ACOUSTIC CHAMBERS
Anechoic vs. Semi Anechoic Rooms.

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During the last 50 years, many anechoic chambers have been built all over the world. The most common construction is a rectangular room with sound absorbing wedges mounted on all six inner surfaces. The floor is often a net of high-tension wires criss-crossed and fastened in a solid frame. Furthermore, the building containing the room with all the sound absorbing elements is often specifically designed, including an elevated floor, to get the entrance to the chamber in the same level as the surrounding floor.

The question is; why do we put so much effort in constructing these kind of rooms simulating a non-existing environment. Is it really necessary? A fully standardized and specified measurement environment gives everybody the possibility to perform the same measurements. - It might not be the correct - but it is the same, and therefore comparable.

When measuring e.g. sound power one must integrate the sound pressure levels emitted from the whole surface at the device under test. This is not an easy task in a fully anechoic environment. Therefore, a reflecting plate is usually placed on the acoustic transparent wire net floor. Already here an unintentional error source could be introduced in the measuring chain. Is everybody using a plate with the same acoustical properties, and does everybody use an adequate size? What about the vibrations emitted trough the jig or object, are they radiated from the plate and thereby contributing to an erroneous measurement? A hard floor integrated in the room is the solution, not only for sound power measurements, but also for a variety of other applications.

Sound radiation from various objects, such as compressors, are often measured at fixed points at one or more positions defined and agreed by the manufacturer and/or customer. Often, these kinds of measurements are performed in sound absorbing rooms with can have very different properties from an acoustic point of view. Often these rooms are chosen because an anechoic solution is too costly. “Sound absorbing” rooms are useful, if initial comparison test and tolerances has been set between the manufacturer and customer, but since it is not a well-defined environment, results can’t be transferred and compared with other measurements from neither anechoic or other sound absorbing rooms.

Again a semi anechoic room would be of great advantage here, with emphasize on the amount of time and money saved when agreeing and determine the tolerances which can be accepted by both parties.

A complete free field simulation is seldom needed for practical measurements. The items often measured are a variety of machinery of all different kinds ranging from cars, punching machines, washing machines and everything else, which is always standing on a floor or other hard surface. Smaller items, such as telephones, household machineries etc. should be placed on a IEC standardized table which again is standing on a hard reflecting floor. Handheld tools and the like should be placed in a jig, simulating a person. The tool should be fitted in a appropriate distance over the hard floor.

In other words, we nearly always want to have an acoustic hard reflecting surface as floor in an anechoic chamber without the need for an acoustically transparent floor, such as a net. Using a net as the floor is also very impractical; if you loose a microphone cartridge you really have trouble to get it up again. You are also working on a net shaking the measuring gear and items under test, so it is very difficult working conditions. Besides, from being the best test environment for so many applications, a hard floor is also a very cost effective solution.

Fig.1. Fully anechoic room shown with door (left), and an elevated floor outside.

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1 A sound absorbing room, often referred to as a hemi-anechoic room, is a - often quite small room, covered with some kind of absorbing material on all 6 surfaces. However, these rooms are not offering a free-field environment, which by standard requires more than 99 % absorption on the absorptive surfaces.
On fig.1 a schematic drawing of a fully anechoic room is shown. It is seen how the room requires a quite large space, since the floor is covered with absorbing elements. The acoustically transparent floor must be raised to at least $\frac{1}{4}$ over the wedge tips at the lowest frequency of interest. On fig.2, the semi anechoic room is shown. The difference in complexity compared to it’s counterpart in fig. 1, is easily spotted. If a semi anechoic room is already installed and full anechoic conditions for some reason is needed, it is possible to insert a grid of single elements on the floor. If the elements are large as the well-known wedges are, it might be a difficult task since the length is in excess of one meter or so. Not only the handling, but also the storage of the elements is a potential problem, as a large number is often needed. However there are ways of avoiding such large elements, and thereby reducing price and troublesome installation.

On fig.3 the relationship between the particle velocity and sound pressure of a sound wave, is illustrated. There is no particle velocity close to the wall and in every multiple of every half wavelength from the wall. The absorption only takes place where particle velocity persists. Thus, the absorption material close to the wall is not very effective. If the absorbing material is placed at a point where the particle velocity is highest, it is possible to archive a very high absorption, with a relatively small amount of material. A plate e.g. 5 cm thick is placed 25 cm from the wall. This construction absorb just as well as if the whole space was filled up with absorbing material. To optimise the absorption, the flow resistance has to be adjusted to a specific value.

Above, is a common wedge, which is used in most anechoic rooms, which has been built until now. It is indeed a very good solution, which performs very well over a broad frequency range. However quite a lot of material is required, furthermore nearly half of the sound absorbing material is placed close to the wall and therefore less effective (see fig.3).

Above, another solution is shown. It is mounted 20-30 cm from the wall. It is therefore quite easy – and thus more economical to install pipes, electric cables etc., and this space might even be used to regulate the temperature in the room. Again, all of this should be of economical benefit, which hopefully can increase the use of acoustic measurements in the industry. Reflections and therefore less absorption, occurs because there is a sudden impedance change between the air and the mineral wool. All this would be avoided if the absorbing material on the tip of the wedge had a relatively low density increasing to higher density into the thicker part of the wedge. Some manufactures have tried to make such wedges, but it is not commonly found in practice. More than 20 years ago, the late professor L.Cremer in Berlin, suggested, instead of wedges, to use a series of absorbing cubes in different sizes and density. Prof. Cremer used thin steel wires to support the cubes. It was arranged in such a way that the smaller cubes with low density was the first layer looked from inside, next layer was larger cubes with higher densities and so forth.

Using separate cubes is an excellent solution, as there are no large plane surfaces. By turning the cubes in random directions the reflections will decrease even more resulting in a increased absorption.
Improved Anechoic Chamber Absorbents

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This paper discusses improving the low-frequency characteristics of anechoic chamber absorbent wedges. The problem of traditional wedge structures is that material properties required for good mid-frequency absorption imply reduced low-frequency absorption. The work presented here suggests applying the gradual transition principle also to the base of the wedge. Forming the base from two triangular wedges, one of which has a substantially higher flow resistance, extends the usable frequency downwards while not adding substantially to manufacturing complexity.

NEW WEDGE STRUCTURE

Anechoic chamber wedges consist typically of a narrowing tip and a homogeneous base. Earlier studies have indicated that the triangular shape commonly used is a good compromise between realisability and low-frequency absorption[1]. The problem of traditional wedge structures is that material properties required for good absorption at higher frequencies imply reduced low-frequency absorption. For good high-frequency absorption the flow resistance of the material should be relatively low, but such materials, regardless of the material thickness, cannot achieve good low-frequency absorption[2].

The work presented here suggests applying the gradual transition principle also to the base of the wedge. Using multi-layered bases improves low-frequency characteristics, but such absorbents are costly to manufacture. Forming the base from two triangular wedges, one of which has a substantially higher flow resistance, extends the usable frequency downwards (by about 10 %) while not adding substantially to manufacturing complexity.

FIGURE 1. Suggested wedge structure. Different colours indicate possible different materials for the wedge; the less dense part of the base may also be of the same material as the wedge tip.

FIGURE 2. Measured absorption coefficient of an experimental plastic foam wedge, total length 80 cm, base length 20 cm.

EXTENDING LOW-FREQUENCY ABSORPTION

Resonators are sometimes applied to extend the low frequency range of the anechoic chambers. There are two alternatives: a Helmholtz resonator where the air space behind the wedges forms the cavity and gaps have to be left between the wedges, and a membrane type resonator, where the wedges are attached to a backing plate to provide the mass of the resonator.

The Helmholtz resonator approach enables more independent design of the resonator function and the
wedge structure. Keeping the losses low enough requires using non-porous lining (sheet metal etc.) to form the resonator space. The denser base material of the dual-wedge structure provides a more solid framework for the resonator, thus keeping the additional losses moderate.

The other way, forming a membrane resonator consisting of a backing plate to which the wedges are attached, appears a reasonable choice for anechoic chamber structures. The mass of the wedges enables tuning the resonance frequency low enough (otherwise a problem of membrane resonators). The problem of this structure is that the velocity of the bending wave in the membrane is low (due to low stiffness and large mass). On the other hand, the wedges provide more damping to the bending waves than to the piston-like mass-spring-resonance, so the effect may in reality be not so alarming. Since it cannot be safely assumed that the incident sound would be a homogenous plane wave (see below) dividing the air space into smaller subvolumes is justified.

An important caveat in using resonators is that they provide improve impedance match in both directions: both for absorbing the sound inside the chamber and for leaking the sound from outside to the interior.

**CURVATURE OF THE SOUND FIELD**

An interesting observation made a few years ago by Don B. Keele is that in an anechoic chamber with realistic dimensions the assumption that the wave field at the lowest frequencies is planar is not well justified[4]. This also implies that standing wave tube measurements of reflection coefficient are not entirely reliable. As an anechoic chamber is typically specified by conforming to a certain deviation from 1/r-law in a measurement performed in the finished chamber, and wedges are typically designed using plane-wave impedance tube measurements, the difference between the two situations should be taken into account.

Taking the curvature of the sound field makes the use of resonators more interesting, since a resonator increases the degrees of freedom for low frequency design, which is important for achieving the necessary control for both phase angle and magnitude of the impedance.

In the mid frequencies the effects of oblique incidence can be ignored in the design, and the absorbent structure can be optimized for normal incidence, because with any reasonable source/microphone placement the direction of sound won't approach values where reflection coefficient would start increasing rapidly.

**NUMERICAL TECHNIQUES**

The simplest way of modeling an absorbent wedge is to treat it as a collection of finite number of finite-thickness layers, each of which is treated as a transmission line[4]. The front surface impedance is computed, starting from the solid backing wall. The reflection coefficient can be computed from this impedance. The transmission line model enables also the analysis of sound insulation and its change due to resonator effects.

The absorbent structure has several degrees of freedom in its design: characteristics of the two different materials, wedge geometry, and the resonator arrangement. This makes numerical optimization essential. Numerical techniques enable including more complex factors, like curvature of the wavefield and resulting non-constant low-frequency impedance. Important design considerations are then the choices of design criteria: whether to minimize low frequency limit or reflection coefficient at higher frequencies, and how to express the desired compromise between the criteria.

**ACKNOWLEDGMENTS**

The authors thank their colleagues, especially Dr Tapio Lahti from Akukon, for valuable discussions during the work.

**REFERENCES**


Source Dependency on Validating Anechoic Chambers

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ISO 3745 (ANSI S12.35) outlines the general procedure for qualification of anechoic and hemi-anechoic chambers and for the acoustic sources used in this effort. A proposed revision of the standard exists and is being evaluated. Application of the standard and its revision at the Georgia Institute of Technology have resulted in observations on the sources recommended by the standard and their impact on validation test results. There exists a procedural flaw with regard to source dependency which may make chamber validation problematic.

SOURCE SPECIFICATION

Procedures for the qualification of anechoic and hemi-anechoic rooms are outlined in the appendices of standard ISO 3745 (and similarly in ANSI S12.35) [1,2]. In a general sense, chamber validation is accomplished by moving a microphone in a straight line, away from a monopole source. The acoustic pressure at the traversing microphone should adhere closely to pressure predicted by free field conditions, i.e. the inverse square law,

\[ L_p(r) = 20 \log_{10}( \frac{a}{r} ) \]  

(1)

where \( L_p \) is the sound pressure level, \( r \) is the distance from the source, and \( a \) is a constant.

ISO 3745 recommends basic source design as a function of frequency, listed in Table 1.

<table>
<thead>
<tr>
<th>Source Description</th>
<th>Frequency Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Enclosed “Box” speaker</td>
<td>&lt;400 Hz</td>
</tr>
<tr>
<td>Face-to-face speakers; radiating “Sphere”</td>
<td>400-2000 Hz</td>
</tr>
<tr>
<td>Baffled speaker with “Pipe” opening</td>
<td>2-10 kHz</td>
</tr>
<tr>
<td>1.5m “Pipe” (proposed revision to 3745)</td>
<td>800-10000 Hz</td>
</tr>
</tbody>
</table>

The current standard specifies variation in directivity of no more than ± 1 dB, whereas the proposed revision to ISO 3745 specifies tolerance in source directivity as a function of frequency (some tolerance limits are shown in Figures 1 and 3). It should be noted that no source design is recommended above 10 kHz. Also, there is no allocation for the use of flat or floor-mounted sources in a hemi-anechoic room.

The sources used for the experimental work presented here are listed in Table 2.

<table>
<thead>
<tr>
<th>Source</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>“Sphere” 7.5cm diameter</td>
<td></td>
</tr>
<tr>
<td>“Sphere” 12.5cm diameter</td>
<td></td>
</tr>
<tr>
<td>“Sphere” 15cm diameter</td>
<td></td>
</tr>
<tr>
<td>“Pipe” 0.5m length, 15mm ID</td>
<td></td>
</tr>
<tr>
<td>“Pipe” 1.5m length, 6.3mm ID</td>
<td></td>
</tr>
</tbody>
</table>

SOURCE DEPENDENCY

Source directivity was measured at a distance of 1 m, making use of physical symmetry in the sources to reduce the required number of measurement points. Directivity was measured in the test chamber, which was the only available free field environment. This points to a significant flaw in the qualification procedure for a single-room facility: Room validation is dependent on the omni-directivity of the test source, while directivity measurement of the source is dependent on the free-field condition of the test room.

Measured directivity for three sources producing a pure tone at 500 Hz is shown in Figure 1. All sources appear to be omnidirectional at this frequency, based

FIGURE 1. Directivity at 500 Hz of three “sphere” sources, measured at a distance of 1 meter from the sources (dashed horizontal lines = specified tolerance).
on the tolerance specified by the proposed revision to ISO 3745 and the assumption that the directivity data are valid.

Chamber characteristics were measured by moving a microphone continuously along a straight traverse line away from a source. Measured acoustic pressure was then optimized to allow for variation in the source center location. Finally, theoretical behavior predicted by the inverse square law was subtracted from the data, resulting in deviation from free-field conditions.

Traverse data for the three “sphere” sources producing a pure tone at 500 Hz are shown in Figure 2. Repeatability, from source to source, of the traverse data is immediately striking. The chamber fails to qualify at a distance just greater than one meter from the source. However, source directivity was measured at one meter and therefore its validity is questionable in light of the traverse results. It follows that the traverse data is also then in question. The proposed ISO 3745 revision specifies directivity measurement at 1.5 meters, which would blur the validity of these measurements even further.

Directivity and traverse data for various sources producing a pure tone at 4000 Hz are shown in Figures 3 and 4. All sources are consistent in construction with the current or proposed standard. It can be seen that none of the sources qualify for chamber validity tests, based on directivity measurements within the chamber itself. Further, chamber qualification tests using these sources yields wildly varying results.

![Figure 2](image1.png)
**FIGURE 2.** 500 Hz traverse data for three “sphere” sources (dashed horizontal lines = specified tolerance).

![Figure 3](image2.png)
**FIGURE 3.** Directivity at 4000 Hz of three sources, measured at a distance of 1 meter from the sources (dashed horizontal lines = specified tolerance).

**CONCLUSION**

Neither ISO 3745, nor its proposed revision, provide a repeatable test method for anechoic chamber validation, particularly in the case of a new, single-room facility. The source recommendations of the standard do not necessarily yield defensible test data. It is the conclusion of this work that more emphasis should be placed on source directivity and its measurement. Additionally, the procedural flaw described above must be eliminated, potentially by third party source validation.

**REFERENCES**


Characterisation and Adjustment of the Reverberation Chamber at the Escuela Politécnica Superior de Gandia

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The aim of this work is the characterisation and adjustment of the reverberation chamber at the Escuela Politécnica Superior of Gandia. We have check the diffusion degree of the sound field in this room in order to guarantee the requirements for carried out of the measurements of absorption coefficients of materials used in acoustic conditioning and the determination of the sound power output of noise sources. Moreover we achieved a comparative study of the results obtained in this room to those from the another reverberation chamber fully operative since many years ago.

INTRODUCTION

This chamber is a room which shape is an irregular polyhedron. The chamber dimensions fulfill the norms related to the requirements for the realisation of the measurements of absorption coefficient of materials and power of the acoustic sources [1],[2].

GEOMETRIC PARAMETERS

In order to fulfill the conditions of the diffuse field this room was built with all opposite planes, including the walls, ceiling and floor that are non-parallel. Figure 1 shows the geometrical development of the chamber in a six-surface irregular polyhedron shape.
The volume of the room is \( V = (238 \pm 2) \text{ m}^3 \). The inner lateral surface is \( S = (236 \pm 2) \text{ m}^2 \).

![FIGURE ERRORRE. L'ARGOMENTO PARAMETRO È SCONOSCIUTO. Dimensions of the six surfaces of the chamber.](image)

ACOUSTIC PARAMETERS

Our chamber is adjusted at dimensions demanded for the measurements of the coefficient absorption. It also verified the ISO 354 recommendation [1] \( 1.9V^{\frac{3}{2}} > D, \) \( S_{\text{Total}} \approx 6V^{\frac{2}{3}} \) (236 m\(^2\) \approx 230 m\(^2\)), where \( D \) is the major diagonal and \( V \) is the volume room.
The global background noise level is 19.4 dBA within the acceptable limits.
The reverberation time in empty room verified the exigencies of RT minimum it also satisfied the requisites of minim absorption (see table 1).
The measures of SPL were performed in 140 measurement points in the room. The SPL maximum fluctuations occur at low frequencies, over 4 and 6 dB's. At upper frequencies, the fluctuations decreased, oscillating over 1.5 and 2.5 dB's.

UNCERTAINTY IN THE COEFFICIENT ABSORPTION AND SOURCE POWER MEASUREMENTS

The absorption coefficient of a material is given by

\[
\alpha_M = \alpha \left[ \frac{S}{S_M} \left( \frac{RT}{RT_1} - 1 \right) + 1 \right]
\]

where \( S \) is area in empty room and \( S_M \) is area of material , \( RT_1 \) in empty room and \( RT_o \) in room with wood material. The error associated is:

\[
\varepsilon(\alpha_M) = \frac{S}{S_M} \left( \frac{RT}{RT_1} - 1 \right) \left[ \frac{\varepsilon(\alpha)}{S} + \frac{\alpha}{S_M} \varepsilon(S) + \frac{\alpha}{S_M} \varepsilon(S_M) \right] + \varepsilon(\alpha)
\]

\[
+ \frac{\alpha S}{S_M(RT_1)} \left[ \frac{\varepsilon(\alpha)}{RT_o(\varepsilon(\alpha)+RT_o(\varepsilon(\alpha)))} \right]
\]

We present in table 2 the results of a sample of wood material.
Table 1. Second column shows the absorption coefficient with your error. In the next columns we compared RT and the absorption area in empty room with the minimum recommended by the norm.

<table>
<thead>
<tr>
<th>F(Hz)</th>
<th>(α ± ε(α)) × 10⁻³</th>
<th>RT(s) Norm</th>
<th>RT(s)</th>
<th>A (m²) Norm</th>
<th>A (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>9.0 ± 0.2</td>
<td>5</td>
<td>18.13</td>
<td>6.88</td>
<td>2.13</td>
</tr>
<tr>
<td>250</td>
<td>8.6 ± 0.2</td>
<td>5</td>
<td>19.08</td>
<td>6.88</td>
<td>2.13</td>
</tr>
<tr>
<td>500</td>
<td>10.9 ± 0.3</td>
<td>5</td>
<td>14.93</td>
<td>6.88</td>
<td>2.6</td>
</tr>
<tr>
<td>1000</td>
<td>14.0 ± 0.3</td>
<td>4.5</td>
<td>11.65</td>
<td>7.41</td>
<td>3.31</td>
</tr>
<tr>
<td>2000</td>
<td>20.0 ± 0.5</td>
<td>3.5</td>
<td>8.16</td>
<td>10.05</td>
<td>4.73</td>
</tr>
<tr>
<td>4000</td>
<td>32.6 ± 0.8</td>
<td>2</td>
<td>5.01</td>
<td>13.75</td>
<td>7.80</td>
</tr>
</tbody>
</table>

Table 2. Determination of the absorption coefficient of a sample of wood material

<table>
<thead>
<tr>
<th>F(Hz)</th>
<th>RT(s)</th>
<th>RT₁(s)</th>
<th>αᵢ and ε(αᵢ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>12.3</td>
<td>9.4</td>
<td>0.077 0.008</td>
</tr>
<tr>
<td>250</td>
<td>11.9</td>
<td>10.0</td>
<td>0.048 0.006</td>
</tr>
<tr>
<td>500</td>
<td>11.0</td>
<td>7.4</td>
<td>0.140 0.015</td>
</tr>
<tr>
<td>1000</td>
<td>9.9</td>
<td>5.2</td>
<td>0.323 0.030</td>
</tr>
<tr>
<td>2000</td>
<td>7.1</td>
<td>4.8</td>
<td>0.254 0.040</td>
</tr>
<tr>
<td>4000</td>
<td>3.7</td>
<td>3.0</td>
<td>0.218 0.070</td>
</tr>
</tbody>
</table>

We suppose in the last calculus an error of 0.1 s for the RT. In the below figure we can show the great influence of the relative error in the determination of absorption coefficient. Its evident the major effects in low frequencies.

![Graph showing relative errors due to imprecision in the reverberation time measure vs. the absorption coefficient of the sample](image)

**FIGURE 2:** Relative errors due to imprecision in the reverberation time measure vs. the absorption coefficient of the sample

The source power is given by (if we do not consider the effect of temperature and atmospheric pressure)

\[ L_w = L_p + 10\log\frac{A}{A_0} + 10\log\left[1 + \frac{S_c}{8V_f}\right] - 6 \text{ dB} \]

the uncertainty in the source power measures it is obtained by simple application of the theory of error propagation at last equation.

\[ \varepsilon(L_w) = \varepsilon(L_p) + \frac{10}{\ln 10} \left(\frac{\varepsilon(A)}{A} + \frac{cVc(S) + cSc(V)}{VcS + 8fV^2}\right) \]

**SOUND POWER MEASUREMENTS**

In figure 3 we have plotted both results obtained of the sound power of an acoustic source in the Gandia chamber and the ETSA chamber.

![Graph showing sound power level versus frequency](image)

**FIGURE 3.** Sound power level versus frequency. a) solid line correspond to our chamber b) dashed line correspond to other chamber.

**CONCLUSIONS**

The geometric parameters of the reverberation chamber verified the ISO 354 recommendation and the reverberation time in empty room verified the norm. The absorption areas are under the maximum recommended values.

The results obtained of the comparative measures of the source power of noise matched very well. But a lower frequencies is necessary to obtain a major diffusion. For this reason we have installing diffusers for increasing the diffusion in lower frequencies. In this mode we achieved the requisites demanded by the norm and our chamber is fully operative.

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