

8. ACOUSTICS OF ROOMS AND ENCLOSURES

8.1 Introduction

This section covers the acoustics of enclosed spaces. Upon completion, the reader should have a basic understanding of how to design spaces with suitable acoustic characteristics for a particular use.

The two fundamental qualities that determine a room's suitability for a particular use are:

- Reverberance or Liveliness: primarily a function of the sound absorption in the room and quantified by the *Reverberation Time*
- Background Noise Levels: predominantly HVAC noise, quantified by the NC or RC value

Typical applications:

- Acoustical spaces such as concert halls, classrooms, churches, offices, etc
- Industrial Environments - occupied spaces, or enclosures around noise sources

8.2 Sound Fields in a Room

Important Concepts:

Near Field

Far Field

Free Field

Reverberant Field

Diffuse Field

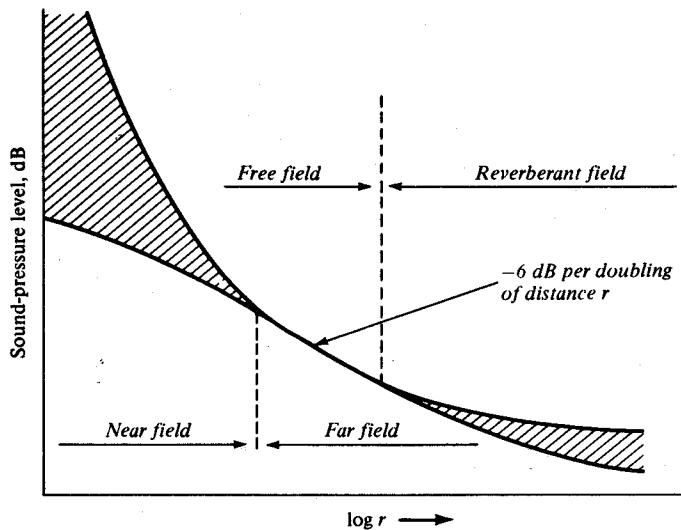


Figure 1. Sound pressure level variation with distance from the source

8.3 Sound Absorption

As sound strikes a wall, some of it is reflected, while some is absorbed by the wall. A measure of that absorption is the absorption coefficient α , defined as:

$$\alpha = \frac{I_{absorbed}}{I_{incident}} = \frac{I_{incident} - I_{reflected}}{I_{incident}} \quad \text{Equation 1}$$

- $\alpha = 1$ if totally absorptive
- $\alpha = 0$ if totally reflective

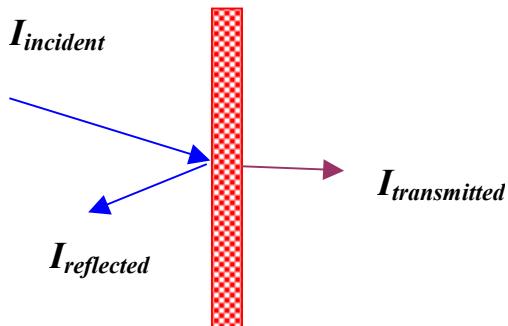


Figure 2. Sound striking an absorbing wall

α is a function of the material, the frequency, and incidence angle

While some of the absorbed sound is dissipated as heat in the material, some re-radiates from the other side. The amount of energy that gets into the next room is quantified by the transmission coefficient: (more on this in Section 9)

$$\tau = \frac{I_{transmitted}}{I_{incident}} \quad \text{Equation 2}$$

Absorption can be obtained by three primary mechanisms:

- porous materials,
- panel resonators or
- volume resonators:

Porous materials: Energy dissipation occurs due to acoustic pressure fluctuations at the surface which pump air into and out of the material. Friction between this air flow and the tortuous passages of the material dissipate energy as friction, and ultimately heat. Materials in this category include fiberglass, open cell foam, carpet and fabric. The frequency dependence for felt (a common absorption material) is shown in Figure 3.

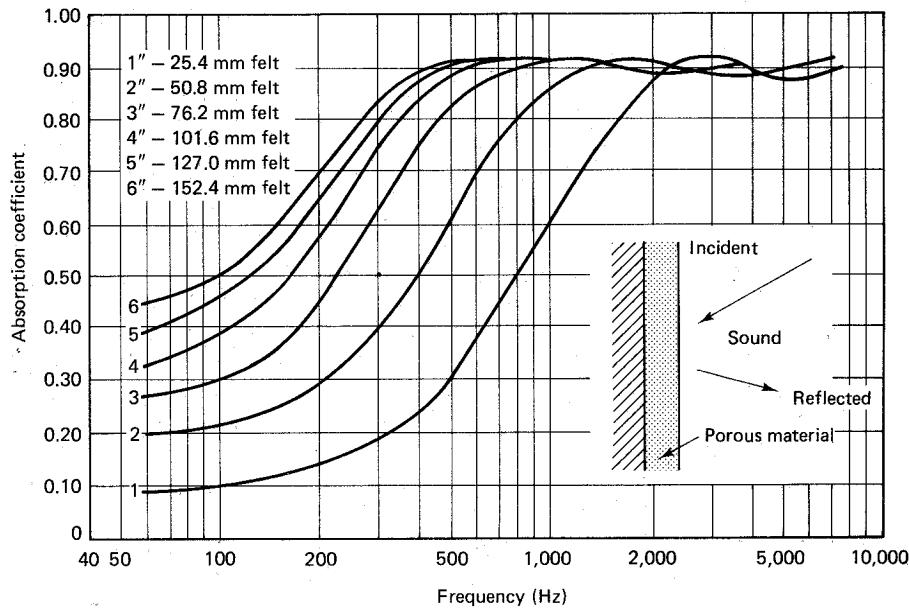


Figure 3. Variation of absorption coefficient with frequency for a porous material – felt

As seen in Figure 3, porous materials are more effective for absorbing high frequency sounds. The effectiveness depends on the thickness, relative to the sound wavelength. In order to be effective (nearly anechoic) at a given frequency, the material thickness must equal to at least $\frac{1}{4}$ of a wavelength. It is difficult to obtain low frequency absorption with porous materials (they would have to be very thick).

Rule of thumb: the lowest frequency that will be effectively absorbed by a porous material has a wavelength of **four (4)** times the absorbent thickness

Example: 152.4 mm (6") thick material will effectively absorb all frequencies above approximately 565 Hz. ($f = c/\lambda = 343/.608$)

Panel Resonators: Any flexible panel which vibrates in response to incident sound will transmit some sound energy to the other side (and therefore decrease the reflected sound). The effect is most pronounced at low frequencies. Typical examples include drywall, plywood, glass panes, sheet metal panels, metal roof decks. Low frequency absorption is usually highly desirable and this is sometimes the only way to achieve it.

Volume Resonators: These are all some variant of a Helmholtz resonator, the characteristic of which is a narrow band of high transmission loss. Bass trap closets are one example which can be designed into a room. Another example is SoundBlox, a commercially available concrete block shown in Figure 4. These are designed to provide low frequency absorption as seen in Figure 5. They work well if you can include them in the original construction, but are not well suited for retrofit.

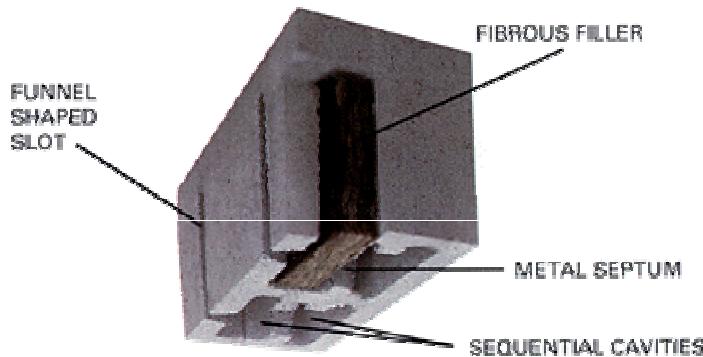


Figure 4. SoundBlox type RSC, a concrete cinder block with enclosed volume resonators for low frequency absorption

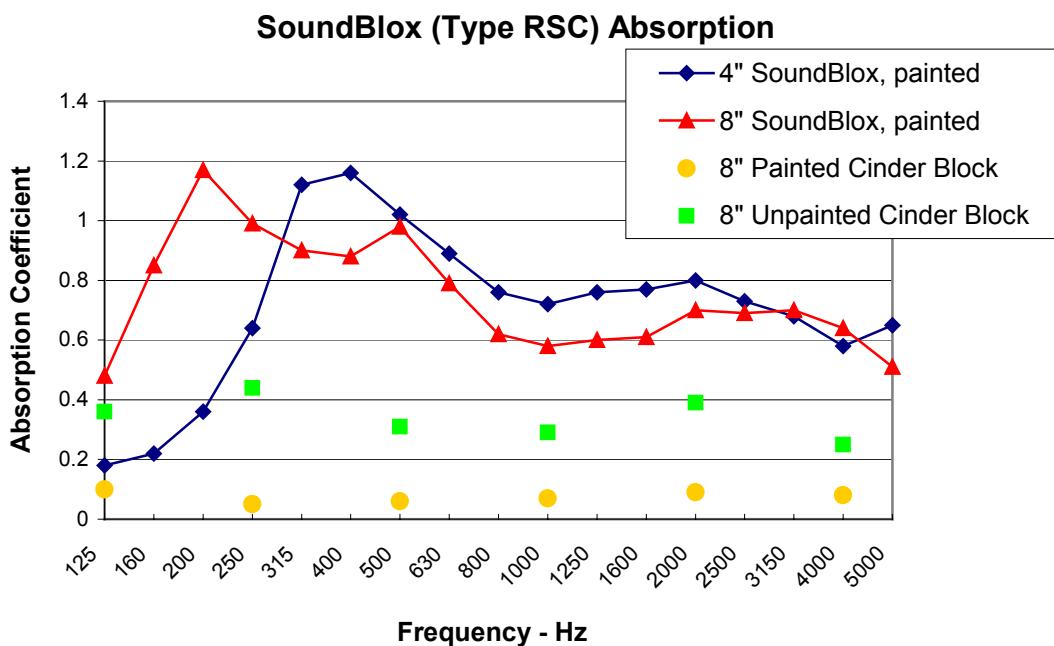


Figure 5. Absorption coefficient of SoundBlox compared to ordinary solid blocks (SoundBlox data from Proudfoot Company).

Published Absorption Coefficient Values

Absorption coefficients for commercially available materials are measured and published by manufacturers. A typical tabulation is shown in Table 1. It is possible to have absorption coefficient values greater than 1.0 for finite sized panels due to diffraction effects at the edges, and the additional absorption caused by the exposed area along the sides.

Table 1. Absorption coefficients of common building materials (ref. NIOSH Compendium of Noise Control Materials, 1975)

COEFFICIENTS OF GENERAL BUILDING MATERIALS AND FURNISHINGS

Complete tables of coefficients of the various materials that normally constitute the interior finish of rooms may be found in the various books on architectural acoustics. The following short list will be useful in making simple calculations of the reverberation in rooms.

Materials	Coefficients					
	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Brick, unglazed	.03	.03	.03	.04	.05	.07
Brick, unglazed, painted	.01	.01	.02	.02	.02	.03
Carpet, heavy, on concrete	.02	.06	.14	.37	.60	.65
Same, on 40 oz hairfelt or foam rubber	.08	.24	.57	.69	.71	.73
Same, with impermeable latex backing on 40 oz hairfelt or foam rubber	.08	.27	.39	.34	.48	.63
Concrete Block, coarse	.36	.44	.31	.29	.39	.25
Concrete Block, painted	.10	.05	.06	.07	.09	.08
Fabrics						
Light velour, 10 oz per sq yd, hung straight, in contact with wall	.03	.04	.11	.17	.24	.35
Medium velour, 14 oz per sq yd, draped to half area	.07	.31	.49	.75	.70	.60
Heavy velour, 18 oz per sq yd, draped to half area	.14	.35	.55	.72	.70	.65
Floors						
Concrete or terrazzo	.01	.01	.015	.02	.02	.02
Linoleum, asphalt, rubber or cork tile on concrete	.02	.03	.03	.03	.03	.02
Wood	.15	.11	.10	.07	.06	.07
Wood parquet in asphalt on concrete	.04	.04	.07	.06	.06	.07
Glass						
Large panes of heavy plate glass	.18	.06	.04	.03	.02	.02
Ordinary window glass	.35	.25	.18	.12	.07	.04
Gypsum Board, $\frac{1}{2}$ " nailed to 2x4's 16" o.c.	.29	.10	.05	.04	.07	.09
Marble or Glazed Tile	.01	.01	.01	.01	.02	.02
Openings						
Stage, depending on furnishings			.25	—	.75	
Deep balcony, upholstered seats			.50	—	1.00	
Grills, ventilating			.15	—	.50	
Plaster, gypsum or lime, smooth finish on tile or brick	.013	.015	.02	.03	.04	.05
Plaster, gypsum or lime, rough finish on lath	.14	.10	.06	.05	.04	.03
Same, with smooth finish	.14	.10	.06	.04	.04	.03
Plywood Paneling, $\frac{3}{8}$ " thick	.28	.22	.17	.09	.10	.11
Water Surface, as in a swimming pool	.008	.008	.013	.015	.020	.025
Air, Sabins per 1000 cubic feet @ 50% RH				.9	2.3	7.2

ABSORPTION OF SEATS AND AUDIENCE

Values given are in Sabins per square foot of seating area or per unit

	125 Hz	250 Hz	500 Hz	1000 Hz	2000 Hz	4000 Hz
Audience, seated in upholstered seats, per sq ft of floor area	.60	.74	.88	.96	.93	.85
Unoccupied cloth-covered upholstered seats, per sq ft of floor area	.49	.66	.80	.88	.82	.70
Unoccupied leather-covered upholstered seats, per sq ft of floor area	.44	.54	.60	.62	.58	.50
Wooden Pews, occupied, per sq ft of floor area	.57	.61	.75	.86	.91	.86
Chairs, metal or wood seats, each, unoccupied	.15	.19	.22	.39	.38	.30

The **Noise Reduction Coefficient (NRC)** is an attempt to get a single number to quantify a material. It is the numerical average of the absorption coefficients in the 250, 500, 1000 and 2000 Hz bands.

$$NRC = (\alpha_{250} + \alpha_{500} + \alpha_{1000} + \alpha_{2000})/4$$

Equation 3

8.3 Experimental Determination Of Absorption

Absorption may be determined by experimental procedures, either:

- normal incidence coefficient α_n using an impedance tube (Figure 8.3)
- random incidence coefficient α_{sabine} using a reverberation chamber

A third theoretical quantity sometimes used in equations is the **statistical energy absorption coefficient α_{st}** . It is defined as:

$$\alpha_{st} = \frac{\text{Sound Energy absorbed by infinite surface in diffuse sound field}}{\text{Incident Sound Energy}} \quad \text{Equation 4}$$

This is an idealized quantity which cannot be measured directly.

8.4 Normal Incidence Coefficient α_N

The normal incidence absorption coefficient is the ratio of energy absorbed/energy incident, for a plane wave, normally incident on an absorptive surface. It is easy to determine using a "standing wave tube" (sometimes called an "impedance tube"). It uses a small sample (typically 4" diameter) and has limited validity and usefulness due to the small sample size and the difference between a true normal incidence condition, and the actual incidence conditions (nearly random) seen in most real installations. But it is still useful for comparison purposes. The diameter of the tube must be smaller than $\frac{1}{2}$ wavelength to insure plane wave sound propagation. A 4" tube is good up to about 3300 Hz. For higher frequencies, a smaller diameter tube is used.

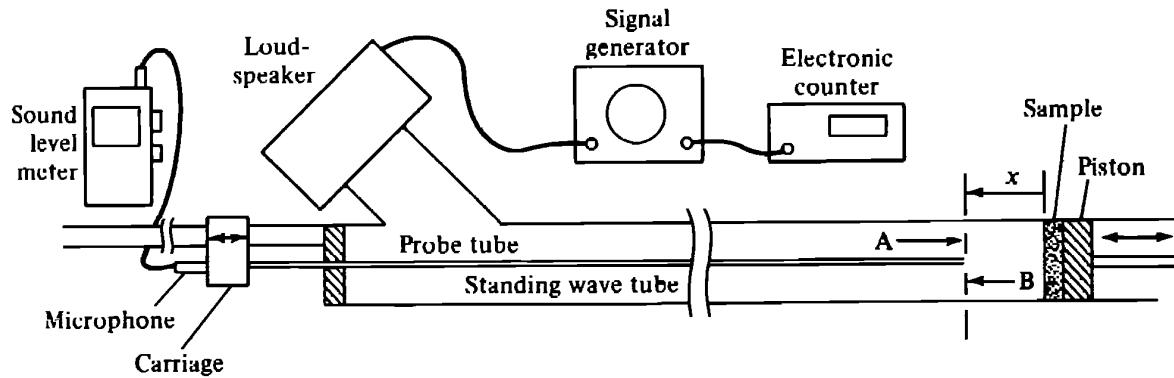


Figure 6. Impedance tube for measuring normal incidence absorption coefficient

We input a pure tone (or band of noise) using a loudspeaker. The incident wave from the speaker combines with the reflected wave from the end of the tube to form a standing wave. The depths of the minima are directly related to the absorption of the sample at the end of the tube. If the sample were perfectly reflective, total cancellation would occur $\frac{1}{4}$ wavelength from the end, and a pressure maximum would occur at $\frac{1}{2}$ wavelength. A totally absorptive sample (anechoic) would exhibit a uniform pressure over the entire tube length. So, the difference in the maximum and minimum pressures is an indication of the absorptive characteristics of the sample.

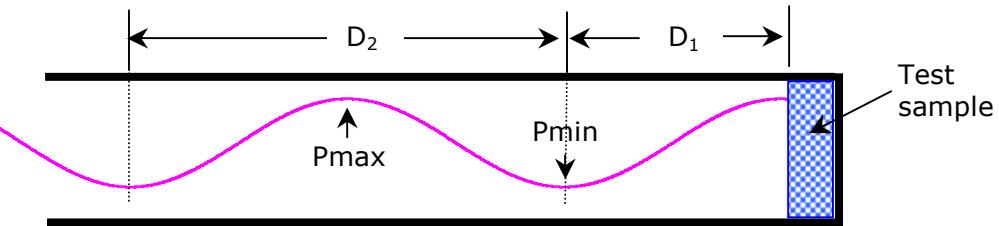


Figure 7. Interaction between incident and partially reflected waves result in a standing wave pattern in an impedance tube. D_1 is the distance from the sample to the first minimum. D_2 is the distance between the first and second minima (equal to $1/2$ wavelength)

We experimentally measure the maximum and minimum pressures inside the tube by sliding a microphone along the centerline, from which we can calculate the normal incidence absorption coefficient, α_n .

$$\alpha_n = \frac{4 \frac{P_{\max}}{P_{\min}}}{\left(1 + \frac{P_{\max}}{P_{\min}}\right)^2} \quad \text{Equation 5}$$

Additionally, if we measure the distance from the sample to the first minimum D_1 , and the distance between consecutive minima (or consecutive maxima) D_2 , the magnitude of the acoustic impedance can be calculated (ref. pg 57 L,G&E). A good check on the data is that D_2 should be equal to one half of a wavelength.

$$|Z| = \left| \frac{P}{u} \right| = \left(\frac{1 + 2R_0 \cos \theta + R_0^2}{1 - 2R_0 \cos \theta + R_0^2} \right)^{\frac{1}{2}} \rho c \quad R_0 = \frac{P_{\max} - P_{\min}}{P_{\max} + P_{\min}} \quad \theta = \frac{2\pi D_1}{D_2} \quad \text{Equation 6}$$

8.5 Sabine Absorption Coefficient α_{Sabine}

A patch of material is placed in a large, highly reverberant room having a diffuse field. α_{Sabine} is calculated from measurements of sound decay (reverberation time) in the room both with and without the material sample in place. It is a better approximation to real installations of absorptive materials, where the incidence angle can be anything.

(reference standards: ISO R354-1963, ASTM C423-84 & AS 1045-1971)

8.6 Room Averaged Coefficient $\bar{\alpha}$

Most real rooms have a variety of surfaces with different materials. The total effect of all these surfaces can be approximated by the average:

$$\bar{\alpha} = \frac{\sum_{i=1}^N \alpha_i S_i}{S} \quad \text{where: } \alpha_i = \text{absorption of the } i^{\text{th}} \text{ surface} \quad \text{Equation 7}$$

S_i = Area of the i^{th} surface S = Total surface area

N = Number of absorbing surfaces

Assuming a uniform intensity (a diffuse sound field) $I \bar{\alpha} S = I \sum \alpha_i S_i$
(the absorbed acoustical energy/unit time = the absorbed power)

If the distribution of α is highly uneven, a better approximation is:

$$\bar{\alpha} = \frac{S}{\frac{S_x}{\bar{\alpha}_x} + \frac{S_y}{\bar{\alpha}_y} + \frac{S_z}{\bar{\alpha}_z}} \quad \text{where } S_{x,y,z} = \text{area of x, y, z faces}$$

$$\bar{\alpha}_{x,y,z} = \text{average absorption of each face}$$

8.7 Sound Buildup In Rooms

If a sound source with power of W is suddenly turned on, acoustic energy flows into the room, with maximum intensity occurring near the source. Waves travel outward and eventually bounce off walls (with partial absorption) back into the room. After several reflections, the sound field approaches diffuseness (if α is low).

Energy builds up until an equilibrium is reached. At equilibrium, the total power input (W) is exactly balanced by the power absorbed by the walls. The power absorbed by the walls is determined by the incident sound intensity:

$$W_{input} = W_{absorbed} = \bar{I} \alpha S$$

Since intensity is proportional to the square of sound pressure, this gives the result that the sound pressure in the room (in the reverberant field) is proportional to the input power and inversely proportional to the amount of absorption present.

$$p^2 \propto \frac{W_{input}}{\alpha S}$$

Another way to think about it is the sound pressure (and intensity) in the room continue to build up until the power absorbed by the walls equals the input power. The higher the absorption, the lower the overall level which results.

A good analogy for this is a leaky water tank filled with a faucet (Figure 8). As the water level in the tank increases, more leaks out of the holes because the head (pressure) which forces water out the leaks is proportional to the water level. Eventually, the level will reach a steady height, where the inflow from the faucet is exactly equal to what leaks out. Now to complete the analogy to our acoustic problem, think of the water level as the sound pressure, the leaks are the sound absorption, and the flow from the faucet is the input sound power.

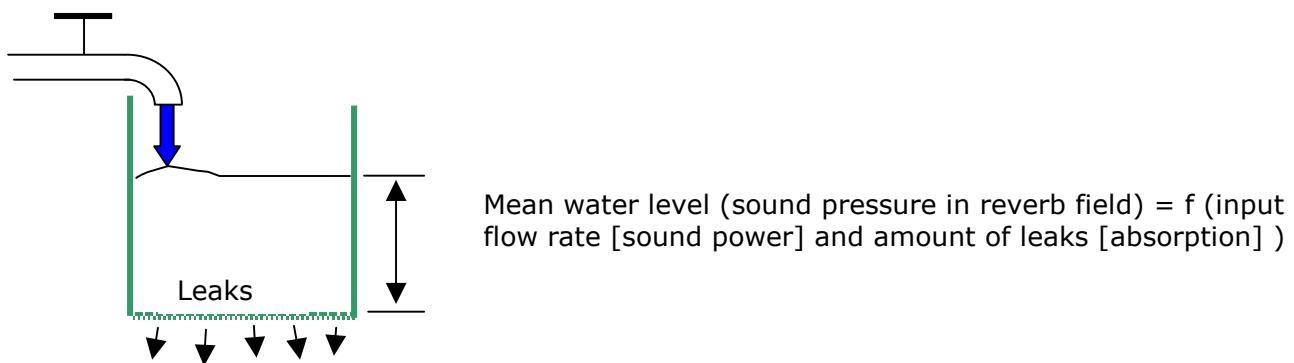


Figure 8. Leaky tank analogy for sound pressure buildup in an absorptive room

8.8 Sound Decay, Reverberation Time

If we now turn off our noise source, the sound level will decay linearly with time. Qualitatively, it's easy to understand that the more absorption a room has, the quicker the sound will decay. We can (and will) use this decay rate to experimentally measure the overall room absorption.

The time required for the sound level to decay 60 dB is called the **reverberation**

time, or T_{60} . It is often difficult (particularly at low frequencies) to put enough sound energy into a room to raise the level 60 dB over the background noise. The typical approach is to fit a straight line to the actual decay and extrapolate to 60 dB. Methods to excite the room include impulse sources such as popping balloons (ok for small rooms) or starter pistols; or a steady source - white or pink noise from amplified speakers.

Reverberation time is the single most important parameter for judging the acoustical properties of a room and its suitability for various uses. (Note, RC or NC criteria are measures of the background noise level of a room)

- High reverberation (long T_{60}) is desirable for music (concert halls 1.8 – 2.0 seconds)
- Low reverberation (short T_{60}) is desirable for speech intelligibility (such as in a classroom, 0.4 - 0.6 seconds)

The reverberation time at 512 or 1000 Hz is typically used as a single number to quantify the acoustic properties of a space. Recommended values for various applications are shown in Figure 9 and Table 2. An equation for calculating the "Optimum" Reverberation Time (according to Stephens and Bate 1950) is

$$T_{60} = K[0.0118 V^{1/3} + 0.1070] \quad \text{Equation 8}$$

V = volume in meters

K = 4 for speech, 5 for orchestras, 6 for choirs

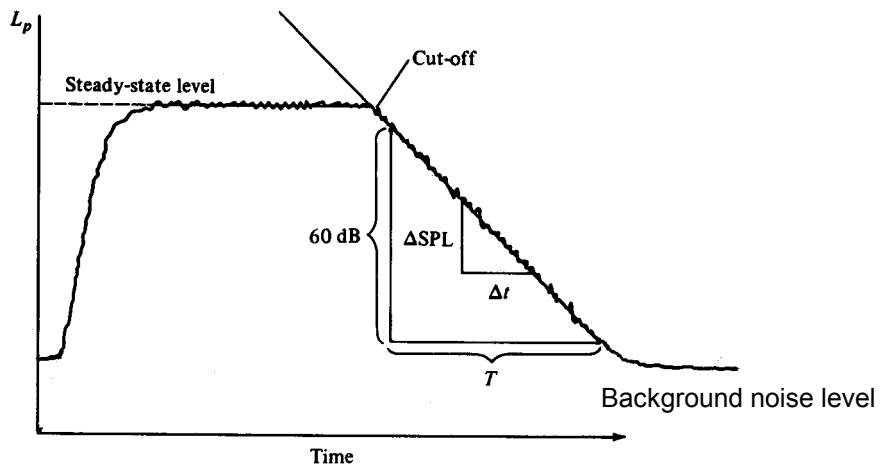


Figure 9. Typical decay of sound in a reverberation time test

example $V = 1000 \text{ m}^3$ for speech, $T_{60} = .9 \text{ sec}$

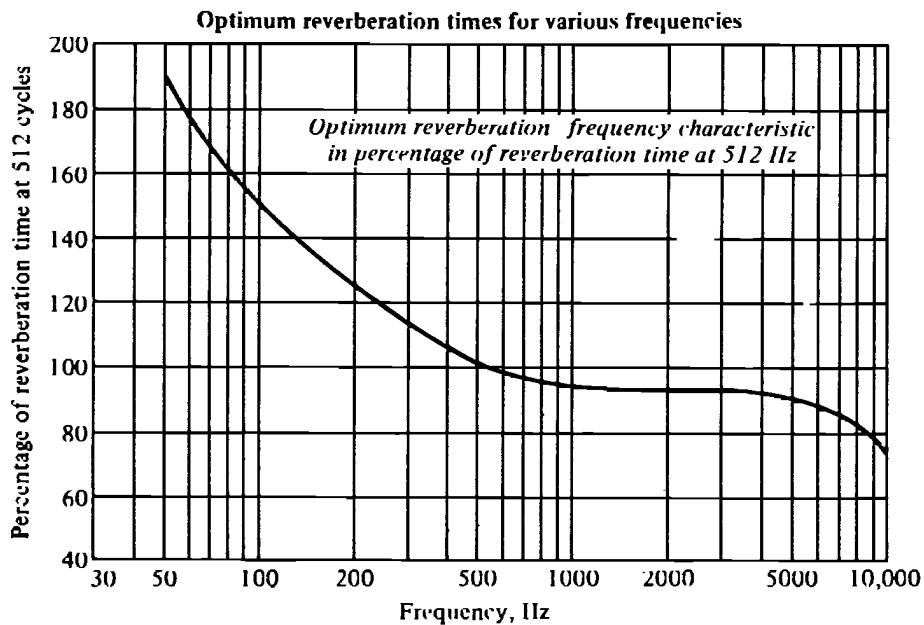
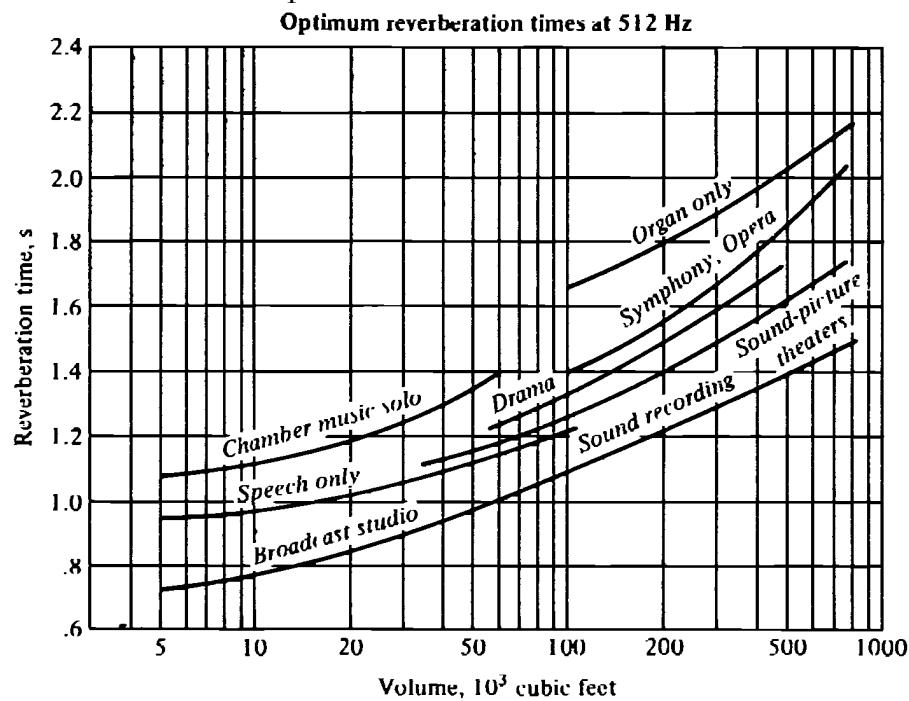


Figure 10. Recommended reverberation times for various uses
(reference Lord, Gatley and Evenson)

Table 2. Suitable reverberation times (seconds) for various rooms typically found in educational facilities. (ref. Classroom Acoustics, Acoustical Society of America, 2000)

Music Rehearsal	0.6 – 1.1
Auditoriums	1.0 – 1.5
Gymnasiums	1.2 – 1.6
Cafeterias	0.8 – 1.2

Classrooms	0.4 – 0.6
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8.9 Relating Reverberation Time to Room Dimensions and Materials

T_{60} is, to a first approximation, proportional to the total room absorption A and the room volume V .

$$T_{60} \propto \frac{V}{A}$$

The simplest relation, from empirical data, called the **Sabine equation** is:

$$T_{60} = .161 \frac{V}{\sum_{i=1}^n \alpha_i S_i}$$

for mks units, or

$$T_{60} = .049 \frac{V}{\alpha S} \quad \text{for English units (feet)} \quad \text{Equation 9}$$

This equation assumes a diffuse field, air at 24° C, and works well if the absorption coefficient of each of the surfaces, $\alpha_i < .20$

For larger absorptions, and a uniform distribution of absorption around the room, the **Eyring equation** can be used (mks units):

$$T_{60} = \frac{.161 V}{-S \ln(1 - \bar{\alpha})} \quad \text{Equation 10}$$

Air absorption is negligible for frequencies < 1000 Hz. However, if the room is very large, and high frequencies are of concern, air absorption cannot be neglected:

$$T_{60} = \frac{.161 V}{-\alpha S + 4mV} \quad (\text{mks}) \quad = \frac{.049V}{-\alpha S + 4mV} \quad (\text{English units}) \quad \text{Equation 11}$$

m = energy attenuation constant for air (see Figure 10).

Table 3. Reverberation time equations for various applications $T_{60} = .161 V/A$ (SI units)

Absorption, A	Application	Comments
$\sum S_i \alpha_i$	Live rooms, all values of $\alpha_i < .20$	Sabine Equation
$-S \ln(1 - \alpha)$	$\alpha' > .20$, Uniform distribution of absorption	Eyring Equation
$\sum S_i \ln(1 - \alpha_i)$	At least one value of $\alpha_i > .20$, Non-uniform distribution of absorption	Millington-Sette Equation
$A + 4mV$	Large rooms, air absorption not negligible, m = attenuation constant from Figure 10	

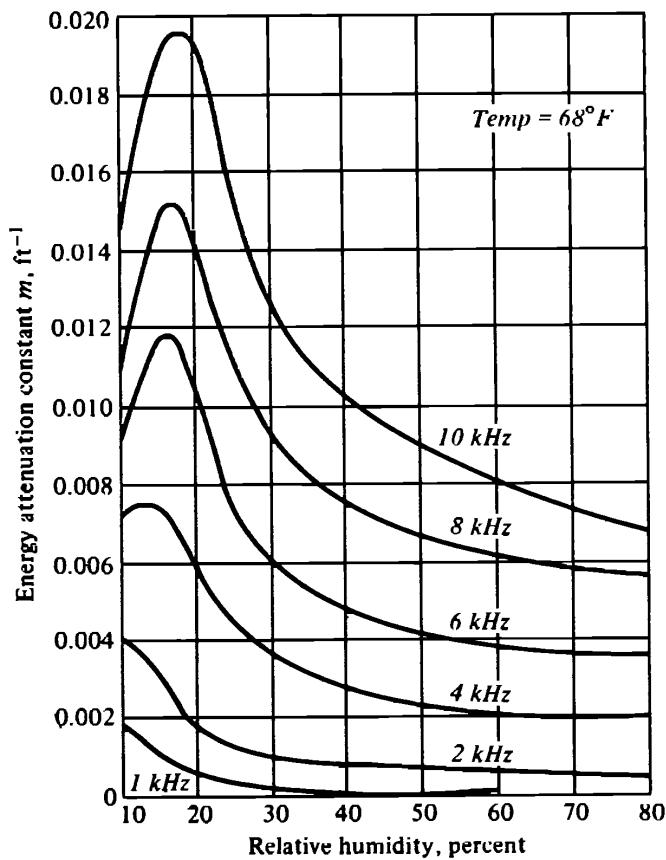


Figure 6.5 Attenuation constant for propagation of sound waves in air.
(From L. L. Beranek, Noise Control, McGraw-Hill, New York, 1960.)

Figure 11. Attenuation for propagation of sound in air

8.10 Steady State Sound Levels In Enclosures

In a direct field, we already know that the intensity varies with distance.

$$I_\theta = \frac{W}{4\pi r^2} Q_\theta \quad \text{and} \quad I = \frac{\langle p^2 \rangle}{\rho c}$$

where : Q_θ = directivity factor

where : $\langle p^2 \rangle$ = mean square sound pressure

In a reverberant field, the intensity is constant everywhere and is related to pressure by:

$$I_{rev} = \frac{1}{4} \frac{\langle p^2 \rangle}{\rho c} \quad \text{Equation 12}$$

Note that the intensity in a diffuse (reverberant) field is only $\frac{1}{4}$ that of a plane wave.

If we assume steady state conditions and a diffuse field, the amount of energy absorbed by the walls must equal the reverberant power supplied. The reverberant power is the sound power of the source minus the sound power absorbed in the first reflection, $W(1 - \overline{\alpha_{ST}})$. The absorbed power is $I_{rev}(S\overline{\alpha_{ST}})$.

The reverberant intensity is then:

$$I_{rev} = \frac{W(1 - \overline{\alpha_{ST}})}{S\overline{\alpha_{ST}}} = \frac{W}{R} \quad \text{Equation 13}$$

Where **R** is called the **room constant**, $R = \frac{S\overline{\alpha_{ST}}}{1 - \overline{\alpha_{ST}}} \quad \text{Equation 14}$

In most cases of low absorption, we typically simplify by assuming:

$$R \approx S\overline{\alpha_{ST}} \quad \text{and} \quad \overline{\alpha_{ST}} \approx \overline{\alpha_{SABINE}}$$

A real room is somewhere between a diffuse and a free field. Therefore the total pressure is the sum of the direct and reverberant fields.

$$\langle p^2 \rangle = \rho c I_\theta + 4\rho c I_{rev} = W\rho c \left[\frac{Q_\theta}{4\pi r^2} + \frac{4}{R} \right]$$

and in terms of levels:

$$L_P = L_W + 10 \log_{10} \left[\frac{Q_\theta}{4\pi r^2} + \frac{4}{R} \right] \quad \text{Equation 15}$$

↓ Direct Field ↓ Reverberant Field

The quantity $L_P - L_W$ is plotted in Figure 12. In the reverberant field, the sound pressure level is independent of location. Note that in a highly reflective room (low R), the reverberant field is very large, and begins very close to the source.

The change in a room's SPL due to changing its absorption is called the **Noise Reduction, NR**:

$$NR = L_{P1} - L_{P2} = 10 \log(R_2 / R_1) = 10 \log \frac{S_2 \overline{\alpha_2}}{S_1 \overline{\alpha_1}} \quad \text{Equation 16}$$

In order to get a decrease of 6 dB, the room absorption must be increased by a factor of 4. (that's a lot !)

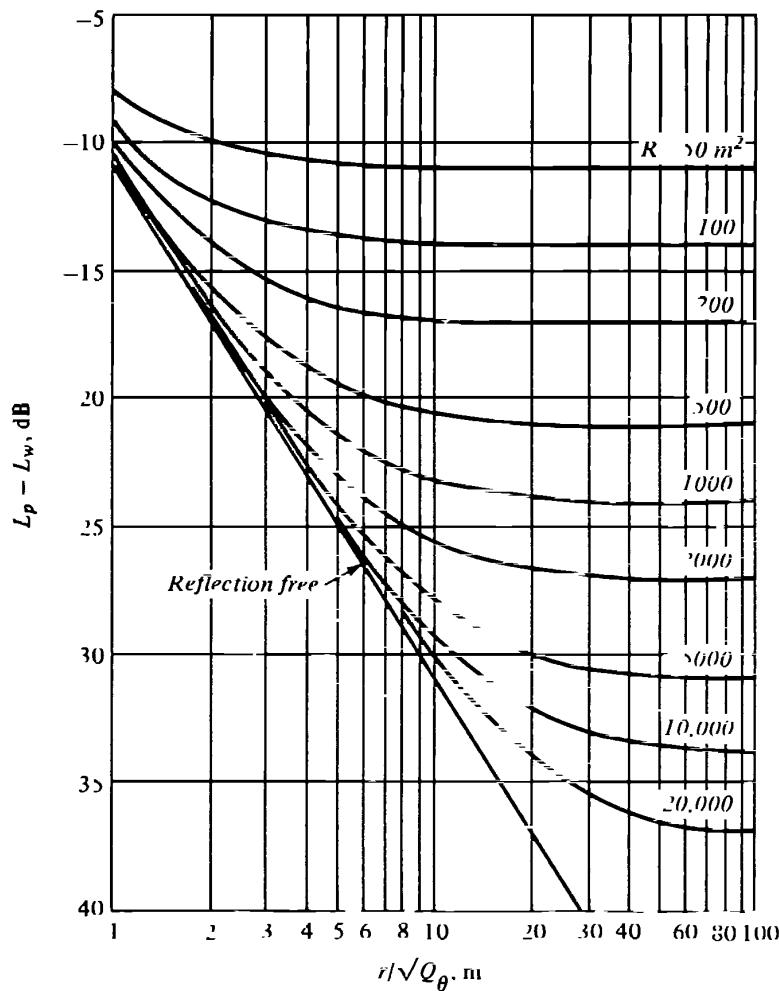


Figure 12. Difference between the sound pressure level and the sound power level in a room as a function of the room constant R_T , the distance from the source r and the directivity factor Q_θ

Other Room Theories:

The preceding relation is based on diffuse field theory. Schultz [ASHRAE Transactions 1983, 91(1), pp 124-153] proposed an empirical formula based on his studies of domestic rooms and closed offices. He found that levels did not ever reach a constant level with distance from the source, as predicted by the diffuse field model. He found that the curves always had a slope of about -3dB/doubling of distance. Notice that there is no term for room absorption!

$$L_p = L_w - 10 \log_{10} r - 5 \log_{10} V - 3 \log_{10} f + 12 \quad \text{Equation 17}$$

where: r = distance (m)

V = room volume (m^3)

f = frequency (Hz)

8.11 Effect of Mounting

The more area an absorbing material presents to incident sound, the more energy is absorbed. In addition, it is possible to make a material more effective at low frequencies by mounting it with an air space between it and the adjacent wall or ceiling (see Figure 13 and Table 4).

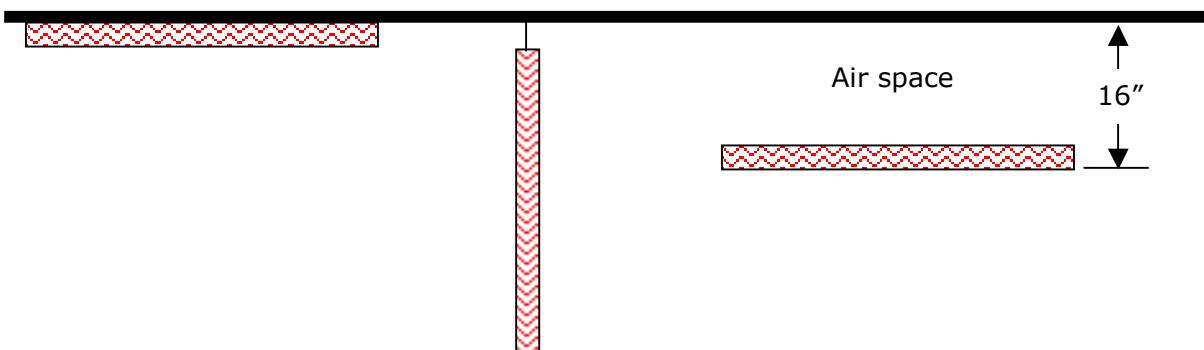


Figure 13. Methods of mounting absorbing panels on walls or ceilings:
 a) hard mounted b) hanging baffle c) air space behind panel

Table 4. Effect of mounting on a 24" x 48" x 1.5" thick fiberglass panel on total absorption (absorption in Sabins) (data from NIOSH Compendium of Noise Control materials)

Mounting Configuration	Frequency - Hz					
	125	250	500	1000	2000	4000
Hanging baffle	4.3	6.6	9.8	13.3	13.6	10.8
Hard mounted on rigid wall (#4 mount)	1.5	3.5	6.2	7.4	6.5	6.2
16" air space (#7 mount)	7.2	6.4	6.0	7.2	6.2	3.6

8.13 Standing Waves

Room modes

Placement of sound sources and absorbing material

Modal density

8.12 Anechoic Rooms

Effectiveness of wedges

8.12 Reverberation Rooms

8.14 Good and Bad Reflections

bad - flutter echo

good – early reflections, reflectors in front of classrooms, orchestra shells

effect of surface roughness on reflection:

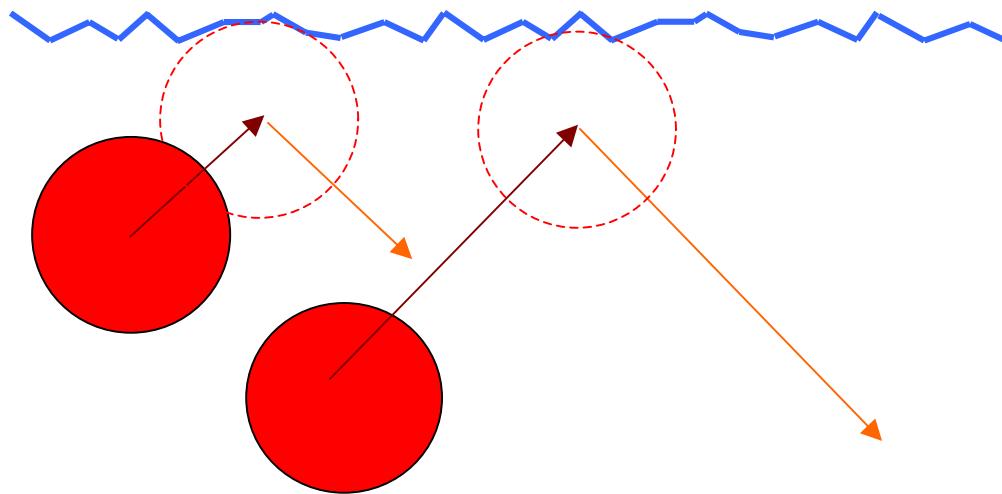


Figure 15. Specular reflection – occurs when the surface roughness length is smaller than the acoustic wavelength (represented by the diameter of the balls)

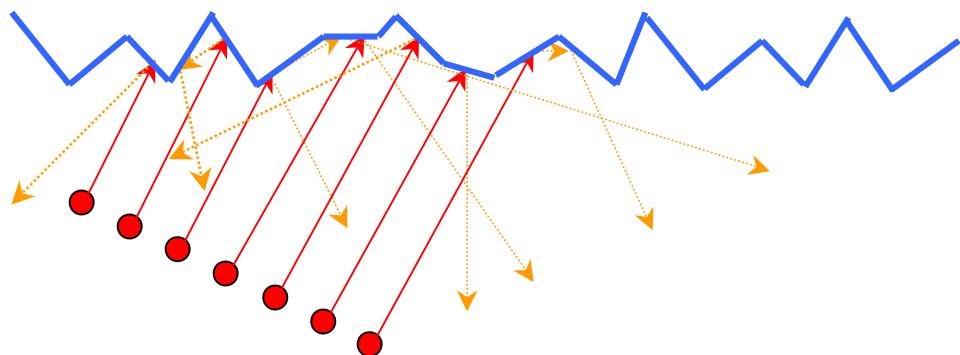


Figure 16. Diffuse reflection – occurs when the surface roughness length is larger than the acoustic wavelength (represented by the diameter of the balls)

8.15 Summary

Adding absorption is only justifiable if the *reverberant* field is dominant. Absorption on walls or ceilings will have little or no effect in the direct field, i.e. in the immediate vicinity of a noise source.

Design guidelines:

1. To have the greatest effect on total absorption (and the reverberation time), add absorption to the **least** absorptive areas first.
2. Distribute absorption around the room as much as possible to minimize local effects.
3. Avoid having two parallel walls that are both highly reflective. This can cause a flutter echo.
4. Low frequency absorption (< 250 Hz) is difficult to achieve with porous materials of reasonable thickness. To be effective at low frequency, porous materials must be thick,
Material thickness $\geq 1/4 \lambda$ for anechoic ($\alpha \approx 1.0$)
5. Low frequency absorption of porous materials can be increased by mounting them with an airspace behind them.
6. Design the room with non-parallel walls wherever possible to break up standing waves and flutter echo.
7. Absorption or a diffusing element on the back wall of a room (the wall directly opposite to the sound source or speaking person) is highly desirable
8. Mount absorbing panels so as to maximize the area exposed to incident sound

8.16 References

- 1) Compendium of Materials for Noise Control, NIOSH, 1975, HEW Publication No. 75-165.
- 2) Sonic and Vibration Environments for Ground Facilities – A Design Manual, NASA , NAS8-11217.
- 3) Classroom Acoustics, Acoustical Society of America, Architectural acoustics technical committee, August 2000.