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PRESENTS

Small Room Acoustics – The Hard Case

by Magne Skålevik

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INTRODUCTION

Below the room acoustical high-frequency region limit, the so-called Schroeder Frequency, the frequency response of the room is dominated by room modes, which is determined by room geometry. Room modes are reported by many authors to cause "boomy" sound and other coloration effects, which can be disturbing in rooms where speech or music is an important part of normal use. Ways to avoid these small room acoustics problem have been suggested, including recommendations for room ratios for cubiods, careful choise of building material, and surface treatment involving absorbers and diffusers. Still, there are many important cases where suggested means are not applicable. This paper is focusing on the hard wall and rigid geometry cases, aiming to define criterias for proper treatment in the hard case: Rectangular room with hard walls and floor, and hopefully allowance for an absorbing ceiling.

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Small room acoustics – the hard case

Magne Skålevik
AKUTEK www.akutek.info
Senior acoustical consultant, Brekke & Strand akustikk, Oslo, Norway

Summary
To avoid disturbing coloration in rooms where speech, music, listening or recording is part of normal use, damping of modes is critical, regardless of their spacing. Modes cannot in general be treated with ordinary broadband reverb measures. Results from the virtual test room described in this paper show that in cuboids with hard plain walls and floor, even a perfect ceiling is not able to control modes travelling in the horizontal plane. The Modal Scattering Coefficient and a way to predict modal reverberation time is suggested. Hard diffusers can provide relatively short modal reverberation times, but necessary treatment may have to cover 50% of wall surface. This paper deals with the many cases with hard walls and no freedom to choose room ratios.

1. Introduction

Below the room acoustical high-frequency region limit, the so-called Schroeder Frequency[1], the frequency response of the room is dominated by room modes, which is determined by room geometry. Room modes are reported by many authors to cause ”boomy” sound and other coloration effects, which can be disturbing in rooms where speech or music is an important part of normal use. Ways to avoid these small room acoustics problem have been suggested, including recommendations for room ratios for cuboids, careful choise of building material, and surface treatment involving absorbers and diffusers. Still, there are many important cases where suggested means are not applicable. This paper is focusing on the hard wall and rigid geometry cases, aiming to define criterias for proper treatment in the hard case: Rectangular room with hard walls and floor, and hopefully allowance for an absorbing ceiling.

2. Previous work

Since Bolt (1946) [1], in the aim for evenly spaced modes, came up with a method for determining preferable room ratios, researchers have continued to search for the optimum room dimensions based on various criteria. A review of papers in this special field of small room acoustics was given in a paper by Cox and D'Antonio[16] in 2004, together with a new approach, aiming for flattest possible frequency response. Among other authors contributing to small room acoustics from the 1940's until present are Volkman [1], Boner [2], Sommerville and Ward [4], Gilford[6], Louden[8], Bonello [11], Walker [12][13], Neuwland [10] and Weber. Splaying one or two walls may improve diffusion, but does not eliminate modal problems[5]. Geometries deviating from rectangular sections (slanted walls, etc) does not make coloration dissapear, only harder to predict (Gilford 1972) [9].

3. The hard case

In many cases the acoustic consultant gets the task of providing or improving acoustics in a room for speech or music at a time or under conditions when the geometry is set, and materials cannot be freely chosen. The reasons for this lack of freedom can be many, and is not necessarily a result of poor planning. Practical reasons, floor area demand, robust surface requirements, and last but not least financial constraints, are very common. Not to mention gravity, making anything but vertical walls difficult to build, and a horizontal floor a human right.
In particular, the following case is to be investigated.

A cuboid room with four hard vertical walls, a hard floor, and place for absorbents somwhere in the ceiling only. A free choice of room ratio in the context of previous work is out of the question. Only hard elements may be added to

wall and ceiling, preferably as high as possible in order to save floor space.

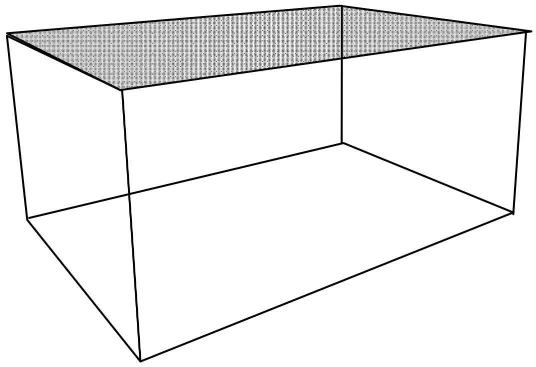

Figure 1 The hard case. Only ceiling is absorbent.

Relevance of the hard case is evaluated and argued for in Table 1 below.
We require that the problem solving can be supported by simple computation methods, however to be verified by measurements, FEM and/or BEM. In short, we require the hard case to be solved easily.

Room category	Relevance of Hard Case
Wherever speech is part of normal use	High. Large percentage of rooms. Coloration annoying, besides discriminating people with difficulies in hearing. Increasingly relevant in light of Universal Design and minimalistic trends in design.
Music rehearsal Recording Critical listening	High. Though special rooms in generel gets higher priority, thus avoiding the Hard Case, the remaining cases are all the more important to solve, due to high demands for acoustical quality.

Table 1 Evaluation of relevance of the hard case

4. Acoustical requirements

As a starting point, we choose to define some criteria for the tonal response of the room. Coloration and boominess shall be controlled, in particular the excess response to complex tones with harmonic (line) spectra typical from voices, strings and wind instruments, etc..

1. Mode decays shall obey the general reverberation time limits for the actual frequency region (e.g. octave band)
2. Harmonic series of room modes shall be eliminated, or at least supressed to prevent detection of pitch, i.e. supression of pitch response

We would like to establish an arrangement to predict the effect of such hard diffusing elements and we therefor try to design a ”virtual test facility”, described below.

5. Exploring axial modes

In this chapter we seek deeper insight by exploring axial modes in a virtual, loss free, test room.
Mode frequencies are given by (1),

$$f=\frac{c}{2}\sqrt{\left(\frac{n_x}{L_x}\right)^2+\left(\frac{n_y}{L_y}\right)^2+\left(\frac{n_z}{L_z}\right)^2} \qquad (1)$$

where n_x, n_y and n_z are integers and L_x, L_y and L_z are the room dimensions in the x, y and z direction.
From theory, the half-power (3dB) bandwith B of modes depends only on reverberation time

$$B=\frac{\log_e 10^6}{2\pi T}=\frac{2.2}{T} \qquad (2)$$

while average mode spacing Δf depends on room volume and frequency

$$\Delta f=\frac{c^3}{4\pi V f^2} \qquad (3)$$

Average mode spacing in the Modal Region should not be confused with average spacing between maxima 4/T in the Schroeder Region[17].
Modal peaks at frequency f can be described by the Q-factor $Q=f/B$.
Empirically, we have results from Sommerville and Ward [4] (1951), that hard rectangular elements on walls will act sound-diffusing when they are thicker than 1/7 of a wavelength, $\Delta x > \lambda/7$.
In practice this limit is, not unexpectedly, equivalent to

$$k\Delta x > 1 \qquad (4)$$

The virtual test room
In a virtual cuboid room with hard surfaces and dimensions L_x, L_y and L_z in x, y and z- directions respectively, modal frequencies can be calculated from (1). We assume no energy losses in air or boundaries. See Figure 2.

Now let the distances L_y and L_z approch infinity. As a result, (1) will simplify to

$$f=0.5\cdot n_x\cdot c/x \quad (5)$$

or the common approximation

$$f=170\cdot n_x/x, \quad (6)$$

modes with wave components travelling perpendicular to the x-direction will vanish, and the *Q*-factor for a mode at frequency will approach infinity. A point source inside the room will have image sources spaced on average L_x apart along a line through the source and parallell to the x-axis.

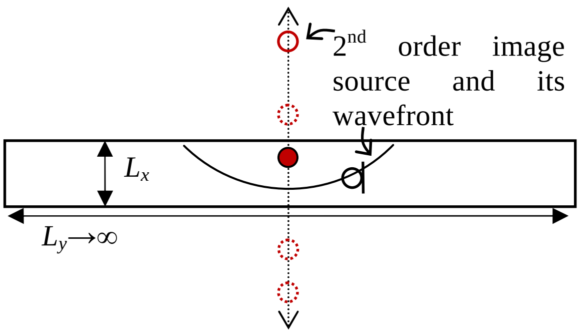

Figure 2 Axial mode study. x-y-plan of virtual cuboid, hard, test room with L_y and L_z approaching zero, containing source (red dot) and a reciever collecting wavefronts from axial image sources.

After sufficient driving-time *t*, image sources in both directions far from the room will send approximately plane waves in opposite directions, equally strong, interfering in the room, building up the so-called standing wave, increasing in amplitude with time, but only inside an ever narrower frequency interval $\Delta f=K/t$, where *K* is a constant. After a time *t*, the amplitude will not increase outside this interval.

Keep in mind that the behaviour of ideal modes, or standing waves, can be described simply by two equally strong plane waves travelling in opposite directions, interfering at a point of interest. So planeness and restricted curvature is essensial. Curvature of the wave at time *t* can be described by measuring the maximum deviation

$$\xi=y^2/(ct) \quad (7)$$

from an ideal plane wavefront disc perpendicular to propagation and width of diameter $2y$ and area πy^2. Planeness can be expressed by the area

$$S_{plane,disc} = \pi y^2 = \pi\, ct\cdot\xi \quad (8)$$

of the largest plane disc from which the wavefront deviates less than ξ , or equivalent, the largest plane square contained by the disc

$$S_{plane,square} = 2ct\cdot\xi \quad (9)$$

Related to curvature is the ratio of transverse component to the axial component in the wavefront having travelled the distance *ct* from its source. In our test room, within the distance *y* from the x-axis, the outward-travelling vector-component must be less than y/ct times the axial component:

$$v_{trans}/v_{ax} < y/ct = \xi/y \quad (10)$$

In a modal study involving diffusing elements of thickness Δx we require wavefronts that are plane compared to the surfaces, and elements involved down to, say 40Hz. We need to combine $k\Delta x > 1$ with $\xi \ll \Delta x$, leading to $\Delta x > 1.4m$ and $\xi <0.14$m if we require 1:10 resolution. Thus, for waves having travelled for more than 1.0 second, the required planeness would by (9) apply to square surfaces of approximately 90 m^2, which is more than large enough for small room studies. Valid area is inside radius of y=6.8m where v_{trans}/v_{ax} <0.020. In terms of intensity this means that there is a non-zero energy dissipation from the test area to be kept in mind. We shall return to this below.

We can conclude this far that later than one second after the source is turned off, the interfering waves would be sufficiently plane for the accuracy required. And this is exactly how we shall proceed – we shall turn the source off and see what happens.

We have established a test facility between two parallell hard surfaces spaced by the distance L_x, with a point source placed centrally in an area of 90 m^2 satisfying our precision demands given above. Note that in this room, no waves travel back towards the source.

For simplicity we shall turn off the source at the time t=0. After 1.0 seconds, the direct (non-modal) sound components will have travelled a distance of 340m away from the source, so inside this radius, all there is left is a standing wave driven by the image sources positioned along the x-axis, as described above. Now return to the possible dissipation of modal energy related to non-ideal plane waves and their transversal component:

If we interpret the ratio v_{trans}/v_{ax} to be the ratio of speed of tranverse energy-propagation to speed of axial energy-propagation, we have

$$c_{trans}/c = y/ct \Rightarrow c_{trans} = y/t\,. \quad (11)$$

And since intensity *I* equals energy density ε multiplied with speed of propagation, we have an expression for the dissipation of modal energy out of the test volume:

$$I < \varepsilon\cdot y/t \quad (12)$$

Since the valid radius of the test area is *y* , the test volume has the form of a sylinder with radius *y* , and height L_x, and the volume can be calculated by

$$V = \pi y^2 \cdot L_x \quad (13)$$

Letting ε be the energy density after the necessary elapse of time after the source stopped, the total energy of the valid test volume is

$$E_0 = \varepsilon V = \varepsilon \cdot \pi y^2 \cdot L_x \quad (14)$$

Around the test volume is an imaginary sylindrical surface $S_{syl} = 2\pi y \cdot L_x$ through which the intensity in (11) causes modal energy to dissipate from the volume by a rate per second of

$$P_d < I \cdot S_{syl} = \varepsilon / t \cdot 2\pi y^2 \cdot L_x \quad (15)$$

Whenever dissipation from a system relates to the momentary energy contained by the system, exponential decay with level changing with time proportional to $B \cdot t$ can be expected, where the half-power bandwith B is proportional to the ratio of dissipated energy to the stored energy,

$$2\pi B = P_d / E_0 < 2/t. \quad (16)$$

Since this leads to $B \cdot t = 1/\pi$ being a constant and thus the level being unchanged with time, we can draw the somewhat surprising conclusion:

While energy is dissipated out of the test volume via large openings, the stored energy of the mode remains unchanged, given lossless boundaries and lossless air.

In order to explain this apparent contradiction we must remember that a mode is an ideal phenomenon. Being mono-frequency involves infinite duration and perfect plane waves, modes can only be approached as time goes to infinity. This is also the reason for the ever-narrowing of the bandwidth $\Delta f = K/t$ with increasing duration t during the build-up phase described above. Combined with the result in (15), $K = 1/\pi$, since the bandwith in question is no other than the half-power bandwidth

$$\Delta f = B = 1/(\pi t) \,. \quad (17)$$

So, in the test volume described above, after a time t after the turning off of the source, the modal energy is conserved, while energi from wave components of frequencies outside the frequency interval centered at the mode, $\Delta f = K/t$, will dissipate. In the spatial domain these are the curved components of the wave, litterally bending their way out of the test volume. Inside Δf the energy is conserved as long as boundaries and air is lossless.

One interesting consequence of the result above, is that the imaginary sylinder surface introduced in (14), defining the test volume, could be replaced by a perfect absorbing surface, since no sound propagates inward from there anyway. After the sufficient time (e.g. one second in the example above) after source-off, the energy dissipated (14) from the volume V would be absorbed by the perfect absorbent equivalently to the absorption from the open window represented by the imaginary sylinder surface.

Returning to the Hard Case, the objective of this paper, one cannot from the results above expect the "perfect component" of a mode to be damped by a plane ceiling, even with perfect absorption coefficient, as long as the waves involved travel tangentially to the ceiling. And this is exactly the common cuboid hard case. Of course, the ceiling will absorb sound, but only the non-perfect components outside the frequency band, $1/(\pi t)$ wide, centered at the mode frequency.

6. Introducing hard diffusers to hard walls

In the virtual test room described above, we shall mount the hard rectangular elements, requirement given in (4), evenly distributed over one of the parallel wall pairs separated by the distance and L_x. Let the hard diffusing elements have rectangular face with surface area S_d and height Δx.

From diffraction theory we can predict the diffracted pressure from a hard wall segment of size $2S_d$, containing the element of area S_d. If we refer to the part of the wall segment outside the element as the *base* an let it's area be $2S_d - S_d = S_d$, the pressure components from base and element face will be in-and-out of phase for wave numbers k such that $k\Delta x > 1$, resulting in an effect similar to the comb-filter from two coherent signals of same strength (the same strength of the two diffraction components here is due to the equal size of the two reflecting surfaces. At frequencies (values of k) where they are in-phase, the element would make no difference at all. On the other hand, at frequencies where they are out-of phase, the pressure would be zero. As a result, the combined reflection in axial direction from element and base will on average be reduced to half the power reflected from the same segment without the diffusing element. However, the power reflection spectrum from the segment would fluctuate sinusodially between 0 and 1.

So, in practice, the diffusing element of the wall is made up by the face and the base, covering the wall area of size $2S_d$. This becomes obvious if one tries to cover the whole wall completely with diffusing elements: Instead of obtaining maximum diffusion, one would have established a new hard plane wall. The face-and-base element has an effect on the power reflection of the plane wave that may be termed the Modal Scattering Coefficient s_m of the diffusing wall segment with surface area $2S_d$. Consistent with related coefficients we define s_m as the fraction of the incident modal power that is neither absorbed, nor reflected back into the mode.

$$s_m = 1 - r_m - \alpha_m \quad (18)$$

In general s_m is a function of frequency, but a convenient single-number value would be the frequency-averaged power ratio. In our test room, α_m=0, so $s_m = 1 - r_m$, and in the case of the face-and-base diffusor above, its single number value is

$$s_m = 0.5 \quad (19)$$

Note that in this modal context, reflection r_m is strictly the fraction of incident power reflected in the direction of the modal plane wave propagation. With the normal reflections occuring for standing waves between paralle walls, the modal reflection is equal to the specular reflection, but this equality is not valid in general.

Now, what happens to the scattered energy after it has been forced out of the mode by the diffuser? Well, the only alternative to conservation in the mode is to be dissipated through the open (virtual sylinder) window, or equivalently, be absorbed by the perfect absorber boundary closing the gap between the parallell hard walls. Any wave component travelling in non-axial direction will sooner or later be dissipated through the window or absorber in our test room.

As a result, scattered energy can be considered equivalent to absorbed energy, which is very convenient, since then one can utilize theory and tools for dissipation from a closed volume developed by Sabine, Eyring and many others.

Using Sabine's Formula, $T = 0.16 \cdot V/A$, where A is the absorption area $A = \Sigma\alpha \cdot S$, applying the equivalence between absorbed energy and scattered energy, valid in our virtual test facility, we replace the absorption coefficient α with the modal scattering coefficient s_m. Absorption area A in the formula can thus be replaced by the equivalent Modal Scattering area $S_{s,m} = \Sigma s_m \cdot S$. Now let $S_{s,m} = \underline{s}_m \cdot 2S$ be the scattering area provided when s_m is the average modal scattering coefficient over the total wall area $2S$ of our modal test room. Since the volume equals $V = S \cdot L_x$, the surface factor cancels out of the formula and $V/S_{s,m} = L_x/2\underline{s}_m$.

Finally, the modal decay can then be predicted by the reverberation time of the axial mode,

$$T_m = 0.16 \cdot L_x/2\underline{s}_m \, . \quad (20)$$

Example: Let the parallell wall spacing be L_x=4.0m. Further, place the face-and-base rectangular diffusors with scattering coefficient equal to 0.5 described above, covering 40% of the surface one of the two walls. Then the average modal scattering coefficients over the total wall area in our test room is $\underline{s}_m = 0.5 \cdot 0.4 \cdot S/2S = 0.1$, which implies $2\underline{s}_m = 0.2$. Now inserting in (19),

$$T_m = 0.16 \cdot 4.0/0.2 = 3.2 \text{ s} \quad (21)$$

To acheive the maximum effect from the modal scattering, one can cover both walls with the face-and-bace diffusers to obtain $\underline{s}_m$=0.5 and $2\underline{s}_m$ =1.0. Subsequently the reverberation time of the axial modes become

$$T_m = 0.16 \cdot 4.0/1.0 = 0.64 \text{ s} \quad (22)$$

This demonstrates that axial modes can be controlled in the Hard Case. However, it requires a substantial amount of wall surface being geometrically treated to obtain sufficient modal scattering.

For more detailed description of hard diffusers, prediction and measurement of their properties, readers are refered to the works of Cox and D'Antonio [20].

7. Large geometry diffusers

Note that the fact that the faces of the diffusing elements cover 50% of the wall area, does not necessarily mean many small elements. In particular, there could be one rectangular box placed on the upper half of each wall, leaving maximum floor space. In this case, a modal reverberation time of 0.64s can be both achievable and acoustically satisfactory.

However, the depth of elements should be considered in relation to the need for frequency range. Most male voice colorations fall in the 100-175 region (Gilford 1959)[6], but occasionally down to 80Hz. Consequently, most rooms for speech needs to control modes down to 100Hz. Rooms for music rehearsal, recording or listening would need to control modes down to 40-50Hz, depending on the particular sources in use.

In this matter the interpretation of the criterion in (4) is critical. As long as the diffusing boxes are rather small and randomly distributed, they would need to satisfy (4) individually. On the other hand, with one large box on the upper half of each wall, oposite of each other, the sum of their heights would determine their lower frequency limit.

8. Conclusion

Modes can be annoying regardless of their spacing. Thus, in rooms where speech, music, listening or recording is part of normal use, mode control is critical. Modes cannot in general be treated with ordinary broadband reverb measures. Results from the virtual test room described above shows that in a cuboid room with hard plain walls, even a perfect ceiling is not able to control modes travelling in the horizontal plane. The Modal Scattering Coefficient and a way to predict modal reverberation time is suggested. Hard diffusers can provide relatively short modal reverberation times, but neccesary treatment may have to cover 50% of wall surface.

9. Further work

In order to explore further the possibilities with large, hard modal diffusers introduced above, it is suggested to study the derivative of f in (1) with respect to varying spacing $x=L_x$ between walls. From $df/dx = -f/x$, we can deduce the relationship between resolution in geometry, Δx, and resolution i frequency $\Delta f = -(\Delta x/x)\cdot f$. If this is interpreted as the bandwidth Δf of the mode frequency f as a function of the building tolerance Δx of the parallell walls with spacing x, a number of interesting applications would follow. Modal reverberation time T_m would be limited to $T_m < 2.2/\Delta f = x/(\Delta x\cdot f)$, slanting of wall could be described by Δx, etc. Further, we ask if the power distribution inside Δf simply is a direct translation of the geometrical distribution inside Δx?
These speculations will be pursued, verified or rejected in further work.
Another task is to explore the significance of pitch detection from complex tones (harmonic spectrum) and how different rooms respond to such tones.
A remaining problem is the full explanation of the acoustical problems in the Modal Region (term defined in [17]). Coloration is repetedly blamed on high peaks, low dips and too wide mode spacing. However, these features are common in the high-frequency region without causing the same trouble.

An important related issue is to determine the frequency range of Modal Region [19][17].

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