Megaron
https://megaron.yildiz.edu.tr - https://megaronjournal.com
DOI: https://doi.org/10.14744/megaron.2025.94580

Article

Variation of the room modes and impulse response according to surface absorption properties in a non-rectangular room

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ARTICLE INFO

Article history
Received: 13 August 2024
Revised: 11 March 2025
Accepted: 14 March 2025

Key words:
ANSYS modal acoustic; Finite element method (FEM); ISO 18233; ISO 3382; room impulse response.

ABSTRACT

Determination of room mode frequencies and shapes in rectangular rooms can be done using calculation methods. Simple calculations cannot be made for non-rectangular rooms and the room must be simulated. In this study, the effects of 4 different acoustic designs on room modes and room responses were investigated using a fan-shaped room with a volume of 85 m^3. The absorption coefficients of the acoustic materials used were calculated based on the reverberation time values obtained as a result of field measurements. T30, EDT, C80 and room response measurements made in the field in accordance with ISO 3382 and ISO 18233 standards and ANSYS modal Data was obtained as a result of comparing the room modes and shapes found with the Finite Element Method (FEM) using the acoustic module. It has been found that if the absorption coefficients of the acoustic devices to be used in the room are greater than 0.5, the room mode shapes and frequencies in the relevant frequency band change, and as a result, the room response becomes smoother. It has been observed that the peaks in the room response in a certain frequency band can only be smoothed out with sound-absorbing materials with an absorption coefficient greater than 0.5 in that frequency band. It has been evaluated that the absorbers, which will be effective in the frequency bands left by the room modes in the room response, will pull these bottom regions lower.

Cite this article as: Tıraş, M., & Ilgürel, M. N. (2025). Variation of the room modes and impulse response according to surface absorption properties in a non-rectangular room. Megaron, 20(1):93–104.

INTRODUCTION

The acoustic parameters to be determined in the arrangement of room acoustics are related to the room volume. Revealing the standing wave shapes of room resonances in small rooms determines at what frequency we can hear sounds depending on the location in the room. The paths of sound waves in the lower frequency region and the contribution of the room to the sounds at this frequency constitute a situation that needs to be examined. Analyzing sounds in the lower frequency region using the normal ray tracing method does not give accurate results (Bai, 1992). Because in these frequency areas where a diffuse sound field does not occur, sounds do not move smoothly and linearly (Beaton & Xiang, 2017). It is important to develop an acoustic design approach that aims to reach optimum

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values determined by taking into account the function of the room (Tıraş & Akdağ, 2024).

Low frequency room modes can act as a filter and can also cause the sound character of the music source to be perceived differently (Hikichi & Miyoshi, 2006; Kelle & Yılmaz Demirkale, 2022). When sound occurs in the frequency regions within the room mode frequency and bandwidth, room resonances take effect and create standing waves (Kleiner & Tichy, 2014). In small rooms, widely spaced modal frequencies in the lower frequency regions that are audible to the human ear, can color the sound in a noticeable, often undesirable way (Rossing & Fletcher, 2004). This coloration can create significant acoustic problems in listening rooms, recording rooms, and music study classes. Considerable effort has been devoted to developing design guidelines and approaches to mitigate these impacts (Boner, 1942; Gilford, 1959; Louden, 1971; Volkmann, 1942).

Small rooms with strong modal behavior will not have a diffuse sound field. In rooms with volumes between 30-200 m^3, the space and time distribution of sound energy will be irregular due to the lack of sound field diffuseness (Prato et al., 2016). In lower frequency regions where room modes occur at wide intervals, the decay curves of sound energy will also vary (Bistafa & Morrissey, 2003). This situation creates problems in determining room acoustic parameters. While determining the acoustic properties of rooms, equations and approaches have been developed considering that the sound inside the room creates a diffuse sound field (Qu et al., 2023).

In small volumes where a diffuse sound field does not occur, it may be useful to determine the available modes by using the room response curve (Das & Abel, 2022). Modal decay times of these modes can be calculated using quality factor and mode frequency data (Kleiner & Tichy, 2014). The room response curve refers to the sequence of sound signals that are emitted from the pulse sound source in the sound field and received at one point. In room response measurement, a sine sweep or MLS signal is usually used as the emitted signal and is generated by the source. (Lim et al., 2016; Wang et al., 2020).

Analytical solutions exist to predict modal frequencies in rooms with parallel walls based on solutions of the wave equation (Jian et al., 2022). The finite element method and the calculation method can be used to determine the room sound field, as well as it can be determined by field measurements (Čurović et al., 2024). Wave-based simulation methods, especially methods that can be formulated to work on unstructured networks such as the finite element method and the boundary element method, can use much more detailed geometric models and thus reduce the associated uncertainty (Bai, 1992; Ekmen et al., 2021). It seems that the finite element method (FEM) is a useful tool for the frequency region below 500 Hz in determining the room acoustic parameters. (Jiang et al., 2011).The use of the finite element method is preferred in the solution of complex geometries (Yoshida et al., 2021). Computer programs are used in the use of the finite element method (Mehra et al., 2012; Qu et al., 2023). With the ANSYS modal analysis module, the modal shapes and levels that will occur in the room can be determined on a frequency basis (Svensson, 2020).

Simulation programs are used in studies on room acoustics (Zhang et al., 2021; Zhu et al., 2022). While simulation programs using the ray tracing method find the volume acoustic parameters close to field measurements at 250 Hz and above, they deviate from these estimates in the 125 Hz and 63 Hz regions (Svensson, 2020). Ray tracing simulations use sound energy without taking the phase information of sound "rays" in consideration. While this provides good approximations, especially at high frequencies where the sound field is considered dispersed, it fails in predicting low frequency phenomena such as standing waves, phase cancellation and diffraction. Commercial FEM software such as Ansys is well suited for these calculations (Svensson, 2020).

It is seen that finite element method (FEM) and simulation programs are used to see the impact areas and quantities of room modes (Jiang et al., 2011; Yoshida et al., 2021). The two most important variables that affect room modes, are the character of the sound source and the absorption factor (Bistafa & Morrissey, 2003). The critical frequency concept introduced by Schroder shows us the limit value of the lower frequency region where room modes may cause problems (Jian et al., 2022). It is seen that sounds at frequencies below this limit value can be heard more strongly by being affected by room resonances. While the coefficient of 4000 was used in the formula published by Schroder in 1954, it was replaced with the coefficient of 2000 in 1964. It is stated mathematically that the use of the 4000 coefficient is more accurate for rooms with high absorbency (Dance & Van Buuren, 2013).

The aim of this study is to determine the areas of use in smoothing the room response by observing the changes in room mode frequencies according to the absorption characteristics of the type of the sound absorbing material used at the boundaries of the room. The main goal is to create an acoustic design guide to obtain an acceptable flat room response for the lower frequency region of the spectrum using the data obtained from the study.

In this study, the effect of absorbing materials on room modes and room response was investigated by using 4 different room versions, observing the change of room modes according to surface absorbance and the improvements in the room response curves that are resulting from this change. By determining the sound absorption coefficients of sound-absorbing materials according to frequency, their impact areas on room response and room modes were determined according to the degree of absorption.

In this study, sound dispersive surfaces were not used. The theory of diffuse sound field is not valid at frequencies below the Schroder frequency (Vorlander, 2013). Since sound dispersive surfaces were not used and the scattering coefficients of the materials used were less than 0.1 below 500 Hz, the scattering coefficient was not taken into account.

EXPERIMENTAL METHOD

In the experiments conducted in a fan-shaped room with an area of 30.78 m^2 and a volume of 86.8 m^3, 4 different room designs were used. As a result of the field measurements, T30, EDT, C80 and room response data were obtained.

In this room, which is planned to be used as a musical instrument study class, the data were evaluated comparatively as a result of the measurements taken from the position of the piano player in accordance with the piano position. Modal analysis was carried out using ANSYS 2023 R1 program for 4 different version room modal analysis. The areas of materials and the versions in which they are used are shown in Table 1.

Table 1. Use and area of absorbent materials

	Area (m^2)	V1*	V2*	V3*	V4*
Curtain	15.5		✓		✓
7 Panel	9.1			✓	✓
Carpet	29.5	✓	✓	✓	✓
Bulk curtain	3.5	✓		✓	

*V1(Curtain bulk), V2(Curtain), V3 (7 panels), V4 (7 panels + curtain).

The effect of absorber elements on room modes was investigated, and mode frequency and shape changes were observed. The modal effect created by the curtains, carpets and 7 different panels used in the room was evaluated and compared with the field measurement results and discussed. The methodological steps followed by the research are shown in Figure 1.

In the first stage, 4 different room designs were determined. For this fan-shaped room, room modes and shapes were found using the ANSYS modal acoustic module. Field measurements were carried out in the 4 room versions shown in Figure 2, and T30, EDT, C80 and room response results were obtained. By comparing reverberation time measurement results, absorption coefficients of the curtain and panels were calculated. At the same time, using the formula in the literature for the panels, the frequency with the most effective absorption was found. In the discussion section, the changes according to panel and curtain use on reverberation and room response were evaluated. In the conclusion section, the data obtained as a result of the discussion are presented.

Field Measurements

Measurements were carried out in accordance with the ISO 3382-2 standard to determine the current acoustic conditions of the instrument working class. 2 source and 3 receiver points were selected (Figure 3), the source positions were positioned 150 cm above the ground, and the receiver points were positioned 120 cm above the ground (TS EN ISO 3382-2, 2008).

A studio speaker, Behringer ECM 8000 measurement microphone, microphone tripod, Arta software and microphone calibrator were used in the measurements. Among the ISO 3382 acoustic parameters, T30, EDT and C80, room response measurement was made for point

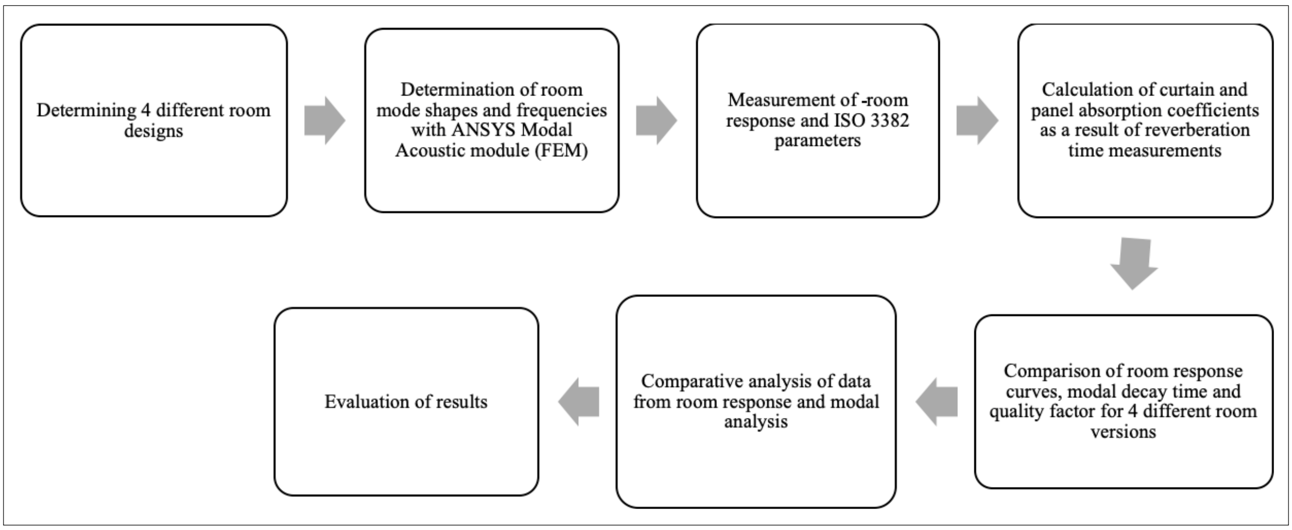

Figure 1. Method steps followed.

Figure 2. 4 different room designs and photos.

A1 in accordance with the TS EN ISO 18233-SS standard (TSE EN ISO 18233, 2010). The microphone positions and distance between the microphones were determined to be 1.2 meters using the formula for the closest distance between two microphones in ISO 3382. Sweep signal was used in room response measurements. Sweep signal has been shown to provide more accurate results in room response measurement.(Lim et al., 2016; Prato et al., 2016).

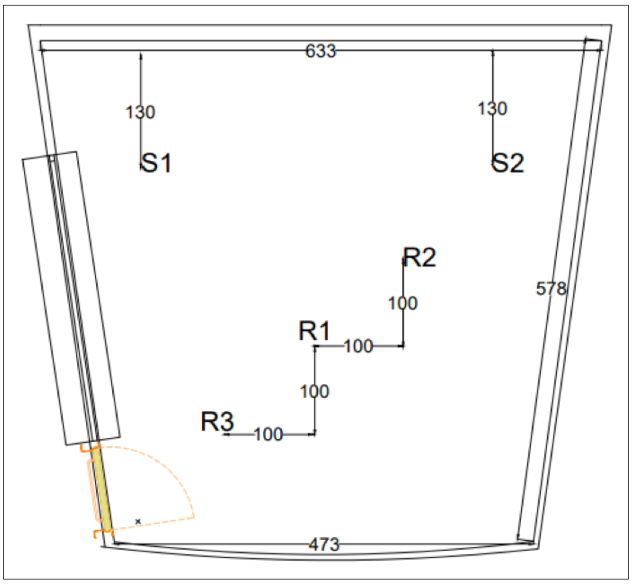

Figure 3. Measurement plan.

ISO 3382 Acoustic Parameters (T30, EDT and C80)

As a result of the measurement, reverberation time values were obtained. When we look at the T 30 values for the 4 different versions, we see that the reverberation time values decrease with the use of absorber elements. While there was no difference in the 4 versions in the 63 Hz region, it was determined that the biggest change was in the 250 Hz center octave frequency band (Figure 4, Table 2). A similar result is

Table 2. Reverberation time of V1, V2, V3, V4 versions

Frequency (Hz)	63	125	250	500	1000	2000	4000
V1 T30	1.48	1.31	1.08	0.76	0.58	0.60	0.60
V2 T30	1.52	1.19	0.83	0.57	0.45	0.43	0.37
V3 T30	1.48	1.08	0.54	0.42	0.36	0.38	0.39
V4 T30	1.52	0.99	0.52	0.35	0.29	0.31	0.35

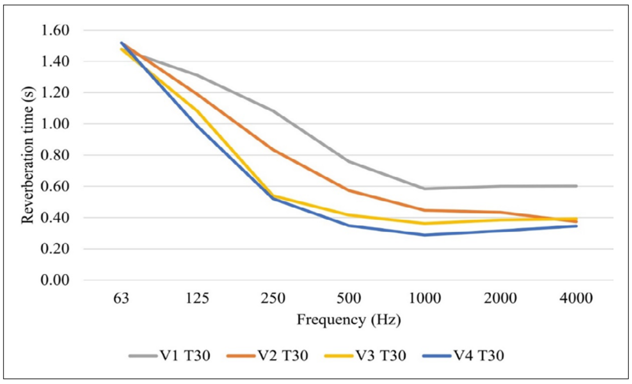

Figure 4. Reverberation time of V1, V2, V3, V4 versions.

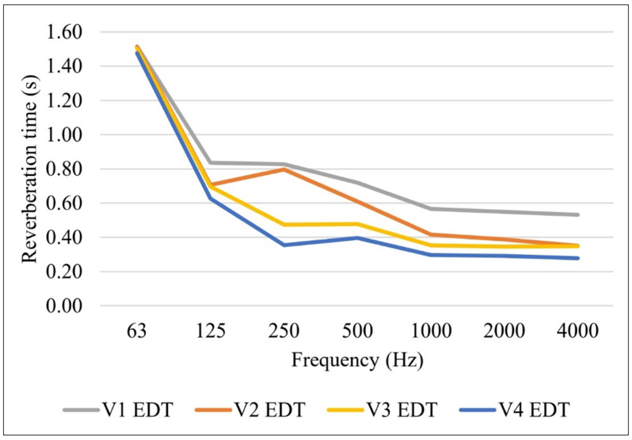

Figure 5. V1, V2, V3, V4 versions EDT.

seen in the EDT data. While there is no change in the 63 Hz central octave frequency, the rate of change is high in the 250 Hz region (Figure 5). The difference in EDT data compared to T 30 data is that the effect of using curtain between V1 and V2 is limited to the 250 Hz band. It was determined that the 0.25 second decrease seen in the T30 chart for 250 Hz was 0.03 seconds for the EDT parameter (Table 3).

When we look at the C80 data, it is seen that it generally increases according to frequency for each version. In the V1 version, it showed a decrease in the 125 Hz region after 63 Hz, and increased continuously at 250 Hz and above. A similar situation occurred in V2 and V3. In the V4 version, as the frequency increased, C 80 values also increased. It is seen that C 80 values increase as we move from V1 to V4 for every frequency region except 63 Hz (Figure 6). It was determined

Table 3. V1, V2, V3, V4 versions EDT

Frequency (Hz)	63	125	250	500	1000	2000	4000
V1 EDT	1.52	0.84	0.83	0.72	0.57	0.55	0.53
V2 EDT	1.51	0.71	0.80	0.61	0.42	0.39	0.35
V3 EDT	1.50	0.70	0.47	0.48	0.35	0.35	0.35
V4 EDT	1.48	0.63	0.35	0.40	0.30	0.29	0.28

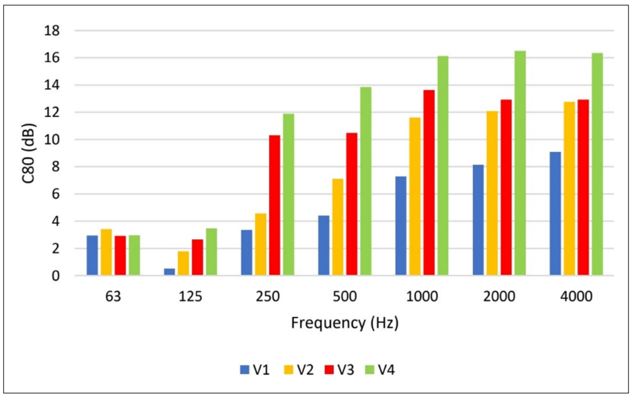

Figure 6. C80 measurement results.

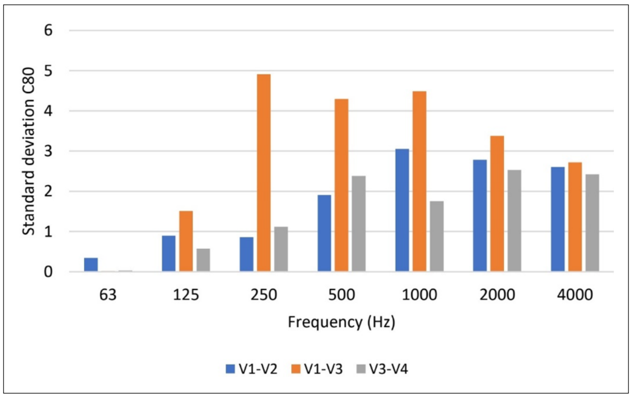

Figure 7. C80 standard deviation values.

that the increase in C 80 values was higher in the V3 and V4 versions where panel was used. It was calculated using equation 4 that the absorption coefficients of the panels would reach their highest value at 345 Hz. When we look at the change in C80 values, we see that the highest standard deviation value in the transition from V1 to V3 is at the 250 Hz frequency (Figure 7). It was determined that the effect of the panels on C80, still continued in the 500 and 1000 Hz regions, but decreased at 2000 and 4000 Hz. It was found that the effectiveness of the panels using wood with 23% perforation in front of the 10 cm sponge, was close to the calculations.

Panel and Curtain Absorption

The absorption coefficients of the panels with perforated fronts and thick fabric curtains which are used in the scenarios created in the room, were determined using the reverberation time measurement results in accordance with the ISO 3382-2 standard. The difference between V1 and V2 room arrangements is the use of curtains. The absorption coefficients of the curtain were found from the difference in reverberation time between the two versions.

$$RT_1 = \frac{0{,}16.V}{A} \tag{1}$$

$$RT_2 = \frac{0{,}16.V}{A + a.S} \tag{2}$$

In Equation 1, the Sabin reverberation time formula is defined for V1. In Equation 2, the value of S is calculated by subtracting the total area of the curtain in bulk position from the total area of the curtain in open position. 15.5 m^2 curtain area is available in the V2 version. In the V1 version, the total area of the curtain is 3.5 m^2. In Equation 2, the area S is taken as 12 m^2. When the absorption coefficient was drawn from these two equations, the formula shown in Equation 3 was obtained. RT1 and RT2 values, obtained from field measurements were used in Equation 3 to find the absorption coefficients of the curtain according to frequencies.

$$a = \frac{RT_1.A}{RT_2.S} - \frac{A}{S} \qquad (3)$$

Here

RT_1: V1 reverberation time

RT_2: V2 reverberation time

A: V1 sabin area

S: Curtain or panel area

V: Room volume

a: Absorption coefficient

Panel sound absorption coefficients were determined in the same way using Equation 3. The difference between V1 and V3 room arrangements is the use of 7 panels. The sound absorption coefficients of the panel were found by using the RT1 and RT3 field measurement results in Equation 3 (Figure 8).

It is seen that the panel sound absorption coefficients exceed 1 in the frequency regions of 250 Hz and above. It is thought that there is an edge effect in the formation of this situation.(Sauro et al., 2009)

Panel Absorbency Calculation and Measurement

The critical frequency value at which the absorption efficiency of a 10 cm thick sponge with a perforated panel in front begins to decrease, is found using Equation 4. (Egan, 2007),

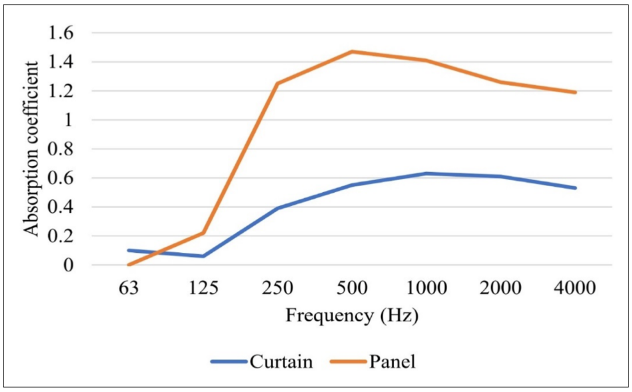

Figure 8. Curtain and panel sound absorption coefficients.

$$f_c = \frac{40P}{D} \qquad (4)$$

where Fc is the critical frequency (Hz), P perforation percentage (%) and D hole diameter (inch). There are a total of 80 holes on the panels with a hole diameter of 7 cm. Panel dimensions are 103-126 cm (Figure 9). Total hole area is 3077 cm^2. Total panel area is 12978 cm^2. The perforation rate was found to be 23.7% and critical frequency was calculated as 345 Hz.

Room Response and Modal Decay Time

The room response is as shown in Figure 10 for 4 different versions. In all versions the floor is carpeted. In the V1 design, no sound-absorbing materials were used on the room walls and glass. In the V2 design, there is a thick curtain on the glass and door section. In the V3 design, there are a total of 7 panels on the walls. In the V4 design, 7 panels and curtains on the door and window section were used. It can be seen that a dip occurs at the frequency of 97.3 Hz in all four designs. At the frequency of 97.3 Hz, the sound level in V3 and V4 designs is 7 dB lower than in V1 and V2 designs. The use of 7 panels caused the dip region

Figure 9. 10 cm sponge front wooden perforated panel.

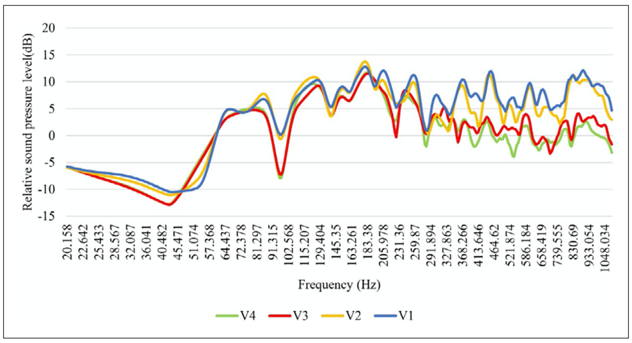

Figure 10. Room response.

at the frequency of 97.3 Hz to shift 7 dB lower. It has been observed that the use of curtains alone does not create a change in the room response up to 300 Hz. In the V1 and V2 versions, there is a change depending on the curtain usage situation.

Q values occurring at the frequency points where the room response curves peak are a criterion used to evaluate the perception of resonances. Q value was calculated using the formula given in Equation 5 (Everest, 2001), where Q is the quality factor, Δf is band width, F_c is mode frequency. It is obtained by dividing the peak frequency by the frequency range that is 3 dB below from the right and left. Q values calculated at the peak frequencies of the room response curves are given in Figure 10.

$$Q = \frac{F_c}{\Delta f} \quad (5)$$

For the V1 version, peak formations that could create a 3 dB difference on the right and left were detected for 8 frequencies. Peak occurrence was determined for 7 frequencies in V2 and 5 in V3 and V4. When V1 and V2 are compared, it is seen that the resonance frequencies are the same, but the peak at 455 Hz for V1 shifts to 445 Hz for V2. It was also observed that the peak at 205 Hz in the V1 version did not occur in V2. A peak occurred at 205 Hz only in the V1 version. The peak occurred at 85.7 Hz in V1 and V2 versions, and at 80 Hz in V3 and V4. Similar shifts occurred at frequencies of 179 Hz, 257 Hz and 454 Hz. It was observed that the 304 Hz and 368 Hz peaks that occurred in the V1 and V2 versions did not occur in V3 and V4. It is seen that the peaks disappear at these frequencies where panel absorbance is high.

At a frequency of 85.7 Hz, the Q value for V1 was 3.06, while it was 5.95 for V2. It has been observed that the use of curtains increases the Q value of the room mode at this frequency. The Q value of this peak, which occurred at 80 Hz between V3 and V4, increased from 2.94 to 3.5. It can be said that there is a similar effect, although less than the change between V1 and V2. Although there were minor changes in other peak frequencies between V1 and V2, no significant difference was observed. It is seen that in the V3 and V4 versions, the Q values decrease at the frequencies of 179 Hz, 241 Hz and 440 Hz compared to V1 and V2. It is seen that the peaks disappear at frequencies of 304 and 368 Hz. As a result of the calculations, it is predicted that the panel absorption coefficient will reach its maximum value at 345 Hz, and Q values also support this prediction. Both the shift and the decrease in the Q value at these frequencies compared to V1 and V2 occur as a result of the increase in panel absorbance. The absence of peaks at 304 and 368 Hz indicates that the panel is effective in the frequency region where its absorbance is highest (Figure 11).

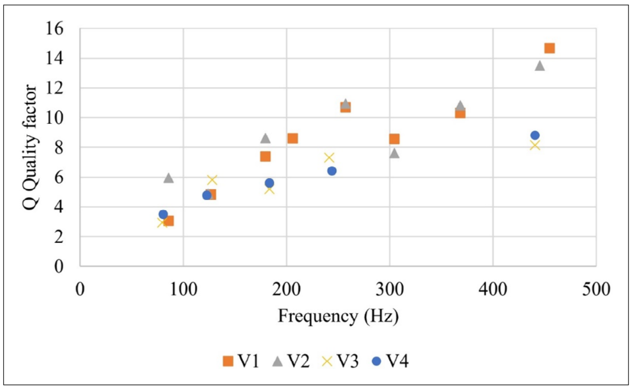

Figure 11. Q values of resonance frequencies.

Using Equation 6, the reverberation time value of the mode frequency is calculated. The reverberation time value of the mode frequency is found by multiplying the quality factor Q value by 2.2 and dividing the result by the peak frequency. In Figure 12, Tmodal values calculated using the Q values.

$$T_{modal} = \frac{2.2xQ}{f_c} \quad (6)$$

It can be seen in Figure 12 that, the reverberation time values of the mode frequencies seen in the room response, form a graph similar to the Q values. Tmodal is calculated using the Q value and as a result, it creates a functional graph depending on the Q value.

ANSYS Simulation Results

As a result of the simulations made with the ANSYS modal acoustic module, room modes in 4 different versions were examined. The degree to which room modes are affected by absorptive acoustic devices has been investigated. Despite its importance in structural dynamics and vibration, modal analysis is rarely performed in acoustics due to the high modal density of sound fields. Modal analysis results become important due to the low diffuse sound field in small volumes. There are differences in room mode shapes and frequencies when the surfaces are hard and reflective

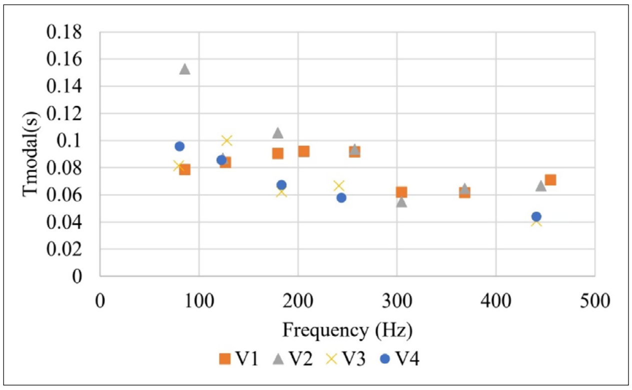

Figure 12. T_{modal} values of resonance frequencies.

or when they are absorbing (Dowell et al., 1977; Meissner, 2008). As the absorption coefficient increases, the change in mode shapes also increases (Dowell et al., 1977). By changing the absorbance coefficient values of the curtain on the door and window and the carpet on the floor of the room where the study was carried out, it was observed how the modal frequencies and shapes changed. The changes in room mode shapes and frequencies were found by giving the absorbance coefficient values as 0, 0.2, 0.5 and 0.9, respectively. ANSYS Modal Acoustic module was used in simulation studies. Figure 13 shows the room modes in the first column, where there are hard walls and the absorption coefficient is 0. Column 2 shows the case where the absorbency of curtains and carpets is 0.2, column 3 shows the case where the absorbency is 0.5, and column 4 shows the case where the absorbency is 0.9. The first 3 axial modes of the x and y dimensions and the first axial mode of the z dimension are shown in Figure 13. When we look at the Figure 13, we see that there is no significant change between the situation where there is no absorbance and when it is at a value of 0.2. There was no deviation in the frequency values and the mode shapes did not change. It was found that when the 0.5 absorption coefficient in the 3rd column was used, the mode shapes did not change and the frequency values shifted less than 0.5 Hz. According to the data in the first 3 columns, it is seen that the room mode shapes do not change if the absorbance values of the surfaces are low. It can be said that when the absorption coefficient is 0.5, frequency deviations begin, but no change in mode shapes is observed.

In the last column, it was found that both mode shapes and frequency values changed when the absorption coefficient was 0.9. When the 4 columns are examined together, it is seen that the change capacity of the room modes increases as a result of the increase in the absorption coefficient. If the absorption coefficients of the absorber elements are high in the frequency band where the room modes are effective, it will be possible to change the frequency and shape of the room modes. It is seen that surfaces with low absorption coefficient do not affect room modes.

As a result of this simulation study, it is revealed that room modes can change as a result of high absorbance values. When 4 different room versions were examined, in cases V1 and V2, the situation in which the empty room was with or without curtains was examined. It is seen that in the case without curtains, only the absorbing areas originating from the carpet and the gathered curtains are formed, but their absorption coefficients are low for the frequency region below the Schroder frequency. For this reason, it can be seen in room response measurements that the effect of using curtains on room modes and room response occurs starting from 300 Hz. It is seen that the room response does not change in the V1 and V2 versions up to 300 Hz, but changes begin from 300 Hz onwards.

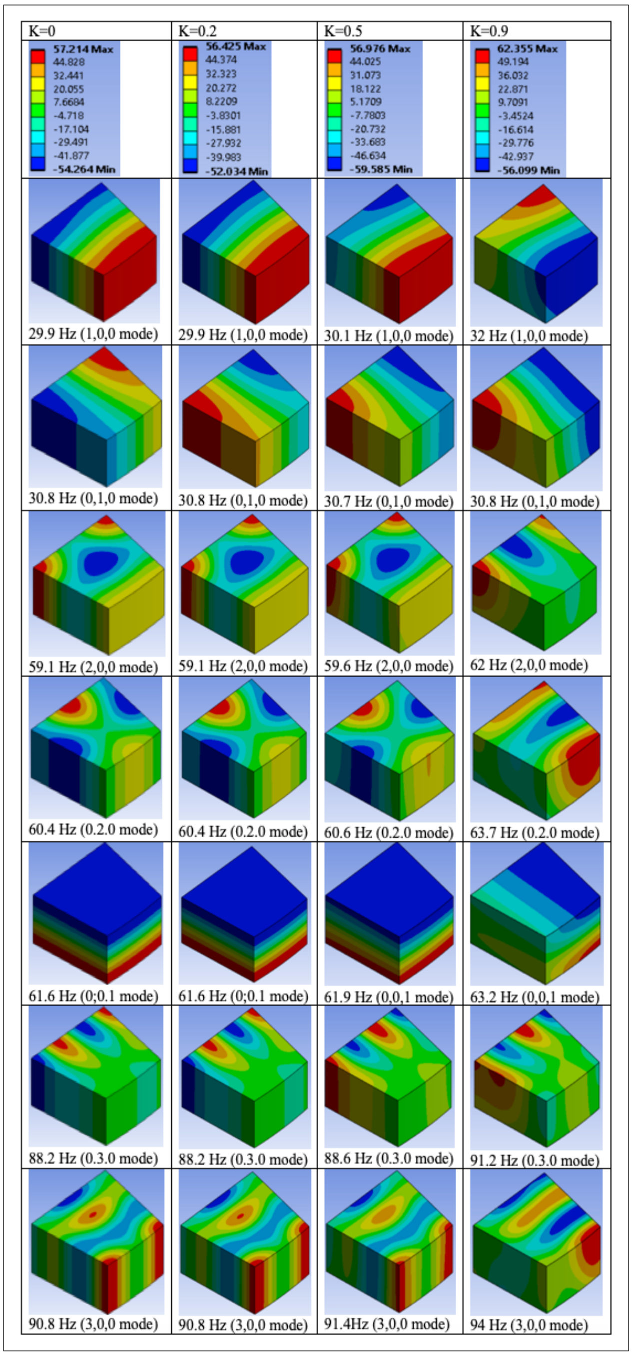

Figure 13. ANSYS modal analysis axial mode shapes and frequencies.

When the absorbance values found from area measurements where the panel and curtain were used for the region below the Schroder frequency, it was observed that there was no change in the simulation results compared to the solid wall. Since the absorptivity values are very low in this frequency band, there is no change in room mode shapes and frequencies for the region below the Schroder frequency. However, it is observed that there is a change in the room response and room mode shapes, especially since the panel absorption coefficients exceed 1 in the 250 Hz region.

DISCUSSION

As a result of this study conducted on a non-rectangular room example, ISO 3382 parametric values, room response, room mode frequencies and shapes were found as a result of field measurement and simulation applications. These values for 4 different room versions were examined and compared. Changes resulting from the use of sound-absorbing materials have been observed (Table 4).

Curtain Effect

In the V1 version, it was used as a curtain roller and a carpet absorber on the floor. In the V2 version, the curtain is drawn across the glass and door. When moving from V1 to V2, the expectation is that the reverberation time in the middle and upper frequency regions will shorten due to absorption. A 23% decrease was observed starting from the 250 Hz region. In the 4000 Hz region, the reverberation time decreased by 37.8% (Figure 14). As seen in the first row of Table 4, it has been observed that absorption is effective starting from the 250 Hz region. By comparing V1 and V2, absorption coefficient was found to be 0.39 in the 250 Hz region and 0.55 in the 500 Hz region. Figure 13 shows that there may be minor changes in room modes with this absorbance value. Absorptivity around 0.5 may cause minor changes in room mode shapes and frequencies. When we compare V1 and V2 in the room response curve, we see that there are decreases in sound level in some frequency regions starting from 280 Hz. It is seen that sound levels decrease and room response changes, depending on the use of curtains between 300-334 Hz, 392-426 Hz, 495-567 Hz, and 631-771 Hz. Absorption coefficient was found to be 0.55 in the 500 Hz center octave frequency region. In the room response, there was a downward change in sound level in 3 ranges in this frequency region.

Table 4. Reverberation time changes according to versions

Frequency (Hz)	63	125	250	500	1000	2000	4000
V1-V2/V1	-2.63	9.26	22.99	24.49	23.56	27.73	37.82
V3-V4/V3	-2.81	8.85	3.37	16.16	20.34	18.05	12.14
V1-V3/V1	0.02	17.46	50.18	45.22	37.93	36.03	34.72
V2-V4/V2	-0.15	17.09	37.49	39.18	35.31	27.46	7.75

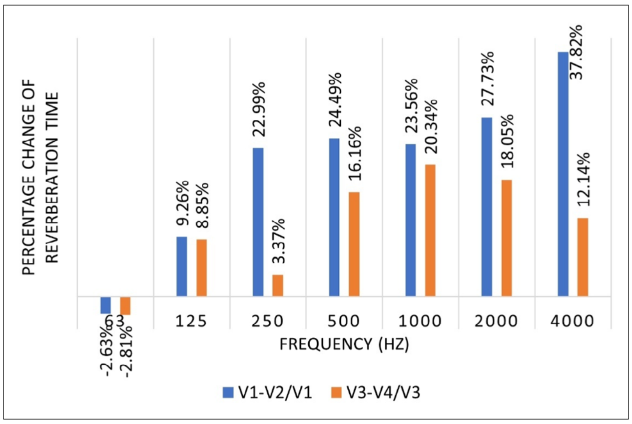

Figure 14. Reverberation time change percentage with curtain effect.

The difference between V1 and V2 and V3 and V4 due to absorption begins to appear starting from 300 Hz. When we compare V3 and V4, there is a decrease in sound levels due to the use of curtains between 300-338 Hz, 388-422 Hz, 474-561 Hz and 770-1100 Hz. In addition, it was observed that in the V3 version, where only panels were used, a dip zone was formed in 3 different frequency regions compared to V4. V3 was found to be at a lower sound level than V4 between 221-231 Hz, 353-364 Hz and 680-724 Hz. While such a situation does not appear between V1 and V2, the reason why it occurs between V3 and V4 may be that there are changes in room modes due to the use of panels in this region and the use of curtains affects the room modes in these 3 frequency regions. It is thought that a thick velvet curtain reduces sound diffusion by increasing absorption on one side. It is seen that reverberation times decrease by 24% in the mid-frequency region between V1 and V2, and decrease by 23% in V4 and V3. It was found that the change between V1 and V2 was similar to the change between V3 and V4.

Panel Effect

Curtain bulk is added in V1 and V3 versions, and 7 panels are added in V3. As a result of examining the reverberation time, room response and room modes together, it was seen that the panels reduced the reverberation time in the mid-frequency region by 42% (Figure 15). In Section 2.3, the frequency range in which the panels are effective was calculated and the frequency with the highest absorbency was found to be 345 Hz. When we look at the reverberation time changes, it was found that the highest changes were in the 250 Hz and 500 Hz regions. It was observed that the panel absorbance calculated based on the reverberation times had the highest absorption coefficient among the

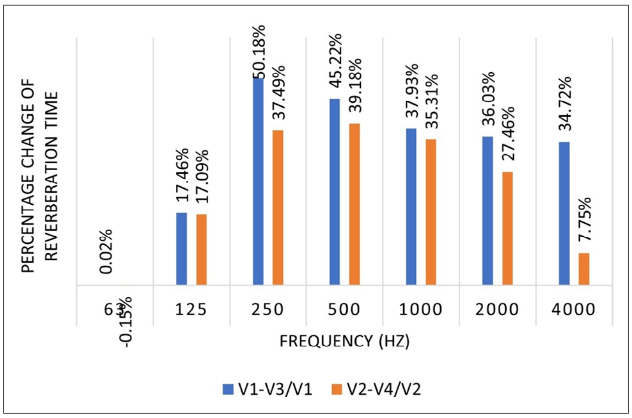

Figure 15. Percentage change of reverberation time with panel effect.

frequency regions with a value of 1.47 in the 500 Hz region.

Panel use lowered the dip point at 97.3 Hz in the room response curve by 7 dB. The same situation occurred for V3 and V4 at 97.3 Hz. As seen in Table 5, after the 95 Hz oblique mode and the 95.1 Hz tangential mode, 102.6 Hz tangential mode comes. It has been observed that the 7.6 Hz gap in between causes the room response to bottom out at 97.2 Hz. It was found that the sound level was 7 dB lower at the frequency of 97.2 Hz in V3 and V4 compared to V1 and V2. It is thought that the reason for this may be that the perforated wooden plate on the front faces of the 7 panels used, has a vibrating plate feature resulting from its own vibration frequency and increases the sound absorption of this frequency. It can be seen in the room response curve that similar situations occur at frequencies of 226 Hz and 356 Hz in panel use.

When we look at the change in the frequencies and shapes of the room modes with the absorption coefficients presented in Section 2.5, it is seen that the mode shapes and frequencies change when the sound absorption coefficients are greater than 0.5. Since the panel absorption coefficients found from both calculation and reverberation time measurement results in Section 2.3 took larger values in the 250 Hz and 500 Hz regions, the peaks in the room response that appeared at 304 Hz and 368 Hz frequencies compared to the case without a panel disappeared. The simulation results for modal analysis and the changes in the room response confirm each other. In the frequency regions where the sound absorption coefficient increased, the peaks formed by the influence of room modes disappeared at 304 Hz and 368 Hz, and the Q values of the peaks at 179 Hz, 241 Hz and 440 Hz also decreased. We can see the panel absorbance effect in the reverberation time, C80, room response measurement results, Q value and Tmodal calculations for the relevant frequency regions.

Table 5. Room mode frequencies and mode ranges

Mode Frequency (Hz)	(x,y,z) mode	Difference (Hz)
29.9	1,0,0	
30.8	0,1,0	0.9
44.3	1,1,0	13.5
59.1	2,0,0	14.8
60.4	0,2,0	1.3
61.6	0,0,1	1,2
67.8	2,1,0	6.2
68.5	1,0,1	0.7
68.9	0,1,1	0.4
72.3	1,2,0	3,4
75.9	1,1,1	3.6
85.3	2,0,1	9.6
86.2	2,2,0	0.9
86.3	2,0,1	0.1
88.2	0,3,0	1.9
90.8	3,0,0	2.6
91.6	2,1,1	0.8
95	1,2,1	3,4
95.1	3,1,0	0.1
102.6	1,3,0	7.5
106	2,2,1	3,4
107.6	0,3,1	1.6
109.1	3,2,0	1.5

CONCLUSION

In this study, which aims at the modal analysis of non-rectangular rooms, reverberation time, clarity and room response measurements were made and compared with the modal acoustic simulation data of the room. Room response is used to determine room modes through measurement. By looking at the room response, the effect of room modes at the measurement point can be observed. One of the features that can affect room response is the sound absorption properties of the surfaces. Although it is known that room modes change with surface absorbance, studies on how the change will occur according to the absorption coefficient remain limited. As a result of the simulations carried out in this study, it was observed that as the sound absorption coefficients of the materials to be used on the surfaces increase, they can change the shapes and frequencies of the room modes in the frequency bands in which they are effective. It has been determined that this change begins when the absorption coefficient is greater than 0.5.

It was evaluated how accurately the empirical formulas produced to calculate the absorption effects of the acoustic devices to be used in the room according to frequency determined the impact area. It has been observed that the measurements and calculations confirmed each other. This experimental study, using variable absorbers, was designed for 4 different versions of the room. In the experiment designed with curtains in use in V1, curtains open in V2, 7 panels in use and curtains not in use in V3, and curtains and 7 panels in use in V4; the reverberation time of the curtains and panels, C80, room response and Q of the peaks were determined. Its impact on the values was investigated. According to the hypotheses generated at the beginning of the study, the panels will have the highest absorbance value at the 345 Hz frequency, and the absorbance values will decrease to the right and left of this frequency. It was hypothesized that the effect of curtain absorbance would also occur in the medium and high frequency regions. It is thought that as the absorbance values increase, the change of room modes will increase and the room response will become smoother and the peaks will decrease.

As a result of field measurements, simulations and calculations, it was observed that the room modes changed with the absorbance and the room response became smoother. It has been determined that these changes occur in frequency bands where absorbing materials are more effective, and that the room response does not change in frequency bands where sound absorption coefficients are less than 0.5. It was found that the panels used in the V3 and V4 versions maximized their effectiveness around 345 Hz, as calculated, and as a result, they created changes in both room response and reverberation time and clarity parameters in the 250 Hz and 500 Hz regions. These panels contributed to the smoothing of the room response by eliminating the 304 and 368 Hz peak regions in the room response and reducing the Q values of the peaks at 179 Hz, 241 Hz and 440 Hz frequencies.

As a result of this study, it was found that the desired changes in room response and room acoustic parameters can occur as a result of the use of acoustic materials with correctly calculated absorbance depending on frequency. As a method, below steps in order were determined:

1. determining the room modes by simulation
2. obtaining reverberation time, clarity and room response curves by making field measurements
3. determining the absorption coefficients, layout and design of the acoustic absorption materials
4. manufacturing and assembling the acoustic absorption materials
5. making the measurements after the application of the absorbers and analyzing them comparatively with the first measurements.

Among the limitations of the research the fact should be expressed that the absorption properties of sound absorbing materials may vary depending on the usage patterns, room characteristics and design goals. Generalizations can be made in room acoustic design by taking into account the differences in these variables.

Based on this research, it is recommended for future research to discover materials with higher absorption capacities and test them in different room sizes and shapes.

ETHICS: There are no ethical issues with the publication of this manuscript.

PEER-REVIEW: Externally peer-reviewed.

CONFLICT OF INTEREST: The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

FINANCIAL DISCLOSURE: This work has been supported by Yildiz Technical University Scientific Research Projects Coordination Unit under project number FDK-2023-5896 BAP Doctoral Dissertation Project.

REFERENCES

Bai, M. R. (1992). Study of acoustic resonance in enclosures using eigenanalysis based on boundary element methods. J Acoust Soc Am, 91(5), 2529–2538.

Beaton, D., & Xiang, N. (2017). Room acoustic modal analysis using Bayesian inference. J Acoust Soc Am, 141(6), 4480–4493.

Bistafa, S.R., & Morrissey, J.W. (2003). Numerical solutions of the acoustic eigenvalue equation in the rectangular room with arbitrary (uniform) wall impedances. J Sound Vib, 263, 205–218.

Boner, C. P. (1942). Performance of broadcast studios designed with convex surfaces of plywood. J Acoust Soc Am, 13, 244–247.

Čurović, L., Murovec, J., Železnik, A., & Prezelj, J. (2024). Estimation of sound absorption coefficient at modal frequencies using decay time measurements and eigenvalue model in a small room. J Low Freq Noise Vib Act Control, 43, 1530–1544.

Dance, S. M., & van Buuren, G. (2013). Effects of damping on the low-frequency acoustics of listening rooms based on an analytical model. J Sound Vib, 332(25), 6891–6904.

Das, O., & Abel, J. S. (2022). Modal estimation on a warped frequency axis for linear system modeling. Retrieved March 28, 2025, from: https://arxiv.org/abs/2202.11192

Dowell, E.H., Gorman, G.F., & Smith, D.A. (1977). Acoustoelasticity: general theory, acoustic natural modes and forced response to sinusoidal excitation, including comparison to experiment. J Sound Vib, 52(2), 519–542.

Egan, M. D. (2007). Architectural Acoustics. J Ross Publishing.

Ekmen, Ş., Karadoğan, C., & Şeker, Ş. S. (2021). Investigation of timbral qualities of guitar using wavelet analysis. Traitement Du Signal, 38(2), 401–411.

Everest, F.A. (2001). Master Handbook of Acoustics: Vol. McGraw Hill.

Gilford, C. L. S. (1959). The acoustic design of talks studios and listening rooms. Proc Inst Elect Engs, 106, 257–258.

Hikichi, T., & Miyoshi, M. (2006). Sound timbre control using estimates of room resonance modes. Acoust Sc Technol, 27(5), 257–263.

Jian, H. M., Chen, Y. S., & Bai, M. R. (2022). Acoustic modal analysis of room responses from the perspective of state-space balanced realization with application to field interpolation. J Acoust Soc Am, 152(1), 240–250.

Jiang, J., Huang, K., & Zhao, Y. (2011). Calculating room acoustic parameters by finite element method. 2011 International Conference on Consumer Electronics, Communications and Networks, Proceedings of CECNet 2011, 4838–4841.

Kelle, D., & Yılmaz Demirkale, S. (2022). Musicians' impressions of low frequency sound field in small music rooms. A/Z: ITU J Fac Archit, 19(3), 113–128.

Kleiner, M., & Tichy, J. (2014). Acoustics of Small Rooms: Vol. CRC Press.

Lim, H., Imran, M., & Jeon, J. Y. (2016). A new approach for acoustic visualization using directional impulse response in room acoustics. Build Environ, 98, 150–157.

Louden, M. M. (1971). Dimension-ratios of rectangular rooms with good distribution of eigentones. Acustica, 24, 101–103.

Mehra, R., Raghuvanshi, N., Savioja, L., Lin, M.C., & Manocha, D. (2012). An efficient GPU-based time domain solver for the acoustic wave equation. Appl Acoust, 73(2), 83–94.

Meissner, M. (2008). Influence of wall absorption on low-frequency dependence of reverberation time in room of irregular shape. Appl Acoust, 69(7), 583–590.

Prato, A., Casassa, F., & Schiavi, A. (2016). Reverberation time measurements in non-diffuse acoustic field by the modal reverberation time. Appl Acoust, 110, 160–169.

Qu, S., Yang, M., Xu, Y., Xiao, S., & Fang, N. X. (2023). Reverberation time control by acoustic metamaterials in a small room. Retrieved March 28, 2025, from: https://arxiv.org/abs/2308.10476

Rossing, T. D., & Fletcher, N. H. (2004). Principles of Vibration and Sound (Second Edition). Springer.

Sauro, R., Vargas, M., & Mange, G. (2009, August 23). Absorption coefficients-part 2: Is "edge effect" more important than expected? 38th International Congress and Exposition on Noise Control Engineering, p. 2112–2122.

Svensson, M. (2020). Simulating Low Frequency Reverberation in Rooms [Degree Project in Mechanical Engineering]. KTH Royal Institute of Technology.

Tıraş, M., & Akdağ, N. Y. (2024). Evaluation of room acoustics in the control rooms of recording studios through a case study. J Fac Eng Archit Gazi Univ, 39(2), 1241–1254.

Turkish Standarts. (2008). Acoustics - Measurement of acoustic parameters of rooms - Part 2: Reverberation time in ordinary rooms. TS EN ISO 3382-2.

Turkish Standarts. (2010). Acoustics - Application of new measurement methods in building and room acoustics. TSE EN ISO 18233.

Volkmann, J. E. (1942). Polycylindrical diffusers in room acoustical design. J Acoust Soc Am, 13, 234–243.

Vorlander, M. (2013). Handbook of Engineering Acoustics. Springer.

Wang, H., Zeng, X., Lei, Y., & Shuwei, R. (2020). Indoor object identification based on spectral subtraction of acoustic room impulse response. IEEE International Conference on Signal Processing.

Yoshida, T., Okuzono, T., & Sakagami, K. (2021). Dissipation-free and dispersion-optimized explicit time-domain finite element method for room acoustic modeling. Acoust Sci Technol, 42(5), 270–281.

Zhang, D., Tenpierik, M., & Bluyssen, P. M. (2021). Individual control as a new way to improve classroom acoustics: A simulation-based study. Appl Acoustics, 179, 108066.

Zhu, P., Tao, W., Lu, X., Mo, F., Guo, F., & Zhang, H. (2022). Optimization design and verification of the acoustic environment for multimedia classrooms in universities based on simulation. Build Sim, 15(8), 1419–1436.