ACOUSTICAL USES FOR PERFORATED METALS:

Principles and Applications

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INDUSTRIAL PERFORATORS ASSOCIATION

ACOUSTICAL USES FOR PERFORATED METALS

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ACOUSTICAL USES FOR PERFORATED METALS:

PART ONE: THE PRINCIPLES

I. Introduction

Of all the markets for perforated metals, acoustical applications have seen the most dramatic growth in the last few decades. There is every reason to expect a further surge of growth in this area in the near future, to match that of the general economy.

Much of the new growth in the acoustical market will come from original equipment manufacturers (OEM) and from architectural firms. OEM's will find that they must reduce the noise of their products to meet consumer demands, and the architects (including highway, airport, and rapid transit designers) are already designing noise control into their projects; these applications will proliferate as commercial building and government construction pick up.

In order to take full advantage of this potential market development, it is important to ensure that the designers of noise control applications give full consideration to the use of perforated materials, and to present a convincing argument that perforated materials are often the best alternative in noise control programs.

The best way of doing this is to present up-to-date, factual information on the acoustical applications of perforated materials and to illustrate these uses with enough practical examples to help specifiers gain a sense of confidence in recommending the right material for the right application, without feeling intimidated by the technical aspects of the design.

It is the purpose of this booklet to provide the necessary technical information in an easy-to-use style, and to provide helpful hints in the choice of perforated metals, so that professionals can recommend these materials to their clients with pride and confidence.

A. How Perforated Metals Are Used in Acoustical Applications

There are three principal acoustical applications for perforated metals:

Figure 1. As an example of the first application, the curved surfaces above the stage of the Orpheum Theatre, Vancouver, B.C., are made of finely perforated metal sheet, not of plaster as they appear. The perforations allow the sound to pass through and to reflect back into the hall at desired locations, from specially designed surfaces behind the perforated metal.

1. *As a Facing for Something Else:*

Here the perforated metal is used as a protective or decorative covering for some special acoustical material; that material may be designed either to absorb sound or to reflect or scatter sound in a special way. It is this special material that does the actual acoustical work, so the purpose of the perforated metal in such applications is to "disappear" acoustically: that is, it must be so trans parent that the sound waves can pass right through it to encounter the acoustical treatment that lies behind.

Our design goal in this case is to choose the perforated metal for greatest sound transparency, for sounds of all frequencies.

2. *In Tuned Resonant Sound Absorbers:*

Sometimes, however, we may wish to absorb sound very selectively, only in a certain band of frequencies but not at frequencies lying above and below that band. For this purpose we design a so-called Resonant Sound Absorber. Here, the perforated metal, instead of disappearing, takes an active part in tuning the aborber, that is, in determining which frequencies of sound are absorbed.

3. *As Airflow Diffusers:*

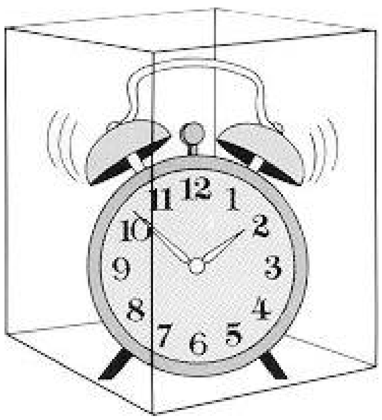
In the acoustical treatment of certain specialized aerodynamic test facilities, such as wind tunnels, perforated metals are often used to break up the turbulence in airflows.

This last application is both highly specialized and highly technical. Moreover, it does not represent a significant portion of the market; therefore, the rest of this book will be concerned entirely with the first two applications.

The main text of this book is intended for readers with no special technical background. It is divided into two parts. The first part deals with the principles of noise control treatments using perforated metals; the second part deals with typical applications. Readers who want more technical detail will find it in the Appendices.

Appendix D also includes worksheets that may be photocopied, filled out and included in the job files for individual projects.

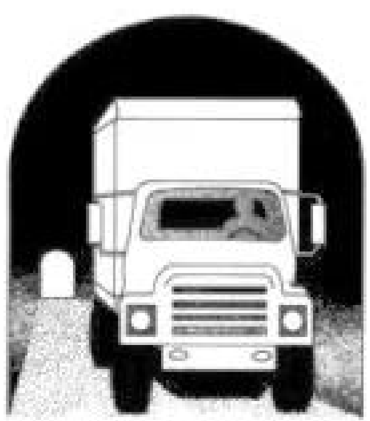
B. Noise Control With Sound Absorptive Treatments Using Perforated Metals



Noise control measures are often applied in order to quiet noisy equipment. We treat various household and office appliances to make them acceptable to the user because excessive noise is annoying; we treat heavy industrial equipment so that it will comply with current OSHA regulations that limit the noise exposure of workers so as to protect them from hearing damage.

In 1970, the Occupational Safety and Health Act (OSHA) set limits on the levels of noise to which workers may be exposed in their work environments. This regulation requires that industries monitor the noise in all worker locations, and, where this noise exceeds the permissible limits, it must be abated by any feasible noise control measures, or by administrative methods such as limiting the employees' exposure time. If such noise control procedures turn out not to be feasible, then hearing protection must be provided for the workers.

One effective and commonly used approach is to CONTAIN the noise by providing an enclosure around the noisy equipment. This approach can work very well, so long as we attend to one very important matter: we must provide sound-absorptive treatment inside the enclosure, to soak up as much sound as possible. This step is necessary because the first thing that happens when we enclose a noise source is that the noise, which can no longer escape, builds up inside the enclosure to levels that are higher than they were without the enclosure. Providing the sound absorptive treatment inside the enclosure prevents this undesirable noise build-up and allows the enclosure to get on with its job of attenuating the noise to acceptable levels outside the enclosure.



In some cases, the "user" is actually inside the enclosure with the noise source, as in a road traffic tunnel. The application of sound-absorptive materials on the walls and ceiling of the tunnel prevents a serious build-up of tire and motor noise, which otherwise could distress and confuse the drivers.

In all such sound absorptive treatments we must take care to match the acoustical performance of the treatment to the frequency range in which the equipment generates the greatest amount of noise. And this is where treatments using perforated metal come in!

II. Matching the Sound Absorption to the Frequencies Where the Noise Problem Lies

Figure 2. In refurbishing the Rotunda at the Rotunda at the University of Virginia, there was a conflict between the architect's wish to preserve the original appearance of Thomas Jefferson's handsome plaster dome and the need for acoustical treatment to quiet the room. The original plaster was replaced with curved, finely perforated sheet metal behind which sound-absorbing blankets were hidden, with a resulting appearance indistinguishable from that of plaster.

Before beginning to design noise control measures using perforated materials, you must decide what kind of noise problem you have.

As suggested under applications 1 and 2, above, perforated metals can be used in two completely different ways in acoustical applications.

In the first application, we want the sheet to be as transparent as possible to sound of all frequencies. This would be the choice if we want to absorb noise that contains energy in a broad range of frequencies, or if we want the sound of an orchestra in a concert hall to pass freely through a false, decorative, perforated surface in order to reach specially designed acoustical treatment behind the sheet.

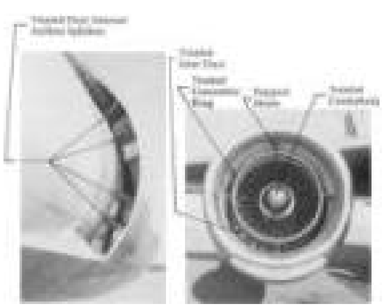


Figure 3 **Inlet and exhaust ducts of jet engine, lined with sound absorptive treatment that is faced with perforated metal.**

If on the other hand, we wish to absorb sound in a relatively narrow band of frequencies, we use the perforated sheet as an integral part of a tuned Resonant Sound Absorber. A common application for this kind of treatment is in the inlet of a jet engine.

The design procedures for these two applications are quite different. They are described in Sections III and IV.

However, before choosing which of the two applications is appropriate, we first have to determine whether our problem concerns broad-band or narrow-band noise: that is, whether we will require the "TRANSPARENCY" or the "TUNED RESONANCE" approach.

A. Frequency Analysis

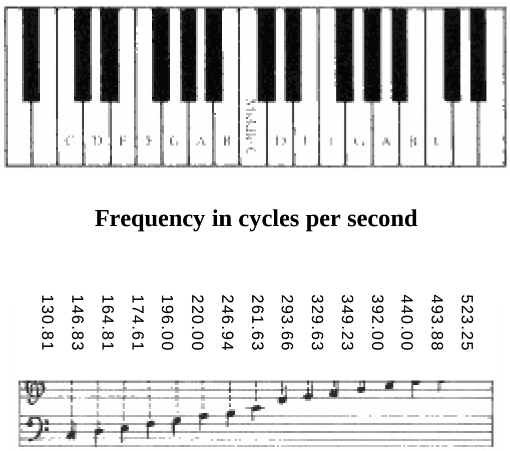


Figure 4. Piano keyboard and musical staff, showing the relations to the frequency spectrum.

For this purpose, we need some kind of frequency analysis, whether measured or estimated, to tell us how the energy of the noise is distributed among the various frequencies.

We can use the analogy of the piano keyboard, here, to represent the range of frequencies of interest: the high pitches lie toward the right, the low pitches to the left.

If we play a single key (or three or four adjacent keys near middle-C, the sound energy will be concentrated around the frequency 250Hz (cycles per second).

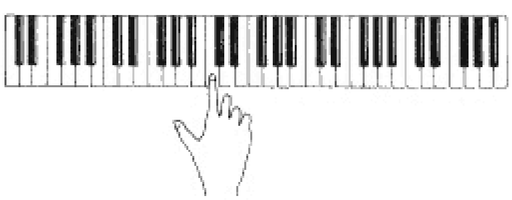


Figure 5. Making a sound spectrum, with the sound energy concentrated around 250 Hz ("middle C").

If we use forearms and elbows to play as many adjacent keys as we can, the resulting "noise" will be distributed over a broad band of frequencies.

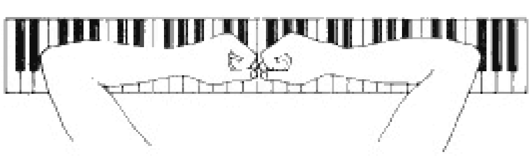


Figure 6. Making a "broad-band" noise with wide, flat spectrum.

A suitable frequency analysis would distinguish clearly between these two conditions, and would guide us to the appropriate choice of design procedure, when we seek to attenuate the noise using perforated metals in an accoustical treatment.

B. Sound Level Meter

Figure 7. We can analyze sounds, showing how the energy is distributed over different frequency bands, by means of a Sound Level Meter (SLM).

Such an analysis is made by means of a Sound Level Meter.

This is a piece of hand-held equipment containing:

- a microphone (to convert the sound wave into an electrical signal);
- an amplifier (to increase the strength of the signal);
- a set of filters (to select different ranges of frequencies for measurement); and
- a meter (or digital read-out device) to indicate the sound pressure level being measured.

If all the filters are switched out, the meter reads the total energy of the noise at all frequencies. If only one of the filters is switched in, the meter responds only to the energy in the band of frequencies passed by the filter.

C. Frequency Spectrum

Returning to our piano example above, where only a few adjacent notes around middle-C were played, if we measure the sound level with the filters successively switched from low to high, we would get a strong meter reading only with the filter for the frequency band centered around 250 Hz; all the other readings would be much lower (corresponding to the ambient room noise).

This would tell us that, if we wish to attenuate this noise, we should use a Tuned Resonant Absorber.

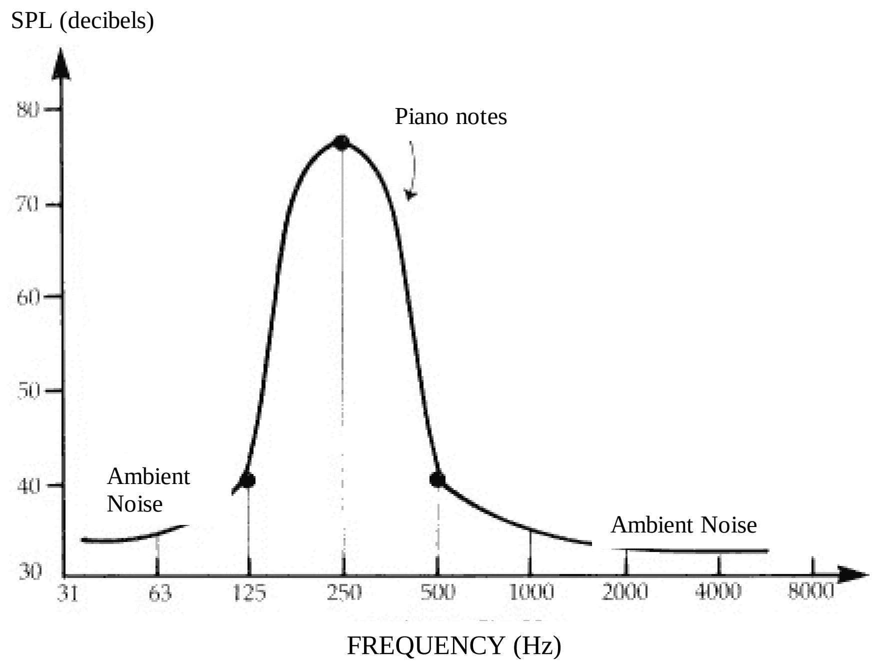


Figure 8. The SLM readings are plotted at the standard octave-band frequencies, in order to exhibit the narrow spectrum from four adjacent piano notes, as in Fig. 5.

For the "all-elbows" piano noise, we would get high meter readings with nearly all the filters, indicating that sound energy of comparable levels in a broad range of frequencies is present.

Here, our acoustical treatment would aim for maximum transparency from the perforated sheet.

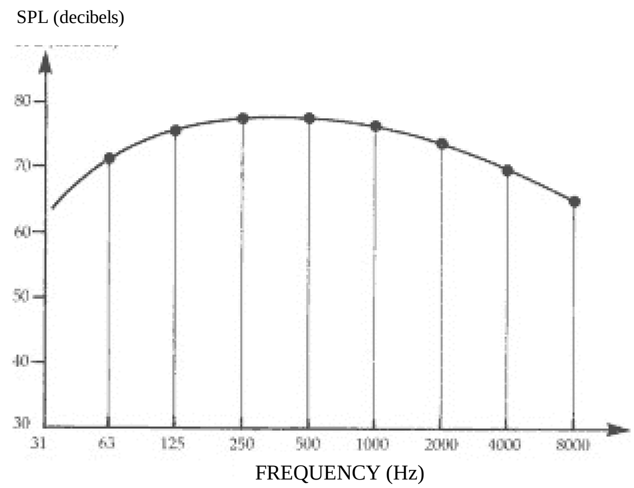


Figure 9. Plot of the SLM readings for the "all-elbows" broadband noise of Fig. 6.

D. Frequency Spectra for Some Household Equipment

The following figure shows how the noise energy is distributed over different frequencies for three typical household appliances. The noise of the stove hood is strongest at low frequencies (125 and 250 Hz), while that of the electric drill is most intense at high frequencies (2000 and 4000 Hz). The dehumidifier noise is distributed about equally over the entire frequency range.

It would require different configurations of sound absorptive treatment to deal with these three noise spectrums.

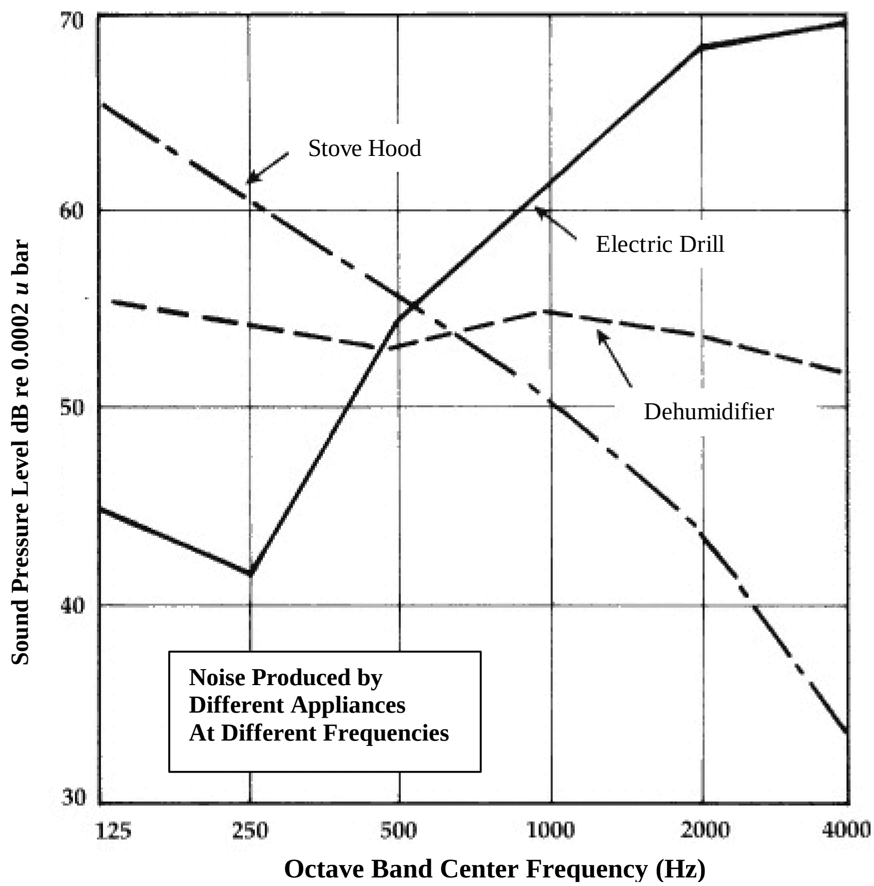


Figure 10. Octave-band Sound Pressure Levels for three household appliances.

Thus, whenever you are called upon to recommend a sound absorptive treatment using perforated materials, your first step must be to determine the frequency distribution of the offending noise. If it is a broad-band noise, you will take the"transparency" approach; if it is a narrow-band noise, you must design a Resonant Absorber tuned to the frequency (or frequencies) where most of the sound energy lies.

These two design approaches are given in detail in sections III and IV, along with illustrative examples.

Take care, when you measure the noise spectrum, that you account for the full range of operating conditions of the equipment and/or the work materials. A change of operating speed or work material can significantly affect the noise energy distribution.

Also, you should allow for a possible change in the dominant noise frequency as the equipment ages and wears.

III. The "Transparency" Approach

A. Perforated Metal Sheet With High Transparency, for Use in Broad-band Sound Absorptive Treatments

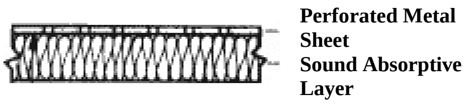


Figure 11. Sketch of perforated sheet over sound absorptive layer.

In this application, perforated metal sheet is used as a sound-transparent protective covering or sound Absorptive als that actually do the work of absorbing the sound.

In this case, because the perforated metal is chosen to be completely transparent to sound, it does not alter the intrinsic performance of the absorptive material in any way.

The following figure shows typical sound absorption efficiency for glass fiber materials at different frequencies.

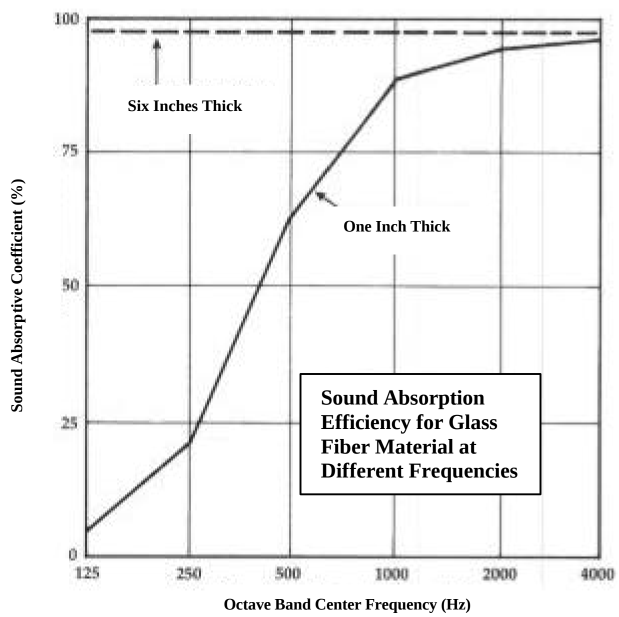


Figure 12. Sound absorption coefficients vs frequency, for glass fiber materials of different thickness.

A layer only one inch thick is quite effective at high frequencies but very poor at low frequencies. It would be a suitable match for quieting noises having frequency spectrums like that of the electric drill that are rich in high frequencies. (The match does not have to be perfect; it is sufficient to follow general trends).

On the other hand, a six inch layer is extremely efficient at all frequencies (about 99% of the incident noise energy is absorbed). The problem is that it takes up a lot of space and is expensive.

If it were not for these drawbacks, ALL sound absorption treatments would utilize thick blankets, with no further worry about targeting specific frequency ranges. Unfortunately, it is not cost effective to provide thick absorbing layers when only a limited frequency range requires treatment. The Tuned Resonant Absorbers, described below, can achieve comparable sound absorption efficiency in a limited frequency range, with lower cost and reduced space requirements.

Material thicknesses, intermediate between the 1" and 6" treatments shown in the figure, exhibit sound absorptive efficiencies intermediate between those two curves, which can be roughly matched to the noise spectrum for which noise control is desired.

A general discussion of the relation between the thickness of the glass fiber blanket and the effectiveness of absorption at low frequencies is presented at the end of this booklet in Appendix A.

The next step is the selection of a suitable protective covering.

B. Why Perforated Metals Are Often The Best Choice

Figure 13. Although it is not conspicuous, the ceiling of this classroom is made of perforated metal with glass fiber blanket in the space behind.

You probably already know that perforated metal sheet is often used as a facing for acoustical treatments, but if more people also realized that for many applications perforated metal is the best available facing material, there would be many more such applications.

A great disadvantage of other commonly used sound absorptive treatments is that they cannot be cleaned or repainted without seriously degrading their sound absorptive properties.

Perforated metals are unique as components of acoustically absorptive treatments because they can be cleaned or refinished without harming the absorptive properties for which they were designed, subject only to the proper choice of perforation size and spacing, described later.

Other important advantages of perforated metals in such applications are:

- inherent structural strength, compared with woven or felted facing materials; they can stand alone, if necessary;
- ability to be formed into complex curved shapes for architectural (visual) purposes;
- resistance to abuse and damage

Finally, the chief architectural advantage of perforated metal is that it is basically uninteresting. It can be made to look like something else: for example, plain plaster. Unfortunately, its neutral appearance creates difficulties for us when we try to illustrate this advantage in this booklet; photographs don't show up what is really going on!

C. Sound Transparency of Perforated Metal

Once a sound absorptive material is chosen to match the noise control task at hand, we must select the proper kind of perforated metal to serve as a protective covering. We must decide which perforation pattern, AMONG THOSE PATTERNS THAT ARE READILY AND CURRENTLY AVAILABLE, provides the greatest transparency.

Most people assume that the greater the percent open area of the sheet, the more easily sound can go through it. In a general way, this assumption is correct… but not always.

For example, we could make a sheet with 10% open area in two ways: either by making a single large hole at the center or by very fine perforations overall.

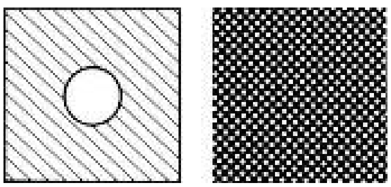


Figure 14. Two samples of perforated metal with the same percentage of open area.

In the first case, instead of a transparent facing material, we would have a small completely open area at the center of the sheet (10% of the total area); but the rest of the sheet would be completely opaque to sound, reflecting ALL of it.

In the second case, the entire sheet is almost completely transparent to sound, because the tiny solid areas between the holes are too small to intercept the sound waves.

For high transparency, the most important consideration is to have many small perforations, closely spaced. It is better to minimize the bar size (the size of the solid portions between the perforations) and (to a lesser extent) to minimize the sheet thickness, rather than to concentrate on percent open area.

In order to help the designer choose a suitably transparent sheet for such applications, we have introduced a parameter called the Transparency Index (TI) given by the following formula:

$$TI = nd^2/ta^2 = 0.04\ P/\pi ta^2$$

where:

n = number of perforations per sq in;
d = perforation diameter (in);
t = sheet thickness (in);
a = shortest distance between holes (in); a = b – d, where
b = on-center hole spacing (in);
P = percent (not fractional) open area of sheet.

The formula is valid for either straight or staggered perforations. An approximation for the value of a, when you do not know the value of b, is:

$a = d[(\text{const.}/P^{1/2}) - 1]$

The value of the constant is 9.5 for staggered and 8.9 for straight perforations.

We can predict from the value of TI the amount by which sound waves at the very high frequency of 10 kHz are attenuated in passing through the sheet, according to the curve in Figure I5, and from this we can develop a curve for the attenuation at lower frequencies. (See Part Two, Section II).

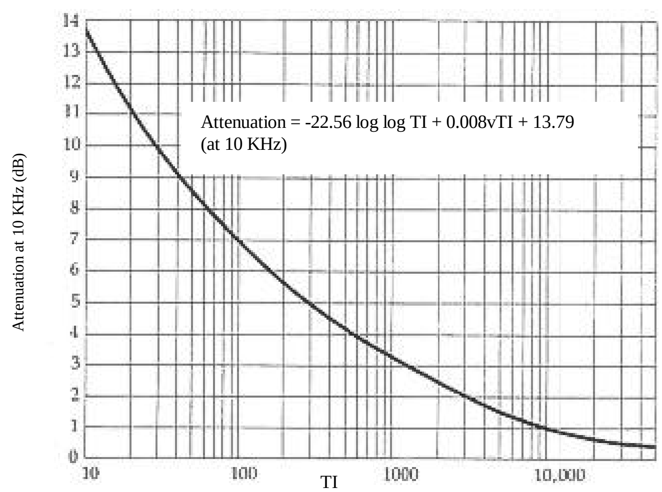


Figure 15. Curve of Sound Attenuation at 10,000 Hz vs TI.

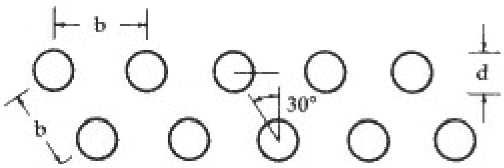
A value of TI upwards of 10,000 should be the goal in choosing a perforated sheet intended for an acoustically transparent facing material. This would lead to an attenuation no greater than one decibel (dB) at a frequency of 10 kHz, and at lower frequencies, the attenuation decreases rapidly: in other words the sheet is essentially acoustically transparent over the entire frequency range of importance.

However, it is not essential to insist on *very* high Transparency Index. For relatively high values of TI, the transparency is not spoiled very rapidly with decreasing values of TI: with TI as low as 5000, the attenuation is only 1.5 dB, and with TI = 2000, the loss is only 2.5 dB.

Therefore, there is no harm in shopping around among the readily available perforated materials to find one whose TI lies between, say 2000 and 20,000. Any value within this range will yield acceptably high sound transparency for most sound absorption applications.

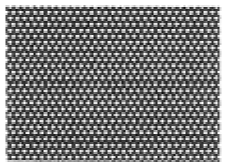
NOTE: The value of TI increases as the hole size and the number of holes per sq in *increases* and as the thickness of the sheet and the distance between holes *decreases*. For values of TI less than 2000, the sound transparency diminishes rapidly, and the perforated metal blocks the passage of sound.

One can also see from the formula that TI generally increases with increasing percent open area P, but NOT if this is achieved with larger holes and an increase of the distance (a) between holes.



This is what the industry refers to as a standard 60° staggered pattern

b = 0.054"
d = 0.023"
t = 0.0184"



Example 1.

A perforated sheet of 26 gauge steel with 0.023-in. holes on 0.054-in. staggered centers leads to the following parameters:

b = 0.054"

d = 0.023"

t = 0.0184"

A = Sheet area per hole = $b^2 \cos 30^0$
= $(0.054)^2 \times 0.87$
= 2.525×10^{-3} sq in;

Then n = 1/A = 396 holes/sq in;

a = b – d = 0.054 – 0.023 = 0.03";
P = $[(\pi d^2/4)/A] \times 100$
= $[(\pi \times (0.023)^2/ (4 \times 2.525 \times 10^{-3})]$
$\times 100$
= 16.45%.

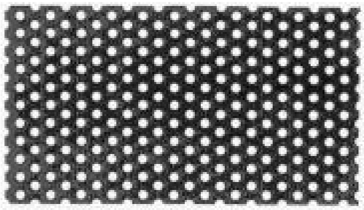
We calculate the Transparency Index by two methods; first by:

TI = nd^2/ta^2
= $[396 \times (0.023)^2/0.0184 \times (0.03)^2$
= 12,650;

or alternatively by:

TI = $0.04P/\pi ta^2$
= $[0.04 \times 16.45/\pi \times 0.0184 \times (0.03)^2$
= 12,648.

The two values of TI agree very well. The sound attenuation at 10 kHz is only 0.9 dB.



Example 2

We have 16 gauge steel sheet, with 0.066" holes on 0.125" staggered centers. Then:

$b = 0.125"$
$d = 0.066"$
$t = 0.0625"$

$A = (0.125)^3 \times 0.87 = 13.6 \times 10^{-3}$ sq in

$n = 1/A = 73.6$ holes/sq in
$a = b - d = 0.125 - 0.066 = 0.059"$
$P = [\pi(0.066)^2/4 \times 13.6 \times 10^{-3}] \times 100$
$= 26.16\%$

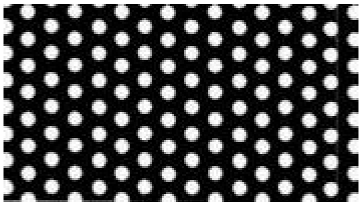
$TI = [73.6 \times (0.066)^2/0.0625 \times (0.059)^2]$
$= 1474;$

or:

$TI = [0.04 \times 25.16/\pi \times (0.0625) \times (0.059)^2]$
$= 1472.$

Again, the agreement between the two values of TI is very good.

But notice that the 10-kHz-attenuation has increased to 2.9 dB, *much* more than the attenuation of Example 1, despite the fact that the open area for this example is 53% greater than in the earlier case!



Example 3

Again, we have 16 gauge steel, but with 7/64" holes on 3/16" staggered centers.

b = 0.1875"
d = 0.109"
t = 0.0625"

$A = (0.1875)^2 \times 0.87$
$= 30.6 \times 10^{-3}$ sq in

$n = 1/A = 32.7$ holes/sq in
$a = b - d = 0.0785"$
$P = [\pi(0.109)^2/4 \times 30.6 \times 10^{-3}] \times 100$
$= 30.49\%$

Then:

$TI = [32.7 \times (0.109)^2/(0.0625) \times (0.0785)^2]$
$= 1009;$

or:

$TI = [0.04 \times 30.49/\pi \times (0.0625) \times (0.0785)^2$
$= 1008.$

Here, even with a percent open area greater than 30%, the 10-kHz-attenuation has increased to 3.3 dB.

You may wonder, then, why it would not ALWAYS be best to choose the most transparent possible material.

We have illustrated by the preceding examples that, generally speaking, perforated sheets with small holes close together give the greatest transparency; but for practical reasons, very tiny holes should be avoided because they may get clogged with dust or filled with paint when the sheet is repainted.

Perhaps more important, very finely perforated sheets tend to be fragile and are much more expensive to manufacture.

Therefore, it is sometimes best to choose the gage of the metal first, based on cost, availability or other reasons; then choose the possible hole size, and jockey the other parameters to achieve the desired transparency.

IMPORTANT NOTICE TO USERS AND SPECIFIERS OF PERFORATED METALS

At this point, large-scale users of perforated metals are probably reaching for their hand-held calculators to find out the values of TI for their most popular products.

DON'T BOTHER! Instead, look at Table 1 on page 37, where the calculations are already done for some commonly produced perforated sheets.

NEXT, DON'T PANIC! You will immediately notice that many popular products have values of TI that are nowhere near the 10,000 recommended above for perfect acoustic transparency.

No matter. Most acoustical problems are concerned with frequencies in the mid-range of 1000 – 4000 Hz. And a glance ahead at Figure 21 (Page 32) will assure you that practically any common perforated metal is nearly totally transparent to sound at those frequencies and below.

So why the emphasis on performance at 10,000 Hz in the Transparency Index? The answer is that a useful *distinction* in the transparency of perforated metals is possible only at very high frequencies. For example, if we decided to rate our Transparency Index at 1000 Hz, say, this would be no good at all, because all the samples would get the same (nearly perfect) rating.

We concentrate here on the 10,000 Hz frequency so that people who are interested in making distinctions in transparency can do so meaningfully.

In all cases, it is important to match the perforated product to the specific needs of the problem at hand.

IV. The Tuned Resonant Absorber Approach

A. Perforated Metal Sheet With Properties Chosen to Target a Limited Range of Frequencies for Optimum Sound Absorption

In the transparency application discussed above, the function of the perforated metal was to act as a protective covering for something else: it must get out of the way and let some other material do its acoustic job.

Now we consider an application where the perforated metal takes an active part in determining the acoustical properties of the treatment.

In many noise control applications, the problem is to reduce noise that occurs only in a limited range of frequencies.

For example, an enclosure around a power transformer must be especially effective at a frequency of 120 Hz (which is the most prominent noise component of the 60-cycle line frequency).

Or, the absorptive lining for the compressor inlet or the exhaust in a jet engine should be most efficient in absorbing sound at the blade passage frequency of the rotor, about 2000 Hz.

One of the great advantages of perforated metal is that it can be used as an element in a "tuned resonant sound absorber" to provide remarkably high sound absorption in the targeted frequency range without requiring a large amount of spacer absorptive material.

Naturally, it sacrifices high absorption efficiency at frequencies outside this range.

In this application, the perforated metal is used in combination with a trapped layer of air, in order to modify the acoustical performance of the absorptive material. This is done by setting up an acoustical resonance condition, which concentrates the sound absorption into a particular frequency range of special interest. It works as follows:

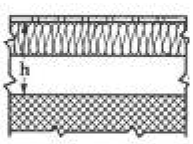


Figure 16. Section through a tuned resonant sound absorber.

All resonant devices have a preferred frequency of operation. For example, a ball suspended on a rubber band oscillates at only one frequency, when disturbed: that frequency is determined only by the mass of the ball and the springiness of the rubber band.

In a resonant sound absorber, the oscillation involves the motion of air particles, in and out of the holes in the metal sheet, in response to an incident sound wave. The preferred frequency of this oscillation is determined by the mass of the air in the perforations and the springiness of the trapped air layer.

At that resonance frequency, the air moves violently in and out of the holes, which pumps the air particles back and forth vigorously within the adjacent sound absorptive layer. There, the acoustic energy (carried by the back-and-forth motion of the air particles) is converted by friction into heat and is thereby removed from the acoustical scene.

The practical advantage of the tuned resonant sound absorber is this: we have seen (page 11) that it requires a six-inch layer of sound absorptive blanket if we wish to attenuate sound effectively at low frequencies. Yet, as we have noted above, the treatment of a power transformer requires maximum absorption around 120 Hz. The one-inch layer of glass fiber (shown in the earlier figure on page 11) is only about 5% efficient at that frequency.

But the use of perforated metal to make a resonant sound absorber especially tuned to 120 Hz can achieve efficient sound absorption at that frequency without requiring so much space and with only a thin layer of absorptive material.

The first clue, to help us decide whether the resonant absorber will be the best approach, is found by listening to the noise. If there is a clearly perceptible pure tone or a prominent frequency (a squeal, hum or whine, as opposed to a whoosh or roar, like a waterfall), this is a good indication that the disturbing noise is concentrated in a limited frequency range, and a tuned resonant sound absorber is called for.

The problem now is to pinpoint that frequency f_R where the maximum sound absorption is desired.

Here one can sometimes rely on the manufacturer's information about the noisy device in question.

Alternatively, one would make a frequency analysis of the noise, using a Sound Level Meter with a set of frequency filters, as described above (page 6).

B. Calculating The Dimensions Of The Tuned Absorber To Give The Desired Resonance Frequency

Having determined the desired frequency f_R of maximum absorption for the tuned absorber, the next step is to calculate the required dimensions for the various elements, in order to make the absorber resonate at the desired frequency.

For this purpose, we use the nomogram on p. 24, where:

f_R = resonance frequency (Hz);

h = distance between the perforated sheet and the solid wall, in inches (see sketch, p. 22);

e = effective "throat length" of the holes; it is given by:

$$e = t + 0.8d,$$

where t is the sheet thickness and d is the hole diameter, in inches;

P = percent (not fractional) open area of the sheet.

For round holes, staggered:

$$P = 0.9\,(d/b)^2 \times 100\%;$$

For round holes, straight:

$$P = 0.8\,(d/b)^2 \times 100\%,$$

where b is the on-center spacing of the holes and d is the hole diameter, both in inches.

[See below, p. 30, for an Important Note qualifying this design procedure.]

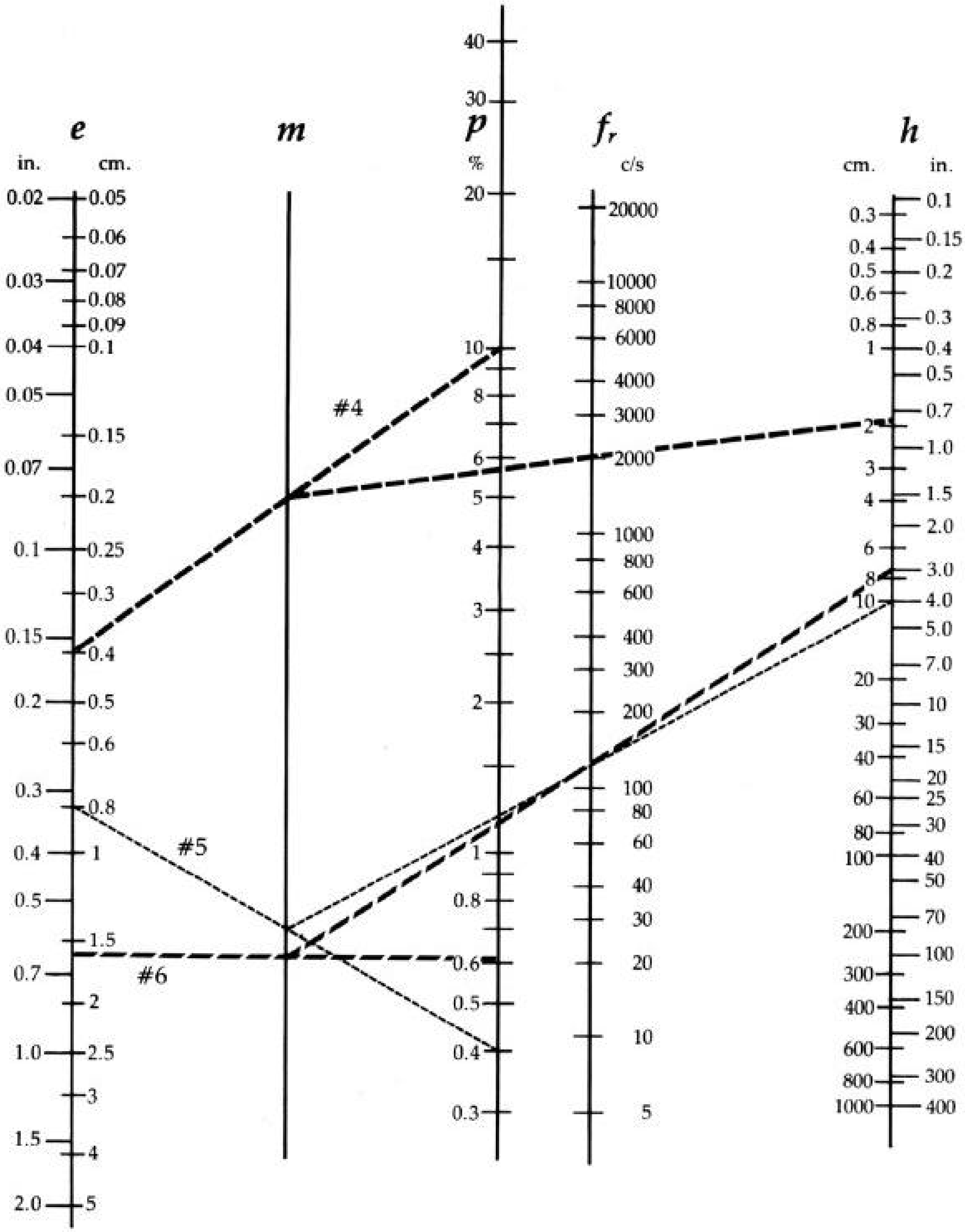
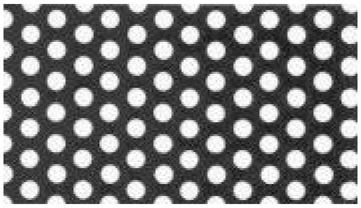


Figure 17. Nomogram for calculating the resonance frequency, with the graphical constructions for Examples 4, 5 and 6. A "clean" version of this Nomogram is included at the back of this booklet, to be copied and used as a worksheet for future design problems.



Example 4: Determining the Resonance Frequency for an Absorber of Specified Dimensions.

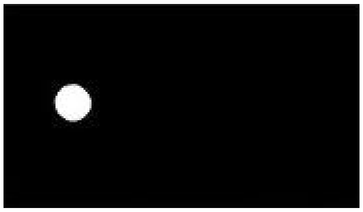
Suppose that we have a sheet of 16 gauge sheet metal, perforated with 1/8-in holes, staggered at 3/8-in on-center (about 8 holes/sq in), which is used as a facing for a glass wool blanket 3/4-in thick, against a solid wall. Determine the resonance frequency.

For this example:

b = 0.375";
d = 0.125";
t = 0.0625";
h = 0.75";
e = 0.0625 + 0.8 x 0.125
= 0.1625;
p = 0.9 $(0.125"/0.375")^2$ x 100
= 10%

We begin by locating the points on the nomogram corresponding to e = 0.16" and P = 10% and connecting these points with a straight line. Mark the point where this line crosses the unnumbered "m"-scale. Now connect that point with the point on the "h"-scale corresponding to the absorber depth, h = 0.75". Read the resonance frequency where this line crosses the f_R-scale: 2000 Hz.

This would be a suitable structure for the jet engine duct lining mentioned above.



Example 5: To Design a Tuned Resonant Absorber to Resonate at a Specified Frequency.

Suppose that we want a structure with a resonance frequency of 120 Hz, to be used as an absorptive enclosure for a large power transformer. The available space behind the perforated metal sheet is 4" and the most readily available sheet stock is 0.125" thick, with staggered pattern. Determine the required hole size and spacing, and the percent open area. We are given:

f_R = 120 Hz;
h = 4";
t = 0.125".

What we must do, given the values for f_R, t and h, is to choose a combination of d and P that will satisfy the nomogram, as follows:

Connect the points corresponding to h = 4" and f_R = 120 Hz with a straight line, continuing it across to intersect the "m"-scale.

Now, as a first guess, let us try perforations ¼" in diameter. With t = 0.125" and d = 0.250" we have:

e = 0.125 + 0.8 x .250 = 0.325"

So we now connect the point corresponding to e = 0.325" to the point found above on the "m"-scale, and continue it to intersect the P-scale at 0.4%.

The plate area per hole is:

$A = [(\pi \times d^2 \times 100)/(4 \times P)$
$= (\pi \times (0.25)^2 \times 100)/(4 \times 0.4)$
. = 12.27 sq in/hole; and

n = 1/A = 0.081 holes/sq in

Finally, the spacing of the (staggered) holes is (See sketch in Example #1, p.17):

$b = (A/\cos 30°)^{1/2} = (12.27/0.87)^{1/2}$
$= 3.76"$

The reader may be surprised that we end up, this application, with a perforated sheet having ¼" holes at a spacing as great as 3.8" on center, and with only 0.08 holes/sq in.

But it is important to realize that, in this case, we are NOT trying to achieve the maximum *exposure* of the sound absorptive blanket, as we did in the "transparency approach". In fact, that approach would be effective in this application only if we could afford a six-inch blanket of glass fiber, in order to get high absorption efficiency at the low frequency of 120 Hz due to its thickness alone .

Instead, we are aiming at a combination of perforation pattern and absorber depth (h) that will encourage maximum air particle motion through the absorptive material at the frequency of interest, by deliberately creating a resonance at that frequency. (See the further discussion of the significance of material thickness for low-frequency absorption, in Appendix A).

Alternative combinations of plate thickness, hole size and percent open area that would achieve the same resonance frequency are illustrated in two further examples.

Example 6:

Suppose we have the same transformer problem, but have only 3" of available depth and a sheet thickness of ¼":

f_R = 120 Hz;
h = 3";
t = 0.250"

Try d = 0.500"; then:

e = 0.250 + 0.8 x 0.500 = 0.65" and

we find;

P = 0.61%
A = 31968 sq in/hole, and
n = l/A = 0.03 holes/sq in.

The hole spacing is;

$b = (32.19/\cos 30°)^{1/2} = 6.1"$.

Example 7:

If we repeat Example 6 with 1" holes, we have:

e = 0.250 = 0.8 x 1 = 1.05"

from which we find:

P = 1.0%;
a = 78.54 sq in/hole;
n = 0.013 holes/sq in; and
b = 9.5".

All these variations of perforation pattern in Examples 5, 6 and 7 lead to the same resonance frequency of 120 Hz, to match the dominant frequency of the transformer noise.

C. Design Refinements

Is there any way to decide *which* of these three treatments (or, perhaps, some other variant) will yield the maximum amount of absorption at that frequency?

And is there any especially effective way of disposing the various elements of the resonant absorber to maximize the absorption?

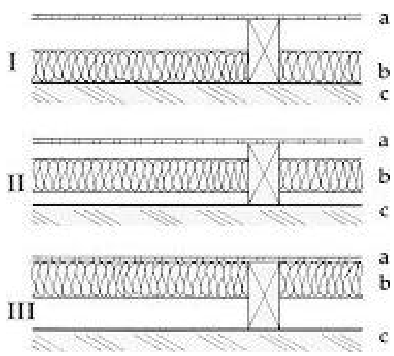


Figure 18. Tuned resonant sound absorbers, showing three possible ways to mount the absorptive material in the airspace.

Once the choice of resonance frequency is made, the actual absorption characteristics can be changed according to the choice of the absorptive material in the cavity and also where the material is located in the cavity, as shown in the sketch of Figure 18.

In each case, a is the perforated sheet, b is the sound absorptive material and c is a rigid backing, such as a wall.

The sound absorptive behavior for these three conditions is shown in Figure 19. The curve with the open circles represents Condition I; that with the filled circles, Condition II; and that with the x's, Condition III.

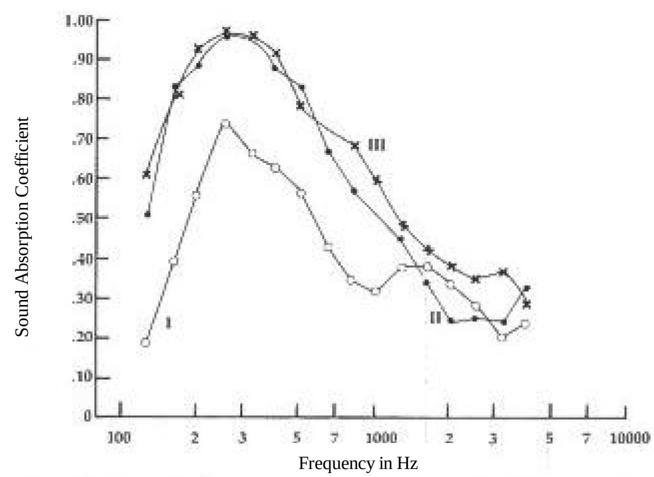


Figure 19. Curves showing sound absorption vs frequency for the three mountings.

Obviously, the most effective arrangement is that with the fibrous material located near the perforated sheet. The worst condition is with the absorptive material close to the wall.

Important Note:

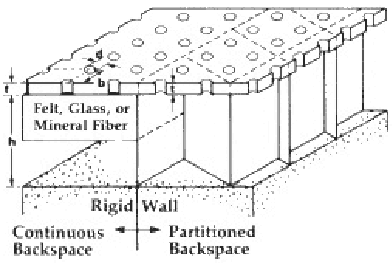


Figure 20. Sketch showing partitioned and non-partitioned airspace behind perforated metal facing.

It also makes a very important difference whether the airspace behind the sheet is continuous or divided into small cells by means of partitions:

When the airspace is continuous, the behavior of the absorber changes greatly at different angles of incidence of the sound. As the sound direction changes from perpendicular to the surface of the absorber (angle of incidence = 0º) to grazing incidence (90º), the resonance frequency changes drastically, rising away from the intended frequency to as much as three octaves higher. In addition, the bandwidth of frequencies within which the high values of sound absorption occur gets smaller and smaller as the angle of incidence tends toward grazing.

By contrast, with the partitioned back structure, not only does the resonance frequency remain the same as the angle of incidence increases, but the bandwidth for high sound absorption actually becomes broader toward grazing incidence.

Finally, there is the effect of the density of the fibrous material used to fill the airspace.

If it is too loose, the sound passes right through the material without being absorbed. But if it is too dense, the sound is reflected and cannot penetrate the material to be absorbed.

More detailed guidance concerning the trade-offs between perforation patterns and depth of airspaces, as well as on the choice of sound absorptive cavity filling, will be presented in PART TWO, below.

ACOUSTICAL USES FOR PERFORATED METALS:

PART TWO: THE APPLICATIONS

I. Introduction

In Part One we were introduced to the principles by which perforated metals are able to serve particularly well in acoustical applications, particularly in noise control treatments.

In Part Two, we show how to use those concepts in a quantitative way, either to analyze an existing application or to design a new treatment in order to meet certain specified requirements.

II. The Transparency Approach

We begin with a closer look at the "Transparency Approach" in the use of perforated metals.

We learned in Part One how to define a Transparency Index as an indicator of how easily sound can pass through a particular sample of perforated metal at high frequencies (see page 14).

We now look at what this means in practice.

Subway station in Vienna; note the perforated metal facing for the sound absorptive ceiling treatment.

A. Sound Attenuation At High Frequencies

The following figure presents laboratory measured data that indicate how high-frequency sounds are attenuated, in passing through samples of perforated metal having different values for the Transparency Index (TI). The horizontal scale gives the frequency in Hz (cycles per second); the vertical scale gives the attenuation in decibels (abbreviated: dB).

It is evident that at frequencies below about 1000 Hz there is little attenuation: the sound passes right through most sheets with no loss whatever.

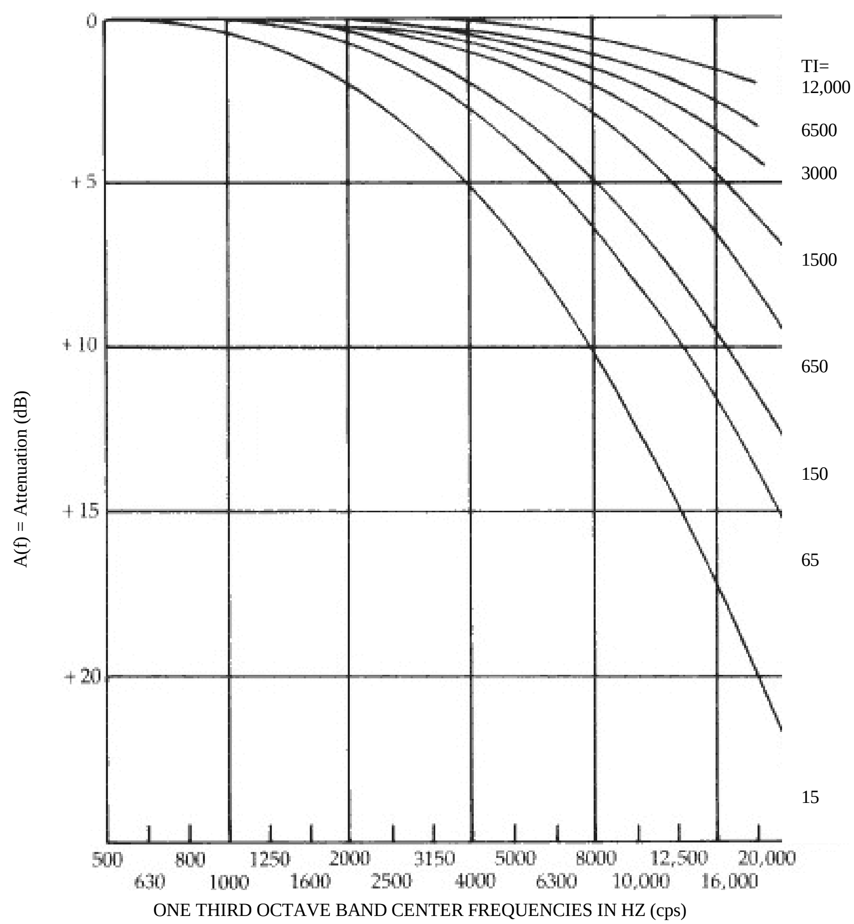


Figure 21. Sound attenuation vs frequency for samples of perforated metal having different TI.

But as the frequency increases, there is more and more attenuation. ...meaning that the sound is reflected from the sheet and fails to get through to reach the acoustical treatment that lies behind.

This condition is more severe, the lower the value of TI. For a sheet with TI = 1500, the attenuation of sound at 16,000 Hz is as much as 4.75 dB; for TI = 12,000, the loss is only 1.5 dB at the same frequencies.

Sometimes the acoustical treatment that lies behind the sheet is a hard, sound-diffusing surface, intended to break up the sound waves and reflect them back to the room, as in a concert hall. Then this attenuation must be counted *twice*: once on the way in and once on the way back.

B. Access To The Sound Treatment

On the other hand, if the acoustical treatment is in- tended to *absorb* the incident sound, then we must determine how much the perforated metal degrades the intrinsic absorptive properties of the material installed behind it, by preventing the sound from getting access to the absorptive material.

For this purpose, we introduce the Access Factor (AF), illustrated in the following figure for the same samples of perforated metal that we saw above, in Figure 21.

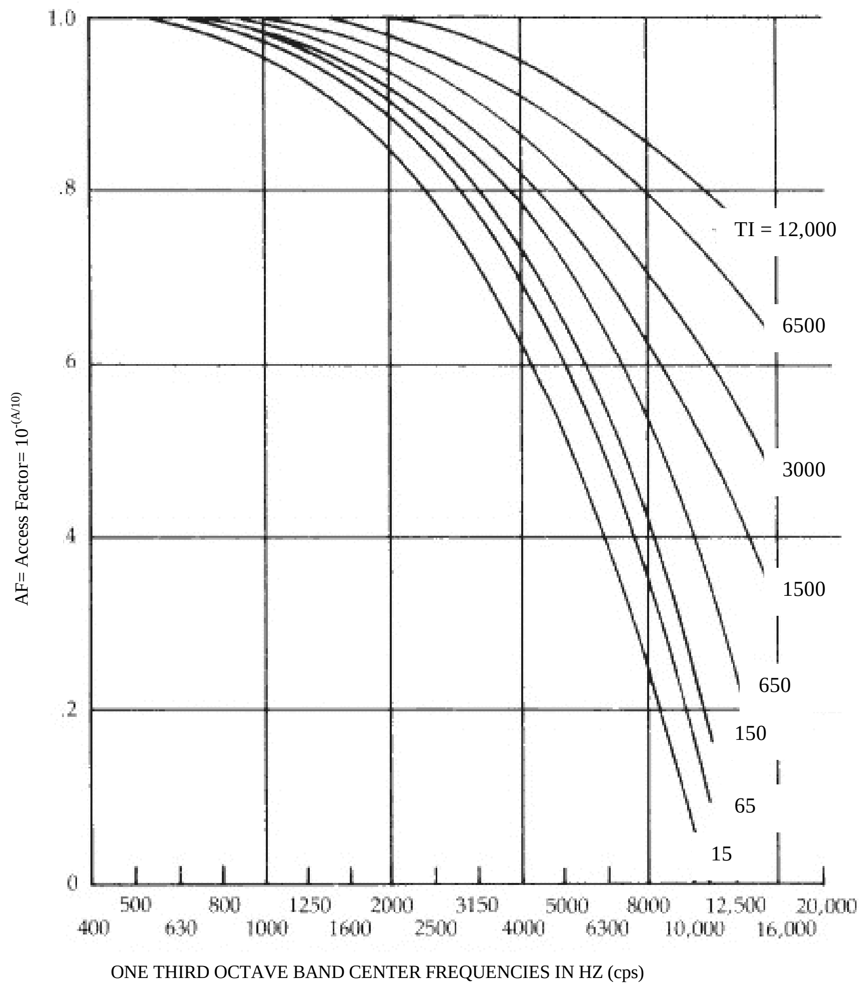


Figure 22. Curves showing the Access Factor vs frequency for the same samples of perforated metal as in Fig. 21.

In general, the Access Factor (AF) at any frequency is related to the Attenuation (A) at that frequency by the following formula:

$$AF = 10^{-(A/10)}$$

C. How To Use The Access Factor

In order to explain how to use the Access Factor, let us recall the definition of sound absorption coefficient, used to characterize the sound absorption efficiency of an acoustic treatment. We saw in Figure 12 that a 1-inch blanket of glass fiber material absorbs about 20% of the incident sound energy at a frequency of 250 Hz, about 65% at 500 Hz, about 87% at 2000 Hz and about 99% at 4000 Hz. On the other hand, a 6-inch layer absorbs about 99% of the incident energy at all frequencies.

All of these numbers assume no *covering* over the sound absorptive material. But when we cover the material with perforated metal, we must expect some degradation of the sound absorptive efficiency. The amount will depend on the frequency, of course, but also on the choice of the perforated metal.

The Access Factor is a measure of this degradation: it describes how much "access" the sound wave has to the underlying acoustical treatment.

If the Access Factor is 1.0, there is complete access and 100% of the sound energy can get through. But if the Access Factor is 0.50, then only half the sound energy can pass through; the other half is reflected from the surface of the sheet and never reaches the acoustic treatment at all.

Therefore, in order to find the effective sound absorption efficiency of an acoustical material covered with perforated metal sheet, we simply multiply the sound absorption coefficient of the basic material at each frequency by the corresponding Access Factor for the metal sheet.

For example, suppose that we cover the 1-inch glass fiber material mentioned above, having a coefficient of 0.99 at 4000 Hz, with a perforated metal sheet having TI = 1500, corresponding to an Access Factor at 4000 Hz of 0.82. Then the effective sound absorption coefficient of the combination is 0.99 x 0.82 = 0.81. The perforated covering has degraded the absorptive performance of the original material at 4000 Hz by 19 percentage points.

Of course, perforated sheet with a TI of only 1500 is a poor choice for this application in the first place. The whole point of the acoustical design in the "transparency approach" is to find a sheet with as high a value of TI as possible, consistent with the other requirements of the project.

It is clear from the figures given above that if we choose a sheet with acceptable transparency at 10,000 Hz (that is, small A and high AF), then everything is much better at the lower frequencies.

The following nomogram allows you to go directly from the calculated value of TI to either the Sound Attenuation or the Access Factor, both at 10,000 Hz.

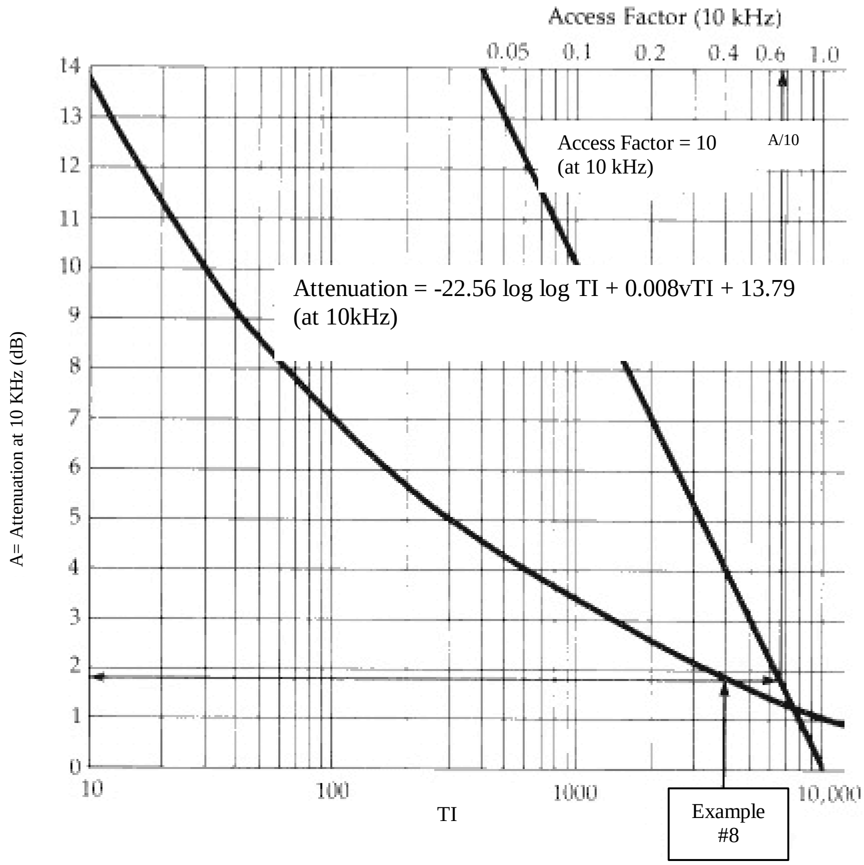


Figure 23. Nomogram for calculating the Attenuation and the Access Factor (10 kHz)

The procedure is as follows:

Enter the lower horizontal scale with the value of TI for your perforated sheet and move directly upward to intersect the lower of the two curves. Move to the left from this intersection point until you strike the vertical scale, where you can read the attenuation at 10,000 Hz in decibels.

Alternatively, if you want the Access Factor, you can move to the right or left from the first intersection point to intersect the upper slanting line, then move upward from the second intersection point to strike the upper horizontal scale, where you can read the Access Factor at 10,000 Hz for the perforated sheet. (See Example 8, next page.)

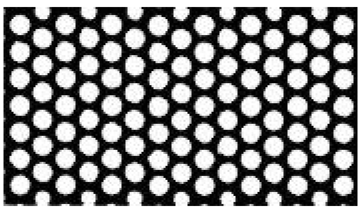
The mathematical formulas corresponding to the curves in this nomogram are, respectively:

$$A_{(10KHz)} = -22.56 \log \log (TI) + 0.008\sqrt{TI} + 13.79 \text{ dB}$$

and

$$AF = 10^{-(A/10)}.$$

The first formula is valid for values of TI up to 50,000, but only for a frequency of 10,000 Hz; the second formula is generally valid for any frequency for which the value of A is known.



Example 8:

Suppose you have a perforated metal sheet with TI = 4000, used as a covering for a sound absorptive glass fiber blanket. What is the effect of the covering?

Enter the lower horizontal scale of Fig. 23 at TI = 4000, move upward to strike the lower curve, then move left to find an attenuation at 10,000Hz of 1.8 dB. Next, move to the right from the first intersection point to intersect the upper slant line, then upward from this point to the horizontal scale to find an Access Factor of 0.66. With this sheet covering an absorptive material, you will realize only 66% of the intrinsic absorption performance of the glass fiber material at 10,000 Hz.

NOTE: See Appendix C for an important technical qualification to the use of the Access Factor, as prescribed above.

NOTE: Full-sized, clean versions of Figs. 21, 22 and 23 are included in Appendix D at the back of this booklet, to be copied and used as worksheets.

Table 1 presents calculated values of the TI, the Attenuation (A) and the Access Factor (AF) at 10,000 Hz, for a group of the most commonly manufactured perforated metals.

Table 1:

Acoustical Properties of Commonly-Manufactured Perforated Metal Products

Item	d (in.)	b (in.)	t (in.)	n (holes/in.^{2})	P (%)	a (in.)	TI	$A_{(10kHz)}$	$AF_{(10k)}$
1.	0.080	7/64" = 0.109	0.030	97	48.5	0.029	24605	0.55 dB	0.88
2.	0.100	5/32" = 0.156	0.030	47	37.2	0.056	4996	1.54	0.70
3.	0.100	3/16" = 0.188	0.030	33	25.9	0.088	1420	2.84	0.52
4.	0.125	3/16" = 0.188	0.030	33	40.0	0.063	4330	1.67	0.68
5.	0.125	1/4" = 0.250	0.030	18	22.5	0.125	600	3.97	0.40
6.	0.156	1/4" = 0.250	0.078	18	36.0	0.094	636	3.89	0.41
7.	0.063	1/8" = 0.125	0.037	74	22.5	0.062	2065	2.41	0.57
8.	3/16" = 0.188	5/16" = 0.313	0.060	12	32.5	0.125	445	4.42	0.36

$a = b - d$; $TI = nd^2/ta^2$; $A_{(10)} = -22.56 \log \log (TI) + 0.008\sqrt{TI} + 13.79$ (dB); $AF = 10^{-(A/10)}$

D. A Case History Illustrating The "Transparency" Approach

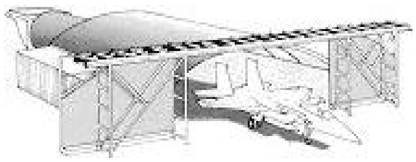


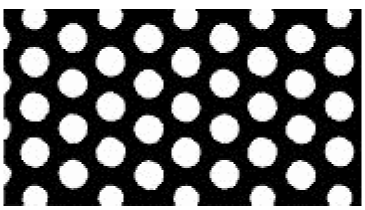
Figure 24. "Hush-house", designed to confine the noise of jet engine tune-ups.

A typical application where widespread use is made of perforated metal is in the acoustical treatment of large "hush houses" for the run-up and testing of jet engines.

In many cases these hush houses are large enough to accommodate an entire airplane for testing.

Since the jet engines on large aircraft are among the noisiest of today's noise sources, it would be intolerable (and a great hazard to hearing) if people had to work in buildings with these engines, unless very effective methods are introduced for controlling and abating the jet noise.

Among the most effective methods is the treatment of the walls and/or ceiling with deep, sound-absorptive material (typically glass fiber blankets or board), covered with perforated metal for protection and ease of maintenance.



Example 9:

If we must choose a very economical wall treatment, it might consist of a 1.5-inch layer of glass fiber board, faced with a perforated metal that has been chosen for the best acoustical transparency consistent with high structural integrity and availability.

For this purpose one might select a stock perforated sheet of 16 gauge steel (t = 0.0598") with 3/16" holes (d = 0.188") on 5/16" centers (b = 0.313"). These dimensions lead to n = 12 holes/ sq in, P = 32.5% and a = b – d = 0.125". We calculate the Transparency Index to be:

$$TI = nd^2/ta^2$$
$$= 12 \times (0.188)^2/0.0598 \times (0.125)^2$$
$$= 445$$

We can already anticipate from this very low value of TI that we will get some degradation of the performance of the glass fiber board; but the sheet dimensions are in this case determined by structural requirements and availability, so we may not have a better choice.

From the nomogram of Figure 23, above (p. 35, or Appendix D), we find the attenuation at 10,000 Hz to be 4.4 dB and the corresponding Access Factor to be 0.36.

We can interpolate in Figure 22 (p. 33, or Appendix D), which gives the curves of Access Factor vs frequency, in order to estimate the Access Factor at octave band frequencies down to 500 Hz, as follows.

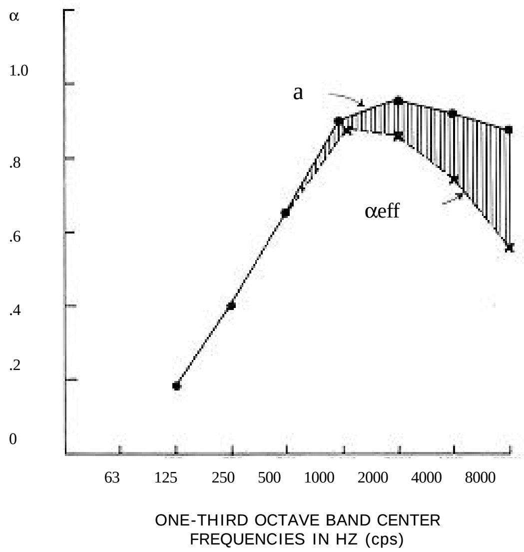


Table 2 gives the sound absorption coefficient at various frequencies for the basic fiber board, as well as the (estimated) Access Factors for the perforated metal, and finally the effective sound absorption coefficients for the composite structure:

Table 2: Effect of perforated metal sheet with a low value of TI on the absorption coefficients for glass fiber board.

Freq	α	AF	α_{eff}
125	0.18	1.0	0.18
250	0.40	1.0	0.40
500	0.65	1.0	0.65
1000	0.90	0.98	0.88
2000	0.95	0.90	0.86
4000	0.92	0.75	0.69
8000	0.88	0.49	0.43

Comparing α and α_{eff} (see sketch), it is evident that the perforated metal covering is hindering the sound absorption at high frequencies. But this may not be a serious drawback, if there is not much high-frequency energy in the spectrum to be controlled in the first place.

Other acoustical applications that use large quantities of perforated metals as facings for sound absorptive treatments include subway tunnels and stations, and street and highway tunnels. Since all of these treatments are trying to cope with noises having broadband spectrums, the acoustical design approach should be the same in all cases: namely, the Transparency Approach. (See Section IV below).

E. Special Considerations: Non-Circular Perforations and "Self-Resistance" Of The Perforated Metal

All of the discussion above has dealt with perforated metal sheet having circular holes, in either straight or staggered patterns. If, instead, the holes are square, we can use the same calculations to a good approximation if we assume an effective hole diameter d' that is equal to $(4/\pi)^{1/2}$ L = 1.13 L, where L is the length of the side of the square perforation. Use the calculation for the straight pattern.

Another somewhat more complicated difficulty arises when, in an attempt to achieve a high value for the Transparency Index, we end up with very small holes. Then, not only is there a risk that the holes will be clogged upon being repainted, but there may even be unwanted energy loss as the air pumps in and out of the tiny holes... just as if it were lost by friction within a sound absorptive blanket.

This condition would cause no harm if our purpose is to use the perforated metal as a facing for an absorptive blanket: it would only add a bit more to the total sound absorption.

But if our goal is to provide a transparent room surface so that sound can pass freely through and back, then we do not want any sound absorbed inadvertently, along the way.

We must, therefore, check our sheet dimensions to be sure that the material is sufficiently sound transparent without adding unwanted sound absorption.

But the further discussion of this problem is slightly complicated; it must wait until we have considered the "Tuned Absorber" application, below. (See Section III, C.3).

III. Resonant Sound Absorbers

In Part One, we learned how to analyze an acoustical treatment in which perforated metal sheet is mounted over an air space containing sound absorptive material, in order to make a "Tuned Resonant Absorber". That is, by the use of a nomogram, we could determine the frequency of resonance where the sound absorption would be especially great; or we could choose the dimensions of the treatment to target a particular frequency range of interest.

Nothing was said there about how much sound absorption would be achieved at the resonance frequency nor about how broad the targeted frequency range would be.

We take up these matters here.

A. Sound Absorption At The Resonance Frequency: $a_{\mathbf{max}}$

In a tuned resonant sound absorber, the sound absorption reaches a maximum value, α_{max}, at the resonance frequency, f_R, falling off to lower values at higher and lower frequencies.

We can control this maximum value of absorption by the choice of the sound absorptive material with which the airspace is filled. Usually, that material will be a kind of porous blanket or board, made of glass fiber or mineral fiber.

The maximum value of absorption depends only on the flow resistance of that material, and not on any of the physical dimensions of the sound absorptive treatment (such as the depth of airspace, perforation diameter, percent open area, etc.).

1. Flow Resistance, Flow Resistivity , and Resistance Ratio

The *flow resistance* of a piece of material tells us how easy it is for air to move through the material. The flow resistance depends upon the density of the fibrous material (lb/sq ft) and the fiber diameter: generally, the heavier the blanket and the finer the fibers, the higher the flow resistance.

And, naturally, thicker layers have more flow resistance than thin ones.

With experience, one can even learn to make a pretty good guess at the flow resistance of a material by seeing how hard it is to blow one's breath through the material.

But for our purposes, we will rely on the measured values of flow resistance for some commonly available fibrous materials.

There's good news and bad news here, however. The bad news is that the manufacturers of fibrous materials don't worry much about the flow resistance of their products, so it's not always easy to find accurate information on this parameter.

The good news is that the acoustical behavior of our tuned resonant sound absorbers isn't critically dependent on the exact value of the flow resistance of the filling in the air cavity. We can miss the design goal quite a bit and it won't make much difference.

But first we have to discuss how to characterize the flow resistance of a layer of material. It is usually done by means of a resistance ratio that tells how much harder (or easier) it is for the sound pressure to push air through the layer in question than to push it through the air itself.

That probably sounds peculiar, because it may not have occurred to you that sound actually encounters some resistance in moving through the air. In fact, there is a "characteristic impedance" that relates the pressure in a sound wave to the corresponding particle velocity in the air: it is given by the product of the density of the air, ρ (gm/cc), and the propagation velocity of sound, c (cm/sec):

Characteristic Impedance = ρc

= 41 cgs rayls.

We always relate the flow resistance, R, of a layer of material to the characteristic impedance of the air, ρc, by forming the resistance ratio $R/\rho c$.

If a layer of material has a flow resistance such that $R/\rho c = 1$, then a sound wave will not recognize the existence of that material when it is encountered, because it can't tell the difference between this material and air.

If the value of $R/\rho c$ is either substantially greater or less than unity, then the sound wave will "notice" the layer, and tend to be reflected from it rather than entering and passing through it.

Important distinctions:

Every fibrous material has a property of its own called the flow resistivity , Ξ which gives the flow resistance per inch of thickness. (We are talking now about the material, itself, not a particular blanket of that material.)

Thus, if a certain type of glass fiber has a flow resistivity Ξ = 60 cgs rayls/inch, then a 2" *blanket* of the material will have a flow resistance of R = 2 x 60 = 120 cgs rayls. And for this blanket the value of $R/\rho c$ = 120/41 = 2.93.

Remember: the flow resistance Ξ is a property of the material, while the flow resistance R is a property of a *blanket* of the material with a particular thickness. The resistance ratio $R/\rho c$ relates the flow resistance of a given blanket to the characteristic impedance of the air.

Now, at last, we are in a position to consider the maximum amount of sound absorption achieved at the resonance frequency of our tuned absorber. As we mentioned above, it depends only on the value of $R/\rho c$ for the filling in the airspace:

$$\alpha_{max} = \frac{1}{\frac{1}{2} + \frac{1}{4}(R/\rho c + \rho c/R)}$$

Table 3 gives values for α_{max} (at the resonance frequency) corresponding to different values for $R/\rho c$ of the cavity filling:

Table 3: Maximum attainable sound absorption (at the resonance frequency), as a function of the flow resistance ratio of the filling material.

$R/\rho c$	α_{max}
0.1	0.33
0.2	0.56
0.5	0.89
0.7	0.97
1.0	1.00
1.5	0.96
2.0	0.89
3.0	0.75
4.0	0.64
5.0	0.56

As we said above, the maximum absorption coefficient at resonance in a tuned absorber is not very sensitive to the filling material: any value of R/ρc from 0.5 to 2.0 will yield a value of α_{max} of 0.89 or greater.

As a practical matter, Table 4 presents, for a number of currently manufactured Owens-Corning Fiberglas products, the value of Ξ (cgs rayls/in.), the value of R (cgs rayls for a 1/2" layer) and the value of α_{max} at resonance for a tuned resonator filled with such a layer:

Table 4: Acoustical properties of typical OCF Fiberglas blankets and boards.

OCF	Ξ	R/ρc (1/2")	α_{max}
700	20	0.24	0.63
701	26	0.32	0.73
702	38	0.46	0.87
703	60	0.73	0.98
704	44	0.54	0.91
705	77	0.94	1.00
PF 105	250	3.05	0.74
TIF	18	0.22	0.59

Sound absorptive treatment, covered with decorative perforated and drawn metal sheet, provides calm acoustical environment in the elegant dining room of the Scandinavia Hotel in Oslo

2. Absorptive Layer Near A Hard Wall.

We come now to a complication that we have already encountered (without an explanation) near the end of Part One: namely, it makes a difference where, within the air cavity, the sound absorptive material is placed (See Figures 18 and 19).

We realize that in order for the absorptive layer to work well, turning the sound energy into heat by the friction of the vibrating air particles within the fine pores of the material, there must be freedom for the air particles to move. If anything impedes this motion, then the energy conversion is less efficient and less sound energy is absorbed.

And that is just what happens at locations near a hard wall: the wall itself, being rigid, cannot move with the sound wave, and this means that the nearby air particles also cannot move. Thus, any sound absorptive material placed against a hard wall is virtually useless, because there can be no air motion within the material to dissipate the sound energy.

Nevertheless, it is common practice to mount sound absorptive layers directly against a wall, because it is very convenient to do so. We must, however, realize that, in such cases, only the outer one-third of the thickness of the layer is effective in absorbing sound. The rest of the material is simply acting as a convenient support!

Therefore, the values of $R/\rho c$ for the 1/2" layer of material given in Table 4, and the corresponding values of α_{max}, assume that this 1/2" layer is mounted near the perforated metal screen with, say, an inch of empty airspace behind it. so that the entite 1/2" layer is effective.

If the layer were mounted directly against a hard wall, the tabulated values of $R/\rho c$ would have to be multiplied by 1/3, and the corresponding values of maximum absorption recalculated.

3. Resonance Frequencies Achievable With Commonly Produced Perforated Metal Sheets

Earlier in this booklet (in Table 1, p.37), we considered the acoustical performance of commonly produced perforated metal products in terms of the Transparency Index, and the corresponding Sound Attenuation and Access Factor at 10,000 Hz. Some were pretty good, some pretty bad.

We now consider four of these same materials in terms of the resonance frequencies that they would produce if mounted in front of a one-inch airspace (the item numbers here are the same as in Table 1):

Item	F_R(Hz)
1	5000
4	3800
5	3000
6	3000

So here's an odd situation! Using common perforated sheet, these "resonant absorbers" all resonate at such high frequencies that the resonance phenomenon adds nothing extra to the natural sound absorption of, say, a 1/2" layer of glass fiber with no covering at all!

Moreover, no reasonable depth of airspace behind these sheets would decrease the resonance frequency below 1000 Hz; for example, samples #5 and #6 would require a 7-inch airspace to make f_R = 1000 Hz.

Tuned resonant sound absorbers evidently require somewhat out-of-the-way perforation patterns, as we saw in Examples 5, 6 and 7, pages 26-28.

B. The Absorption Bandwidth

Not only are we interested in the *maximum* value of sound absorption that occurs at the resonance frequency of a tuned absorber, but we want to know whether the resonance peak is broad or narrow.

In fact, in designing a tuned resonant sound absorber, we want to *achieve* the *required* bandwidth.

As a practical matter, we can characterize the absorption bandwidth of a resonant sound absorber by determining the two frequencies, f_2 and f_1 (above and below the resonance frequency, respectively) at which the absorption has dropped to half its value at resonance. For frequencies below f_1 and above f_2, the absorption of the tuned absorber is relatively insignificant.

The difference between f_2 and f_1 is called the "Half-Power Bandwidth" because at all frequencies within this band the sound absorption exceeds half the (maximum) value at resonance:

$$\Delta f_H = f_2 - f_1;$$
$$= 2\pi[1 + (R/\rho c)]\,(h/c)\, f_R^2;$$

and

$$f_{1,2} = f_R \pm (\Delta f_H/2)$$

These quantities are shown in Fig. 25.

$$\frac{\alpha}{\alpha_{max}} = \frac{1}{1 + (f_R/\Delta f_H)^2\,(f/f_R - f_R/f)^2}$$

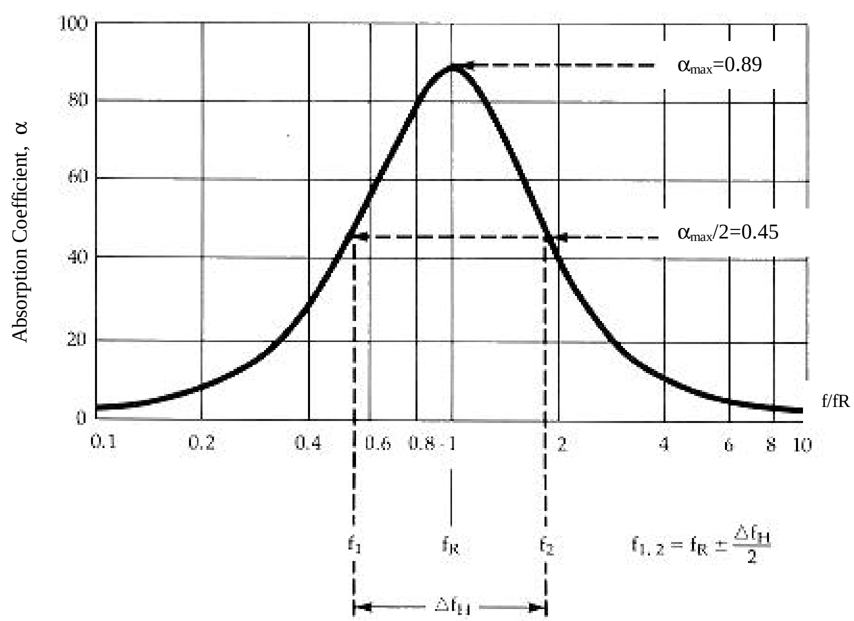


Figure 25. Bell curve of absorption defining α_{max}, f_1, f_2 and Δf_H.

Once the resonance frequency f_R is chosen, the breadth of the absorption curve depends only on (1) the depth of the airspace behind the perforated metal and (2) the flow resistance of the filling material.

Assuming that the latter is chosen so as to maximize the maximum absorption at resonance, or to suit some other criterion, then the airspace depth alone governs the resonance bandwidth.

If Δf_H is small, the absorption band is narrow, and we target a very limited range of frequencies. To absorb a wider band of frequencies effectively requires greater depth for the airspace.

C. Combined Effects of Flow Resistance, Filling, and Absorber Dimensions

Figure 26 will help us understand the respective roles of the filling material and the absorber dimensions in determining how the absorber behaves.

$R/\rho c = 1$
$\beta = 2$

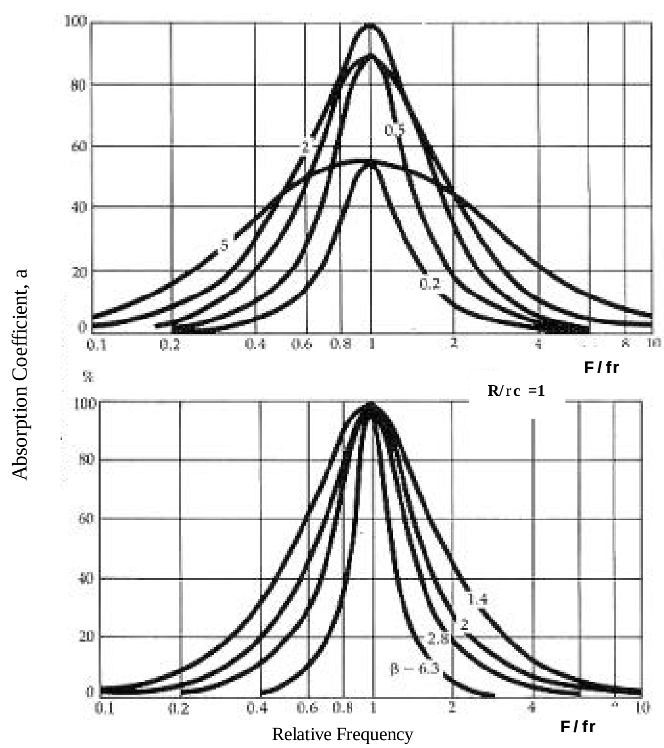


Figure 26. Frequency dependence of the absorption coefficient of perforated sheet in front of an air cushion filled with absorptive material: various values for R/rc and b (see text).

The lower part of the figure shows frequency plots of the sound absorption coefficient for several tuned resonant absorbers, all of which are filled with an absorptive material having $R/\rho c = 1$, but with different values for the construction parameter β.

For the frequency, we use the ratio, f/f_R, of actual frequency to the resonance frequency, on a logarithmic scale. On such a scale, the absorption plots show up as bell-shaped curves, symmetrical about the resonance frequency $f/f_R = 1$.

The parameter β, here, is related to the depth of the absorber and to the "half power bandwidth" Δf_H (introduced above) by the following two equations:

$$\beta = c/2\pi f_R h;$$

The greater the depth of the airspace h, or the higher the resonance frequency f_R, the smaller the value of β.

$$\Delta f_H = (f_R/\beta)\,[1 + (R/\rho c)].$$

The smaller the value of β, or the higher the resonance frequency, or the greater the flow resistance of the filling, the wider the frequency band over which high sound absorption will occur .

In the lower part of Figure 26, the choice of $R/\rho c = 1$ causes the sound absorption at resonance ($f/f_R = 1$) to be 100% in all cases. But smaller values for β lead to absorption curves that are broader; and high values for β lead to narrower curves.

In the upper part of the figure, the value of β is 2 for all the curves, but different values for $R/\rho c$ are chosen. Again, for $R/\rho c = 1$ the absorption at resonance is 100%. But for $R/\rho c = 0.5$ and 2.0 the absorption at resonance is nearly as great ($\alpha_{max} = 0.89$).

Also, note that values of $R/\rho c$ greater than unity lead to broader absorption curves, while values less than unity give narrower curves.

We have seen that the bandwidth of high absorption is proportional to the airspace depth, h. If only a restricted depth is available, and we wish to achieve reasonably high absorption over a wider band, we must design a number of tuned resonant absorbers having different resonance frequencies to cover the whole required frequency range:

$$f_2 - f_1 = n\,\Delta f_H$$

in such a way that, for every frequency, a reaches near to α_{max} on at least one partial surface area.

However, further analysis shows that the necessary construction volume remains the same: what we save in depth of air cushion must be made up in additional area of absorptive surface.

In each case, the choice depends on the available space. Certainly, it is more expensive to enlarge the treated area than to increase the depth of the treatment, so h should be chosen as great as the available space will allow.

1. The Proper Choice for R

The choice of flow resistance for the filling material is not quite so straightforward as it seemed when we were considering only the sound absorption at the resonance frequency.

Certainly, it would not be favorable to choose R smaller than ρc, because then we would decrease both the maximum absorption at resonance and the width of the absorption curve (Fig. 26).

But if R is greater than pc, then again the maximum absorption at resonance decreases, but we get a *broader* curve, which may be desirable.

If we try to optimize *both* the maximum absorption at resonance and the half power bandwidth, by forming their product:

$$\alpha_{max}\, \Delta f_H = 2\pi[4(R/\rho c)/(1 + R/\rho c)](h/c)f_R^2$$

we see that it would still be good to choose a large value for R.

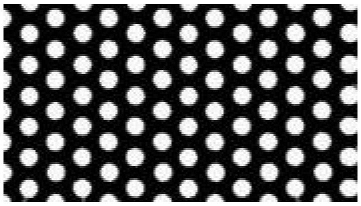
However, even a choice of infinitely high R would yield a result for the product that is only twice that for the matching case, R/ρc = 1. And if we make R *too* great, we invalidate our whole theory of resonating absorbers: a too-strongly damped resonator is no resonator at all!

We conclude, then, that a choice of R/ρc around 2 to 3 will give the best compromise between a high maximum sound absorption at resonance and a broad half power bandwidth.

2. Further Illustrative Examples

In Part One, we presented several examples (#4 - #7) in which we calculated the resonance frequency for different combinations of perforated metal sheet and airspace (pp. 23-28).

We return to those treatments now to determine the half-power bandwidths and to see the effect of different choices of filling material.



Example 10:

Example #4 in Part One concerned a sheet of 16 gauge sheet metal, perforated with 1/8" holes, staggered on 3/8" centers, as a facing for a 3/4" glass fiber blanket against a solid wall. We found that the resonance frequency is 2000 Hz.

We now assume that the glass fiber material has a flow resistivity, Ξ = 55 cgs rayls/in; accordingly, R = 3/4" x 55 = 41 cgs rayls, and $R/\rho c = 1$.

We first calculate the maximum sound absorption coefficient reached at the resonance frequency:

$$\begin{aligned} \alpha_{max} &= 1/[1/2 + (1/4)(R/\rho c + \rho c/R)] \\ &= 1/[1/2 + (1/4)(1 + 1)] \\ &= 1.0. \end{aligned}$$

Next we calculate the half-power bandwidth:

$$\begin{aligned} \Delta f_H &= 2\pi[1 + (R/\rho c)](h/c)f_R^2 \\ &= 2\pi[1 + 1](0.75/13560)(2000)^2 \\ &= 2794 \text{ Hz.} \end{aligned}$$

The lower and upper frequency bounds for the half-power bandwidth are:

$$\begin{aligned} f_1 &= f_r - (\Delta f_H/2) \\ &= 2000 - 1397 = 603\text{Hz} \\ f_2 &= f_R + (\Delta f_H/2) \\ &= 2000 + 1397 = 3397 \text{ Hz} \end{aligned}$$

The value of the absorption coefficient at frequencies f_1 and f_2 is 0.50 (open triangles). The entire frequency plot of absorption for this example is given on Figure 27.

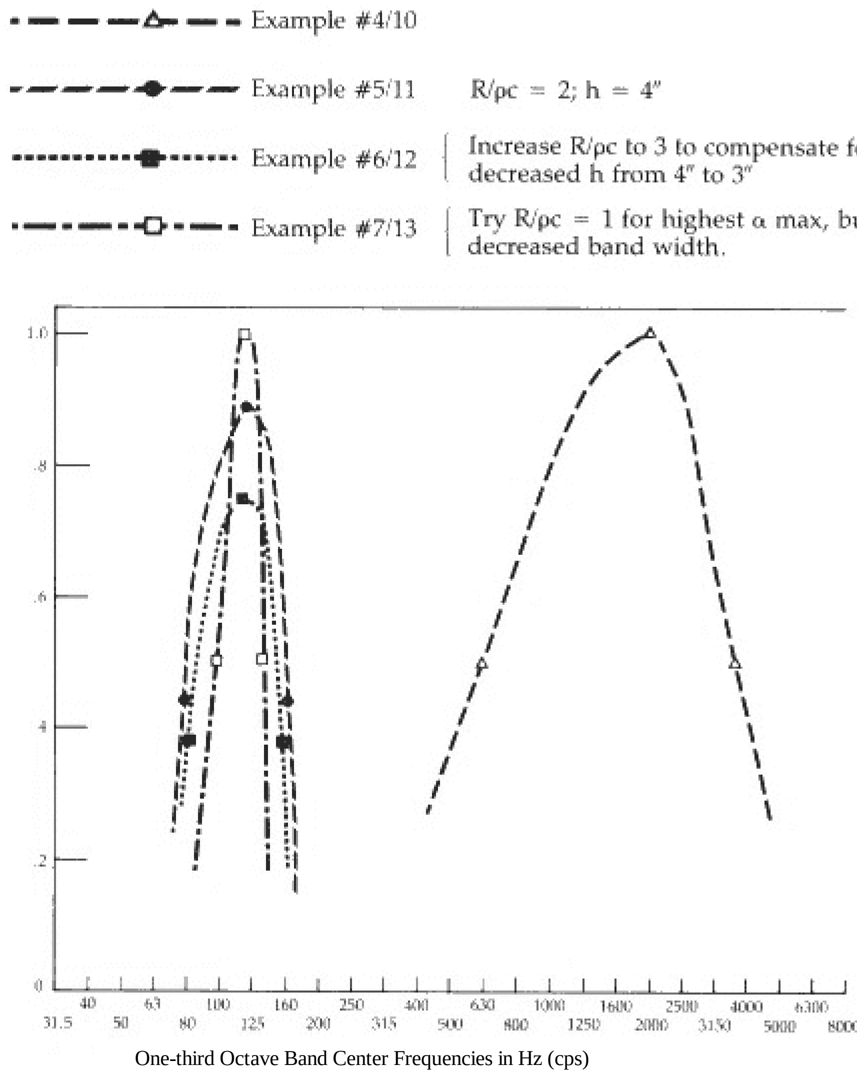


Figure 27. Frequency dependence of the absorption coefficients for the tuned resonant absorbers of Examples 10-13.

Example 11:

The next example (Example 5, p. 26) involved a resonator tuned to f_R = 120 Hz; the available air depth was 4", the thickness of the sheet was ? ", and it turned out that a hole diameter of ¼", a percent open area of 0.4%, with only 0.081 holes per sq in satisfied the nomogram.

This time let us assume a value of R/ρc = 2 for the filling; then the maximum absorption at resonance is (as we already know from the table on page 43):

$$\alpha_{max} = 0.89.$$

The half-power bandwidth is:

$$\begin{aligned}\Delta f_H &= 2\pi[1 + 2]\ (4/13560)(120)^2 \\ &= 80 \text{ Hz.}\end{aligned}$$

Then

$$\begin{aligned}f_1 &= 120 - 40 = 80 \text{ Hz} \\ f_2 &= 120 + 40 = 160 \text{ Hz}\end{aligned}$$

and at these two frequencies, the absorption coefficient is 0.89/2 = 0.44. The entire frequency plot for this example is also given on Figure 27.

Example 12:

The next example (#6, page 28) was the same as the previous example except that the airspace was limited to 3”.

In order to compensate for this reduced airspace, we choose a higher value of the flow resistance of the filler: $R/\rho c = 3$.

Then (again from Table 3 on page 43)

$$\alpha_{max} = 0.75,$$

and:

$$\begin{aligned}\Delta f_H &= 2\pi\,[1 + 3](3/13560)(120)^2 \\ &= 80 \text{ Hz.}\end{aligned}$$

Because of our compensation with the higher flow resistance, the half-power bandwidth remains the same. But in this case, the sound absorption coefficients at $f_1 = 80$ and $f_2 = 160$ Hz are only $0.75/2 = 0.38$ (filled squares in Figure 27).

Example 13:

The final example (#7, page 28) achieved the same resonance frequency of 120 Hz with 1” holes, a percent open area of 1.0%, with a hole spacing of 9.5” and 0.013 holes/inch.

This time, let us assume $R/\rho c = 1$, so again the absorption coefficient at resonance is 1.0.

The half-power bandwidth now is:

$$\begin{aligned}\Delta f_H &= 2\pi\,[1 + 1](3/13560)(120)^2 \\ &= 40 \text{ Hz,}\end{aligned}$$

which leads to values for f_1 and f_2 of 100 and 140 Hz, respectively; at these frequencies the value of α is 0.5 (open squares in Figure 27).

3. "Self-Flow-Resistance" of Fine Perforated Metal Screens

We mentioned above, that, although it is desirable, from the point of view of trying to achieve high transparency from perforated sheet, to aim for tiny perforations closely spaced, there is a danger in over-doing it.

First, if the sheet is painted, the fine holes may get clogged and this would spoil the transparency altogether.

Second, if the holes are fine enough, they will act like the fine pores in a glass fiber absorptive blanket, and may introduce unwanted sound absorption. This would be particularly undesirable if all we want from the perforated metal is acoustical transparency.

Therefore, it is a good idea, once you have finished your design for acoustical transparency, to follow through with a calculation of the self-resistance of the perforated sheet that you have chosen, and calculate the absorption coefficient of the sheet *without any filling.* This is explained below.

Considering that there is a large range of possible perforation patterns, there are two extremes that we could consider, in calculating the self-resistance of the sheet. It depends on whether the holes are *wide or narrow*, compared with the length of the so- called "viscosity waves" that cause the unwanted absorption.

Since the theory for coping with "in-between" situations is not developed, we must calculate *both* values and use the higher of the two results. In addition, there is an end correction to be taken into account, as in our calculations of resonance frequency, above.

So our self-resistance is given by:

$$R_{self} = R_o + 2\Delta R_o$$

where the value for R_o is the larger of the two following formulas, depending on whether the flow resistance is dominated by (1) boundary layer effects or (2) laminar flow:

$$R_{01} = 4.24(b^2t/d^3)\sqrt{f} \text{ x } 10^{-3} \text{ cgs rayls}$$

or

$$R_{02} = 2.88(b^2t/d^3)\sqrt{f} \text{ x } 10^{-3} \text{ cgs rayls}$$

The value for $2\Delta R_0$ is given by:

$$2\Delta R_0 = 4.19(b^2/d^2)\sqrt{f} \text{ x } 10^{-3}.$$

Example 14:

To take a rather extreme case, let us look at the effect of the self-resistance, at a frequency of 1000 Hz, of a sheet of 0.1" metal with 0.16" holes at 0.8" on center, when used in a tuned resonant absorber.

We must calculate both R_{01} and R_{02}:

$$R_{01} = 4.24 \text{ x } (0.8^2 \text{ x } 0.1)/(0.16)^3 \text{ x } \sqrt{1000} \text{ x } 10^{-3}$$
$$= 2.1 \text{ cgs Rayls.}$$
$$R_{02} = 2.88 \text{ x } (0.8^2 \text{ x } 0.1)/(0.16)^4 \text{ x } 10^{-3}$$
$$= 0.28 \text{ cgs Rayls.}$$

Here, the dominant factor in the flow resistance is the boundary layer, R_{01}.

Also:

$$2\Delta R_0 = 4.19(0.8^2)/(0.16)^2 \text{ x } \sqrt{1000} \text{ x } 10^{-3}$$
$$= 3.3 \text{ cgs Rayls.}$$

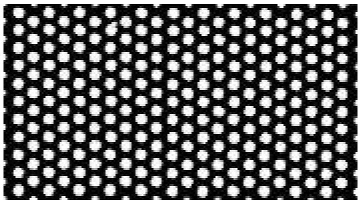
The total self-resistance is:

$$2.1 + 3.3 = 5.4 \text{ cgs Rayls,}$$

which at first glance doesn't seem like much flow resistance.

But the corresponding value for the resistance ratio $R/\rho c$ is $5.4/41 = 0.13$; and, according to Table 3 on page 43, this already yields a maximum value of 0.41 for the sound absorption coefficient at resonance, even with no deliberately added filling material in the air cavity!

Note that for many practical perforated sheets, the end-correction term dominates the contributions from the holes, themselves, in the total flow resistance.



Example 15:

As a more typical example of commonly used perforated sheet, let us repeat the calculation for Item #7 in Table 1 at a frequency of 8000 Hz. Here the hole diameter d is 0.063", the on-center spacing b of the holes is 0.125", and the sheet thickness t is 0.037". (Of the examples in Table 1, this one will yield the *greatest* self-resistance).

We find:

R_{01} = 0.88 cgs Rayls;
R_{02} = 0.11 cgs Rayls;
$2\Delta R_0$ = 1.48 cgs Rayls.

Again, R_{01} dominates.

Then:

R = 0.88 + 1.48 = 2.36;
$R/\rho c$ = 0.58; and:
α_{max} = 0.2.

As, suggested above, this value is about the highest that one would expect to find in typical perforated metal sheets. If the resonator were to be filled with any reasonable absorptive material, the acoustical performance would be governed by the filling, not by the self-resistance of the sheet.

IV. Practical Large-Scale Application of Sound-Absorptive Treatments Using Perforated Metals

We conclude this section with some illustrative photographs showing sound absorptive treatments for the control of roadway and subway noise.

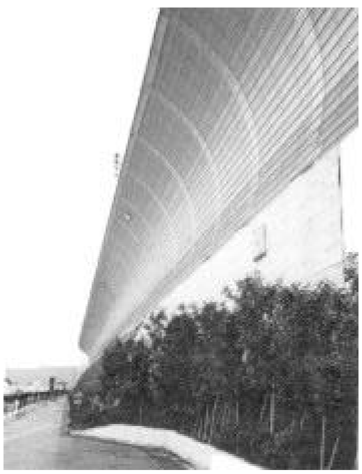
A. Barrier Screen for Tokyo Roadways

Because of the extremely crowded conditions in the large cities of Japan, the highways and elevated roadways often pass quite near residential communities, and cause considerable annoyance to the residents because of the noise.

Considerable protection can be afforded to these communities by erecting sound barriers along the roadways, which shield and absorb the sound of the motor vehicles.

In Tokyo alone, there are 90 running miles of such barriers, ranging from 8 to 15 feet in height!

The following photographs show some typical sights along the Tokyo roadways.

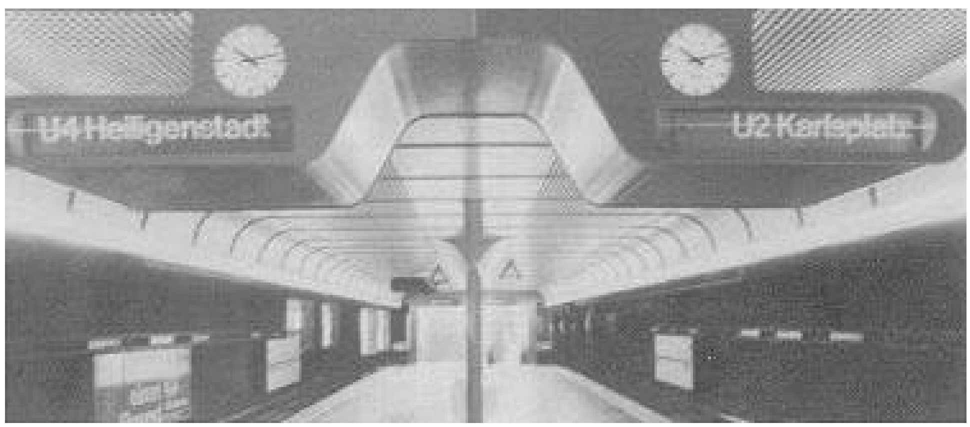


Shown are details of the Tokyo roadway that appears on page 59. These perforated curved barriers are more acoustically efficient and present a more pleasing aesthetic appearance than the hard barriers presently being used in the United States.

B. Absorbtive Barriers and Ceiling Treatments in the Vienna Subway System

A similar large-scale application of perforated metals occurs in the newly built sections of the subway in Vienna (Austria)

The following photographs show some of these treatments.



C. Acoustical Effects of the Sound Absorptive Treatment in the Vienna Subway

Figure 28 shows measured data from the Vienna subway in the course of successively more extensive sound absorptive treatment of the ceiling and side walls.

Part A of the figure indicates, on the cross-section of the subway tunnel, the various areas where sound absorptive treatment was applied:

1. On the ceiling over the central platform;
2. On the lower, outer parts of the side walls;
3. On the lower walls beneath the central platform;
4. On the overhang of the central platform floor;
5. On the ceiling, above the tracks;
6. On the main side walls of the tunnel.

Part B of the figure indicates the progressive reduction of the reverberation time in the tunnel (as a function of frequency) as successively more sound absorptive treatment was added.

Part C of the figure indicates the progressive reduction in the A-weighted sound level of the passing trains as successively more treatment was added.

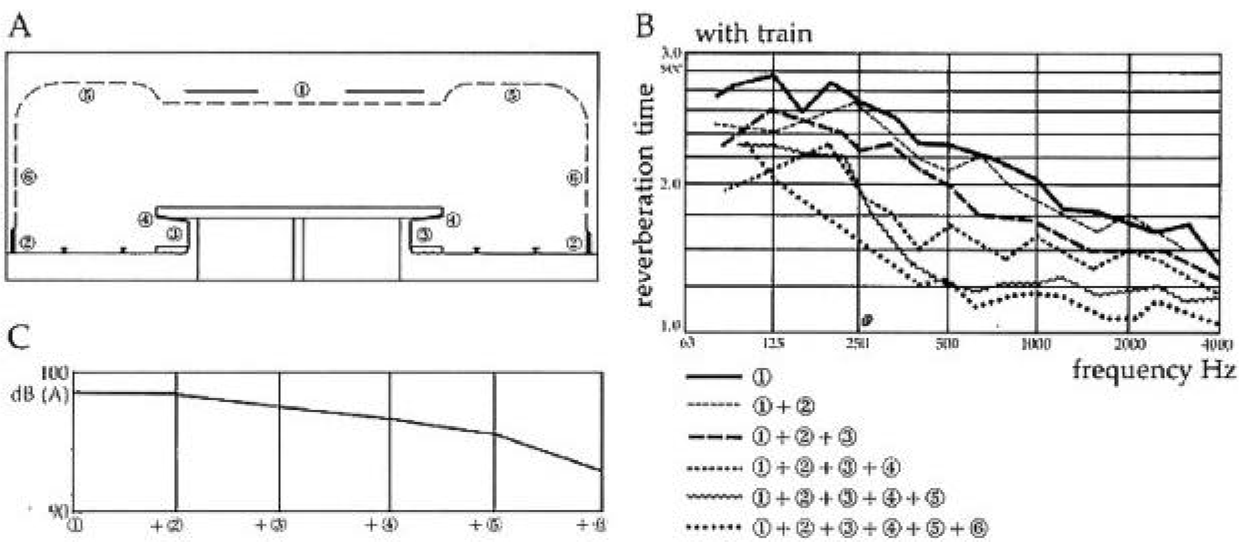
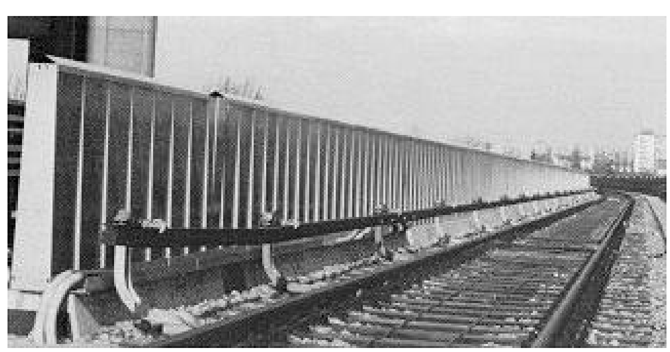
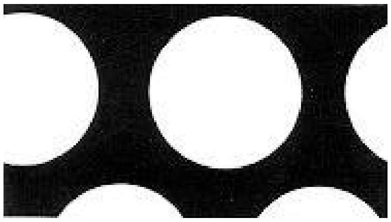


Figure 28. Sound absorbtive treatments in the station. A. Indicates the absorptive sufaces -- j to m, as shown; n, 35 meters of sound absorptive walls in the tunnel. B. Reverberation time in the station. C. Effective sound level of the passing trains.

Exterior acoustical shielding in the Vienna subway system: a sound absorptive barrier, with perforated metal facing, runs along the tracks to shield the neighbors from the noise of the wheels and tracks and the traction motors.



D. Sound Attenuation and Access Factors for These Treatments



It will be noticed that some of the perforated metal sheets in these treatments have quite large perforations, rather widely spaced (few perforations per inch).

In view of our earlier discussion of the advantages of numerous small holes, it is of interest to calculate the TI for these large-scale sheets and the corresponding acoustical parameters.

Let us assume the 1/16" sheet has 1" holes, staggered at 1 ¼" on centers. The Percentage Open Area is 58% and the number of holes per sq. in. is 0.73; that is:

n = 0.73; d = 1.0"; b = 1.25";
t = 0.063"; and a = b – d = 0.25".

Then TI $= nd^2/ta^2$

$= 0.73 \times (1)^2/0.063 \times (0.25)^2$
$= 185.$

Such a low value of TI implies poor transparency at high frequencies. Figure 23 indicates an attenuation of 5.9 dB at 10 kHz, and a corresponding Access Factor of only 0.26 at the frequency.

But the important frequencies in the noise of roadway and subway traffic are below 2000 Hz. At that frequency, according to Figures 21 and 22, the attenuation is only 0.2 dB and the Access Factor is up to 0.91; and, of course, the situation improves for all lower frequencies.

Other properties of the perforated sheet in these applications, such as ruggedness and durability are more important than the highest acoustical transparency.

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APPENDIX A: Thickness of Sound Absorptive Treatment and Sound Absorption at Low Frequencies

Sound is a disturbance in the air characterized (among other things) by the fact that the air particles move back and forth locally, in a restricted region. The back-and-forth motion is slow at low frequencies (say, 100 cycles per second), fast at high frequencies (say, 5000 cycles per second).

This disturbance propagates through the air as a wave, moving at a constant speed, independent of frequency, called the speed of sound: about 1130 ft/sec at normal temperatures, or about one mile in five seconds.

The wave nature of the sound means that there is a definite relation between the air particle motion at different locations in space, along the direction of sound propagation. At positions separated by a distance equal to the wavelength of the sound, the air particles move together in synchronism, back and forth exactly in step.

The wavelength, λ, is determined by the ratio of the speed of sound to the frequency of the sound:

$$\lambda = c/f.$$

For air at normal temperature, $c = 1130$ ft/sec; so for a frequency of 100 Hz, the wavelength is $\lambda = 1130/100 = 11.3$ ft and for a frequency of 5000 Hz, the wavelength is $\lambda = 1130/5000 = 0.23$ft. or 2.7 in.

In general, *low* frequencies mean *long* wavelengths and *high* frequencies mean *short* wavelengths.

In the vicinity of a solid reflecting surface (such as the wall or ceiling of a room), the sound wave is reflected back into the space from which it came, and the incident and reflected waves interfere with each other. The result is that directly *at* the reflecting surface the two waves add together to create a sound pressure double that of the incident wave. But since the reflecting surface is rigid, there cannot be any air particle motion near the wall; the particle velocity there is zero.

Farther from the surface, however, these two waves are out of phase with one another, such that at a certain distance the pressure waves may cancel one another, leaving nearly null pressure; but at this same location the particle velocity is greatly increased to almost twice that of the incident wave. This location of maximum particle velocity occurs at ¼ wavelength from the surface: this is 11.3/4 = 2.83 ft for a frequency of 100 Hz, or 0.68 in for 5000 Hz.

These phenomena govern the sound absorptive behavior of the porous blankets that are often mounted against room surfaces in order to absorb some of the sound energy in the incident sound waves.

The absorption takes place by a process of friction between the moving air particles and the fibers of the sound absorptive material; the sound energy in the incident wave is converted by this friction into heat, and it therefore disappears as sound energy from the acoustical scene.

It follows that the most effective sound absorption will occur where the air particle motion is greatest. This behavior is quite different at low frequencies than at high frequencies because of the differences in wavelengths, and the corresponding effect on the interference pattern between the direct and reflected waves.

For this reason, a thin blanket of sound absorptive material (say, ½ inch) placed *against* the wall would have almost no effect on a sound wave at 100 Hz, because near the wall there is practically no air particle motion and therefore no friction.

The maximum particle motion for this low frequency would be at a distance of ¼ wavelength (34 inches) from the surface. In order for the blanket to absorb the 100 Hz sound effectively, it would have to be mounted at this distance, where it would be very effective, even with nothing but airspace between it and the wall. (It would be even more advantageous if this back space is partitioned into closed cells, as we have seen on page 30.)

One way to place absorptive material far from the wall is simply to use a very thick layer. In this case, the part of the material near the wall takes little part in absorbing the sound, but is simply a means of supporting the rest of the material farther away from the wall, where it can do a very good job. Even a blanket thickness of only six inches yields very effective sound absorption at 100 Hz (see Figure 12, page 11).

For much higher frequencies, however, the quarter-wavelength location is much nearer the wall, and relatively thin blankets of fibrous material can absorb the sound very effectively (see again Figure 12).

The reason for using sound absorbers with perforated metal facings is to create an acoustical resonance condition, such that even at low frequencies the location of maximum air particle velocity is made to be near the wall surface. This saves on space and on the amount of sound absorptive material required for the installation.

APPENDIX B: Background for the Transparency Index and the Attenuation of Perforated Metal at High Frequencies

A number of years ago an experiment was conducted for the purpose of evaluating various kinds of transparent and semi-transparent material to determine how readily they would transmit sound.

The experiment was conducted in a so-called "anechoic chamber," namely, a room whose interior surfaces are all covered with highly sound-absorptive material, in order to suppress all sound reflections as completely as possible.

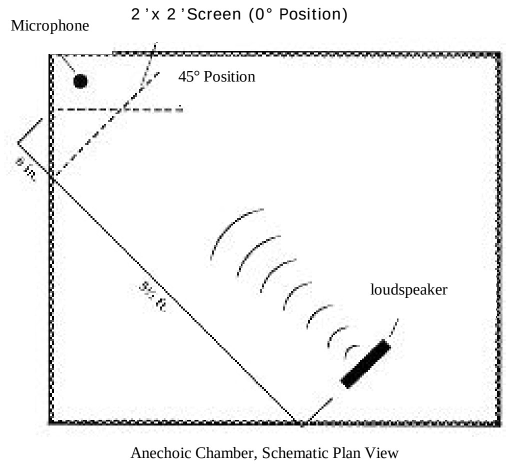


Figure B-1. 'Direct Path' test arrangement.

Figure B-l shows the setup. A loudspeaker out in the room radiates broad-band sound (containing all the frequencies of interest) toward a microphone located in a corner of the room. The sound at various frequencies received by the microphone in this situation is regarded as a baseline. Then, when a sample of perforated material is interposed between the loudspeaker and the microphone and the measurements are repeated, the differences, frequency by frequency, between the new measurement and the baseline measurement give an indication of the amount by which the sample attenuates the sound passing through. The measurements were made with two angles for the incident sound: perpendicular (0º) and 45º.

Figure B-2 shows the result of such a measurement with a solid (i.e., acoustically opaque) plastic panel in the sample position, to illustrate that there is very little “sound-spill” around the sample. In other words, the measurement accurately assesses the attenuation of the test sample, since there is no contamination by sound leaking around the sample.

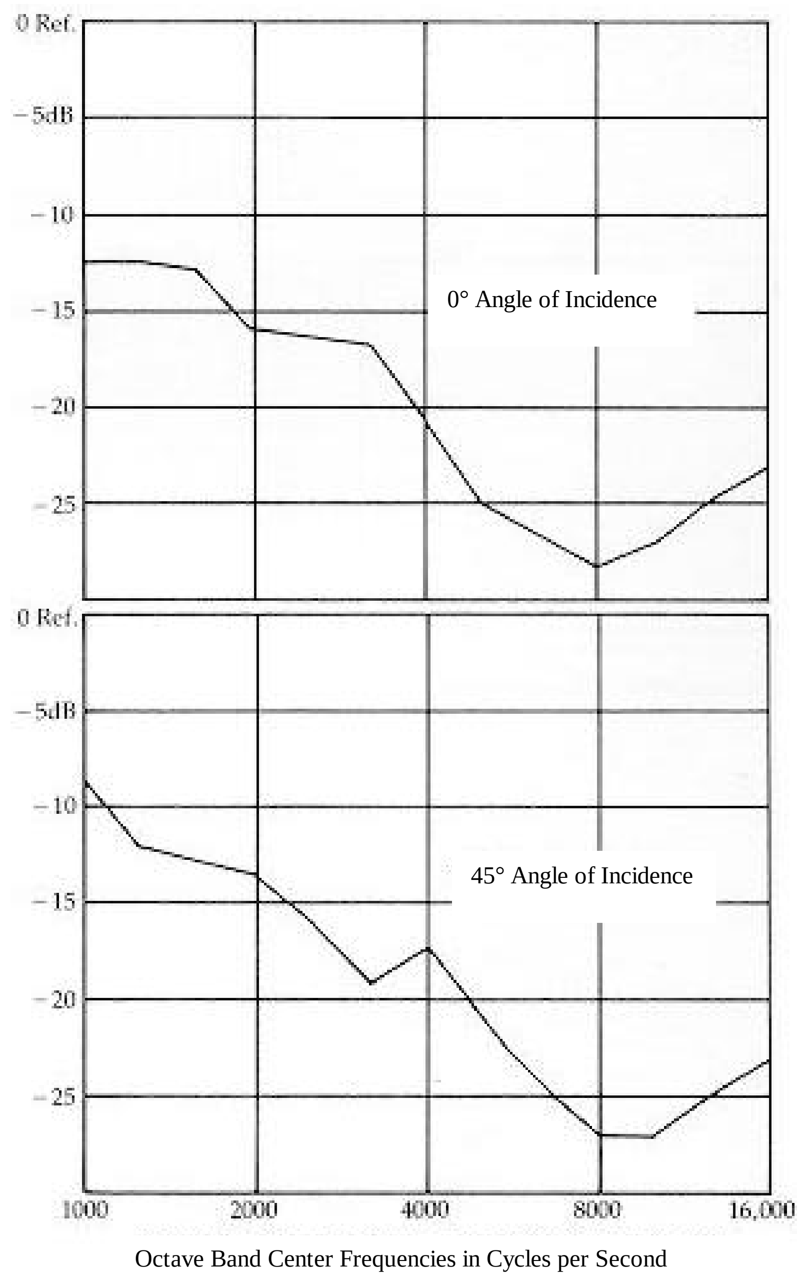


Figure B-2. Difference in sound levels, showing insertion loss of 2’ x 2’ plastic panel (i.e. ‘spill’)

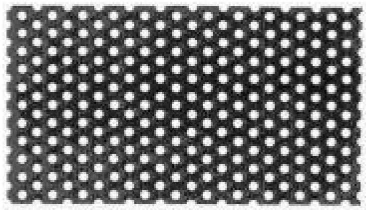


Figure B-3. Material G.

One sample that was tested at that time, Material "G," was perforated metal sheet having No. 8 perforations, 0.066" in diameter, 1/8" on center (73 holes per sq. in.). The sample was 1/16" thick and 2' x 2' square. It is shown full-scale in Figure B-3. The direct path test was used with 0° and 45° angles of incidence for the incoming sound. The test results are shown here in Figure B-4. The attenuation at a frequency of 10,000 Hz was 2 dB for 0° incidence, and 1.5 dB for 45°.

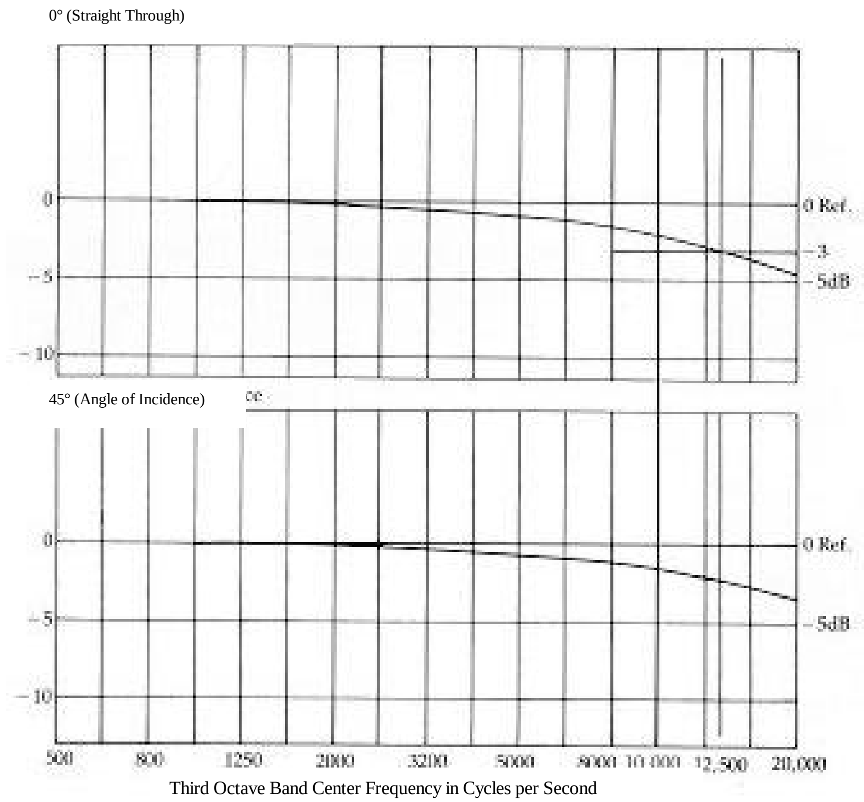


Figure b-4. Test results, insertion loss vs. frequency, Material G.

The results of ten such measurements are given in Figure B-5, where the values of the Transmission Index (TI = nd^2/ta^2) for the various test samples are plotted on the horizontal scale, while the corresponding values of sound attenuation for a frequency of 10,000 Hz are plotted on the vertical scale. The data point “G” is for the sample described above.

The curve shown on Figure B-5 was empirically fitted to the measured data points; it corresponds to the following formula:

A (10 kHz) = -22.56 log log TI + 0.008 $\sqrt{\text{TI}}$ + 13.79 dB.

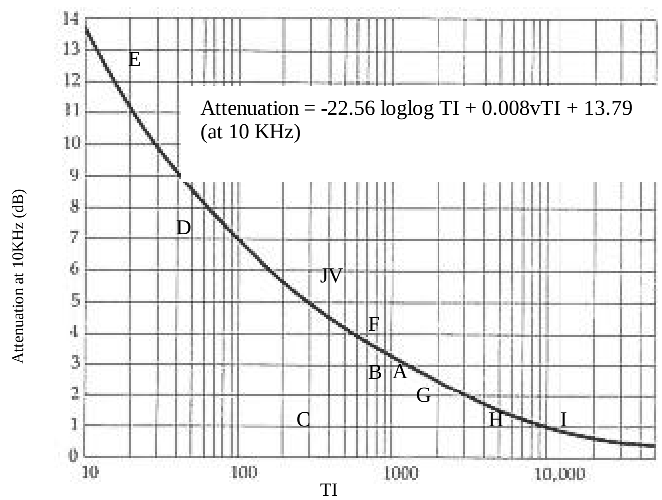


Figure B-5.

As for the formula for the Transmission Index, itself:

$$TI = nd^2/ta^2,$$

it was adopted in a slightly arbitrary manner. First, the four quantities involved are those (and only those) that *ought* to govern the sound attenuation through the sheet. And the particular formula chosen was the simplest combination of those quantities that yielded a monotonic function for the attenuation, with no peculiar, sudden changes in slope.

The slight arbitrariness of the formula is no disadvantage, since the TI, as a quantity, has no significance except as it is related to Figures 21, 22 and 23 in this booklet.

APPENDIX C: The Access Factor, and Access of the Sound Wave to the Sound Absorptive "Treatment" Lying Behind the Perforated Metal

In Section II.B of Part Two, we describe the use of the Access Factor as a means of accounting for the access that the sound wave has to the acoustical treatment lying behind the perforated metal, taking into account the attenuation suffered by the sound in passing through it.

If this treatment is a sound absorptive blanket, whose absorption coefficients are known for the various frequencies of interest, we assess the degradation of the sound absorptive capability of that blanket, due to the perforated metal covering it, by multiplying the absorption coefficients of the blanket by the Access Factor for the perforated metal at the various frequencies. In this case, we treat the Access Factor as a quantitative measure of how much of the incident sound energy gets through to encounter the blanket.

In principle, this procedure is not technically correct; the matter is much more complicated than that. If we were to carry out the correct procedure, it would require adding the mass impedance represented by the perforated metal screen to the impedance of whatever combination of materials lies behind it, and then recalculating, from the impedance of the whole ensemble, the net absorption coefficient presented to the incident sound. This is a complicated procedure, indeed, and one that (even so) does not necessarily give the right answer, since a number of questionable assumptions are involved.

It is believed that, for the range of perforated materials likely to be used in these applications, and for the types of sound absorptive treatments that they will be used to cover, the use of the Access Factor, as prescribed in Section B, will give answers with acceptable accuracy.

APPENDIX D: Work Sheets

This appendix contains clean work sheets corresponding to figures in the main report. They are intended to be photo-copied and used for the calculations required in the design of acoustical treatments using perforated metal. The completed sheets should be kept with the job file for the project in question.

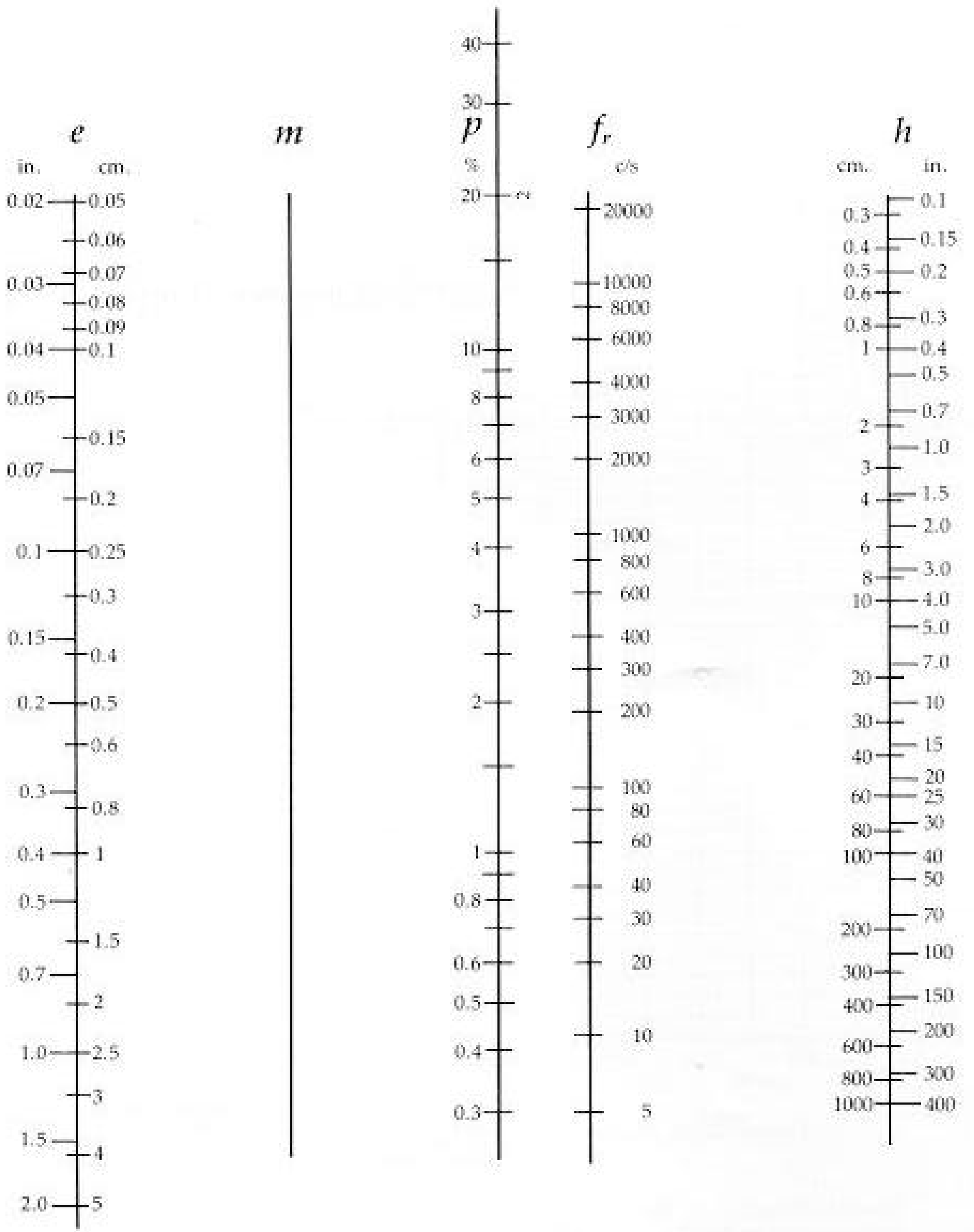
The work sheets are as follows:

Figure 17. Nomogram for Calculating the Resonance Frequency of a Tuned Resonant Sound Absorber.

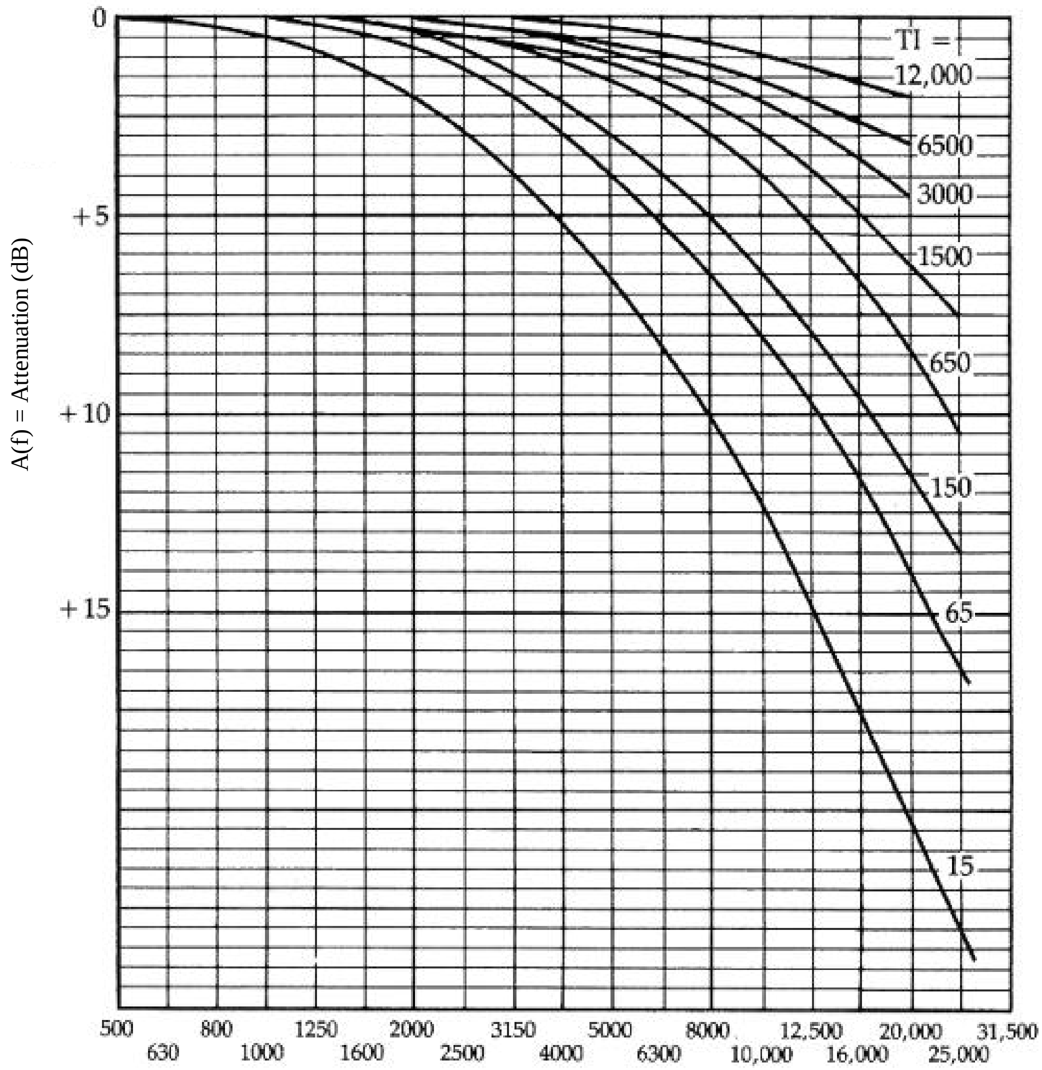
Figure 21. Sound Attenuation *VS* Frequency for Samples of Perforated Metal Having Different Values of Transparency Index (TI).

Figure 22. Curves Showing the Access Factor *VS* Frequency for the Samples of Perforated Metal Having Different Values of TI.

Figure 23. Nomogram for Calculating the Sound Attenuation and the Access Factor at a Frequency of 10,000 Hz (cps).

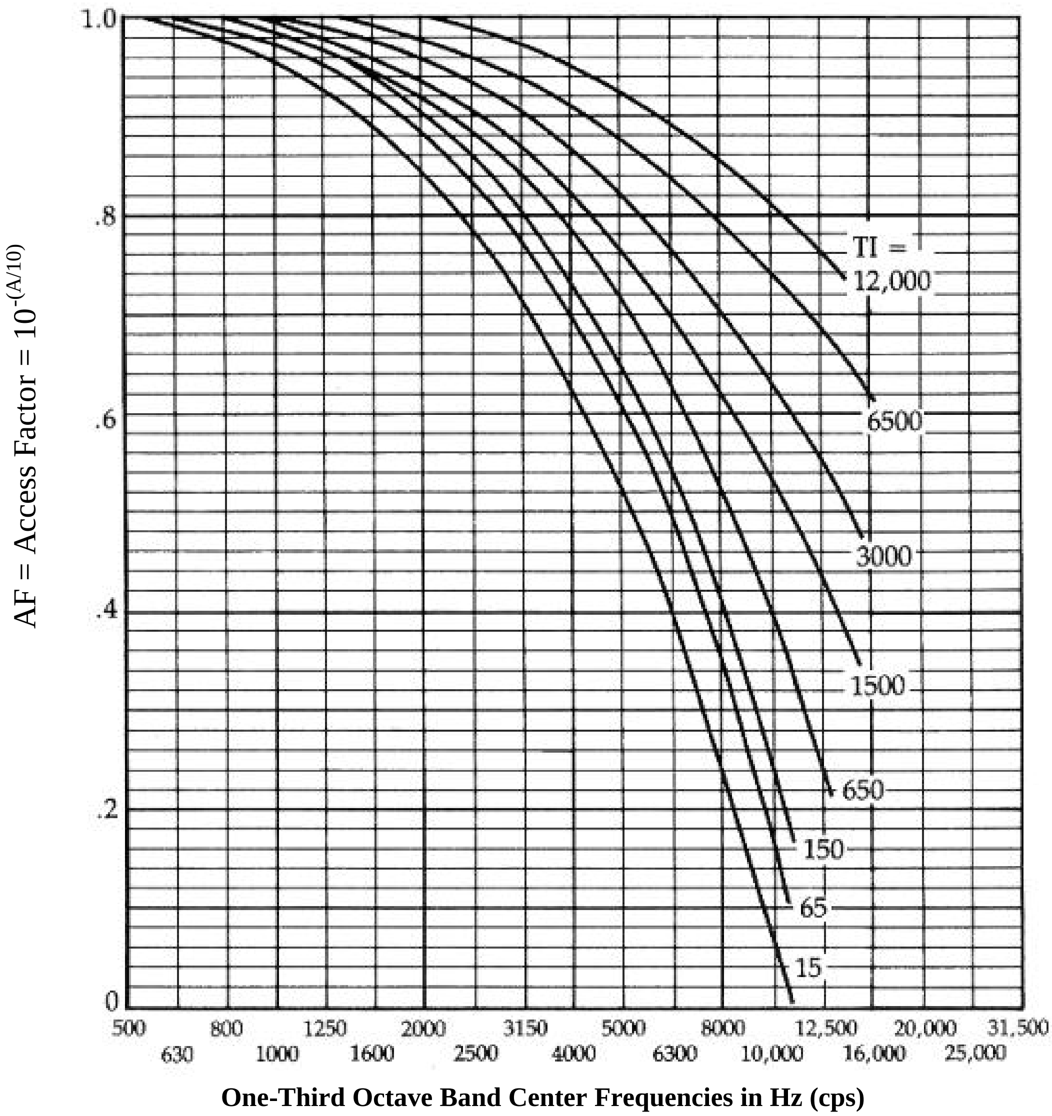


Nomogram for Calculating the Resonance Frequency of a Tuned Resonant Sound Absorber.

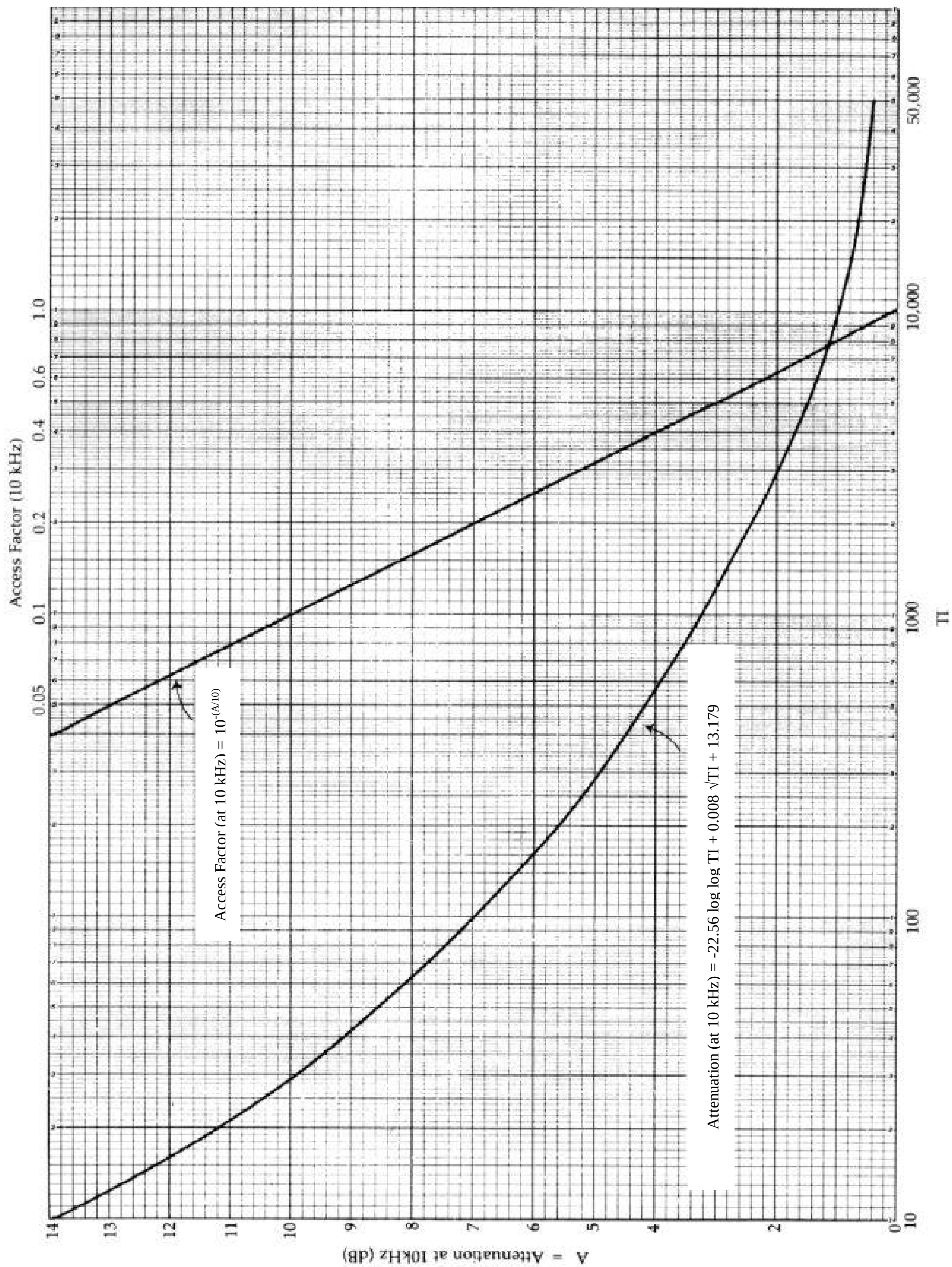


One-Third Octave Band Center Frequencies in Hz (cps)

Sound Attenuation vs Frequency for Samples of Perforated Metal Having Different Values of Transparency Index (TI).



Curves Showing the Access Factor vs Frequency for the Same Samples of Perforated Metal Having Different Values of Transparency Index.



Nomogram for Calculating the Sound Attentuation and the Access Factor at a Frequency of 10,000 Hz (cps).

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Dr. Schultz began his career as a professional musician, studying bassoon at the Eastman School of Music in Rochester, N.Y. Later he switched to Electrical Engineering at The University of Missouri, but ended up in acoustics, in which field he holds Master of Science and Doctor of Philosophy degrees from Harvard University.

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