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Characterization of low frequency reverberation room behavior using simulation

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ABSTRACT

Primary metrics for assessing the performance of acoustic materials are the normal incident and random incident sound absorption. Measurement of the former is performed in an impedance tube whereas the latter is measured in a reverberation room. It is well known that the reverberation room can greatly impact the measured random incident absorption especially at low frequencies. To determine the random incident absorption coefficient, the material is positioned on the floor of a reverberation room and the decay rate is measured after a sound source is switched off. Reproducibility and repeatability are often suspect in the low frequency range where the sound field is dominated by room modes. The low sound absorption coefficients of the walls and test samples often result in long reverberation times that lead to a less than ideal diffuse field assumption. To better understand the reverberation room at low frequencies, a study correlating experimental mapping and finite element simulation was undertaken to validate a model to guide future design changes to the room. It is hoped that the model will enable the engineers to better select diffusers or low frequency sound absorbers to improve the room performance.

1 INTRODUCTION

Sound absorbing materials are regularly applied to room or enclosure walls to reduce noise to more acceptable levels or prevent unwanted reverberation. Materials are generally characterized

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by their sound absorption which is defined as the ratio of absorbed to incident sound power. The sound absorption is a function of frequency but also the incident angle of the sound wave. The two standard metrics for assessing the sound absorption are the normal and random incident sound absorption coefficient. The normal incident sound absorption coefficient is relatively easy to measure. A sample is placed at the end of an impedance tube. Plane wave behavior is ensured in the tube, and results tend to be highly reproducible and repeatable for well-cut samples. The more commonly published metric is the random incident sound absorption. The goal of the test is to produce a diffuse field on the sample. This requires a special testing facility commonly referred to as a reverberation room. To quantify the random incident absorption coefficient, an absorber is positioned on the floor of the room and the resulting change in reverberation time is recorded.

Though the method has been standardized in ASTM C423[1] and ISO 354[2], the method does not perfectly determine the random incident absorption. The standards are based on room acoustics theory which assumes that the sound absorption is evenly distributed in the room. That is not the case in reality. Edge and refraction effects lead to an overestimation so that the sound absorption coefficient often exceeds 1.0 in the mid frequencies where the theoretical range is from 0 to 1. Though clearly not realistic, this artifact of the measurement is accepted so that a fairly repeatable standard practice can be established.

Nonetheless, it turns out that the test is not as repeatable as desired due to differences between reverberation rooms. This is especially the case at low frequencies where room modes tend to dominate the response. When one side of the room is treated, waves propagating in different directions do not decay at the same rate. In order to reduce the effect of room modes, a small amount of sound absorption is sometimes added to the room. In addition, rotating diffusers are often used to slightly modulate room modes at low frequencies.

Reverberation room modifications are normally performed in a trial and error fashion. This research documents an attempt to develop a simulation model that may be employed in addition to experimentation to drive treatments. Normally, finite or boundary element methods are preferred at low frequencies whereas ray tracing methods are used for higher frequencies. There have been a few studies. Some of the more notable work is by Hasan and Hodgson[3] who varied size and shape of the room in the model. Similarly, Ayr et al.[4] used a validated finite element model to qualify a reverberation room based on ISO 3741[5].

In this work, an existing reverberation room is simulated and results are compared with measurement. A measure of the diffuseness, the standard deviation of the sound pressure level in dB is reported. A study coupling experimental mapping and simulation techniques was commenced to better characterize the low frequency sound field. The model is used to investigate the effect of the rotating diffuser.

2 BLACHFORD REVERBERATION ROOM

Figures 1 and 2 show a schematic and photograph respectively of the ~213 m^3 reverberation room. The reverberation room is a part of the Blachford Acoustic Laboratory and is situated adjacent to a small anechoic room. The room serves two main purposes. It is used (1) as a source room for panel transmission loss testing and (2) for the determination of random incident sound absorption.

Walls are filled concrete block and it is isolated from other rooms. To enhance the diffusivity, five stationary and one rotating diffusers (with two angled panels) were introduced. The diffusers are four corrugated fiberglass and three flat wooden panels. There is one primary source: a loudspeaker that consists of three speaker cones generating a broadband frequency spectrum.

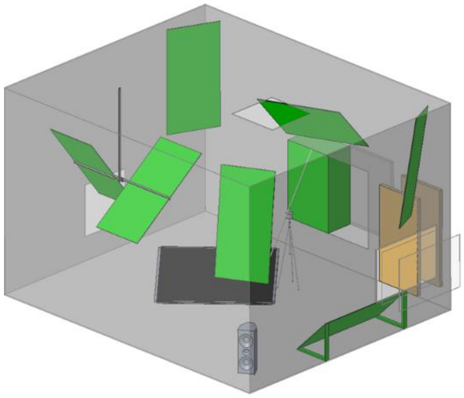

Fig. 1 – Reverberation room model.

Fig. 2 – Reverberation room internal view.

3 FINITE ELEMENT MODELING

The focus of this research was on low frequency behavior of the room so the finite element method (FEM) was used. The Siemens Virtual.Lab[6] software (Version 13.7) was used where both modal and direct frequency response finite element analysis were performed. Altair Hypermesh[7] was used to discretize the volume into eight linear tetrahedral elements per wavelength with a maximum frequency of 500 Hz. The rule of thumb as recommended by Marburg[8] is to have over 6 elements per acoustic wavelength. This results in a finer mesh than previous investigations by Hasan and Hodgson[3] where mesh convergence was studied. Results were performed in narrowband and then summed in 1/3 octave bands.

The corrugated diffusers were modeled as flat surfaces since corrugations are small compared to an acoustic wavelength. The rotating diffuser was approximated as a stationary diffuser at 60-degree increments until a full revolution is achieved. The sound field resulting from each of the different angles was then averaged. A constant real surface impedance was applied to all surfaces such that the normal incident sound absorption coefficient is approximately 0.02. This value was averaged from normal incident sound absorption coefficient measurements on room surfaces using an impedance tube. The approximated values compared well to Long's[9] tabulated concrete values.

Fluid dissipation was determined by beginning with an altered Sabine equation for reverberation time (T_{60}) presented by Cox and D'Antonio[10],

$$T_{60} = \frac{55.3V}{c\alpha S + 4Vm} \tag{1}$$

where m represents the air attenuation constant in 10^{-3} m^{-1}. Using this expression, the speed of sound is given an imaginary part so it will have the same decay characteristics.

4 COMPARISONS TO MEASUREMENT

The simulation is compared with measurement results. Measurements were made at 157 points on two planes parallel to the floor at approximately 6 cm and 120.5 cm respectively. The measurement grid and a corresponding photograph of the grid drawn on the floor are shown in Figure 3. There is interest in the 6 cm plane due to proximity to a proposed test absorber. A height of 120.5 cm is more representative of diffuse field sound absorption measurements. Two diffuse microphones were placed on a stand and sound pressure level measurements were made at the center of each gridded area. The temperature and humidity of the room were kept steady. The rotating diffuser was set to a stationary position which is referred to as 0 degrees for the remainder of the paper.

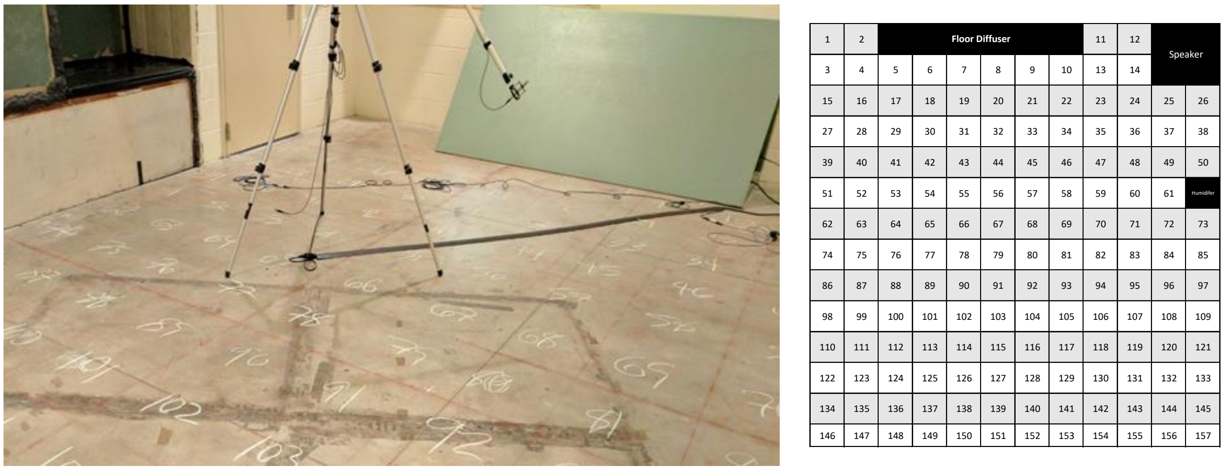

Fig. 3 – Discretization of the room.

The standard deviation of the spatially averaged sound pressure level, which will not be affected by source level, was calculated for each plane. Figures 4 and 5 compare simulation and measurement for the planes located 6 cm and 120.5 cm from the floor respectively. Results show similar trends though standard deviation in the measurement is on the order of 1-2 dB higher. This level of accuracy should be appropriate for evaluating the effect of potential room improvements.

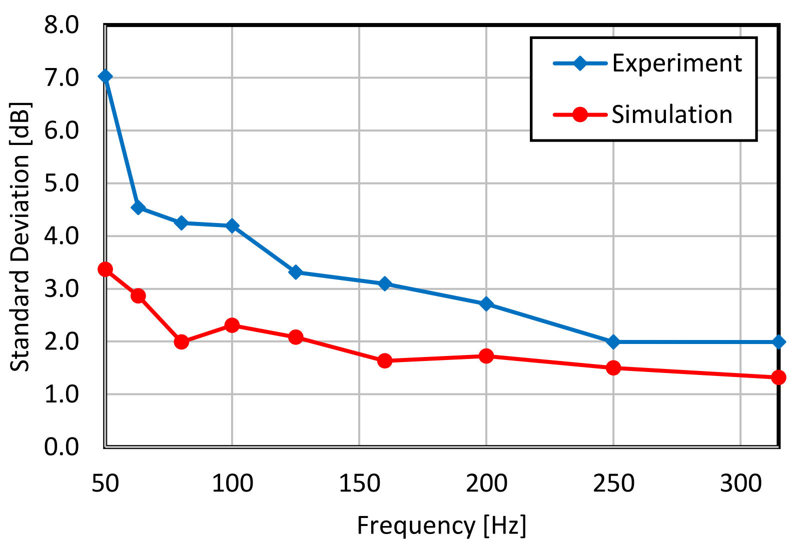

Fig. 4 – Comparing simulated and experimental spatial variance at 6 cm plane in one-third octave band.

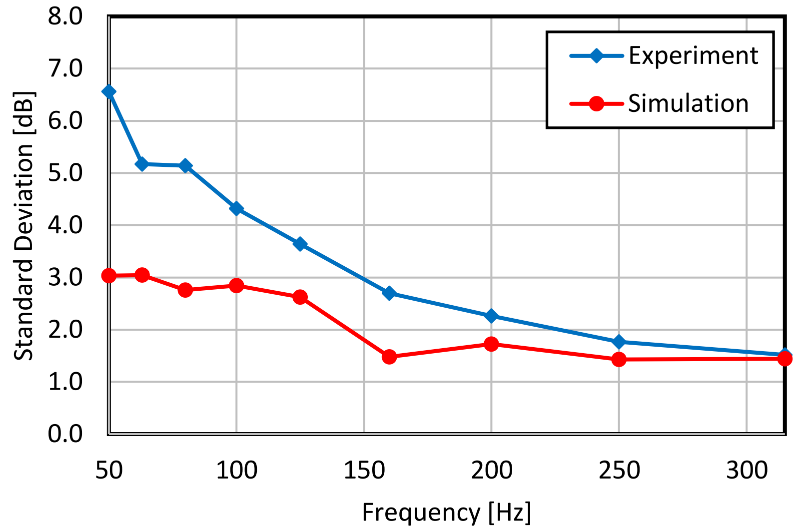

Fig. 5 – Comparing simulated and experimental spatial variance at 120.5 cm plane in one-third octave band.

5 DIFFUSER EFFECTS ON ROOM BEHAVIOR

Though the application of diffusers is an art, some recommendations can be made. For example, it is normally recommended that surface or volume diffusers be applied to at least 3 boundaries and that these boundaries should not be facing one another[10]. Diffusers work by splitting or varying the modal frequencies of a room during the test. High frequency dispersion is straightforward but low frequency is more problematic.

To evaluate the effectiveness of the existing diffuser application, the reverberation room was simulated with and without diffusers. Diffusion criteria have been recommended. For example, Ramakrishnan and Grewal[11] defined the Schroeder frequency or minimum threshold for diffusiveness as

$$f_s = 2000\sqrt{{T_{60}}/{V}} \tag{2}$$

where V is the room volume. This criterion tends to be highly restrictive and at times conflicts with more practical parameters on spatial variance. A more relaxed criteria is to define the diffusiveness threshold as the one-third octave band where there exist 20 acoustic modes per one third octave band.[3] The cut-off frequency is defined as,

$$f_c = \frac{c}{\sqrt[3]{V/4}} \tag{3}$$

where c is the speed of sound and V is the room volume.

Acoustic modal analysis was performed on the reverberation room for 8 cases including a room without diffusers and then at 60 degree increments of the rotating diffuser. Results are summarized in Table 1. Using criterion 2, a reasonably diffuse field can be expected at 100 Hz and above. If diffusers are used, the mode count increases slightly and modal frequencies slightly shift.

Table 1 – Effect of diffusers on reverberant room diffusion through acoustic modes.

Center Frequency [Hz]	Number of Modes								
	No Diffusers Applied	Stationary Diffusers (x5)							
		Stationary Only	Rotating Diffuser Orientation (Degrees)						
			0	**60**	**120**	**180**	**240**	**300**
50	5	6	6	6	6	6	6	6
63	7	7	7	7	7	7	7	7
80	13	13	13	13	14	13	13	14
100	22	23	24	24	23	24	24	23
125	41	43	42	43	43	42	43	44
160	82	82	84	83	81	85	83	82
200	160	162	161	160	161	160	161	162
250	295	301	298	299	298	299	298	299
315	575	578	590	589	588	588	592	587
400	1085	1072	1127	1130	1116	1114	1128	1106

Another approach to evaluating the effectiveness of the diffuser treatments is to examine the spatial variance of the sound field. Following a similar methodology to that discussed in the previous section, three planes, each having 168 receiver points, were examined as indicated in Fig. 6. It is anticipated that the spatial variation should decrease as diffusers are added to the room. Results are shown in Fig. 7. It is evident that there is a noticeable decrease in the standard deviation above 160 Hz. Results indicate that the treatments are likely effective at higher frequencies but may not provide a great deal of benefit at lower frequencies aside from adding some absorption to the room.

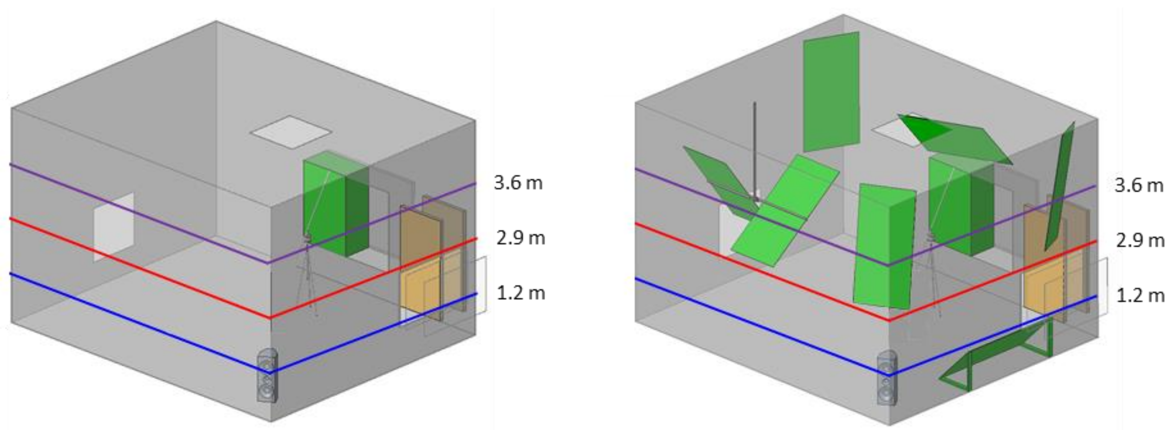

Fig. 6 – Location of receiver planes.

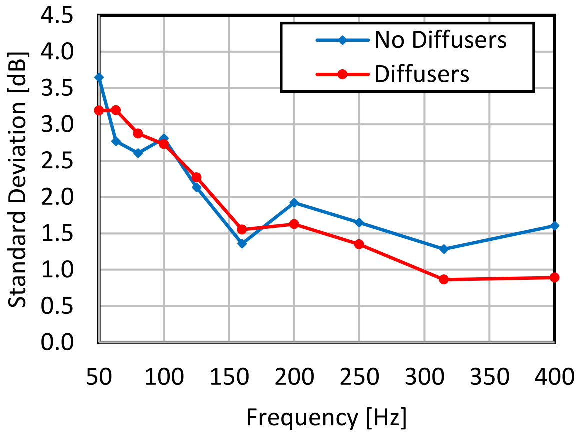

Fig. 7 – Diffuser application on spatial variance at 1.2 m plane in one-third octave band.

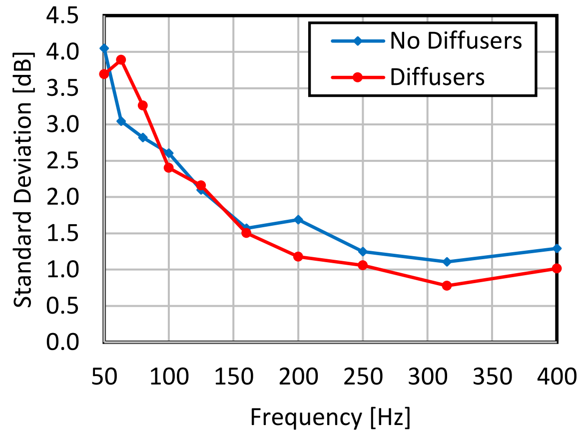

Fig. 8 – Diffuser application on spatial variance at 2.9 m plane in one-third octave band.

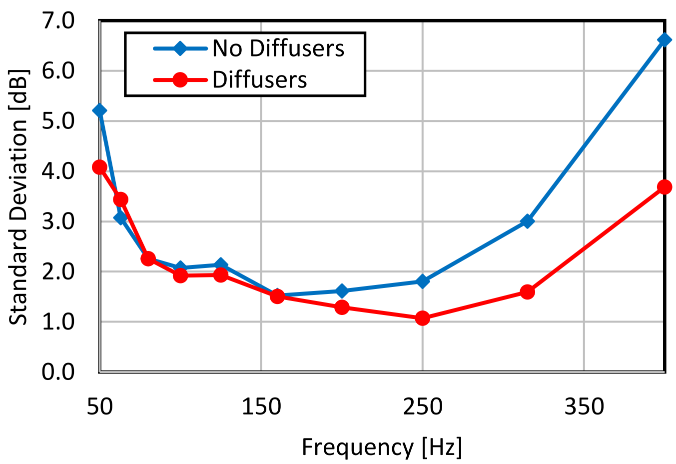

Fig. 9 – Diffuser application on spatial variance at 3.6 m plane in one-third octave band.

6 ROOM BEHAVIOR WITH A TEST SPECIMEN

The impact on the spatial standard deviation of the reverberation room when placing a sound absorption sample on the floor was investigated. It was anticipated that the application of an absorber to one of the boundaries of the reverberation room should lead to substantial changes in the spatial uniformity. A test absorber was modeled, where a surface impedance was prescribed using Wu's empirical model[12] assuming a thickness of 2 in and flow resistivity of 49,000 Rayls. Results are shown in Fig. 9 with and without diffusers. The spatial variation in the room is significantly higher without diffusers.

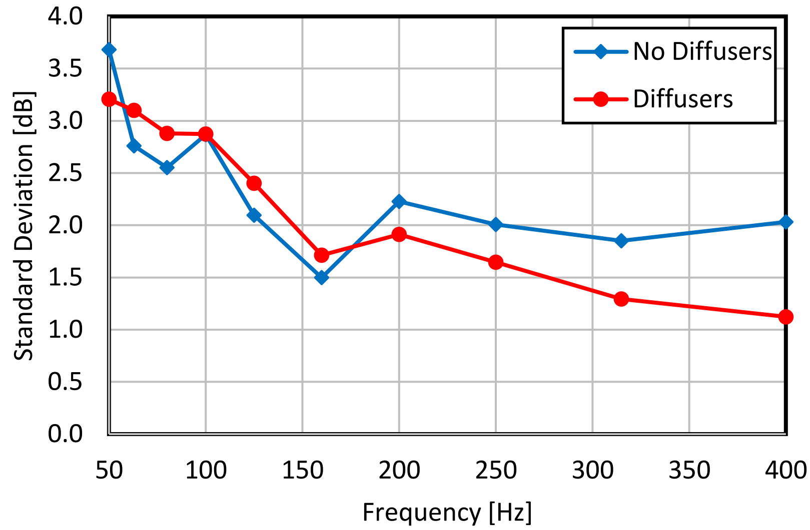

Fig. 10 – Spatial variance at 2.9 m plane with a test absorber placed on the ground.

7 CONCLUSIONS

Finite element analysis provides an alternative to often cumbersome trial and error approaches to improving reverberation room performance by adding diffusers. The model includes diffusers, wall sound absorption, and dissipation in air. This paper details the development of a model. From the model, it was observed that the existing diffusers are effective above the 160 Hz center frequency especially when an absorber sample is introduced. The model will be used to drive further modifications to the room.

8 ACKNOWLEDGMENTS

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