeter is used to monitor the power sent to the source. The Fluke 45 Dual Display Multimeter can monitor both voltage across the source and the current sent to the source at the same time. Doing so ensures that the source
isn't being driven above its maximum power handling. A picture of the Fluke multimeter is shown in Figure 3.5.

The last piece of equipment used is a National Instruments USB-4431. The National Instruments interface was used in order to employ LabVIEW software. The interface has four inputs and one output, all of which use BNC type connectors. The interface acts as a AD/DA converter connecting to a computer via a USB connection. A picture of the USB-4431 is shown in Figure 3.6.

### 3.1.2 Experimental Signal Flow

All of the experiments preformed can be classified into two different types of signal flows. These classifications are based on whether the AD/DA conversion is done on the EASERA GATEWAY or the NI USB-4431.
If using the EASERA GATEWAY for AD/DA conversion the signal flow can be seen in Figure 3.7 and is called Sig Flow A. The source signal is generated on a computer which is sent to the GATEWAY. The GATEWAY converts the digital signal into a voltage which is sent to the Crown amplifier. The Crown amplifier then amplifies the voltage signal to an adequate level for the B&K sound source. The Goldline Microphones pick up the sound signal produced by the source, which is sent back to the preamps in the GATEWAY. The GATEWAY then converts the voltage from the preamp into bits, which are then recorded on the computer.

If using the NI USB-4431 to do the AD/DA conversion, the signal flow looks as pictured in Figure 3.8 and is called Sig Flow B. The source’s signal is created on the computer which is sent through the output of the NI USB-4431 to the Crown amplifier. The amplifier raises the voltage to drive the B&K source. The two Goldline microphones then pick up the sound signal generated by the source, which is then carried to the microphone preamps. Next, preamps raise the voltage signal of the microphones to a higher level which is then carried to the inputs of the NI USB-4431. Finally the signal is returned to bits and recorded on the computer.
Figure 3.8. Flow graph of Sig Flow B.

3.2 Experiments

3.2.1 Modal Analysis Experiment

The modal analysis experiment is designed to find resonance frequencies and loss factors for low frequency modes in the reverberation chamber and calculate the corresponding reverberation times for each mode. Loss factors can be related to reverberation time using:

\[ \eta(f) = \frac{2.20}{f T_{60}} \]  

(3.1)

a detailed derivation of this relation is found in Appendix A.

Typical modal analysis measures an FRF of displacement over force. Due to the nature of the experiment it is chosen to create an FRF of pressure over volume velocity (Q). The pressure is detected using the Goldline microphones, while the volume velocity is produced from the B&K source. The value of volume velocity is estimated using the calibration sheet of the source and assuming a monopole
point source using:

\[ Q = \frac{2\lambda r}{\rho m c} p(r), \]  

(3.2)

where \( \lambda \) is the wavelength, and \( p(r) \) is the pressure at a distance \( r \) from the source. The calibration sheet gives the pressure in one third octave bands at a distance of 1 meter from the source when driven by 100 W of power. Thus a calibration factor for volume velocity can be obtained.

Normally modal analysis is done using a 2-Dimensional plane created across an object. The reverberation chamber isn’t 2-D, so three different grids are used to account for all directions. A grid in the x-y plane, the x-z plane, and the y-z plane, are used to account for the three dimensions of the chamber. Recall the chamber dimensions are 5.69 \times 6.71 \times 3.47 \text{ meters} (18.7 \times 22 \times 11.4 \text{ ft}). The grids are set up with a spacing of 1 ft to the next adjacent point in the perpendicular directions. A picture of the horizontal grid is shown in Figure 3.9. The x-y grid \((21 \times 18)\)

![Figure 3.9. Horizontal grid used in modal analysis experiments](image)

has a total of 378 points, positioned five feet from the floor. Due to restraints of
the microphone stands, the two planes in the vertical direction only make it 8 ft into the air. This skips the two highest rows that would make the grid entirely fill the plane to the ceiling. The x-z plane \((18 \times 8)\) has a total of 144 points while the y-z plane \((21 \times 8)\) has 168 points. The x-z plane is positioned 8 ft down the y-dimension wall, and the y-z plane is positioned 7 ft down the x-dimension wall.

For the experiment an FRF is created from a microphone at each location on the grid with respect to two different source positions. The first source position is on the floor in a tetrahedral corner of the chamber, the bottom left corner of the chamber in Figure 3.9. The second position is 5 ft down the x-direction wall and 1 ft off the wall, also positioned on the floor.

The experiment uses Sig Flow B. The signal used to drive the source is generated using LabVIEW program Function Generator V3p3, and chosen to be white noise. The signal is acquired using Modal Impact V6p3. A BNC cable running directly off the output of the NI interface returns to an input of the NI interface and is captured as volume velocity.

### 3.2.2 Reverberation Time Measurements

Two different experiments are designed in order to validate the reverberation time of the empty chamber. A third experiment is also performed using commercial software EASERA, and is discussed in Appendix D.

#### 3.2.2.1 Reverberation Test 1

The first experiment looks at the reverberation time following the standard procedures found in the Appendix A.3 of ASTM C423 [7]. This procedure requires the measurement of 20 decays at five or more locations, with at least one loudspeaker location. Other requirements of the standard are discussed in Appendix B, in which the actual measurement of the decay is analyzed.

In the ASTM C423 based experiment five microphone locations are used with 20
decays at each location. Only one source position is used; the B&K source placed in a tetrahedral corner of the chamber, which should excite all room modes. The source signal is chosen to be broadband white noise, being driven to the source at approximately 20 Watts.

The signal flow of the experiment follows Sig Flow A. The source signal is generated on a MacBook Pro played through Logic Pro 9, which links up with the GATEWAY. The source signal is played for 40 seconds before it is terminated, and the decay recorded. Logic Pro 9 is also used to record time series data that microphones detect inside the reverberation chamber. After all the decays are recorded they are post processed in MATLAB following the procedure in Appendix B.

### 3.2.2.2 Reverberation Test 2

The second experiment is similar to the first test, but uses a source signal of band filtered white noise. White noise is filtered into one third octave bands between 100 Hz and 5000 Hz following the ASTM C423 standard. Using band limited noise allows for more power to be injected into the specific one-third octave band of interest, without harming the source. This experiment ensures a higher signal to noise ratio, and should yield similar results as the broadband white noise tests.

This test uses fewer decay measurements at each location, but more locations are used. A total of twelve locations are used with four decays at each location. The source is again placed in a tetrahedral corner of the chamber. Sig Flow B is used for this experiment. The one third octave band limited white noise is produced by LabView program Function Generator V3p3. The signal is played for 35 seconds before it is turned off and the decay recorded. The signal is then captured using the LabVIEW program TimeDataAcquisition V4p2. The acquired data is then post processed using the same MATLAB program as used in the previous experiment.
3.3 Modal Analysis Pre-study

After the collection of the first set of data, a horizontal plane with frequency spacing of .3 Hz and two source locations, validation of the results acquired using modal analysis was attempted using the 3 dB down method. Figure 3.10 shows the pressure at a location close to the corner of horizontal plane normalized to the volume velocity of the source. Figure 3.10 shows that the lowest resonant peak appears to be off bin, due to the shape of the peak. Data points extracted around the lowest mode of approximately 26 Hz clearly shows that the 3 dB down method doesn’t work accurately; as ideally more that four points on each side of the resonant peak will be used to compute the 3 dB down locations. If needed, extrapolation between points is used to find the actual value of the 3 dB down location.

Due to the issue of not being able to resolve the peak using the 3 dB down
method at lower modes, a study was developed to check how well the modal analysis software is able to extract loss factors using different frequency resolutions.

### 3.3.1 Theory and Methods of Modal Analysis Pre-study

A mock set of data was created in order to check the modal analysis software. The modal analysis software requires knowledge of a FRF at specific points in a geometry. Mock FRFs of $p/F$ are created using:

$$p(r_r) = \rho_m c^2 \omega Q \sum_n \frac{\Psi(r_r)\Psi(r_s)}{[2\delta_n \omega_n + i(\omega^2 - \omega_n^2)]K_n},$$

(3.3)

found in Kuttruff [13], and assuming a forcing function of white noise. Equation 3.3 describes the pressure between a point and receiver in a room with damping where $\Psi(r_r)$ is the eigenfunction of the receiver position, $\Psi(r_s)$ is the eigenfunction of the source position, $\delta_n$ is the modal damping factor $[1/s]$, $\omega_n$ are the resonance angular frequencies of the room $[1/s]$, and $K_n$ is a constant related to the volume of the room and whether the mode of interest is axial, tangential, or oblique $[m^3]$. The modal damping factor can be related to loss factor by:

$$\eta_n = \frac{\delta_n}{\pi f_n}.$$  

(3.4)

A more detailed derivation of the previous can be found in Kuttruff and Morse [13, 20].

A MATLAB routine was set up to find the FRF at each grid location in the room assuming two different source positions, the same positions as used in the modal experiment. Two different frequency spacings of .05 Hz and .3 Hz are chosen to assess the performance of the modal analysis software at low frequencies. The grid consisted of 378 points in a horizontal plane 5 ft above the floor with a spacing of 1 ft to the closest point in the x or y direction, the same horizontal grid used in
the actual experiment.

The parameters used in the model are common values for air at room temperature; \( \rho_m = 1.21 \text{ [kg/m}^3\text{]} \) and \( c = 343 \text{ [m/s]} \). The volume velocity of the source was assumed to be \( Q = 1 \text{ [m}^3\text{/s]} \). A modal damping factor of \( \delta_n = 2 \text{ [1/s]} \) was used for every resonance angular frequency \( \omega_n \), which is determined by the chamber’s geometry using:

\[
\omega_n = c\pi \left[ \left( \frac{n_x}{L_x} \right)^2 + \left( \frac{n_y}{L_y} \right)^2 + \left( \frac{n_z}{L_z} \right)^2 \right]^{1/2},
\]

where \( n_x, n_y, n_z \) are indexes starting at one, and \( L_x, L_y, L_z \) are corresponding chamber dimensions.

### 3.3.2 Results of Modeled Modal Analysis Study

After running the two sets of data with different frequency spacings, the results of extracted loss factors are very similar. As shown in Figures 3.11 and 3.12 major differences between the two cases are not apparent. Although the 3dB down method doesn’t work to find the loss factor of the lowest frequency mode with a .3 Hz frequency spacing, the modal analysis software, which used a more complex routine, is able to extract the value of the loss factor of the lowest mode to within 1%. The results validate the modal analysis software for use at a frequency spacing of .3 Hz, even though the 3 dB down method can’t be used to check the computed loss factor at low frequencies.

Extraction of modes higher than 130 Hz was not attempted as the density of modes above this frequency makes accurate extraction difficult. The use of more source positions would aid in the extraction or resonance frequencies, but the experiment only has two source positions so the developed model only contains two locations. Even with .3 Hz frequency spacing being adequate for tests the remainder of the data is collected using a smaller frequency spacing.
3.4 Validation of Recorded Decay Analysis

Standards ISO 354 and ASTM C423 deal with the measurement of absorption in reverberation chambers. In order to make sure that the MATLAB code developed works accurately in finding absorption, i.e. decay rate, it is checked against another method.

3.4.1 Method to Validate Decay Analysis

In order to check the measured decays, an experiment is setup to determine the reverberation time, related to decay rate by a constant, using two different methods. The first method is developed using computer code in MATLAB. A description of the MATLAB code and why some of the parameters are chosen is found in Appendix B. The second method uses the built in software on a Larson Davis System 824. The Larson Davis System 824 is a small sound level meter that includes real-
Figure 3.12. Simulated loss factor with a frequency spacing of .05 Hz

time analysis capabilities, and has the ability to measure reverberation times [27]. The system records the third octave band spectra at 400 samples/sec, and then finds the slope of the decay and extrapolates the full reverberation times based on the first portion of the decay. The slope is determined using a least squares method for a dB down setting chosen by the user, i.e. 15dB 20dB, 25dB, 30dB. In order to best match the Matlab routine a dB down setting of 25dB is selected.

As a result of spatial variability in the reverberation chamber, the location of the recordings has to be the same. This is achieved by placing an extension probe for the Larson Davis System 824, directly next to one of the TEF Goldline microphones. This is shown in Figure 3.13

The source for this experiment is chosen to be popped balloons. This is so that the Larson Davis System 824 can be monitored to make sure it triggers from the correct source and not the opening or closing of the chamber door. It has been shown that balloon pops can be quite directional [28]; for this reason the balloons
are popped in a tetrahedral corner of the chamber using a pair of scissors.

For the experiment, only three locations are used with five balloon pops at each. The Larson Davis System 824 is setup to calculate the reverberation time in third octave bands, and triggered to begin calculations after an impulse louder that 95 dB. Reverberation times found using the MATLAB routine and the Larson Davis System 824 are then compared to validate the MATLAB code.

3.4.2 Results of Validation Decay Analysis

The reverberation times calculated using both methods at locations 1-3 are plotted in Figures 3.14 - 3.16. The reverberation times are plotted for third octave
frequency bands in the range of 50 Hz -12.5 kHz. From the results it is clear that the MATLAB routine works well in most third octave bands, as the two methods are within each other’s standard deviation. There are a few outliers which can be accounted for due to the small sample size.

It should be noted, when comparing locations that reverberation times are different, especially in the lower third octave bands. This is caused by spatial variability in the reverberation chamber, which is why the recorded locations had to be the same. If comparing the two methods at locations that aren’t the same, results would be different and comparisons would be meaningless.
Figure 3.15. Reverberation Times calculated at second location

Figure 3.16. Reverberation Times calculated at third location
Chapter 4

Results

4.1 Chamber diffusivity

Before presenting the main results involving loss factors and reverberation times, the diffusivity of the chamber will be discussed. A quick test was designed to find the mean pressure of the background noise in the chamber as well as the mean pressure when the source is driven strongly. 50 random locations are used to find the mean pressure for both the background noise and sound pressure level when the chamber is excited by the source. From the 50 locations, the five closest to the mean are marked to be used for future tests.

Figure 4.1 presents the background noise present in the chamber when no tests are being performed during daytime hours. The overall mean pressure during the measurements was 57.8 dB, which is higher than those of a commercial chamber. This is likely due to the location of the chamber, being in the vicinity of mechanical, electrical, and plumbing (MEP) equipment used in the building. Although the walls and floor are very massive, the ceiling is metal sheeting and it is unknown what type of backing is behind the metal material. It is likely that most of the background noise is coming through the ceiling. Fortunately, the modest noise floor isn’t an issue as long as the source is driven high enough to exceed the noise
Figure 4.1. Background noise in the reverberation chamber.

Figure 4.2 shows the mean pressure in the chamber when white noise drives the B&K source at ~13.8 Watts of electrical power, placed in a tetrahedral corner of the chamber. The overall mean pressure when the room is excited is 102.1 dB. It appears that the room’s response is influenced by the B&K sound source, noted by the decline around 400 Hz, see Appendix C.1. This is the same slight rolloff in the source’s specifications.

The five locations that most closely yield a mean that matches the mean of the 50 locations are shown in Table 4.1 with the geometry defining their coordinates shown in Figure 4.3.

These five locations should be used for future tests as they most closely match the chamber’s average pressure, and are the five locations used in Reverberation Test 1. These locations were found by plotting each pressure and choosing the five locations that most closely matched the mean pressure. Figure 4.4 displays results
Figure 4.2. Mean pressure in the reverberation chamber over 50 locations, with the source placed in the corner.

Table 4.1. Table of five best locations in the reverberation chamber

<table>
<thead>
<tr>
<th>Location</th>
<th>Coordinates</th>
<th>x / total x distance</th>
<th>y / total y distance</th>
<th>z / total z distance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>13,18,6</td>
<td>.70</td>
<td>.82</td>
<td>.52</td>
</tr>
<tr>
<td>2</td>
<td>8,5,4</td>
<td>.42</td>
<td>.23</td>
<td>.35</td>
</tr>
<tr>
<td>3</td>
<td>7,17,7</td>
<td>.37</td>
<td>.77</td>
<td>.61</td>
</tr>
<tr>
<td>4</td>
<td>14,8,7</td>
<td>.75</td>
<td>.36</td>
<td>.61</td>
</tr>
<tr>
<td>5</td>
<td>13,9,6,5</td>
<td>.70</td>
<td>.27</td>
<td>.57</td>
</tr>
</tbody>
</table>

for the best five locations compared to the mean as well as the improvement in standard deviation when compared to all 50 locations. These five locations as well as the overall standard deviation are lower than the requirements of ISO 3741, in which the standard deviation must be below 3dB in third octave bands 100Hz - 160Hz, 2dB from 200Hz - 315Hz, 1.5 dB from 400Hz - 630Hz, and 1 dB above 800Hz [9].
Figure 4.3. Defining coordinate system of the reverberation chamber.

4.2 Modal Analysis Results

The loss factors computed for modes under 100 Hz that were detected in at least two planes are shown in Table 4.2. The results are not exactly uniform from plane to plane, but are reasonable. Scatter plots for all loss factors extracted for each plane are shown in Figure 4.5. Loss factors for all planes display the same trend, higher at low frequencies and lower at high frequencies.

A look at the averaged room response in each plane is given in Figure 4.6. Figure 4.6 shows that the density of modes rises very quickly as seen in Plot a. Plot b, which zooms into a closer scale, shows that most modes are detected in all the planes, but some are missed or not fully detected when there is little pressure along the plane, as in the case of the low response in the x-y plane around 74 Hz. Additionally, the plots show that the signal to noise level is relatively high for the average frequency response function. This is a limitation of the measurement and couldn’t be enhanced in any way.
Figure 4.4. Comparison of the best five locations to the 50 locations at which data was taken: a) Spatial averaged mean pressure, b) Spatial variability of pressure levels.

4.2.1 Mode Shapes

Figures 4.7 - 4.13 display the different mode shapes extracted using the modal analysis software. It should be noted that the Figures with the vertical mode pictures don’t have the top two rows due to the experimental limitation with the microphone stand. An additional two rows would need to be visually added to the top to complete the entire grid filling the chamber. Also each figure is captioned with the approximate resonance frequency. A common frequency couldn’t be set due to slight differences in temperature during the several days of data measurement. Additionally Figures 4.12 - 4.13 display sample extracted mode shapes for only the x-y plane. Figures 4.12 -4.13 show some extra shapes of modes extracted from only the x-y plane.
One surprising result is that the modal software is able to, in some cases, pick up modes that travel perpendicular to the plane. The result is the whole plane moving not just a portion of the plane. Yet, in other cases the software is unable to find the mode, as in Figure 4.10 where the horizontal plane is unable to pick up the 0,0,1, or the first mode in the vertical direction.

### 4.2.2 Modal Overlap Factor and Schroeder Frequency

Using the data collected in the modal tests the modal overlap factor (MOF) is calculated using:

$$\text{MOF} = \frac{\pi}{2} \omega \eta(\omega) n(\omega),$$

(4.1)

where $n(\omega)$ is the modal density. The modal overlap factor is the ratio of half-power bandwidth to local average interval between natural frequencies [29]. The $\pi/2$ factor refers to the random bandwidth, and comes from the 3 dB bandwidth. In the field of structural acoustics the practical lower limit for a statistical energy analysis (SEA) application is often considered to be a modal overlap factor of one.

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Approx. Frequency</th>
<th>x-y plane</th>
<th>x-z plane</th>
<th>y-z plane</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>0,1,0</td>
<td>26</td>
<td>0.020</td>
<td>0.025</td>
<td>0.026</td>
<td>0.024</td>
</tr>
<tr>
<td>1,0,0</td>
<td>30</td>
<td>0.030</td>
<td>0.024</td>
<td>0.029</td>
<td>0.028</td>
</tr>
<tr>
<td>1,1,0</td>
<td>40</td>
<td>0.022</td>
<td>0.018</td>
<td>0.017</td>
<td>0.019</td>
</tr>
<tr>
<td>0,0,1</td>
<td>51</td>
<td>0.009</td>
<td>0.016</td>
<td>0.016</td>
<td>0.013</td>
</tr>
<tr>
<td>0,2,0</td>
<td>52</td>
<td>0.012</td>
<td>0.014</td>
<td>0.011</td>
<td>0.012</td>
</tr>
<tr>
<td>0,1,1</td>
<td>57</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
<td></td>
</tr>
<tr>
<td>2,1,0</td>
<td>66</td>
<td>0.012</td>
<td>0.012</td>
<td>0.012</td>
<td>0.012</td>
</tr>
<tr>
<td>0,2,1</td>
<td>72</td>
<td>0.012</td>
<td>0.012</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0,3,0</td>
<td>77</td>
<td>0.008</td>
<td>0.007</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1,3,0</td>
<td>83</td>
<td>0.011</td>
<td>0.012</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3,0,0</td>
<td>91</td>
<td>0.007</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0,3,1</td>
<td>92</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3,1,0</td>
<td>94</td>
<td>0.010</td>
<td>0.010</td>
<td>0.010</td>
<td></td>
</tr>
<tr>
<td>2,3,0</td>
<td>98</td>
<td>0.008</td>
<td>0.007</td>
<td>0.008</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.2. Loss factors extracted for matching modes under 100 Hz.
Figure 4.5. All valid loss factors extracted for the modal analysis.

[30]. The modal density for an acoustic volume is given by:

$$n(\omega) = \frac{V\omega^2}{2\pi^2c^3} + \frac{S\omega}{8\pi c^2} + \frac{L}{16\pi c},$$

(4.2)

where $L$ is the room perimeter [m]. The first term dominates the modal density while the second and third terms are considered one and two dimensional correction factors. In room acoustics the room is considered sufficiently diffuse and modeled statistically above the Schroeder frequency, which corresponds approximately to the frequency above which there are at least three modes within a half-power bandwidth [12]. The Schroeder frequency can be defined as:

$$f_{sch} = 0.329c^{3/2} \sqrt{\frac{T_{60}}{V}} \approx 2000 \sqrt{\frac{T_{60}}{V}},$$

(4.3)

where $c$ is the speed of sound, $T_{60}$ is the overall reverberation time, and $V$ is the volume of the room.
Figure 4.6. Averaged pressure in the reverberation chamber for each plane: a) the spectrum up to 475 Hz, b) the spectrum up to 100 Hz.

Figure 4.7. First mode of the chamber 0,1,0 at ~26 Hz.

For all the loss factors extracted the reverberation chamber was found to have a modal overlap factor of one at a frequency of 119 Hz. This was found by calculating the MOF for all extracted modes and fitting a curve to the results to find the lowest frequency where MOF is 1. The Schroeder frequency calculated from the broadband reverberation time of 4.71 seconds is found to be 378 Hz. Note that
the Schroeder frequency is higher than the frequency where the MOF is 1 due to its definition, using the mean T60 time. Here, MOF is computed as a function of frequency, which accounts for room loss factors which vary with frequency. Since the room loss factors are higher at lower frequencies, MOF is increased, and reaches 1 at a lower frequency. Figure 4.14 shows the density of modes in the reverberation chamber, both theoretical and the volume only term, along with markers indicating MOF=1, Schroeder Frequency, and first chamber mode. Based on the calculated modal loss factor of 119 Hz it is reasonable to assume that the room is diffuse enough to be used in the 125 Hz third octave band.
4.3 Reverberation Time Results

4.3.1 Reverberation Test 1 Results

The first results presented are for the first reverberation test in which the chamber is excited with broadband white noise, and then filtered to third octave bands, before the decay rate is found. The decay rates are found using the MATLAB routine described in Appendix B. Also found are sample line fits to decays. The average and standard deviation for each third octave band of 100 decays, 20 decays at five different locations are found using:

\[
\begin{align*}
    d_{\text{avg}} &= \frac{1}{N} \sum_{i=1}^{N} d_i, \text{ and} \\
    d_{\text{stdev}} &= \left( \frac{1}{N-1} \sum_{i=1}^{N} (d_i - d_{\text{avg}})^2 \right)^{\frac{1}{2}},
\end{align*}
\]  

(4.4)
Figure 4.12. Modes extracted from the x-y plane.

where \( N \) is the total number of decays and \( d_i \) is the \( i \)th decay.

The raw decay rates measured are displayed in Figure 4.15. The ASTM C423 standard requires adjusting the reverberation time for the relative humidity in the chamber. The decay rate caused by the relative humidity is:

\[
d_{\text{air}} = \frac{mc}{1000},
\]

where \( m \) is the absorption coefficient (dB/km) defined in ANSI S1.26 [31]. The recorded humidity and temperature in the chamber at the time of the first test are given in Table 4.3. Figure 4.16 shows both the adjusted decay rate and the

<table>
<thead>
<tr>
<th>Relative Humidity</th>
<th>60%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>23.1°C</td>
</tr>
</tbody>
</table>

Table 4.3. Table displaying the temperature and relative humidity during the first test.
raw decay rate. It is seen that the humidity in the reverberation chamber causes a modest addition to the decay rate especially at high frequencies. This adjustment in decay rate is made so that when making absorption measurements, in which the difference in decay rate is compared, the decays compared don’t reflect any differences in humidity.

ASTM C423 also has a tight requirement for variation in decay in the reverberation chamber. This is determined by comparing the standard deviation of the decay rates with the average of the decay rates. Figure 4.17 shows the measured standard deviation over the average versus the requirements of the ASTM C423 standard. The test only meets the standard in some of the eighteen third octave bands of interest. It appears that the chamber meets the requirements in the 630 Hz third octave band and above, but not 500 Hz and below.

The conclusion is that the chamber doesn’t meet the standard due to variations
Figure 4.14. Mode count in the reverberation chamber with marked MOF=1, Schroeder Frequency, and first fundamental mode of the chamber.

in decay rate with changing position. Even using the best five locations found in Section 4.1 the requirement isn’t met. However, figure 4.4 shows that using these five locations produces an overall standard deviation of the chamber of less than 1 dB at and above 200 Hz. Therefore refer to Figure 4.4 as proof of diffusivity rather than Figure 4.17.

The decay rate can be converted to a reverberation time using:

$$T_{60} = \frac{60}{\text{decay rate}} \quad (4.6)$$

Doing so leads to a more common plot of reverberation time. The adjusted reverberation time for the empty chamber is shown in Figure 4.18
4.3.2 Reverberation Test 2 Results

The second results come from the test in which the reverberation chamber is excited with band limited white noise. The decay rates are calculated in the same manner as the first reverberation test except third octave filtering is skipped because the chamber is only excited by band limited noise. A sample of the decay fitting for the 1000 Hz third octave band is displayed in Figure 4.19. The signal to noise ratio is approximately 10 dB higher when compared to the previous case in which broadband noise was used to excite the room.

The temperature and humidity in the chamber during the test are given in Table 4.4. The raw decay rate and the adjusted decay rate calculated during

<table>
<thead>
<tr>
<th>Relative Humidity</th>
<th>62%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>23.2°C</td>
</tr>
</tbody>
</table>

Table 4.4. Table displaying the temperature and relative humidity during the second reverberation test.
Reverberation Test 2 are shown in Figure 4.20, with the average and standard deviation calculated using Equation 4.4.

Reverberation Test 2 results for variation in decay are plotted in Figure 4.21. The measured standard deviation over average for the band limited test hangs the required ASTM C423 level more closely than the results from Reverberation Test 1; likely due to the number of microphone positions used in the test. Five locations are used in Test 1 versus twelve locations in Test 2. At each location decay rates are found to be very similar decay to decay; while variation in location causes more of a difference in decay. It is likely that the locations used in Test 1 didn’t provide enough variance, even if using the ideal five locations. It appears that the variance requirement is met in third octave bands 800 Hz and above, but not met for bands below this level.

The reverberation time for the band limited tests are very similar to the the
results of Reverberation Test 1. Figure 4.22 shows the calculated reverberation time using band limited white noise.

4.4 Discussion of Reverberation Tests and other Parameters of the Reverberation Chamber

Figure 4.23 shows the results of the two reverberation tests plotted on the same graph. As speculated both methods yield similar answers. There is some variance at lower frequencies, yet this is to be expected as the sound field isn’t truly diffuse. Since Reverberation Test 1 takes much less time to preform, only having to record 100 decays as opposed to hundreds, the results from that test will now be used to compute other common room parameters of reverberation chambers.

Figure 4.24 shows the calculated reverberation times computed with the Sabine,
Eyring, and Fitzroy equations all with air absorption (Equations 2.12, 2.16, and 2.19), as well as the results of Reverberation Test 1. Standard text book values for absorption coefficients were used for all building materials, and are given in table 4.5, with air absorption computed with a temperature of 23°C and 70% relative humidity.

<table>
<thead>
<tr>
<th>Octave Band</th>
<th>Painted concrete walls</th>
<th>Sealed concrete floor</th>
<th>Metal deck ceiling</th>
<th>Doors</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>0.01</td>
<td>0.01</td>
<td>0.08</td>
<td>0.02</td>
</tr>
<tr>
<td>250</td>
<td>0.01</td>
<td>0.01</td>
<td>0.10</td>
<td>0.02</td>
</tr>
<tr>
<td>500</td>
<td>0.015</td>
<td>0.015</td>
<td>0.10</td>
<td>0.05</td>
</tr>
<tr>
<td>1000</td>
<td>0.02</td>
<td>0.015</td>
<td>0.10</td>
<td>0.08</td>
</tr>
<tr>
<td>2000</td>
<td>0.02</td>
<td>0.02</td>
<td>0.05</td>
<td>0.10</td>
</tr>
<tr>
<td>4000</td>
<td>0.02</td>
<td>0.02</td>
<td>0.08</td>
<td>0.10</td>
</tr>
</tbody>
</table>

Table 4.5. Table of coefficients used to compute Sabine, Eyring, and Fitzroy reverberation times.

The majority of the absorption in the reverberation chamber comes from the
metal deck ceiling. The Fitzroy reverberation approach displays a much higher reverberation time because of the undistributed absorption, matching the claims he made in this paper [18]. However, none of the theoretical methods match the measured data, which falls generally between the Fitzoy and other two methods. This is likely due to the actual room surface absorption coefficients differing from those given in Table 4.5.

The total absorption in the reverberation chamber is a measure of how much total surface area of the reverberation chamber can be considered as an open window, or area or total absorption. The total absorption is related to the decay
Figure 4.20. Raw decay rate and adjusted decay rate when chamber is excited by band limited noise.

Figure 4.21. Requirement of diffusivity in chamber, for Reverberation Test 2.
rate by constants and the volume of the chamber given by the equation:

$$A = 0.9210 \frac{Vd}{c}.$$  \hspace{1cm} (4.7)

The total absorption for each third octave band is shown in Figure 4.25.

Another important parameter of rooms is the critical distance \(r_c\). The critical distance is the distance away from the source at which the direct-field level is equivalent to the reverberant-field level. The critical distance is defined as:

$$r_c = \sqrt{\frac{A}{16\pi}}.$$  \hspace{1cm} (4.8)

Typically a reverberation chamber has a small critical distance, as they have very little absorption. The critical distance for each third octave band for the reverberation chamber is shown in Figure 4.26. The results range around 0.3 meters and
Figure 4.23. Two reverberation tests plotted against each other.

seem reasonable, as in a reverberant room with many echoes and little absorption it becomes difficult to identify the location of the noise source.

The last parameter computed is for the free field radiated sound power of the B&K sound source when driven by white noise at 13.8 Watts of electrical power and placed in the corner of the room. The relation is defined by:

$$L_w = \bar{L}_p + \left[10 \log_{10} A + 4.34 \frac{A}{S} + 10 \log_{10} \left(1 + \frac{S \cdot c}{8 \cdot V \cdot f}\right) - 25 \log_{10} \left(\frac{427}{400} \sqrt{\frac{273}{273 + T}} \cdot \frac{B}{B_0}\right) - 6\right],$$

where $L_w$ is the sound power level of the source, $\bar{L}_p$ is the mean pressure in the chamber, $T$ is the temperature of the chamber in °C, $B$ is atmospheric pressure [Pa], and $B_0$ is 101300 [Pa] [9]. The correction factor, term in brackets, is made up of a correction for absorption by walls, a correction for friction on walls or Waterhouse correction, and a corrections for absorption by air. The sound power level is found from the average pressure with a correction factor that depends on absorption, temperature, and atmospheric pressure. Figure 4.27 plots correction
Figure 4.24. Modeled chamber reverberation times with calculated reverberation times using Sabine, Eyring, and Fitzroy Equations.

The reverberation times and loss factors can be related by:

\[ \eta(f) = \frac{2.20}{f T_{60}} \]  

as derived in Appendix A.

Figure 4.28 displays all the extracted loss factors and the reverberation times from Reverberation Test 1 converted to loss factors. The comparison of the two methods is reasonable. The loss factors that come from reverberation times tend to be a little bit lower, which may have several causes. The reverberation times
are calculated using the first 25 dB of the decay of a third octave band, while the loss factor is extracted looking at the whole decay of one resonant peak. In any given one third octave band it is unknown what modes dominate the room response, as some modes could dominate the first few dB of a decay while other modes only add in to the later portion of the decay. Another reason for the lower
Figure 4.27. Plot of the correction factor to find the free field sound power level of a source in the reverberation chamber.

reverberation loss factors could be due simply to the modal density. As shown in the modal pre-study in Section 3.3 the modal analysis software tended to extract slightly higher loss factors than what was input.

For completeness Figure 4.29 displays loss factors averaged into third octave bands and converted to reverberation times.
Figure 4.28. Plot All extracted loss factors with the reverberation times form Test 1 converted to loss factors.

Figure 4.29. Computed reverberation times from the loss factors and from Reverberation Test 1.
Chapter 5

Conclusions

5.1 Discussion of Results

In this thesis the characteristics of the reverberation chamber in the CAV facilities at PSU have been found several different ways. The first two ways found the reverberation time by typical source-on source-off tests with different band limited white noise. The third method relates loss factors and reverberation times using modal analysis. The results match fairly well with the high modal densities likely biasing modal loss factors higher.

Studying the chamber using modal analysis was an interesting approach to finding reverberation times. It provided additional results other than reverberation times, such as the mode shape pictures. However, testing is time consuming as the grid requires many more locations than the other methods, and is limited to the low frequency characteristics of the room.

The reverberation tests yielded good results showing a high reverberation times in the range of 7-3 seconds, however, the chamber only meets the diffusivity requirement in ASTM C423 above the 630 Hz third octave band. The addition of diffusers will aid in the diffusivity of the chamber. Analysis of the Schroeder frequency finds that above 380 Hz the room can be looked at statistically. The
modal analysis test found a MOF of one below the 125 Hz third octave band. This specifies that the chamber has a modal spacing close enough to be looked at statistically in the 125 Hz third octave band and above. Yet when looking at the mean pressure in the chamber the standard seems to be met at any frequency range. The deviation in level from position to position is low and below 1dB 160 Hz and above.

Additionally, other specifications of the reverberation chamber were found and discussed. The background noise level in the reverberation chamber is found to be 57.8 dB. While higher than most commercial chambers, it will still provides a low noise floor for the tests to be preformed. Also the corrective term to change a measurement taken in the reverberation chamber to a free field sound power measurement was found following ISO 3741. The correction varies with frequency between ±3 dB. Additionally the critical distance or reverberation radius for the chamber was found. This term also varies with frequency and is less than half a meter in all third octave bands.

5.2 Future Work

Reverberation times in the reverberation chamber in the CAV area at PSU are high and well within the workable requirements for tests. However, the diffusivity in the chamber is not up to ASTM C423 standard’s requirements. Ideally the chamber would not have a rectangular geometry, however changing this would be extremely expensive. By adding some hanging diffusers or wall diffusers the requirement for diffusivity will likely be met. Additionally the absorption coefficients of the materials used to build the chamber are not known. Investigation should be done to find out what the exact coefficient of each surface is. By using the Fitzroy reverberation equation and covering up the entire floor, ceiling, or one entire wall the absorption coefficients of the actual surfaces could be extracted.
Changing the chamber from its empty state by adding diffusers will likely change the current results discussed in the thesis. The five ideal locations discussed in Section 4.1 will likely change, as well as the reverberation times, as materials will be introduced to the reverberation chamber. After the installation of diffusers the entire process of finding the ideal locations and reverberation times must be repeated, however, the results should meet the standard’s requirements. If diffusers are added the modal analysis results would also change. It would interesting to see how the mode shapes change with the addition of diffusers.

One question through the entire study of the chamber is what is the material of the ceiling and why is it in the shape it is. It appeared to be a steel deck in a corrugated pattern. Perhaps by covering the deck with some other flat nonabsorbent surface the high frequency properties will change for the better, by eliminating any Helmholtz resonator effects.
Appendix A

Relationship between Reverberation Time and Loss Factor

The purpose of this Appendix is to clarify the relationship between a loss factor \( \eta \) and a common used value of decay time \( T_{60} \). The Appendix first contains the mathematical derivation of the equation relating the two values, followed by a validation of the mathematics by extraction of values from plots.

A.1 Math of the Simple Exponential Decay

The general definition of a loss factor is the energy lost per radian divided by the total energy:

\[
\eta = \frac{D}{2\pi} \frac{1}{W_T},
\]

where \( D \) is the energy lost per cycle, and \( W_T \) is the total energy. Assuming that the rate that energy is dissipated is constant at all times leads to:

\[
-\frac{dW_T}{dt} = \frac{D}{T} = \frac{2\pi \eta W_T}{T} = \omega \eta W_T,
\]
where \( T \) is the period of the vibration. Rearranging Equation A.2 into a first order homogeneous differential equation:

\[
\frac{dW_T}{dt} + \eta \omega W_T = 0. \tag{A.3}
\]

The solution to this type of equation is well known, with this particular case being:

\[
W_T(t) = W_{T0} e^{-\eta \omega t} \tag{A.4}
\]

Now define \( T_{60} \) as the time it takes the signal to reach a value of one millionth its initial value, or by a value of -60dB. Such an assumption leads to:

\[
W_T(T_{60}) = \left( \frac{1}{1,000,000} \right) W_{T0} = W_{T0} e^{-\eta \omega T_{60}}. \tag{A.5}
\]

Now rearranging the mathematics to relate \( \eta \) to \( T_{60} \):

\[
\left( \frac{1}{1,000,000} \right) = e^{-\eta \omega T_{60}}
\]

\[
\ln \left( \frac{1}{1,000,000} \right) = -\eta \omega T_{60}
\]

\[
\eta(\omega) = -\frac{\ln \left( \frac{1}{1,000,000} \right)}{\omega T_{60}}
\]

\[
\eta(f) = \frac{2.20}{f T_{60}}, \tag{A.6}
\]

where \( f = \omega/2\pi \), and is the frequency of interest.

### A.2 Validation of the Mathematical Derivation

The validation of the mathematics will be done by finding the decay time by the extraction from a graph. An example is shown in Figure A.1. The graph was
found by applying a decay using Equation A.4 to a \( \cos(\omega t) \) signal, with \( W_{T0} = 1, \eta = .1 \) and \( \omega = 2\pi \cdot 5 \). An even cleaner version of the graph can be seen by adding in the envelope and noticing that the signal’s decay follows the envelope exactly, as shown in Figure A.2.

When looking at Figure A.1 or Figure A.2 the pictured decay visually seems to occur quickly. However the value of \( T_{60} \) can’t be determined by simple inspection, due to the scale of the plot. A better way to look at the decay is on a logarithmic scale, where the decay is normalized and much clearer. Figure A.3 shows the decay on a logarithmic scale. The graph shows that \( T_{60} \) can be found from visual inspection by locating -60dB. Rather than from visual inspection, which could be done, \( T_{60} \) is found by looking at the vector of normalized energy amplitude, and choosing the corresponding value from the time vector. Doing so yields the following value:

\[
T_{60 \, \text{Graph}} = 4.40 \, [s]. \quad (A.7)
\]
Figure A.2. Energy signal being dissipated by a constant amount every cycle with included decay envelope.

Comparing the value found from the graph to the value found using Equation A.6:

\[
T_{60}^{\text{Equation}} = \frac{2.20}{f \eta} = \frac{2.20}{5 \cdot 1} = 4.40 \text{[s]}, \quad (A.8)
\]

It has been shown that the equation derived to relate \(T_{60}\) and \(\eta\) is in fact correct.
Figure A.3. Decay displayed on a logarithmic scale.
Appendix B

MATLAB Routine used to Identify Decay Rates

This appendix focuses on the code developed to find decay rates from times series data. The code developed was written to follow the requirements found in ASTM C423 [7].

B.1 Description and Requirements

Standard ASTM C423 has several requirements pertaining to the number of microphone positions and the number of decays to be measured. The standard also contains requirements to be followed when measuring the decay of sound using the speaker-on speaker-off method. The standard’s requirements of measurement of decay are not strict, and are easily met using the modern equipment available today. If integrating the time signal, the integration time must be between 90 -100% of the output interval time. The output interval time must be small enough for at least five measurement points, from a time 100 ms after the source is shut off to when the signal has reached a level of 25 dB below that level. The standard suggests ten points if possible.
From all of the data acquired, the sound pressure levels in each corresponding output interval are then averaged together; starting with the output interval 100 ms after the source is turned off to the last output interval before reaching 25 dB below the starting level. The decay rate (dB/s) is then calculated from the averaged sound pressure levels using a first order regression. This analysis of decays must be done for third octave bands in the range of 100 Hz - 5kHz.

The MATLAB routine developed goes above and beyond the requirements of the standard. Instead of having only five to ten output intervals through the entire decay, the code developed uses an output interval of 25 ms. Using this output interval a reverberation time of 5 seconds would have 83 output intervals before reaching a point of -25 dB.

Due to the way that the data was acquired, there was no way to accurately trigger and know the exact time at which the source was turned off. This presents a difficulty, and an exact following of the standard isn’t possible. It would be most impossible to link up the corresponding output intervals in over 50 decays and average them correctly. Therefore the solution to this problem is to find the decay rate of each decay and average them, instead of averaging the sound pressure levels in corresponding output intervals.

**B.2 Code Snippets**

Below is the code used to find the decay rates of time series. The code is very basic but can be modified to automatically load in files, and average all the decays.

**B.2.1 Detailed Description of MATLAB Code**

The first step in the MATLAB code is the loading of the .wav file. Since the decay is going to be measured the signal doesn’t need to be converted to actual pressure. The second step begins a loop in which the original time series is filtered to correct
one-third octave bands. Once filtered the time series is squared making it one sided and all positive.

Third is the integration, or averaging, of the time series, into output intervals of 25ms. For all data collected, a sample rate of 44.1 kHz, and an integration of 2205 points are used. This means that 2250 original data points are averaged together to make one point. This is done with a 50% overlap, meaning that the second integration starts with the 1126 data point acquired. This process is then repeated for the full time series.

The fourth step takes $10 \log_{10}$ of the integrated signal over the max of the signal; this normalizes the signal to the maximum value, and changes the vertical scale to deciles. A new time vector is then created to match the time spacing of the new integrated time signal.

In the fifth step, a line is fit to the decay portion of the signal. The decay portion is found in-between levels from -9 dB to -29 dB. Fitting a line to this portion of the decay is found to work well with all third octave bands. From the slope of the fit line, the decay rate is found as well as the reverberation time. Lastly, a graph of the integrated signal and the fit line are plotted to visually check that a decent fit was found.

Figures B.1 - B.2 show different one-third octave bands with the fit line to the integrated signal. It is seen that the fit lines match the decays well. The 100 Hz third octave band signal is choppy when compared to the 1000 Hz third octave band. This is simply the nature of lower frequencies, as they oscillate at a slower rate. Despite the choppiness, the small output interval and fit line still identify the decay well in both cases. It is observed that the fit lines do not seem to match overly well in the very last portions of the decay. This is because the decay is fit to the first 25 dB of the decay, as is called for by the standard. Even though the decay should be linear, strange effects seem to occur in the last portion of the decay in most cases.
Figure B.1. Fit line with integrated signal of 100 Hz third octave band.

The decays in Figures B.1 - B.2 are found by filtering third octave bands from a broadband decay signal. This causes a higher signal to noise ratio than if the room was excited by band limited white noise. The results are still valid as the decays are fit to the first 25 dB of decay. Sample plots with a higher signal to noise ratio can be found in Section 4.3.2.

B.2.2 Code

% Code to take .wav files and extract the reverb time using only a moving average for each 1/3 octave band, then a curve is fit to the first part of the decay

[x,fs] = wavread(some_wav_file.wav);
% Background info on the signal
N = length(x);
Figure B.2. Fit line with integrated signal of 1000 Hz third octave band.

\[
\begin{align*}
n &= 1:N; \\
t &= n./fs; \\
\% \text{ Filter the signal into } 1/3 \text{ octave bands} \\
f_c &= \{100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250, \ldots, 1600, 2000, 2500, 3150, 4000\}; \\
\text{for } l = 1:\text{length}(f_c) \\
[bb, aa] &= \text{third_octave_filter}(f_c(l), fs); \\
x_\text{filtered} &= \text{filtfilt}(bb, aa, x_s); \\
\% \text{ Find the just the pressure squared (positive part)} \\
sig &= x_\text{filtered}^2; \\
\% \text{ Create the integrating (averaging) of the pressure squared signal} \\
\% \text{ First create a matrix of pressure that will be averaged}
\end{align*}
\]
\( N_r = \text{floor}(fs/20); \) % Number of points in the column
\( ol = 50; \) % Amount of overlap (percent)
\( N_c = \text{floor}(\text{length}(\text{sig})/((1-.01*ol).*N_r))-1; \) % Number of points in each row
\( x_{\text{fil}} = \text{zeros}(N_r,N_c); \)
for \( p = 1:N_c \)
  if \( (p-1 == 0) \)
    \( x_{\text{fil}}(:,p) = \text{sig}(1:p*N_r,1); \)
  else
    \( x_{\text{fil}}(:,p) = \text{sig}((p-1)*N_r*(1-.01*ol)+1:(p-1)*N_r*(1-.01*ol)+N_r,1); \)
end
\%
% Find the average
\( \text{rms}_p = \text{sqrt}(\text{mean}(x_{\text{fil}}.^2)); \)
\%
% Normalize and move to log space
\( \text{Lp} = 10*\log10(\text{rms}_p/\max(\text{rms}_p)); \)
\%
% Now make the time vector which works with the 50% overlap
\( \text{time} = \text{ones}(1,\text{length}(\text{rms}_p)); \)
for \( m = 1:N_c \)
  \( \text{time} = t(1:m)*\text{floor}(N_r/2); \)
end
\%
% Find the point that the decay drops 25dB
\( \text{index1} = \text{find}(\text{Lp}>-9); \) %choose beginning point
\( \text{index2} = \text{find}(\text{Lp}>-29); \) %Choose ending point
\( \text{t}_f = \text{time}(\max(\text{index1}):\max(\text{index2})); \)
\( \text{Lp}_f = \text{Lp}(\max(\text{index1}):\max(\text{index2})); \)
\%
% Fit a curve to the point between -9 and -29 dB
fit = polyfit(t_f,Lp_f, 1);
% Now use the given slope to find decay and T.60
decay(l) = abs(fit(1,1));
T.60(l) = 60./abs(fit(1,1))
% Plot fitted curve with decay
figure(1)
plot(time,Lp,’r’,time,fit(1,1)*time+fit(1,2),’b’) % Change to plot of abs
ylim([-80 0]); grid;
title(’Example of Moving average 1/3 octave filtering’)
ylabel(’Pressure Envelope (dB)’); xlabel(’Times (s)’)
end
Other Reference Material

C.1 Crown Amplifier and EASERA Preamp Calibration

Figure C.1 shows the calculated gain for the Crown XTi2000 power amplifier for every click on the amplifier knob. Figure C.2 displays the calibrated gain for the microphone preamp enclosed in the EASERA Gateway for every click on the gain knob.

C.2 Calibration Sheet for B&K Source

Figures C.3 - C.4 show the calibration curves for the Brüel & Kjær Type 4292 OmniPower Sound Source used in all experiment work.

C.3 Goldline Microphone Calibration Curves

Figure C.5 displays one of the microphone calibration curves. Both microphones are phase matched so it is assumed that both microphones have the same response.
Figure C.1. Calibration curve for the Crown XTi2000 Amplifier used to power the B&K source.
Figure C.2. Calibration curve for the EASERA microphone preamps.
Figure C.5. Calibration curve of one of the TEF Goldline microphones used in all tests.
EASERA Reverberation Times Tests

D.1 Setup of EASERA Reverberation Tests

An attempt to triple check the reverberation times calculated is preformed using commercial software EASERA. EASERA stands for Electronic & Acoustic System Evaluation & Response Analysis and has the ability to compute the reverberation times using built in algorithms. Ten random locations are used for the EASERA tests, with 20 averages at each location, and the source is placed in a tetrahedral corner of the chamber. The EASERA software gives the ability to use several different types of source signals. Three different source signals of white noise, maximum length sequence (MLS), and weighted sweeps are used to compute reverberation times. EASERA also computed the reverberation time four different ways [32] shown in table D.1

<table>
<thead>
<tr>
<th>Early Decay Time (EDT)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>T 10</td>
<td>60 dB extrapolated for the level between -5 dB and -15 dB</td>
</tr>
<tr>
<td>T 20</td>
<td>60 dB extrapolated for the level between -5 dB and -25 dB</td>
</tr>
<tr>
<td>T 30</td>
<td>60 dB extrapolated for the level between -5 dB and -35 dB</td>
</tr>
</tbody>
</table>

Table D.1. Description of the four different reverberation times computed by the EASERA software
The signal flow for the EAERA test follows the Sig Flow A type, letting the GATEWAY do the AD/DA conversion. The computer software EASERA then generates all signals and computes all the reverberation times using its build-in algorithm.

### D.2 EASERA Reverberation Time Results

The different results found using commercial software EASERA are displayed in Figures D.1-D.3. Figure D.1 shows the four different reverberation times for the results of the white noise test, while Figures D.2 and D.3 show the MLS signal and weighted sweeps respectively.

![Reverberation Time Graph](image)

**Figure D.1.** Reverberation time of the chamber found using commercial software EASERA with white noise source signal.

The results of the test are not promising. It appears that EASERA has some difficulties in all tests, computing something incorrectly in all cases. It is difficult to pinpoint the exact error as no literature states exactly what processes occur in order to compute the reverberation times. Perhaps somewhere in the third
Figure D.2. Reverberation time of the chamber found using commercial software EASERA with MLS source signal.

Averaging octave band filtering or and incorrect triggering of a parameter to calculate the reverberation time, causes these errors.

The results are not a total disappointment. If looking at only the good data, in which the error bars are tighter, trends are noticed. One thing to notice are the similarities in reverberation times with the white noise signal and MLS signal using EASERA and the results from Reverberation Test 1 as is shown in Figure D.4. The EASERA times compared in the Figure are for the extrapolated times computed using T 20, which closely matches the Matlab routine used to compute the times in Reverberation Test 1. All have similar results with the higher frequencies tailing off to below two seconds. Another thing to notice is the lower reverberation times calculated using weighted sweeps, this is likely due to the chamber not being excited for some time, as the sweep travels through frequencies quickly.
Figure D.3. Reverberation time of the chamber found using commercial software EASERA with weighted sweep source signal.
Figure D.4. Comparison of EASERA tests to Reverberation Test 1.
Appendix E

Absorption Measurements in Reverberation Chamber

Typical absorption measurements were done on a sample provided by Blachford. Four acoustic panels with a total area of 6.27 m² along with calibration curves for absorption coefficient measured in their reverberation chamber were sent to Penn State’s CAV. The sample is a 0.46 lb/ft² vertically lapped black fiber, which is nominally 2” thick. It is composed of two compressed fiber layers approximately 0.08 thick with a 1.8 thick lower density middle layer.

Absorption measurements using ASTM C423 were attempted in the reverberation chamber along with tests in an impedance tube.

E.1 Impedance Tube Tests

In an impedance tube test, involves measuring the absorption coefficient of a sample by determining the ratio of reflected sound to incident sound on the sample. The Brüel & Kjær Standing Wave Apparatus Type 4002 uses is a long 10 cm diameter pipe with a speaker at one end and a sample placed against a rigid block at the other. The absorption coefficient is determined by finding differences in the
peak and null of the standing wave created in the pipe.

### E.1.1 Theory of Impedance Tube Test

The theory of a impedance tube test lies on the principles of standing waves in a tube [33]. Pressure in a pipe is a function of a left and right traveling plane wave, which follows the equation:

\[
p = Ae^{i[\omega t+k(L-x)]} + Be^{i[\omega t-k(L-x)]},
\]

(E.1)

where \( L \) is the length of the pipe and \( A \) and \( B \) are determined by boundary conditions. By setting:

\[
A = A \quad \text{and} \quad B = Be^{i\theta},
\]

(E.2)

Equation E.1 can be modified into:

\[
|p| = [(A + B)^2 \cos^2 [k(L - x) - \theta/2] + (A - B)^2 \sin^2 [k(L - x) - \theta/2]]^{1/2}
\]

(E.3)

Define the ratio of pressure maximum to pressure minimum as the Standing Wave Ratio (\( SWR \)):

\[
SWR = \frac{A + B}{A - B}
\]

(E.4)

The sound power reflection coefficient (\( R \)) is the ratio of reflected sound against incident sound:

\[
R = \frac{B}{A} = \frac{SWR - 1}{SWR + 1}
\]

(E.5)

Relating sound power reflection coefficient to absorption coefficient:

\[
\alpha = 1 - R^2 = 1 - \frac{(SWR - 1)^2}{(SWR + 1)^2}
\]

(E.6)

The Brüel & Kjær Standing Wave Apparatus Type 4002 used contains a small tube that enters the pipe. The tube is fixed to a sound level meter outside the pipe,
where maximum and minimum pressure are displayed. The SWR is calculated from the ratio of the maximum pressure to minimum pressure. From the SWR the absorption coefficient can be found using Equation E.6. The impedance tube test is run using single frequencies corresponding to third octave band center frequencies of bands 200 Hz to 5000 Hz. The test is run using four different drive voltages to the speaker in the apparatus. This is done to ensure accurate results.

### E.1.2 Results of Impedance Tube Test

The results of the impedance tube test are shown in Figure E.1. Driving the

![Figure E.1. Results of impedance tube test driving with four different amplitudes.](image)

tube at different amplitudes doesn’t change the results much at all. There is one discrepancy in the data around the 3150 Hz third octave band. Around this frequency radial modes begin to appear in the cross section of the pipe. Figure E.2 shows the results of the impedance tube corrected to account for a random incidence compared to the specifications sent from the manufacturer [34]. London
corrects to random incident using:

\[
\alpha_{\text{rand}} = 4 \left[ \frac{1 - (1 - \alpha_0)^{1/2}}{1 + (1 - \alpha_0)^{1/2}} \right] \times \left[ \ln(2) - \frac{1}{2} \ln[1 - (1 - \alpha_0)^{1/2}] - \frac{(1 - \alpha_0)^{1/2}}{2} \right],
\]

(E.7)

where \(\alpha_0\) is the normal incident absorption coefficient experimentally found using the impedance tube.

\[\text{Figure E.2.} \quad \text{Results of impedance tube absorption coefficients corrected for random incident.}\]

When comparing the measured absorption coefficients in the impedance tube to the data sent from Blachford agreement is close. The Blachford calibration comes from a random incident test in a reverberation chamber, while an impedance tube test corrected to give the normal incident value.
E.2 Reverberation Chamber Absorption Coefficient Test

E.2.1 Theory of Absorption Measurements following ASTM C423

The absorption coefficients calculated using ASTM C423 are found using a change in decay rates [7]. The decay rates are calculated for both the empty chamber and with the sample installed. From the decay rates the total absorption in the reverberation chamber is calculated using:

\[ A = 0.9210 \frac{Vd}{c}, \quad (E.8) \]

where \( d \) is the decay rate. The total absorption of the sample is found using:

\[ A_{\text{panel}} = A_2 - A_1, \quad (E.9) \]

where \( A_2 \) is the absorption of the room with the sample in place, and \( A_1 \) is the absorption of the empty reverberation chamber. The absorption coefficient of the sample can then be found using:

\[ \alpha = (A_2 - A_1)/S + \alpha_1, \quad (E.10) \]

where \( S \) is the total surface of the sample, and \( \alpha_1 \) is the absorption coefficient of the surface behind the sample.

The decay rate were calculated in the same way as in Section 3.2.2, except two decay rates are measured, one with the sample installed and one with the chamber empty. Tests using broad band white noise and band limited white noise are performed.
E.2.2 Results of Absorption test in the reverberation Chamber

The results of the ASTM C423 absorption coefficient tests are shown in Figure E.3 along with the results of the adjusted impedance tube test. The results don’t match the known values measured in the Blachford Lab. Exact reasoning is not known as the standard was followed rigorously. Figure E.4 shows the difference in the decay envelope for the 500Hz third octave band It appears that a large enough difference is there to decipher an absorption difference.

The test preformed in the Blachford used angle iron and plastic trim around the edges of the sample. This wasn’t done in the tests performed in CAV’s reverberation chamber. The performance of a test without the plastic trim would likely yield results of a higher absorption coefficient, as more area of the sample
Figure E.4. Decays of empty chamber and chamber with sample for the 500Hz third octave band.

...can absorb sound. As found in the results the diffusivity of the chamber isn’t high enough at lower frequencies, however, this doesn’t completely explain the inconsistent results between the CAV facilities and Blachford’s facilities. Investigation of this problem was attempted using a power injection approach. Using the mean pressure in the room with and without the sample, differences in pressure are related to differences in absorption. This analysis also yielded poor results. Further investigation is need to decipher exactly what is happening and why the results don’t match.
Bibliography


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