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# ROOM MODELING IN COMSOL USING FINITE ELEMENTS METHOD

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## ABSTRACT

The main purpose of this project is to develop a validated acoustic model that closely resembles the reverberation chamber of the Campus Sur at UPM, focusing on the low-frequency response. To achieve this, it is employed COMSOL Multiphysics software, which uses formulas that describe the physics of sound behavior through the Finite Elements Method (FEM), due to its effectiveness and widespread use in solving this type of problems.
Linked to this simulation process is the in-situ measurement phase, during which is obtained the actual response of the room for its modeling. To this end, they were distributed multiple positions within the room, separated from each other for measuring representative points of the room being measured to obtaining the room's impulse response and reverberation time, respectively. With the impulse measurement it is obtained the frequency response of the room for later comparison with the data from the simulation. In addition to the acoustic measurements of the room, several impedance estimation methods were tested in order to incorporate the boundaries information into the simulation. After completing these processes, the measured and simulated data are compared to validate the acoustic model.

**Keywords:** *Room acoustics, Finite Element Method (FEM), Reverberation chamber, Impedance estimation, Low-frequency sound absorption.*

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## 1. INTRODUCTION

When we delve into the field of room acoustics, there is a need to find solutions that can be applied in the real world to address low-frequency acoustic problems. Low-frequency sound absorption is required in many engineering applications where noise reduction is desirable. In architectural acoustics, this is normally the case when strong modes dominate the frequency response of the room in the frequency range of interest. The effectiveness of conventional porous materials depends on the relation between its thickness and the wavelength of the incident sound; hence, for low-frequency absorption thick materials are required. Due to the limited space normally available to install sound absorption solutions in real applications, there is an increasing interest on the design of effective solutions for low-frequency sound absorption with limited thickness based on acoustic metamaterials. To characterize the sound absorption of these systems, the standardized method defined in [1] in reverberant chamber is normally used. This assumes that the sound field inside the room is diffuse; however, low frequencies this is dominated by the room modes, and it cannot be considered diffuse. Moreover, large sample sizes are required, that can be costly and time consuming when the acoustic metamaterial is manufactured using 3-D printing techniques. Numerical simulations can help to reduce the cost of carrying out test in large reverberant chambers. An example can be found in [2], where a numerical model as a reverberant chamber using Finite Element Method (FEM) was carried out and validated with measured frequency responses. The room surfaces were modelled as rigid boundaries and the damping factor of the room was accounted for by a volumetric approach, that is, by using a complex speed of sound that depends on frequency and reverberation time. Murillo et al. presented in [3] a numerical model of a reverberant room was made in COMSOL Multiphysics® where the specific acoustic impedance of the room surface is obtained from the diffuse absorption coefficients obtained applying Sabine equation from measured reverberation times. The accuracy of the results in low frequencies can be compromised due to the lack of diffusiveness in the room. To overcome this limitation, Soares et al. proposed a Boundary Element Method (BEM) to calculate the low-frequency sound field of a small room [4]. This approach also requires knowing the low-frequency impedance of the room surfaces which is approximated as real-valued surface impedance from direct inversion using the available absorption coefficients and using optimization to compute complex-valued surface impedance. As numerical methods rely on the correct definition of the impedance of the room surfaces, and this is difficult to obtain for low frequencies, different estimation methods were recently applied such as eigenvalue-based inverse method for estimating locally reacting surface impedance [5] or a diffusion model [6].

In this work, a FEM model of the reverberant chamber of the Universidad Politécnica de Madrid (Campus Sur) is built in COMSOL Multiphysics®. A three-dimensional model of the room was made in Blender to closely replicate the actual dimensions and intricate geometry of the room. The frequency range of interest is from 30 Hz to 100 Hz. To estimate the absorption of the room surfaces, several methods are employed: direct measurements using the Microflown Technologies Impedance Gun; estimation using Sabine equation; an inverse method based on a diffusion-based impedance estimator; and a volumetric approach using complex-valued speed of sound. The frequency responses at different points in the chamber and the spatial average reverberation time obtained from the numerical simulations are compared with measurements.

This paper is organized as follows: Section 2 details the measurement and simulation procedures Section 3 outlines the different methods used for estimation the room surface sound absorption. Section 4 shows the room frequency response and reverberation time obtained by simulations and measurements. Section 5 discusses the findings of this research. Finally, the conclusions of the paper are presented in Section 6.

## 2. MEASUREMENTS AND SIMULATIONS

The reverberation chamber under study is shown in Figure 1. The chamber consists of two rooms connected by a large open window, and the walls forming the entire enclosure are non-parallel. Normally, the reverberation chamber is equipped with a series of diffusers hanging from the ceiling to provide a diffuse and uniform sound field throughout the room as far as possible. In this work, these diffusers were removed to simplify the room modelling. Measurements were conducted using a full-range loudspeaker (Figure 3) and multiple microphone positions distributed within the room. Additionally, both the temperature and relative humidity were measured to enhance the accuracy of subsequent simulations.

### 2.1 Measurement setup

The chamber is excited using an exponential frequency sweep signal covering a frequency range from 20 Hz to 200 Hz. A duration of 43.7 seconds was used to excite with enough energy the first room modes. That signal is emitted by the loudspeaker, and the resulting signal is captured by an omnidirectional microphone. The positions were chosen following the guidance given in [7] to minimize the effect of the spatial non-uniformity of reverberation time estimates at low frequencies due multiple modal decay rates, by selecting measurement positions based on knowledge of the mode shapes. The microphone positions are evenly spaced within the room to obtain a representative mapping of the acoustic field. The 12 microphone positions used during the test are specified in Table 1.

### 2.2 Simulation setup

The sound field inside the reverberation chamber is calculated using a Finite Element Method (FEM) in COMSOL Multiphysics®. The "Pressure Acoustics, Frequency Domain" interface is utilized, based on the application of wave theory to calculate the pressure variation during the propagation of acoustic waves in fluids. It is well-suited for frequency domain simulations with harmonic variations in the pressure field. This interface can be employed for linear acoustics, which is described by a scalar pressure variable. It includes domain conditions for modeling losses in a homogenized manner, for porous materials, as well as for losses in narrow regions. Additionally, the domain features support background incident acoustic fields and incorporate monopole sources. The attenuation behavior of plane acoustic waves can be entered as a user-defined quantity or characterized as environment-determined losses. The physics interface solves the Helmholtz equation in the frequency domain for specified frequencies, or as part of a modal analysis or eigenfrequency study.

**Figure 1**. Photography showing both sides of the reverberation chamber under study.


**Figure 2**. Full-range loudspeaker JBL EON315. Set-up used during on-axis driver measurements.

**Table 1.** Measurement receiver positions (in m).

| Id. | x | y | z |
|---|---|---|---|
| **M1** | 6.81 | 9.82 | 1.71 |
| **M2** | 6.99 | 4.94 | 0.76 |
| **M3** | 4.15 | 10.47 | 1.23 |
| **M4** | 3.12 | 3.75 | 1.52 |
| **M5** | 0.20 | 5.21 | 2.14 |
| **M6** | 2.00 | 10.26 | 0.76 |
| **M7** | 4.24 | 5.71 | 1.08 |
| **M8** | 2.23 | 3.22 | 2.18 |
| **M9** | 1.54 | 1.61 | 0.78 |
| **M10** | 2.43 | 9.36 | 0.00 |
| **M11** | 3.16 | 1.44 | 2.07 |
| **M12** | 3.29 | 8.42 | 0.88 |

Figure 3 shows the 3D model of the reverberant chamber used in COMSOL for the FEM calculations, including the chamber dimensions, coordinates reference and the mesh used for the calculations. The 3D model was made in Blender—a free and open-source 3D computer graphics software tool—and imported to COMSOL. Figure 4 shows the top view of the 3D model including the loudspeaker (S) and measurement positions (M). The positions selected to plot the frequency responses are numbered.

The mesh for the simulation elements has been defined by balancing the size of the elements with the desired computational accuracy, so as not to excessively burden the simulation's computational load. Consequently, the maximum element size is set to one-fifth of the wavelength corresponding to the highest frequency to be simulated (i.e. 100 Hz), while the minimum element size is defined as one-sixth of that same wavelength. The chosen element type for the mesh is a Free Tetrahedral mesh, as shown in Figure 3, which provides a good geometric approximation and adapts to any type of surface.

For the implementation of the sound source, a monopole point source was employed. Given that the frequencies of interest—form 30 Hz up to 100 Hz—are low, this approach is appropriate since sound radiation at such frequencies can be considered omnidirectional. The source sound power was adjusted as explained in Section 4.

Environmental measurements of temperature and relative humidity are used to determine the effects on the speed of sound. A baseline speed of 331.45 m/s is assumed for dry air at 0 °C, and percentage modifiers based on the environmental data are applied to account for changes in the speed of sound. Temperature affects the movement of particles, while variations in relative humidity alter the water vapor pressure, both of which modify the speed of sound [8]. The resulting sound speed calculated considering the aforementioned conditions—with a temperature of 24 °C and 45% relative humidity—is 346.53 m/s.

## 3. ABSORPTION COEFFICIENT ESTIMATION

Once the in-situ measurements, the room geometry, and its implementation in the simulation software are available, the next step in the acoustic room modeling process is the accurate estimation of the absorption coefficients of the surfaces comprising the room. Accordingly, several methods are applied to approximate the sound absorption in the room: in-situ measurements of surface acoustic impedance using a Microflown impedance gun, an estimation of the mean absorption coefficient using Sabine equation, an inverse method based on the acoustic diffusion equation, and a volumetric approach using a complex equation for calculating the speed of sound in terms of the frequency and the measured reverberation time.

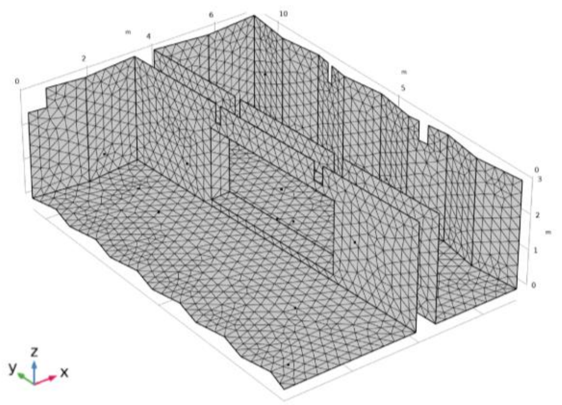

**Figure 3**. View of the 3D model representing the reverberation chamber under study (with mesh).

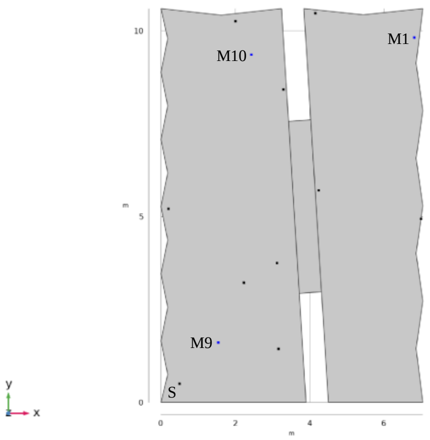


**Figure 4**. Top view of the reverberation chamber. The nearest point to the origin corresponds to the pulsating sphere simulating the loudspeaker, located at S (0.5, 0.5, 0.2) [m]. The remaining points, represent the microphone positions.

It should be emphasized that, on one hand, the impedance gun method has low-frequency limitations—arising from both the distance between the probe and the loudspeaker and the dimensions of the latter, which can result in insufficient low-frequency radiation, sample size (edge diffraction), etc. This method was found to be comparable with the impedance tube method for frequencies above 500 Hz [9]—and on the other hand, Sabine equation, which is a method based on statistical theory and therefore valid assuming diffuse field [10] (for frequencies above the Schroeder frequency, which is 285.7 Hz). The third method employed is an inverse method based on the diffusion equation [11], which is also not accurate when predicting the sound field for frequencies below the Schroeder frequency, while the fourth approach [2] tries to account for the energy losses due to absorption in the chamber surfaces using a complex form of the speed of sound that depends inversely on the frequency and reverberation time.

### 3.1 Impedance Gun

The Impedance Gun is a technology developed by Microflown Technologies for in situ measurement of impedance and absorption coefficients. In this study, it is employed to assess the sound absorption of the surfaces within the room, namely the floor and walls, assuming that the ceiling is composed of the same material as the walls. The impedance gun is pointed toward the material to be measured (see Figure 5). A loudspeaker is placed on one side of the gun to produce a sound field (white noise) that reach the material with normal incidence, and sound pressure and particle velocity sensors placed in the opposite side of the gun are located very close to the material to be characterized. A first measurement pointing to the free space is required for system calibration. The mathematical model used to obtain the sound absorption is the Mirror Source Model. Although the results obtained with the gun are considered reliable from 300 Hz up to 10 kHz as shown in the manufacturer specifications, as indicated by the manufacturer, they are deemed potentially useful for obtaining estimations applicable to low-frequency simulations. Figure 6 shows the absorption coefficients obtained with the impedance gun for the walls and floor between 31.5 Hz and 100 Hz. Values between 0.3 and 0.5 are obtained for all frequencies and surfaces, except for the wall at 31.5 and 40 Hz, with lower values. Considering that the room surfaces have low porosity they are expected to be highly reflective, and hence the obtained data seems not reliable. Therefore, these results are disregarded.

### 3.2 Sabine's equation

Using the impulse response measured at each microphone position, a filtering process is applied employing second-order Butterworth band-pass filters, with a one-third octave bandwidth centered on the central frequencies of the one-third octave ranging from 31.5 Hz to 100 Hz. The reverberation time is then obtained from these signals using the integrated impulse response method [12].

**Figure 5**. Microflown Technologies impedance gun. Set-up used during measurements for the estimation of the wall absorption coefficient.

Subsequently, the absorption coefficients per one-third octave band are determined using Sabine's equation [10].

$$\bar{\alpha} = \frac{0.161 \cdot V}{S \cdot TR} \tag{1}$$

where *TR* is the reverberation time, *V* is the volume (203.11 m³) and *S* is the surface area (301.34 m²) of the reverberation chamber.

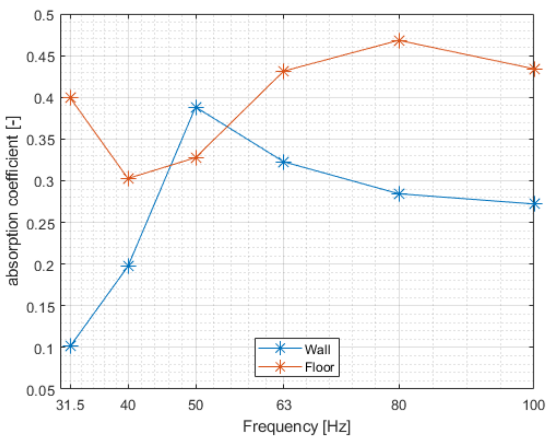


**Figure 6**. Absorption coefficient obtained with the impedance gun for the walls and floor.

Although the estimation provided by this equation is based on statistical acoustic theory, it can serve as an initial approximation to the absorption. This theory assumes that the sound field within the room is diffuse, with energy distributed uniformly in all positions. Under this assumption, Sabine's equation can be used for estimating the reverberation time as a function of the room's volume and the total absorption of its surfaces. It is important to note that Sabine's equation is most accurate in rooms with low absorption and highly diffuse sound fields. Considering that the frequencies we are working hinder the fulfillment of these conditions—since achieving a diffuse field below 100 Hz is challenging—we assume that the conditions are sufficiently acceptable to provide a preliminary approximation to the solution of the problem at hand.

### 3.3 Acoustic diffusion model

The absorption coefficients are obtained using a mathematical method based on a diffusion equation, which forms the diffusion-based impedance estimator method proposed by Prinn et al. [11]. This article presents an acoustic estimation of the low-frequency surface impedance of the same reverberation chamber considered in this study. In that case, an inverse method is followed to derive the surface sound absorption coefficients using the diffusion equation method and measured impulse responses.
For future simulations, it would be of particular interest to employ impedance estimation via the Helmholtz-based model [11], which is based on eigenfrequencies and is also developed in the article that underpins the absorption coefficient values we use. As this method is based on the modal response of the room, it is expected to be more accurate in low frequencies.

### 3.4 Volumetric approach

Additionally, simulations of the room response were carried out by implementing sound absorption due to air friction losses, rather than modeling the absorption on the room's surfaces [2]. This approach was adopted to provide additional data for comparison and to draw further conclusions. Given the reverberation time, the loss factor can be determined from the room's volume. In principle, losses due to these effects are generally small for low frequencies, but in situations where the surfaces of a room are highly reflective, viscous (and thermal) losses may occur that constitute a non-negligible fraction of the acoustic energy transported. The complex sound speed term proposed in [2] can be obtained as follows:

$$c = c_0 \left(1 + 0.5i \frac{2.2}{f \cdot RT}\right) \tag{2}$$

where $c_0$ is the sound speed, *f* represents the frequency and *RT* the reverberation time.

## 4. RESULTS

The sound power of the point source used in the FEM model is adjusted—ensuring that measured and simulated signals are directly comparable— as follows: an impulse response was measured at 18 cm from the microphone positioned along the loudspeaker's axis, as shown in Figure 5, and a corresponding simulation was conducted in COMSOL Multiphysics® obtaining the impulse response at the same position using an initial sound power of 0.1 W. The RMS power of the measured and simulated impulse response are compared, and accordingly, a correction is applied to the source power of the point source of the FEM model to obtain the same value as measured. In this case, a sound power of $2{\cdot}10^{-4}$ W was required.

Once the impulse responses at different positions are obtained, the reverberation time is determined using the procedure outlined in the ISO 3382-2:2008 [12], known as the integrated impulse response method. This method involves generating a decay curve of the sound energy by performing the reverse integration of the squared impulse response, as shown in Eq. (3).

$$E(t) = \int_t^{\infty} p^2(\tau)d\tau = \int_0^{\infty} p^2(\tau)d\tau - \int_0^{t} p^2(\tau)d\tau \qquad (3)$$

where $E$ represents the energy decay curve as a function of time, $t$, and $p$ denotes the acoustic pressure.

In Figure 7, a comparison is shown between the frequency response of the in situ measured signal and those obtained through simulation—using the different methods explained in the Section 3—for the microphone position M1, located at the corner opposite to the sound source position.

While the results are similar to those obtained with the other two methods, over nearly the entire frequency range the pressure peaks obtained by the volumetric approach are higher than those of the measured signal, suggesting that the sound absorption predicted may be underestimated. The results for the methods employing Sabine equation and the diffusion-based approach are almost identical and follow a similar trend as those obtained by the volumetric approach, but the amplitude of the first peaks of the frequency response are closer to those of the measurements. For all the predictions, the frequency peaks are slightly shifted to higher frequencies that those of the measurements.

Figure 8 shows that the absorption coefficients derived from Sabine's equation and those from the diffusion-based model (taken from [11]) are similar. As expected, the frequency responses obtained from the simulations are very similar in both cases; therefore, only the simulations performed using the latter model are presented in subsequent results. Figure 9 shows the frequency response derived from the measured impulse response and those obtained from simulations based on the diffusion model.

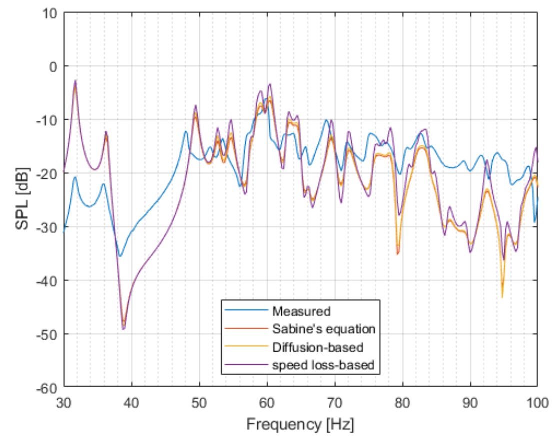


**Figure 7**. Comparison between frequency response measured and predicted using the Sabine's equation, the acoustic diffusion model and volumetric approach at the measurement position M1.

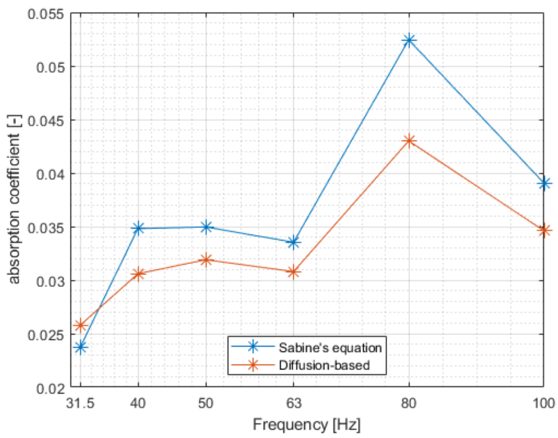


**Figure 8**. Comparison between Sabine-based and diffusion-based absorption coefficient.

Figure 9(a) corresponds to the measurement at microphone position M9, which is the closest to the source among all positions. Figure 9(b) corresponds to a position located at the opposite end within the same side of the chamber as the sound source, one of the positions of greatest interest, as will be discussed below. When comparing the frequency responses for all measured and simulated points, it is evident that the simulated responses are shifted by between 1 Hz and 2 Hz, which is particularly notable at low frequencies.

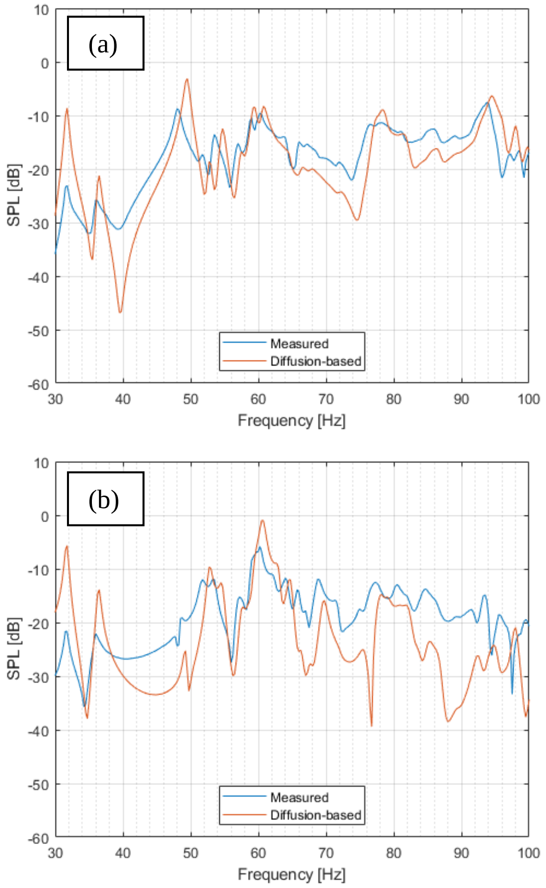


**Figure 9**. Comparison of measured and predicted frequency response functions using the diffusion-based model. (a) Measurement position M9. (b) Measurement position M10.

Figure 10 displays the spatial average reverberation time for both the measured and the simulated impulse responses using the acoustic diffusion method for each one-third octave band. The standard deviation for both simulations and measurements is also included. It can be observed that the simulated reverberation time is lower than the measured time for most of the frequency bands, which appears to indicate that the sound absorption obtained using the acoustic diffusion equation method is too high.

## 5. DISCUSSION

In considering the possible causes for this discrepancy between the resonance frequencies obtained from measurements and simulations, the speed of sound could be a contributing factor, although this aspect was considered in this study. Another factor could be the definition of the room geometry; although it is fairly accurate, there remain elements that deviate from reality. Thirdly, it should not be overlooked that the absorption of the room surfaces is one of the most critical factors in the simulation, indicating that further work is needed in its estimation. A more accurate source model including the cone membrane and the actual driver's velocity could be used instead of a pulsating sphere.

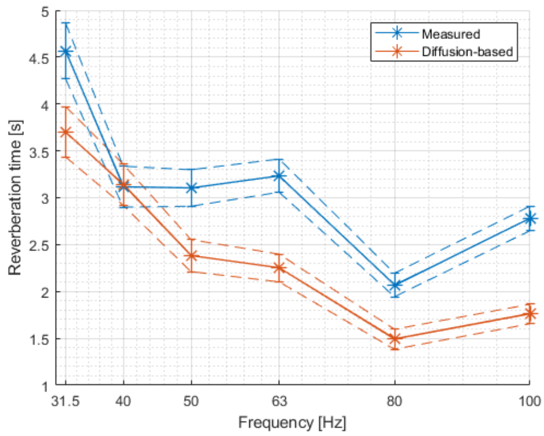


**Figure 10**. Measured and simulated spatial average reverberation time (-) with standard deviation (--).

Figures 11 and 12 show the mode shapes or two modes with close frequencies: 49.25 Hz and 49.46 Hz. Relating These results can be related with the frequency responses shown in Figure 10 for points M9 and M10. The point M9, located near the source, exhibits a high peak in the frequency response at 49 Hz (see Figure 9.a) probably due to the overlap of the two modes mentioned above, as point M9 is placed in a pressure maximum for both mode shapes. In contrast, the frequency response of point M10 has a small peak at 49 Hz (see Figure 9.b) in the frequency response. In this case a pressure maximum for the mode shape in Figure 11 overlaps with that shown in Figure 12 of a neighboring mode, which exhibits a minimum, resulting in a valley in the frequency response. It is curious how the room's geometry induces a pressure minimum over a rather extensive area for this particular mode.

Considering the obtained results, the work carried out is encouraging, as—even though it is evident that the model still requires adjustment—the simulation results are sufficiently close to suggest that we are moving in the right direction.

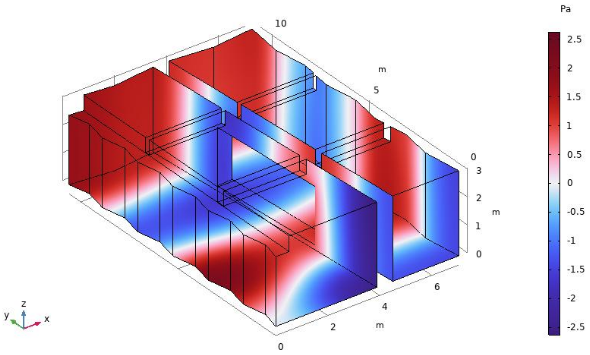


**Figure 11**. Mode shape at 49.25 Hz.

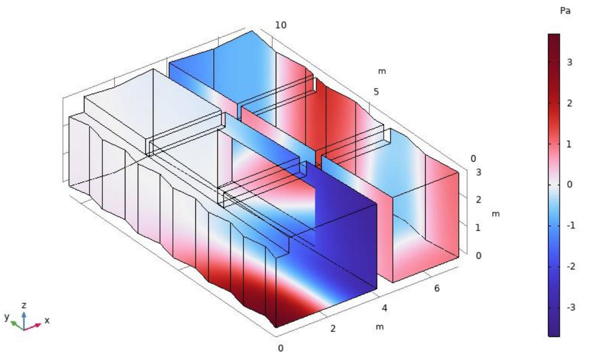


**Figure 12**. Mode shape at 49.46 Hz.

## 6. CONCLUSIONS

A Finite Element Model of the UPM reverberant chamber is made in COMSOL Multiphysics®. Frequency responses at several points are computed in the frequency range from 30 Hz to 100 Hz. Four different methods are used to estimate the room surfaces absorption coefficient: measurement with the Microflown Technologies impedance gun, estimation using Sabine equation and an inverse method based on the acoustic diffusion equation and a volumetric approach. The results obtained with the first method are discarded it turned out that the impedance gun method is limited to higher frequencies. Similar results are obtained with the other three methods, but the one based on the acoustic diffusion equation provide slightly more accurate results. The frequency of the modes is shifted by around 2 Hz when comparing simulations with measurements. The predicted reverberation time is significantly lower than that measured for frequencies above 50 Hz, indicating the sound absorption is overpredicted. Improvements can be applied in future studies, such as refinement of the model geometry and a more detailed model of the sound source. To improve the estimation of the sound absorption, a Helmholtz-based impedance estimator could be implemented.

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