April 8, 2021

Design of Acoustic Enclosures

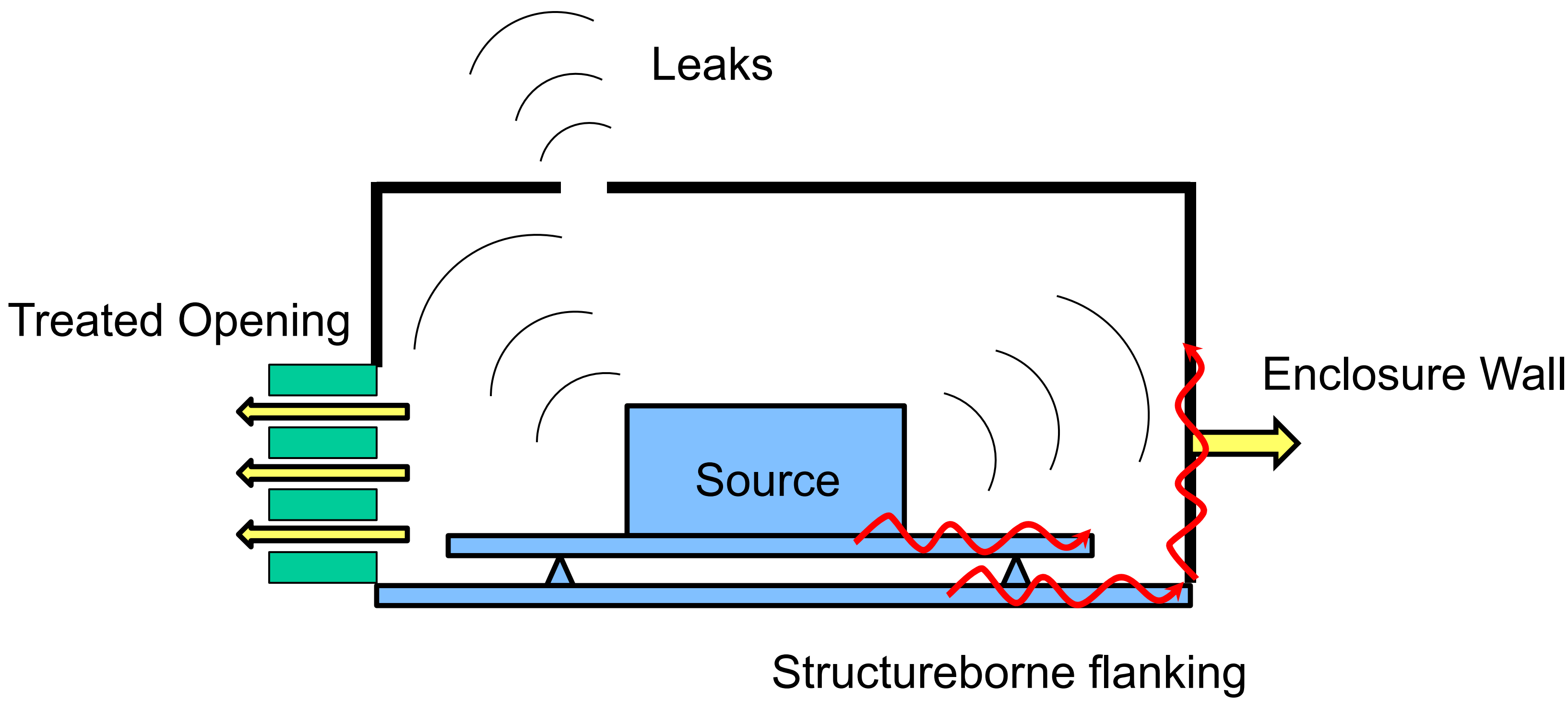
David Herrin
University of Kentucky

Vibro-Acoustics Consortium

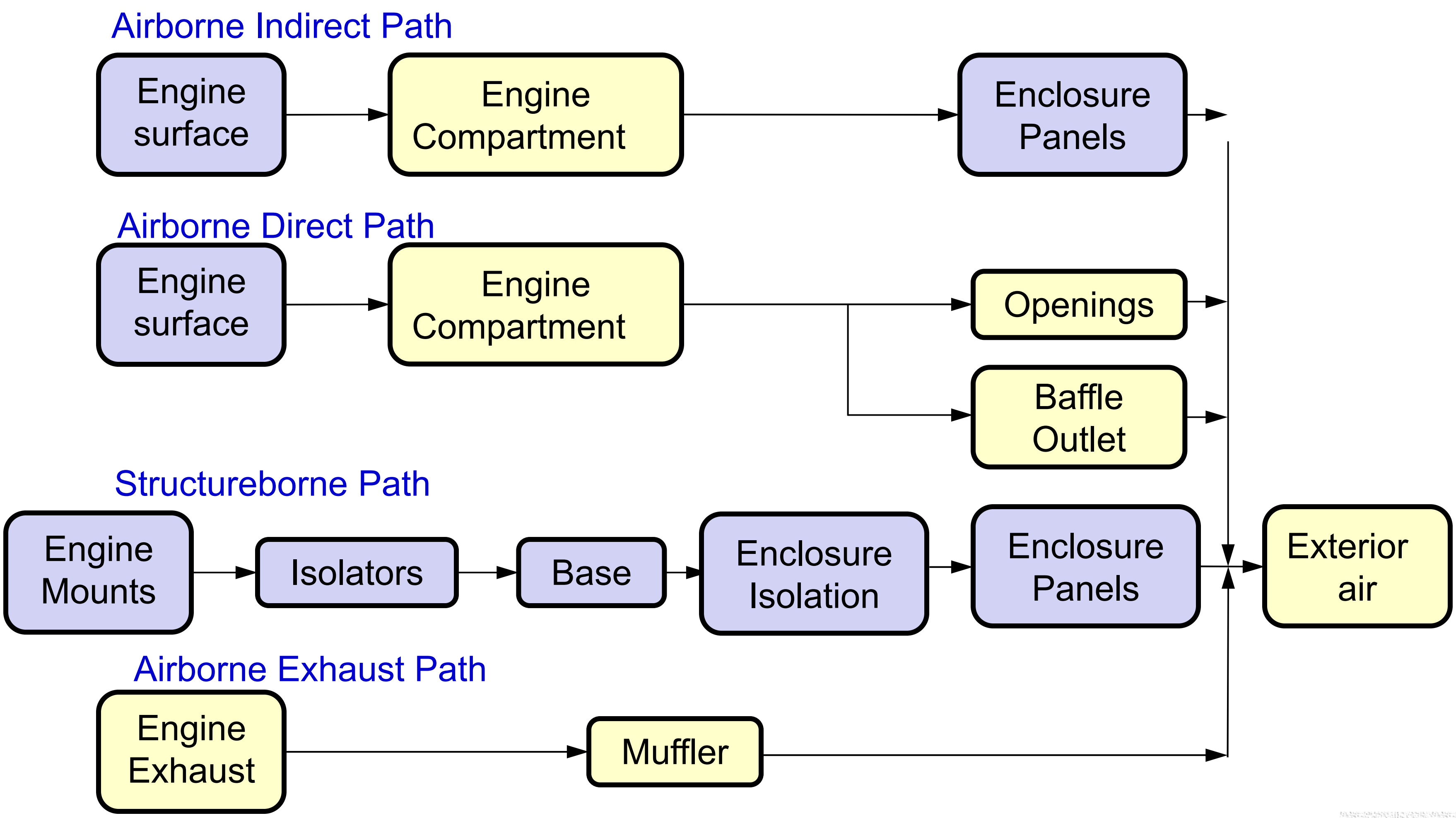
Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

Partial Enclosure



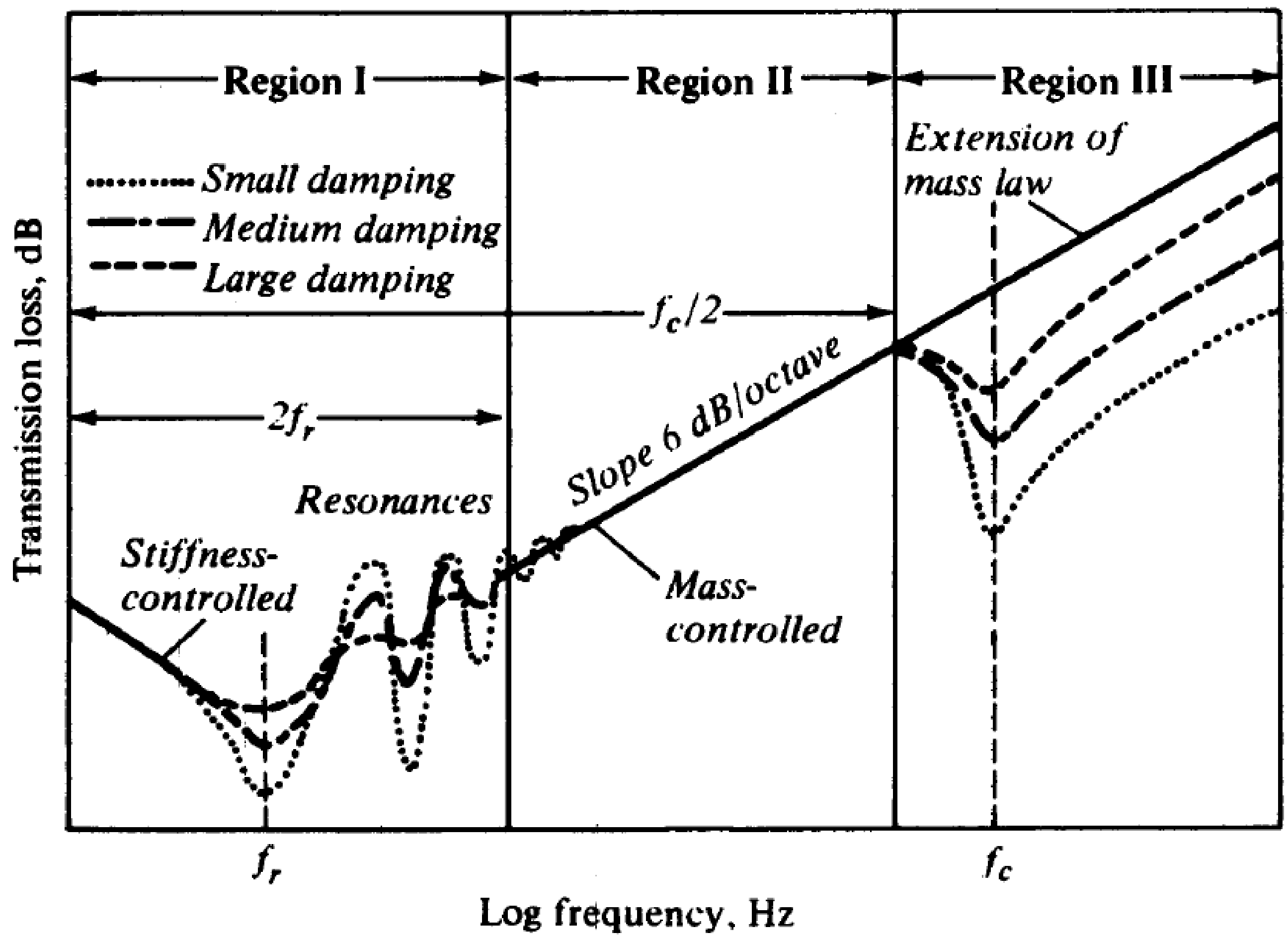
Source Path Receiver Map



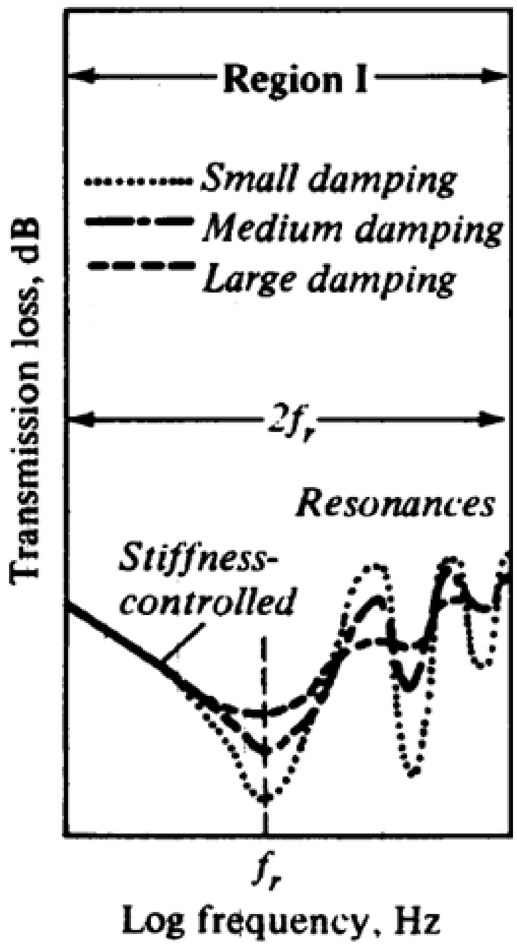
Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

Sound Transmission Through Thin Panel



Region 1 Resonance Controlled



Below 1st Panel Resonance

- The response is determined by the panel's static stiffness.
- Higher stiffness, higher transmission loss.

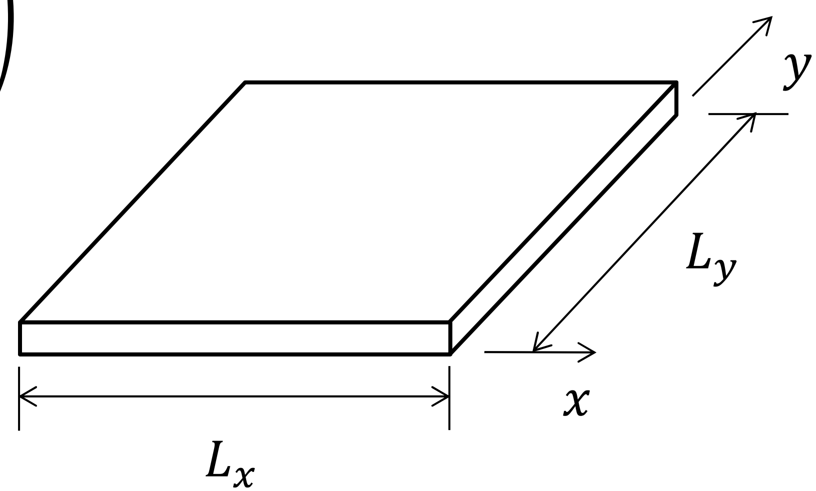
At and Above 1st Panel Resonance

- The response is determined by the resonant modes.

1st Panel Resonance

For simply-supported rectangular panel:

$$f(n_x, n_y) = \frac{\pi}{2}\sqrt{\frac{Eh^2}{12\rho}}\left(\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2\right)$$

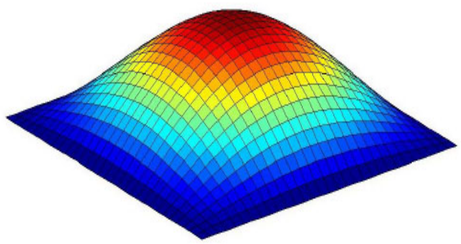
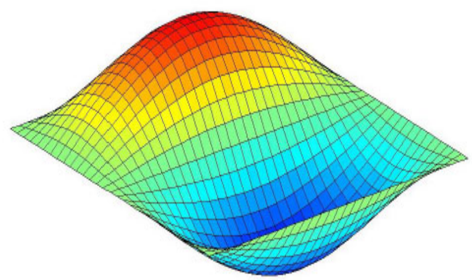
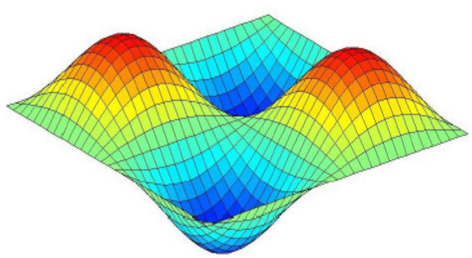
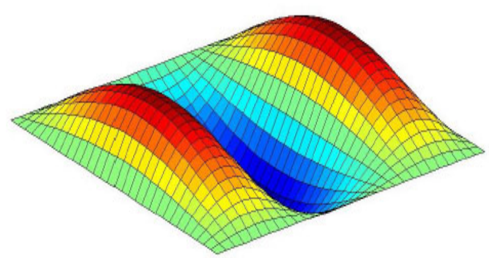


where:

E	Young's modulus
h	plate thickness
ρ	density
n_x	x mode index
n_y	y mode index
L_x	plate width in x direction
L_y	plate width in y direction

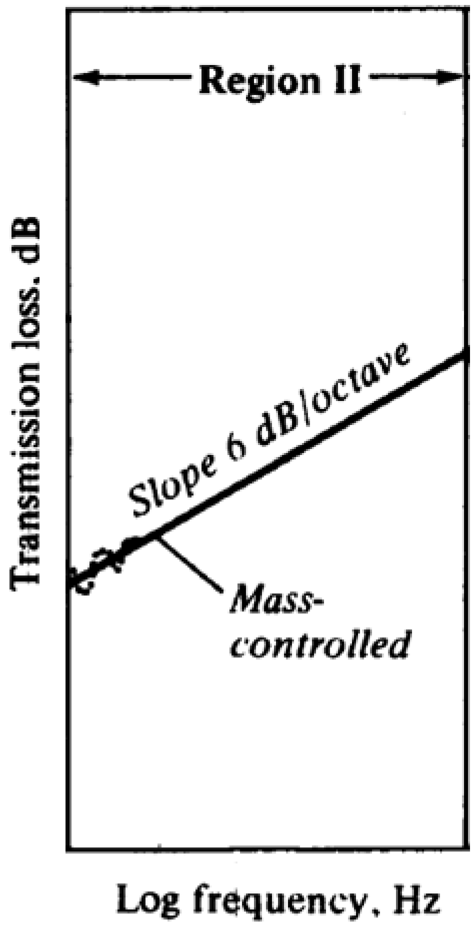
Panel Resonances

First 4 modes of a 30 inch square steel plate which is 1/8 inch thick.

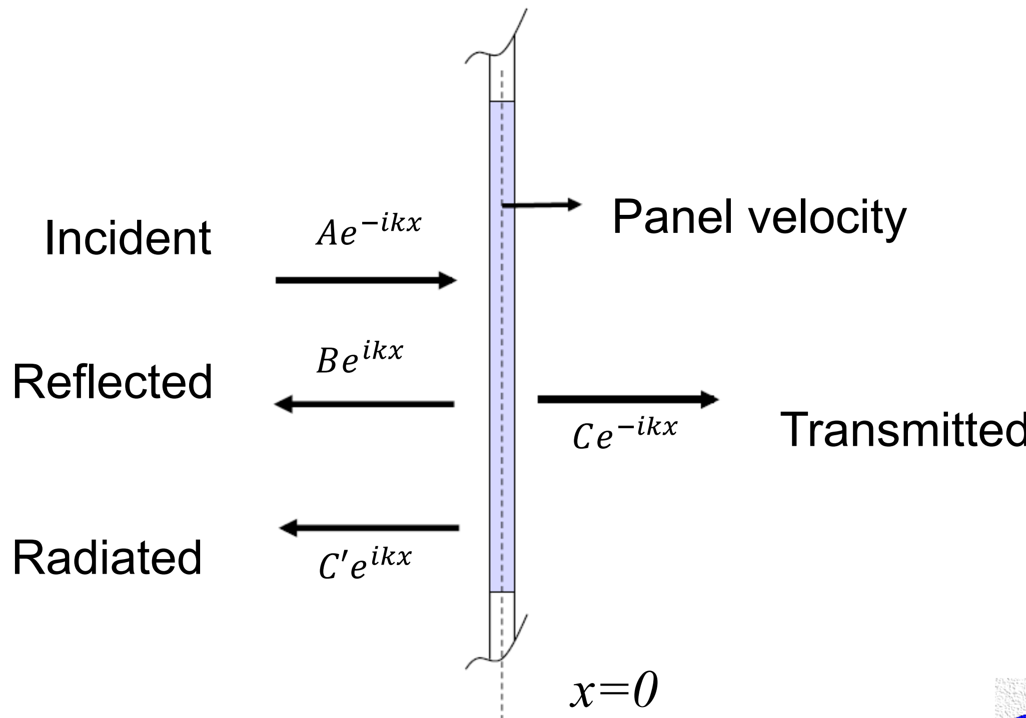
Index	$n_x = 1, n_y = 1$	$n_x = 2, n_y = 1$
Mode shape		
Frequency	25.5 Hz	63.7 Hz
Index	$n_x = 2\ n_y = 2$	$n_x = 1, n_y = 3$
Mode shape		
Frequency	101.9 Hz	127.4 Hz

If possible, avoid first several resonances in the frequency range of interest.

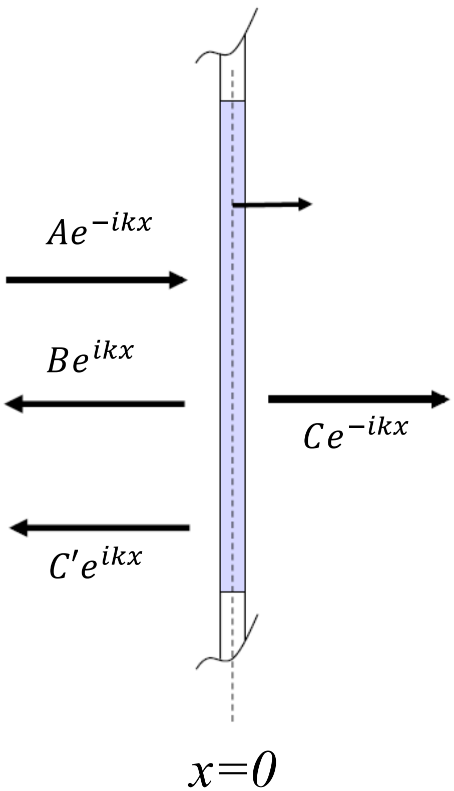
Region 2 Limp Panel Theory



Assumption:
- Panel is homogeneous
- Stiffness and damping ignored – mass only



Normal Incidence Transmission Loss



Define τ transmission coefficient:

$$\tau = \frac{I_t}{I_i} = \frac{|C|^2/\rho_0 c}{|A|^2/\rho_0 c} = \frac{|C|^2}{|A|^2} = \frac{1}{1 + (\omega \rho_s / 2\rho_0 c)} \approx \frac{1}{\omega \rho_s / 2\rho_0 c}$$

where:

$\rho_s = m/S$ Panel surface density

$$TL_0 = 10\log_{10}\left(\frac{1}{\tau}\right) = 20\log_{10}(\rho_S f) - 42 \quad \text{(dB)}$$

Mass Law: Higher surface density, higher TL.

Oblique Incident Sound Transmission

Diffusive sound field: plane waves of the same average intensity travelling with equal probability in all directions.

$$\tau = \tau(\varphi)$$

$$\bar{\tau} = \frac{\int_0^{\varphi_{\lim}} \tau\,(\varphi)\cos\varphi\,\sin\varphi\,d\varphi}{\int_0^{\varphi_{\lim}} \cos\varphi\,\sin\varphi\,d\varphi}$$

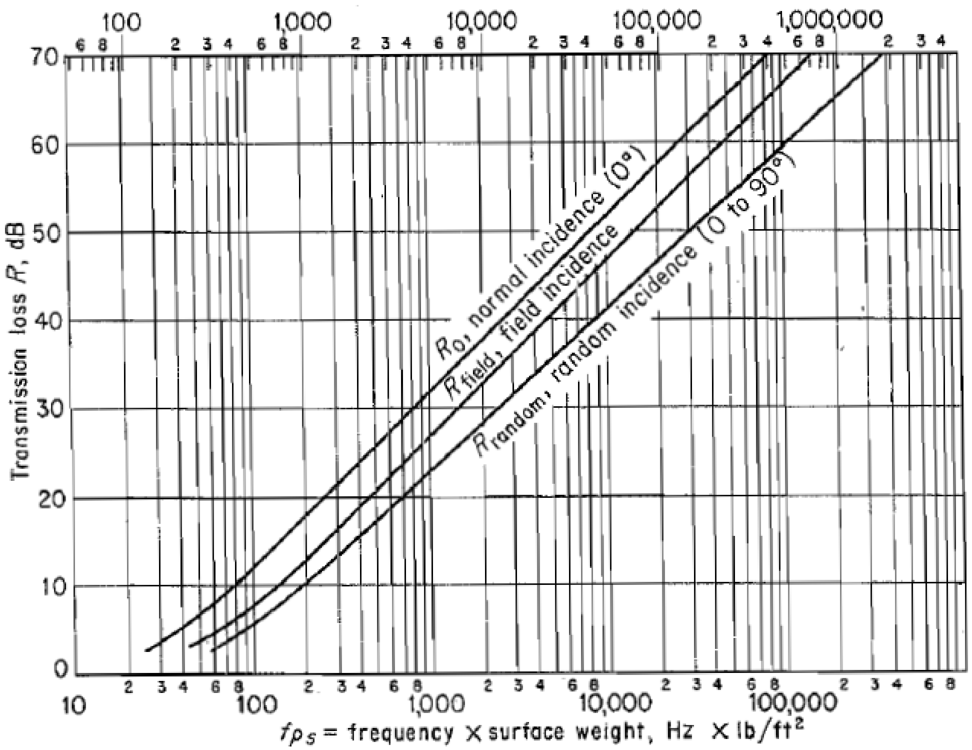
For field incidence (better agreement with measurement) $\varphi_{\lim} = 78°$

$$TL_{Field} = TL_0 - 5.5\ \text{dB}$$ 1/3 octave bands

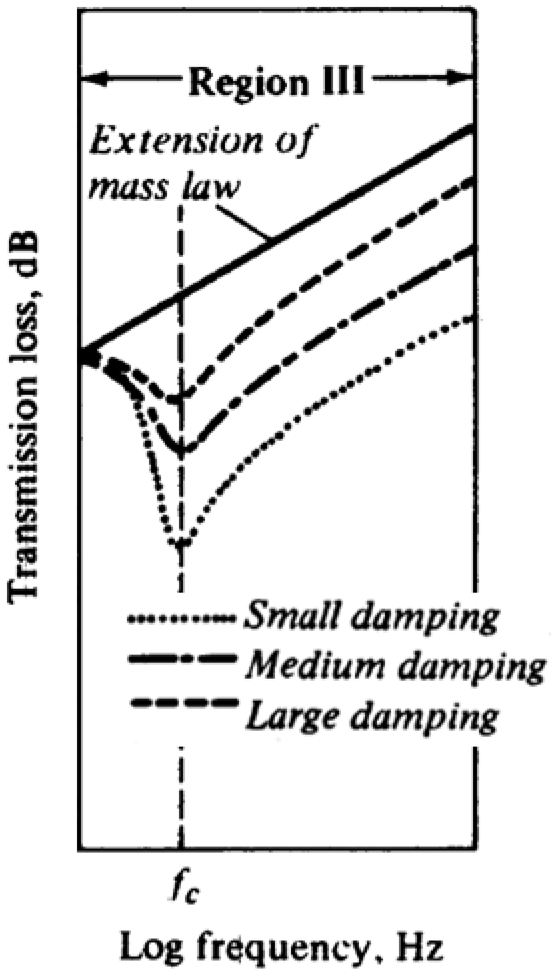
$$TL_{Field} = TL_0 - 4.0\ \text{dB}$$ octave bands

Field Incidence

Theoretical sound transmission loss of large panels for frequencies in Region 2:



Region 3 Coincidence Effect



This pronounced dip in transmission loss curve occurs when the wavelength of sound in the air coincides with the structural wavelength. This frequency is called critical frequency.

$$f_C = \frac{c^2}{2\pi}\sqrt{\frac{\rho_S}{D}}$$

where:

$\rho_s = m/S$ Panel surface density

$D = \frac{Eh^3}{12(1-v^2)}$ Bending stiffness of plate

Radiation Efficiency

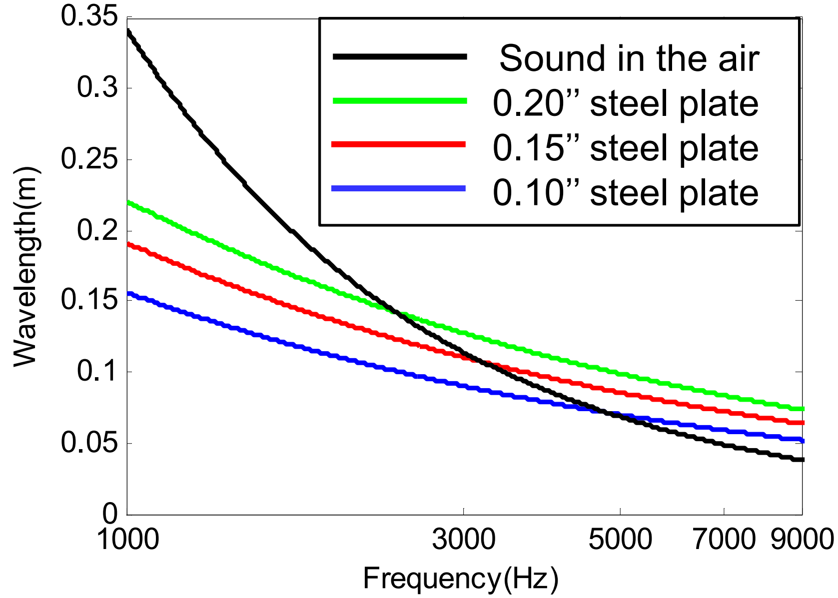
In thin plates, the dominating vibration will be bending vibration. Unlike an acoustic wave, bending wave speed is dependent on frequency.

Plate bending

$$c_p = \sqrt[4]{\frac{D\omega^2}{\rho_S}} \qquad \lambda_p = \sqrt{\frac{2\pi}{f}}\sqrt[4]{\frac{D}{\rho_S}}$$

Sound in air

$$\lambda_a = \frac{c}{f}$$

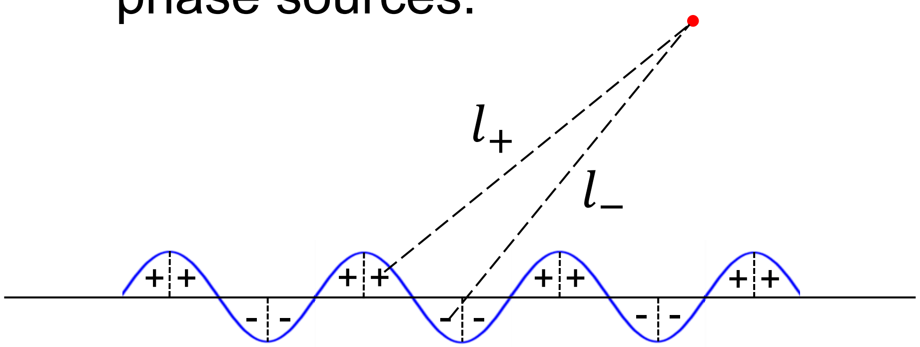


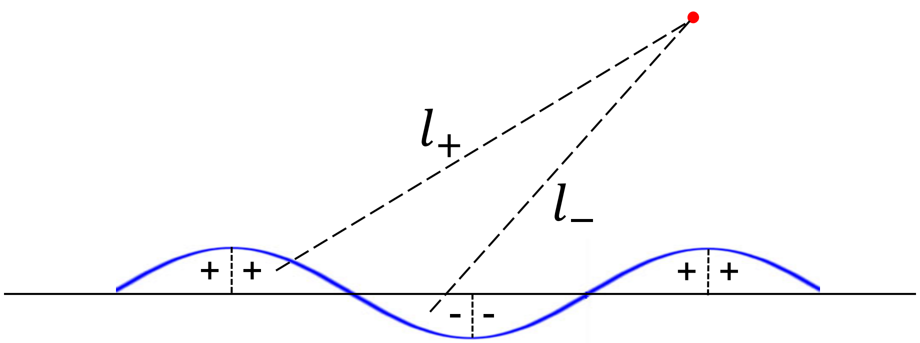
Radiation Efficiency

$\lambda_p \ll \lambda_a \qquad \Delta l = l_+ - l_- \approx 0$

$\lambda_p \approx \lambda_a \qquad \Delta l \gg 0$

The plate will perform like closely distributed out-of-phase sources.





Radiation Efficiency

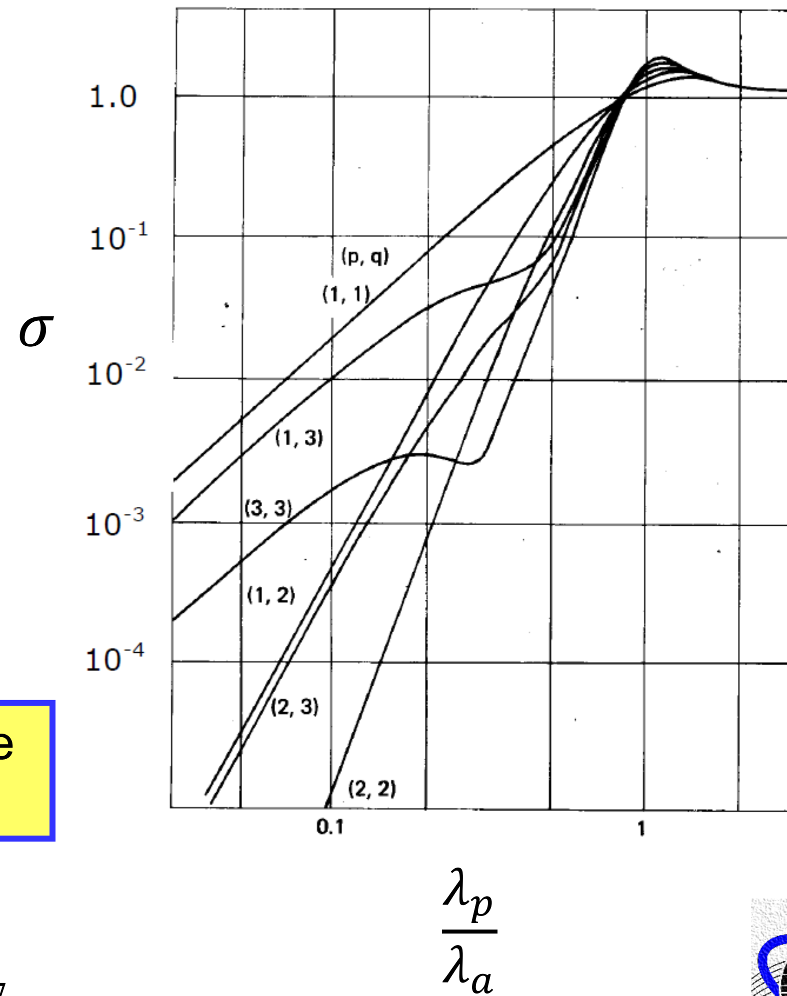
Define radiation efficiency:

$$\sigma = \frac{W}{\rho c S \langle v_n^2 \rangle}$$

where:

W	Actual sound power radiated
v_n	Mean square normal velocity
S	Panel area

Around and above critical frequency, the thin panels are very efficient radiators.



Effect of Thickness

Increase TL according to Mass Law

$$TL_0 = 20\log_{10}(\rho_s f) - 42 \qquad h \uparrow$$

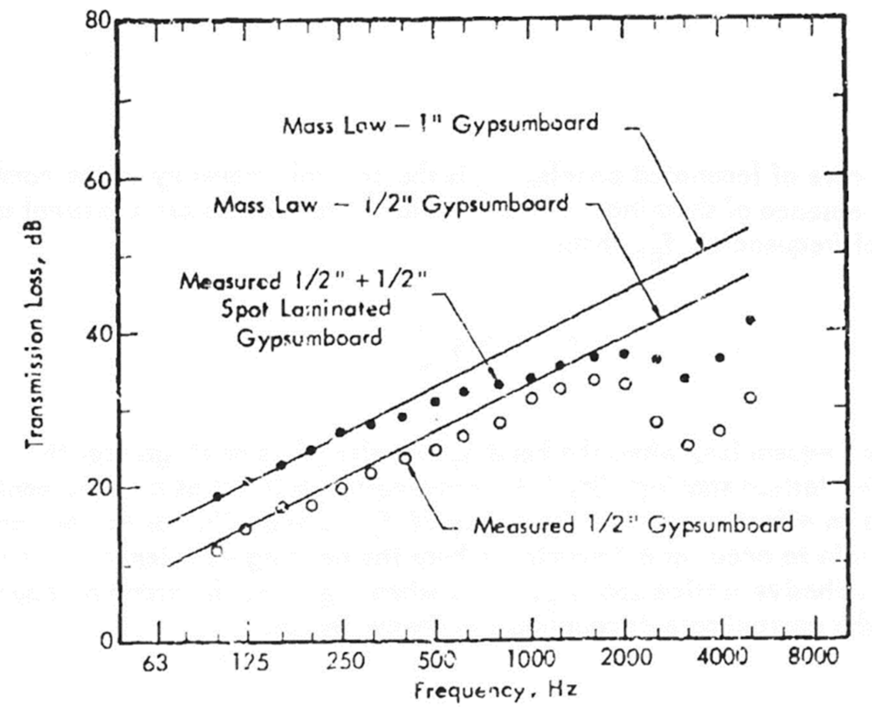
Shift critical frequency above range of interest

$$f_C = \frac{c^2}{2\pi}\sqrt{\frac{12(1-v^2)\rho_S}{Eh^3}} \qquad h \downarrow$$

Poses a dilemma due to inconsistent requirement.

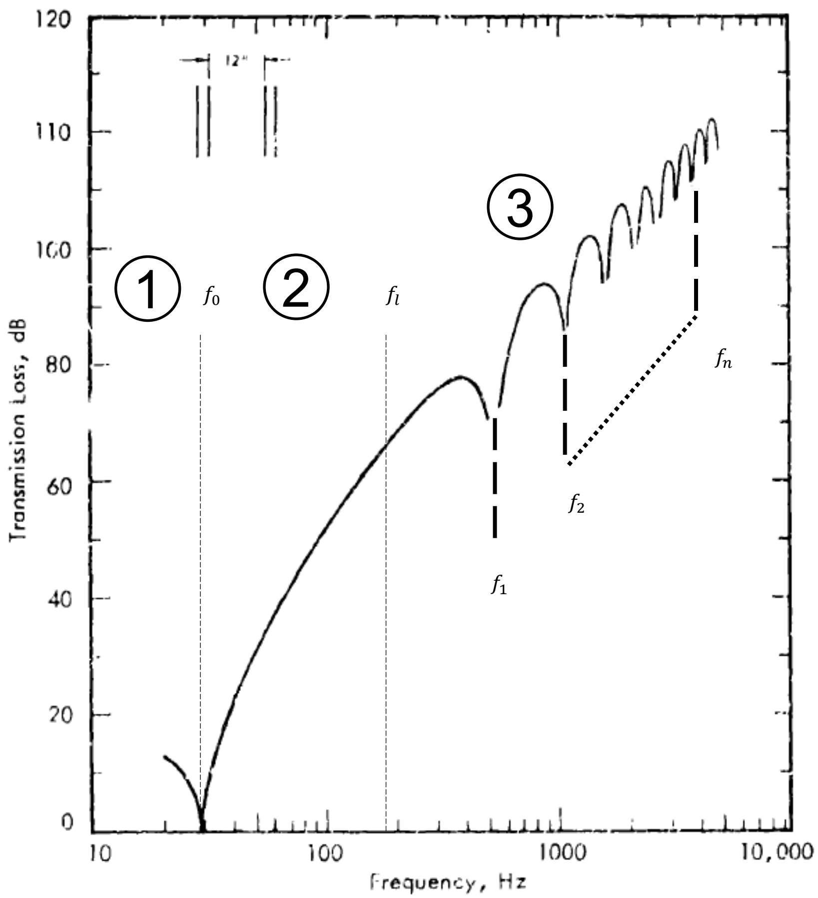
Various Designs

Laminated Panels



Single 1-inch and two 1/2-inch spot laminated sheets of gypsum board

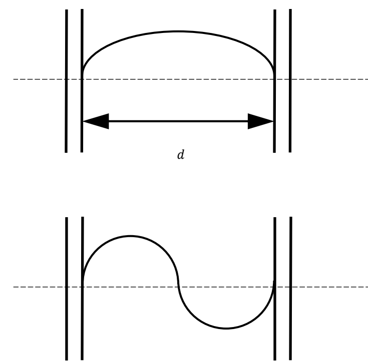
Double Panels



Fundamental resonant frequency:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{\rho_0 c^2}{\rho_S' d}} \qquad \rho_S' = \frac{\rho_{S1}\rho_{S2}}{\rho_{S1}+\rho_{S2}}$$

Cavity resonant frequency:



$$f_1 = \frac{c}{2d}$$

$$f_2 = \frac{c}{d} \cdots\cdots\cdots f_n = \frac{nc}{2d}$$

$$f_l = \frac{c}{2\pi d} = \frac{f_1}{\pi}$$

Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

For Composite Panels

For each part of the panel

$$\tau_i = \frac{1}{1 + \left(\frac{\omega m_i}{2\rho_0 c S_i}\right)^2} \approx \left(\frac{2\rho c S_i}{\omega m_i}\right)^2 = \left(\frac{\rho c}{\pi f \rho_{si}}\right)^2$$

For composite panels

$$\tau_{total} = \frac{\sum \tau_i S_i}{S_{total}}$$

Transmission loss

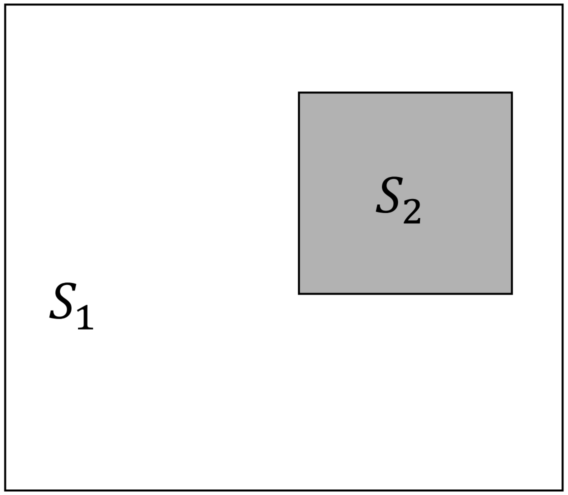
$$TL_{total} = 10 \log_{10} \frac{1}{\tau_{total}}$$

Transmission Loss of a Composite Panel

n	S_n	TL_i	τ_n
1	2	20	0.01
2	2	10	0.1

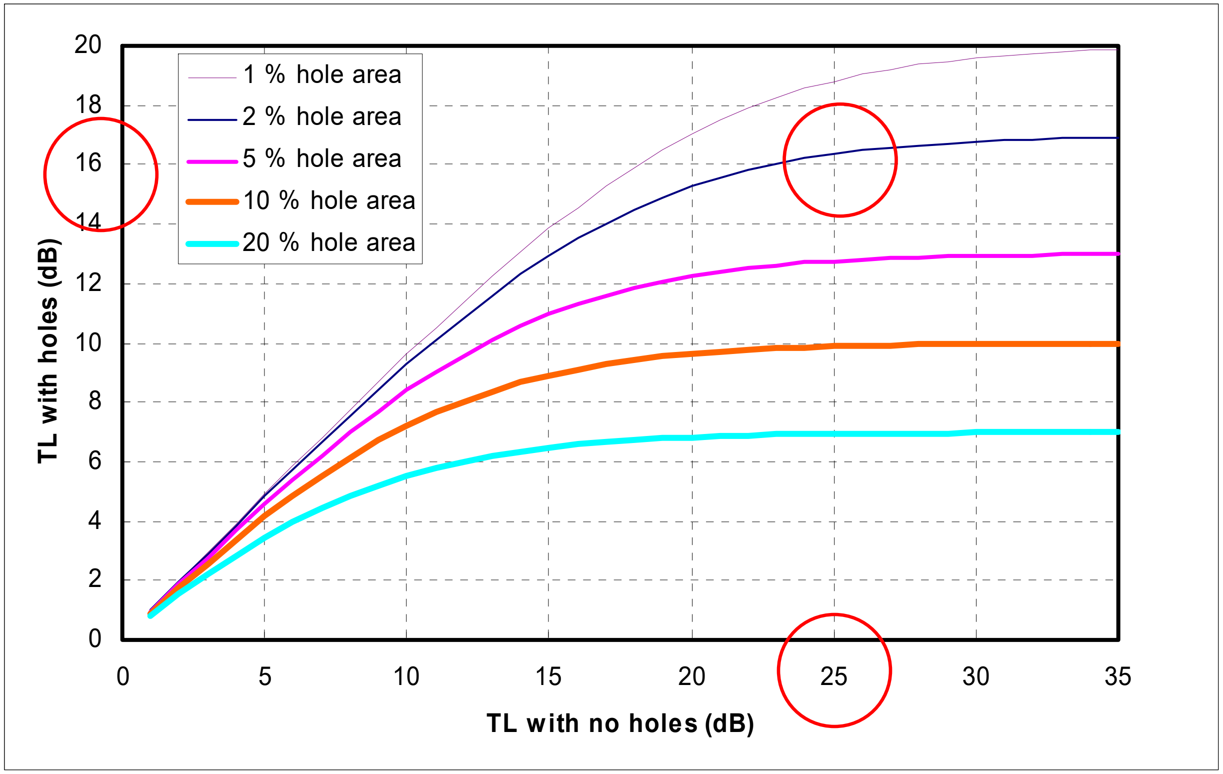
$$\tau = \frac{1}{4}(0.01 \cdot 2 + 0.1 \cdot 2) = 0.055$$

$$TL = 10 \log_{10} \frac{1}{0.055} = 12.6 \text{ dB}$$



Transmission Loss of a Composite Panel

Assume $\tau = 1$ for openings.



Transmission Loss of Slits

For $kw < 0.5$

Slit in the center of a wall

$$\tau = \frac{2w}{kl^2}$$

Slit on edge of wall

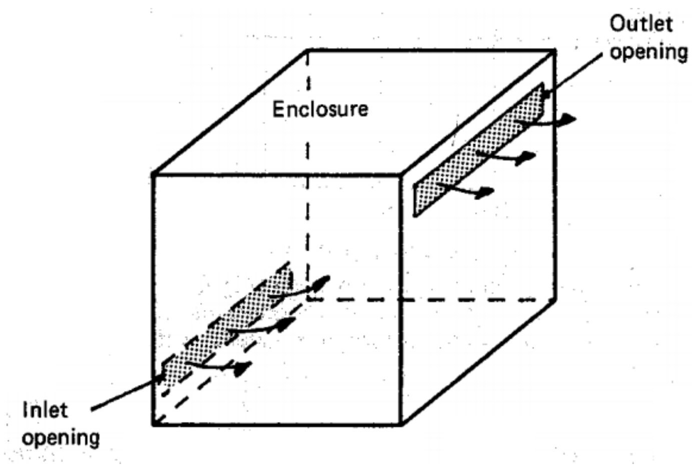
$$\tau = \frac{4w}{kl^2}$$

w width or height of slit (use 1.2 mm for doors with no seals)

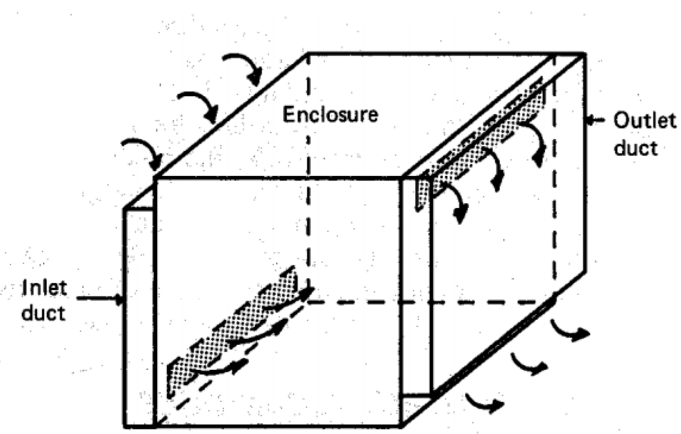
l depth of the slit

k wavenumber

Treatment with Ventilation Openings



Untreated

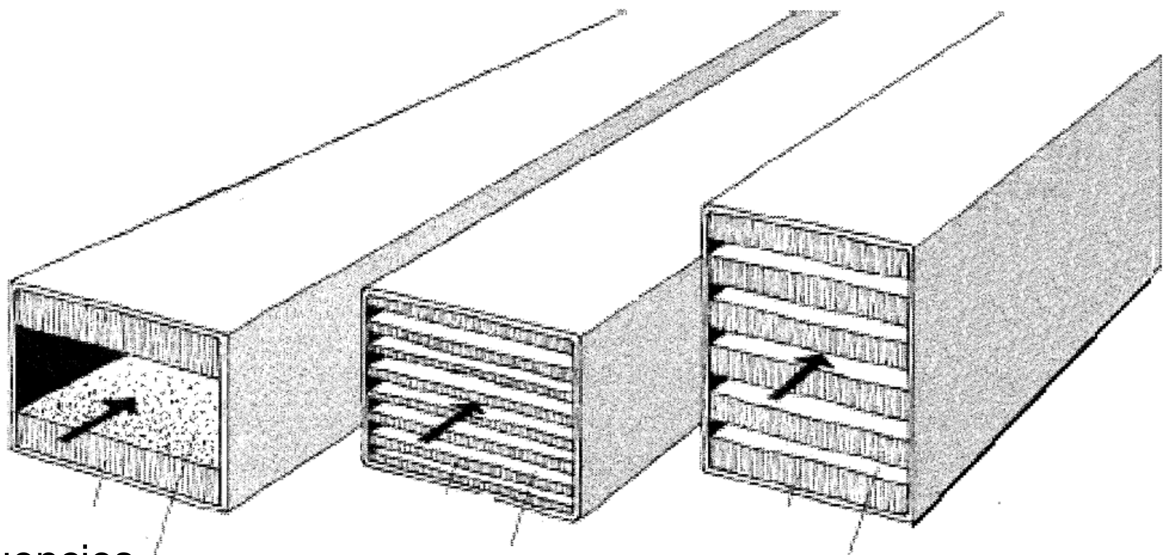


Treated

Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

Baffle Silencers Examples

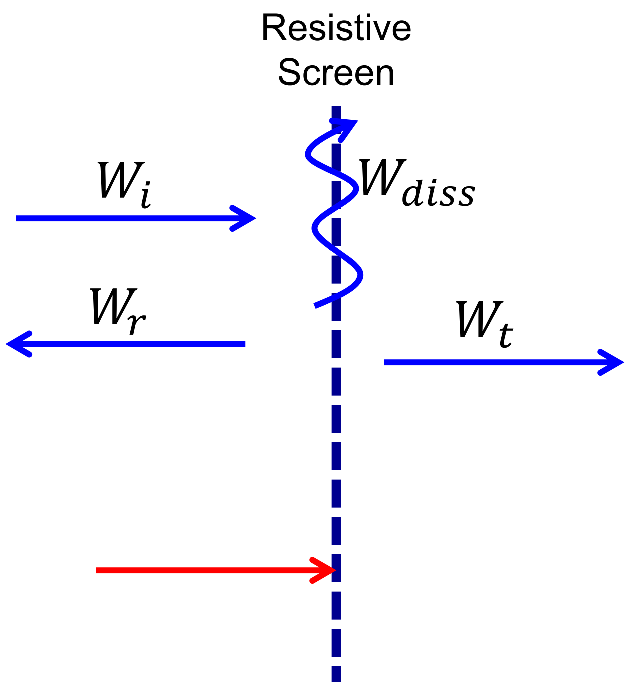


Low frequencies
Thick absorbent
Large air gap

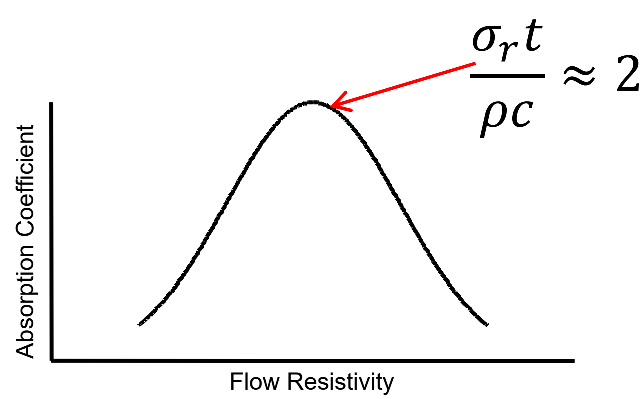
High frequencies
Thin absorbent
Small air gap

Both low and high frequencies
Thick absorbent
Small air gap

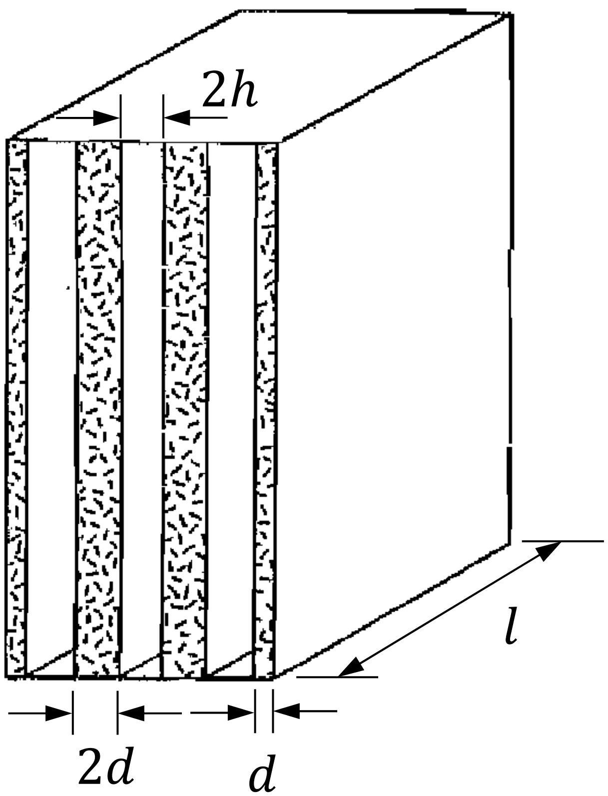
Absorption materials Flow Resistivity σ



In theory, the dissipated power (W_{diss}) is a maximum when $\sigma_r t = 2\rho c$. A general rule of thumb is that a sound absorber will be effective when $\sigma_r t \approx n\rho c$ where n is on the order of 2. This assumes that the acoustic resistance is equal to the static flow resistance.



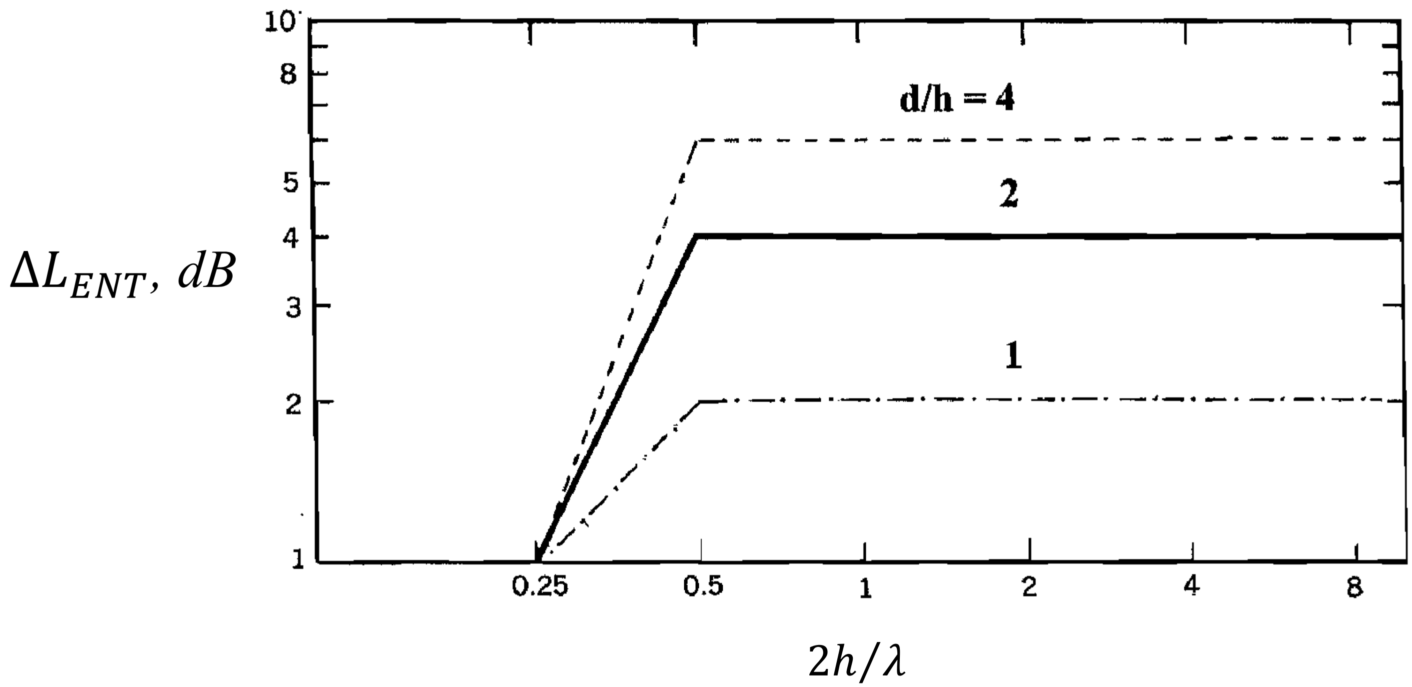
Baffle Silencers Design Parameters



$$IL = \Delta L_{ENT} + L_h \frac{l}{h}$$

ΔL_{ENT}	Entrance loss
L_h	Normalized attenuation

Baffle Silencers Entrance Loss



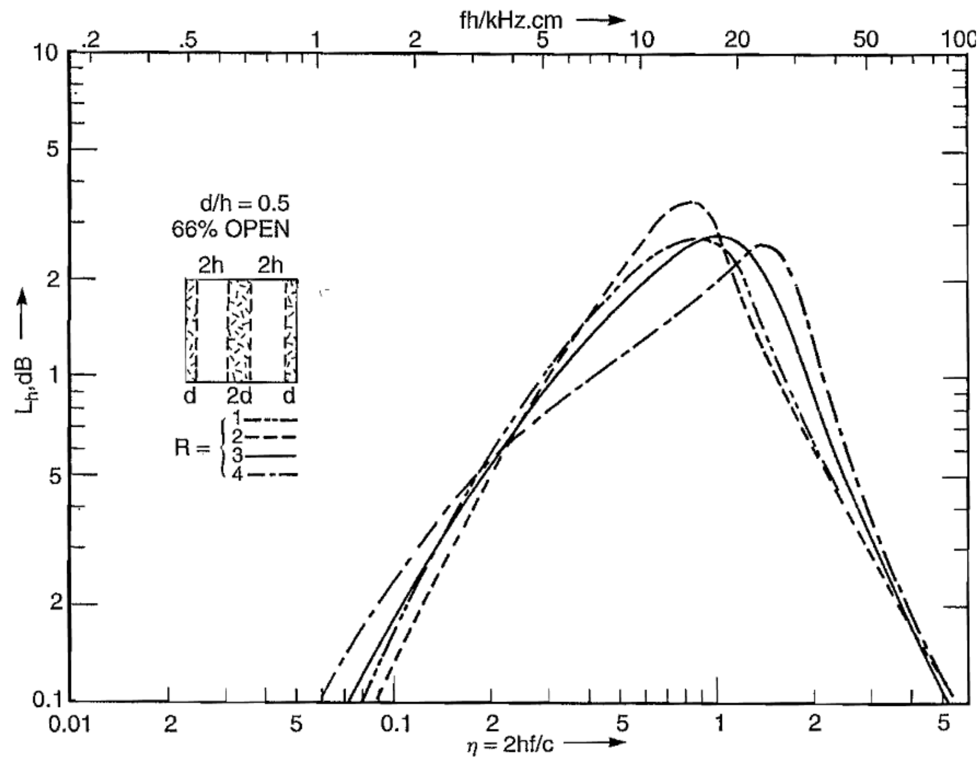
The entrance loss is high when the air gap is comparable to a wavelength.

Baffle Silencers Normalized Attenuation L_h

Normalized flow resistivity:

$$R = \frac{\sigma d}{\rho c}$$

The normalized attenuation L_h has been computed for various percentage of open area of silencer cross section and for various values of the normalized flow resistivity of absorption materials in the baffles.



Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

Sealed Enclosure

$$L_p = L_W - TL_{Field} - 10\log_{10} S_e + 10\log_{10}\left(0.3 + \frac{S_e(1-\bar{\alpha}_i)}{S_i\bar{\alpha}_i}\right)$$

L_W sound power of the source (dB)

S_e external surface area

S_i internal surface area

$\bar{\alpha}_i$ average sound absorption

Sealed Enclosure (Simplified Equation)

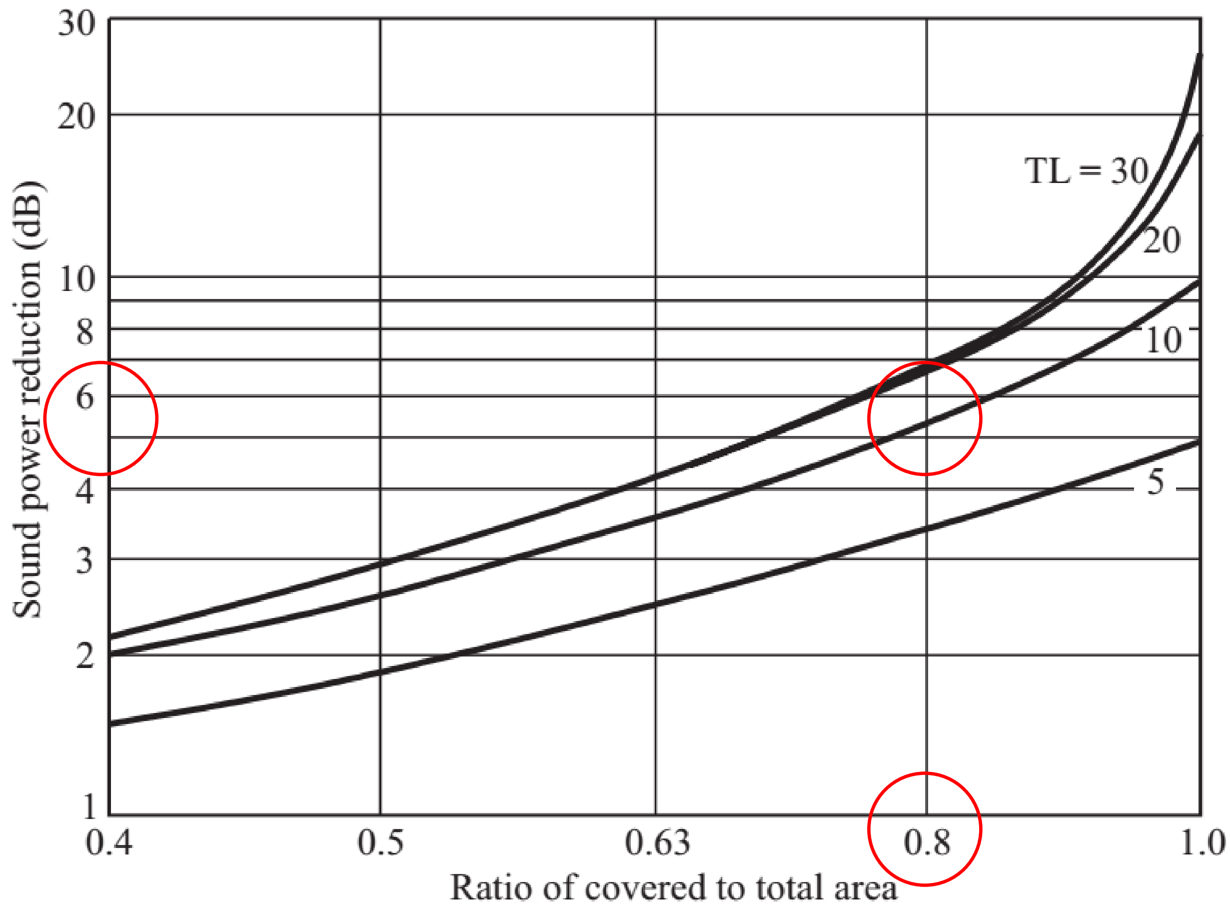
$$L_p = L_W - TL - 10\log_{10} S_e + C$$

TABLE 7.8 Values of coefficient, C (dB), to account for enclosure internal acoustic conditions. The following criteria are used to determine the appropriate acoustic conditions inside the enclosure

Enclosure internal acoustic conditions	Octave band centre frequency (Hz)							
	63	125	250	500	1000	2000	4000	8000
Live	18	16	15	14	12	12	12	12
Fairly live	13	12	11	12	12	12	12	12
Average	13	11	9	7	5	4	3	3
Dead	11	9	7	6	5	4	3	3

Live: All enclosure surfaces and machine surfaces hard and rigid.
Fairly live: All surfaces generally hard but some panel construction (sheet metal or wood).
Average: Enclosure internal surfaces covered with sound-absorptive material, and machine surfaces hard and rigid.
Dead: As for '**Average**', but machine surfaces mainly of panels.

Partial Enclosures (2 inch lining)



Partial Enclosure Insertion Loss

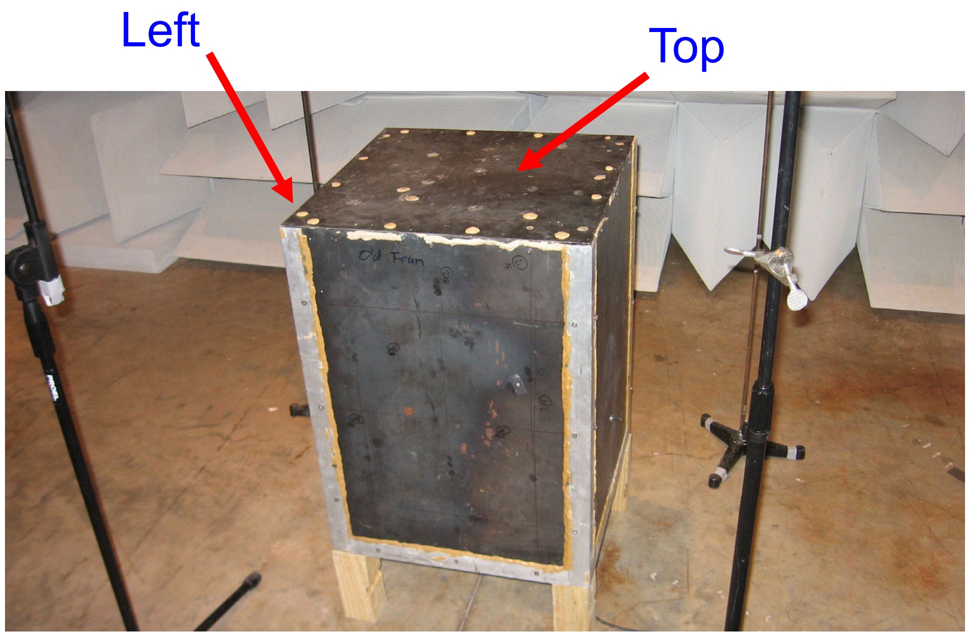
$$IL = 10\log_{10}\left[1 + \bar{\alpha}\left(\frac{\Omega_{total}}{\Omega_{open}} - 1\right)\right]\mathrm{dB}$$

$\bar{\alpha}$	average sound absorption
$\Omega_{total} = \frac{S_{total}}{r_{avg}^2}$	solid angle of enclosure
$\Omega_{opening} = \frac{S_{open}}{r_{open}^2}$	solid angle of enclosure opening
r_{avg}	average distance from source center to enclosure
$r_{opening}$	distance from source center to opening center

Overview

- Introduction
- Sound transmission through panels
- Sound transmission through leaks
- Baffle silencers
- Rudimentary equations for enclosure design
- Numerical simulation of partial enclosures

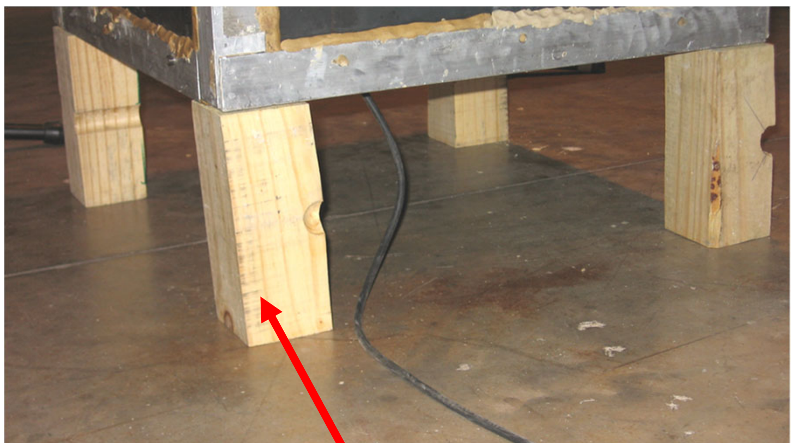
Test Case



- $0.48 \times 0.48 \times 0.66$ m^3
- Opening of radius 0.051 m
- Top and left panels (1 mm thick steel)
- All other panels (2 mm thick steel)

Measurement Setup

Sound Absorption Material



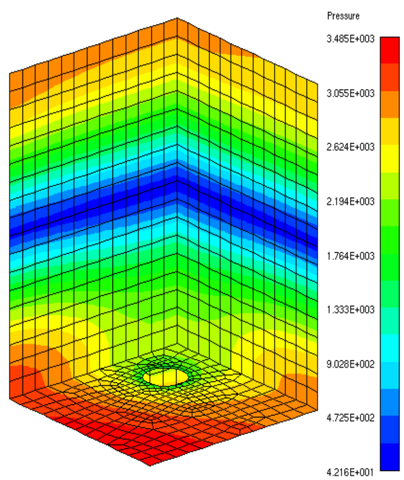
Wood Blocks

Opening Area

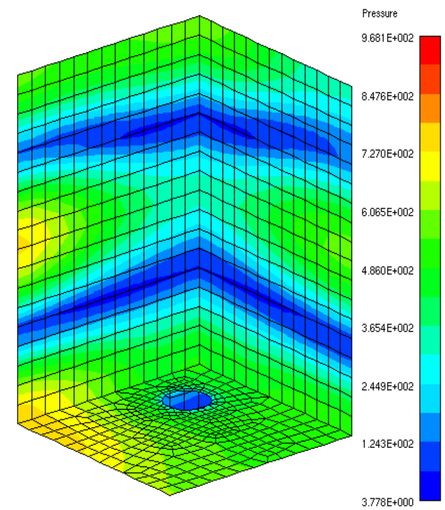
Enclosure Modes

$$f_{lmn} = \frac{c}{2}\sqrt{\left(\frac{l}{L_x}\right)^2 + \left(\frac{m}{L_y}\right)^2 + \left(\frac{n}{L_z}\right)^2}$$

$$l, m, n = 0, 1, 2, 3 \ldots N$$

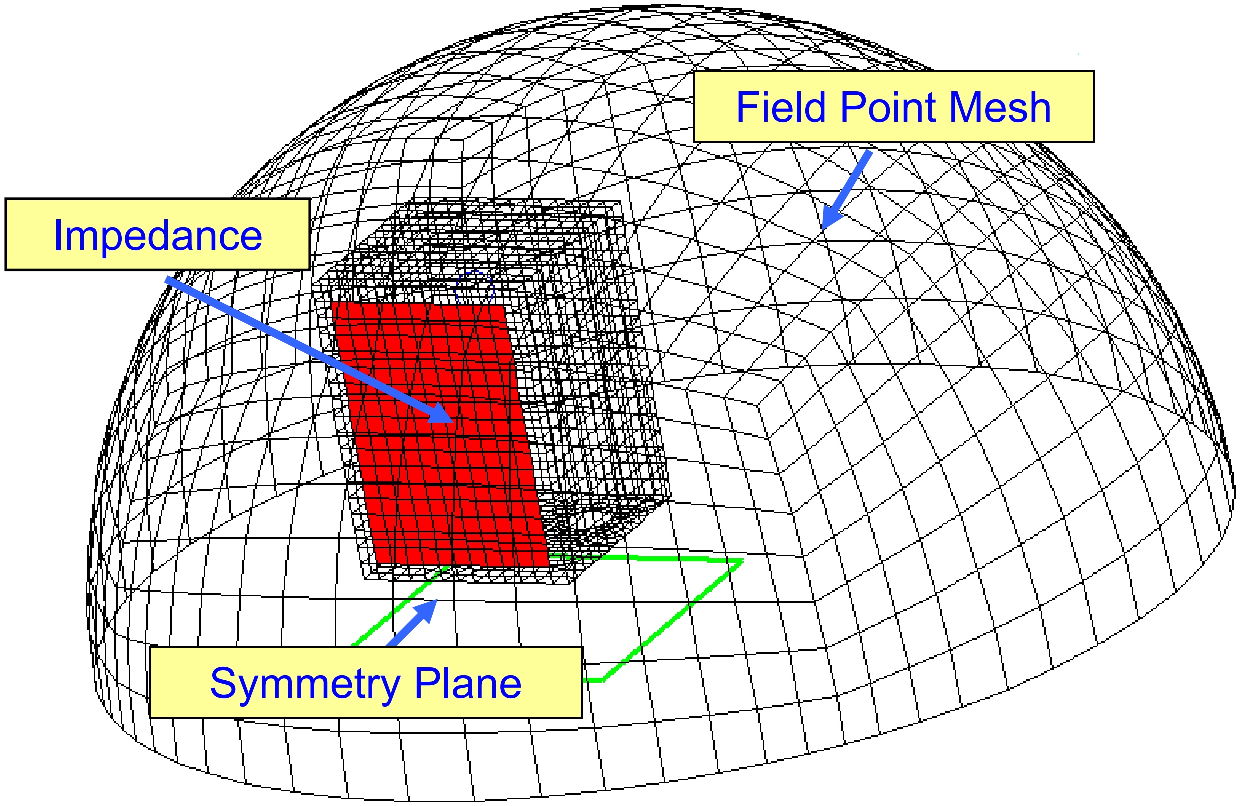


$f_{0,0,1} = 260$ Hz

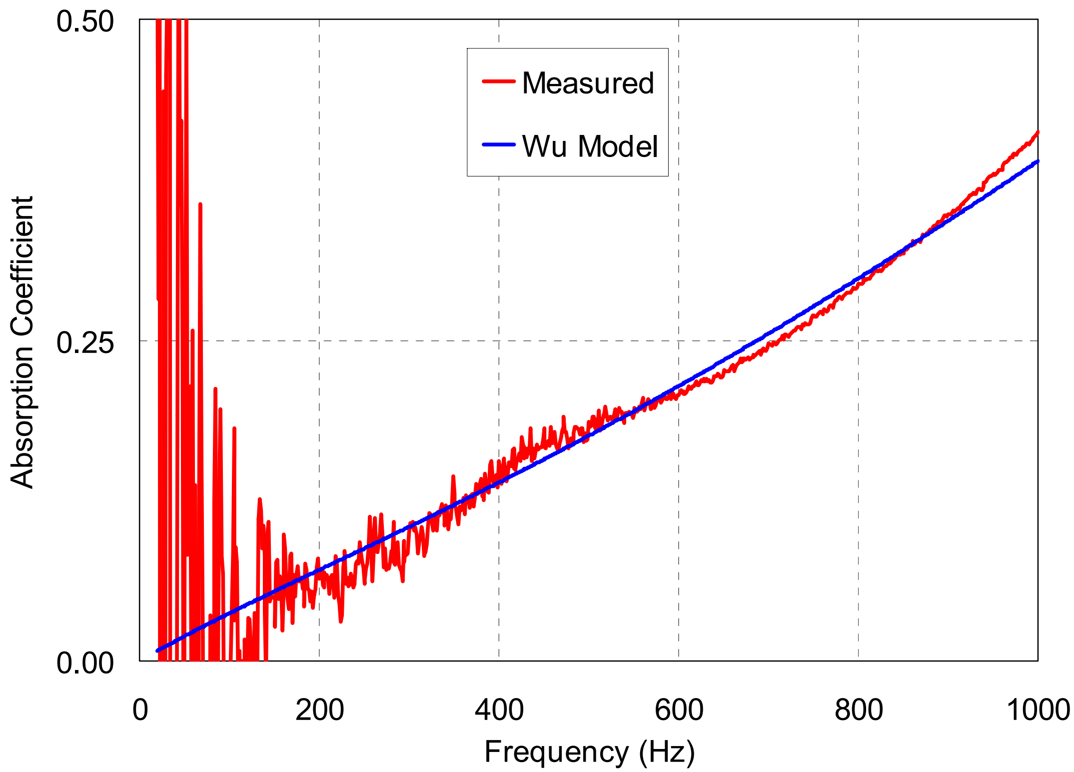


$f_{0,0,2} = 519$ Hz

Indirect BEM

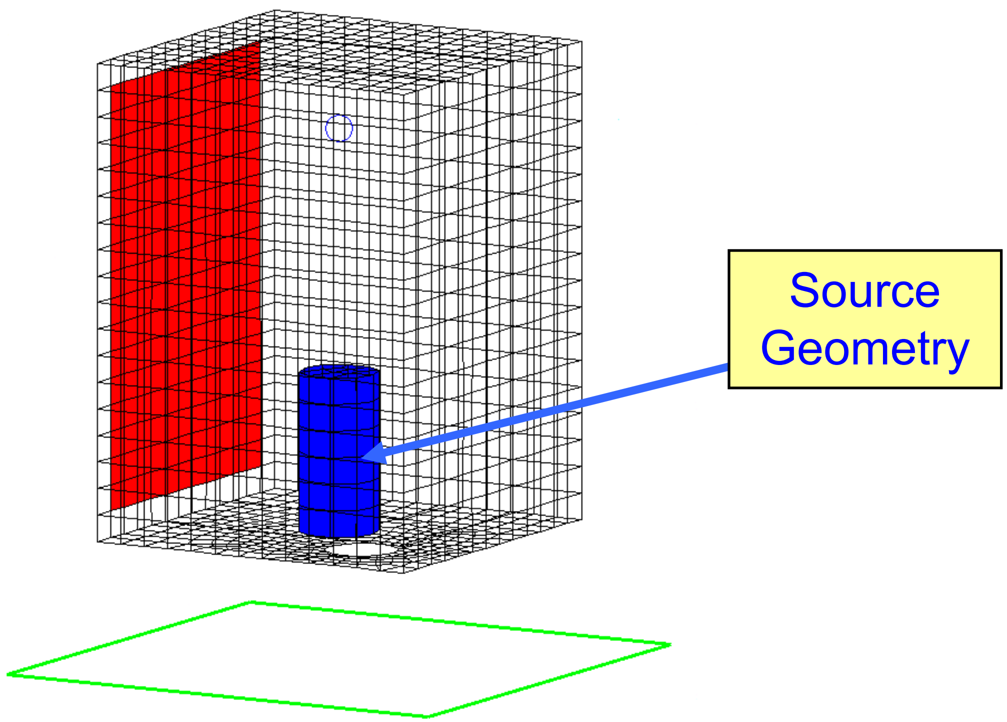


Modeling Approach Impedance

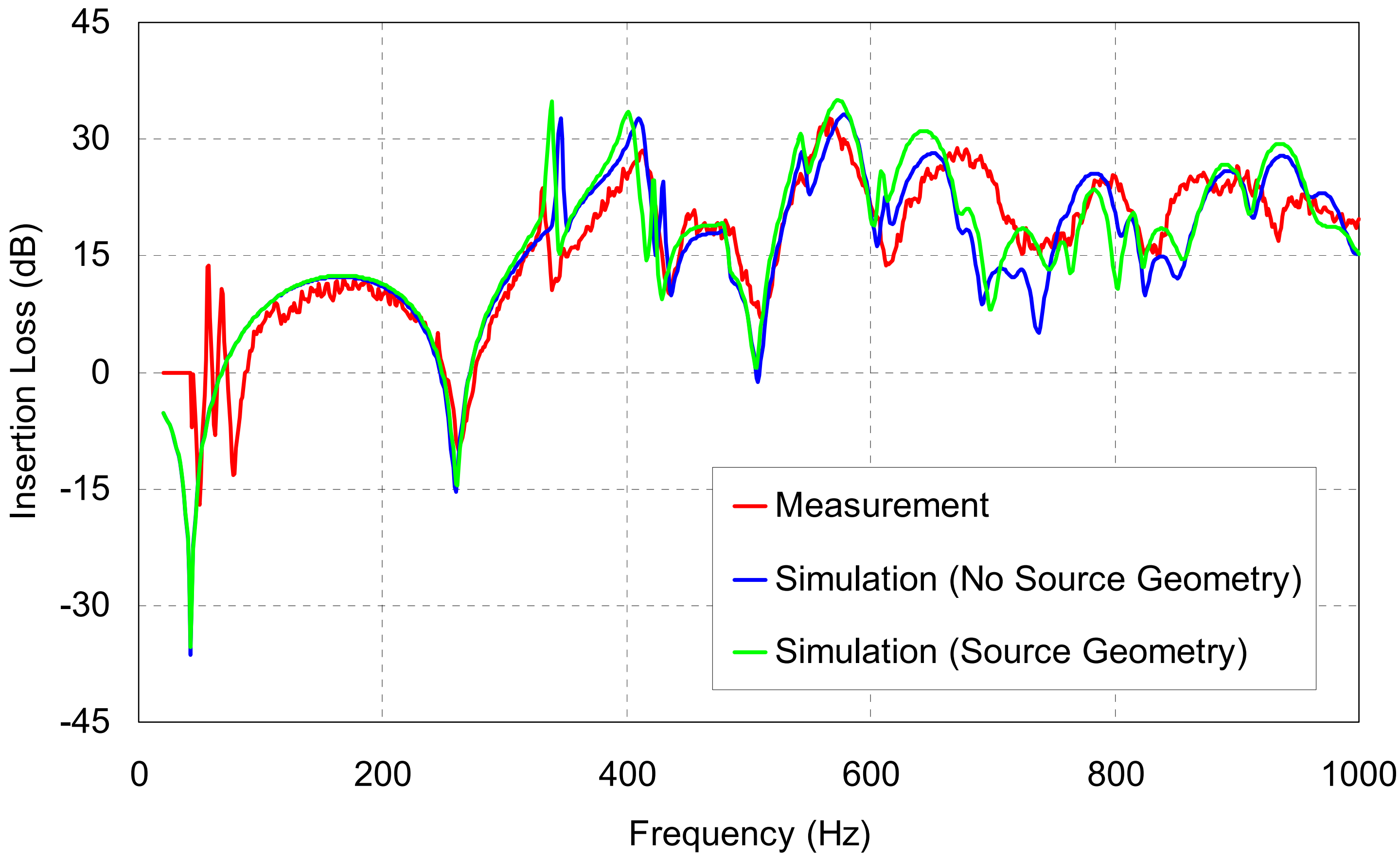


Empirical model provides a more reasonable estimate of the sound absorption at low frequencies.

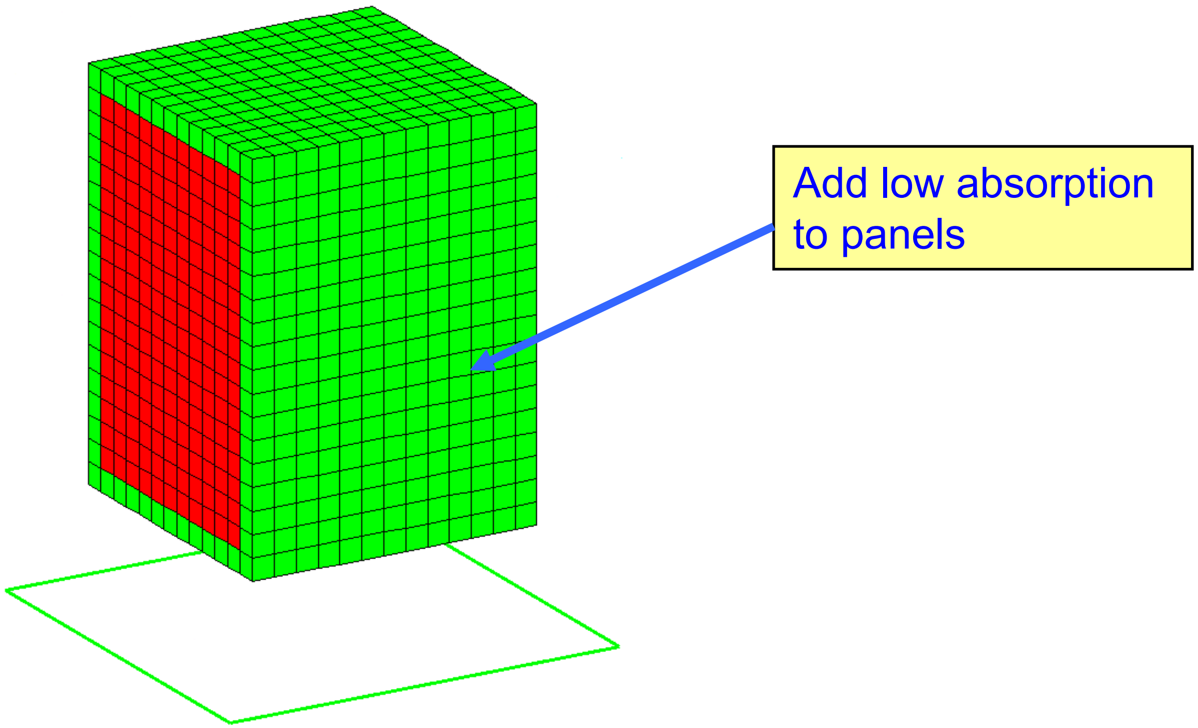
Modeling Approach Source Geometry



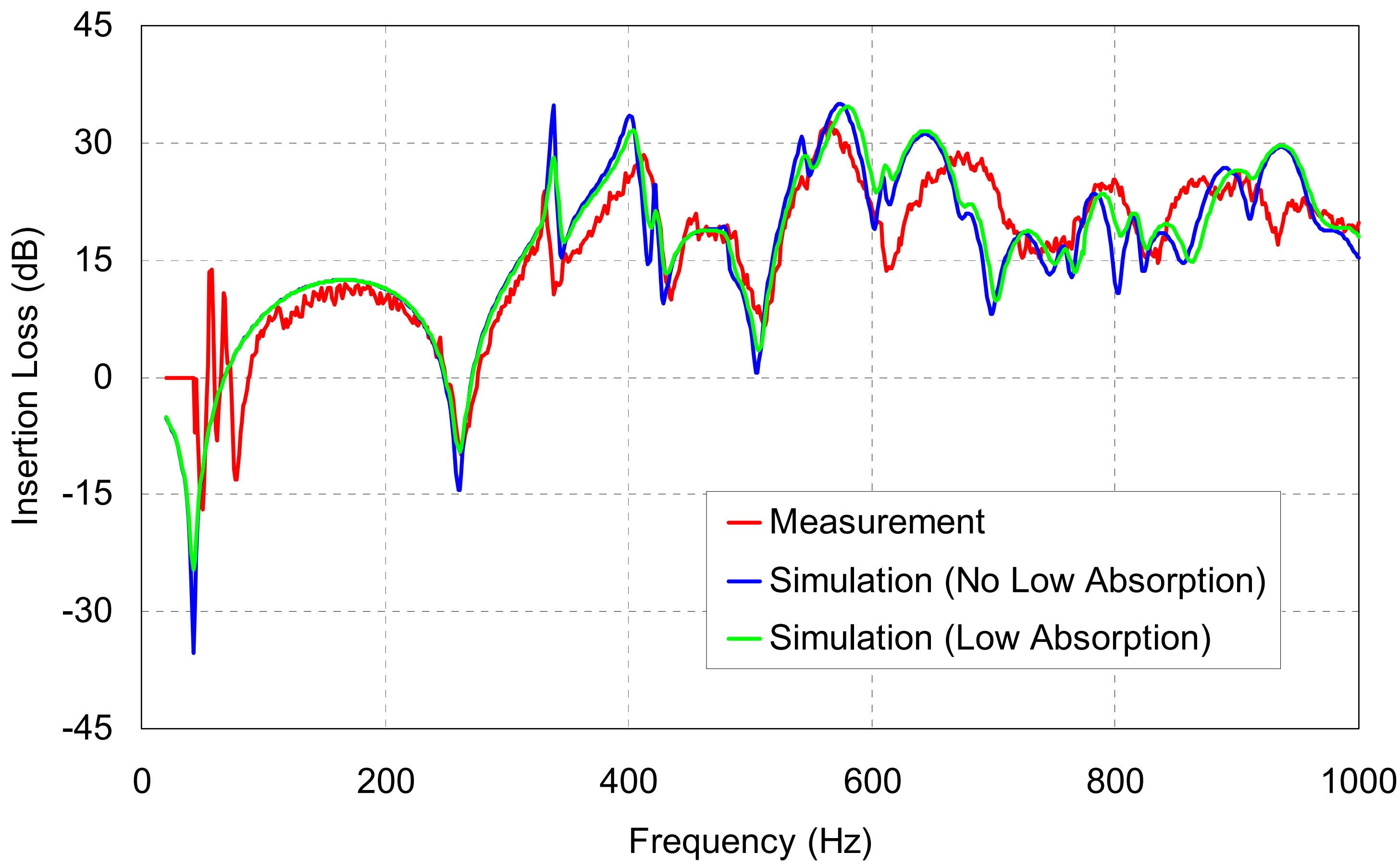
Modeling Approach Source Geometry



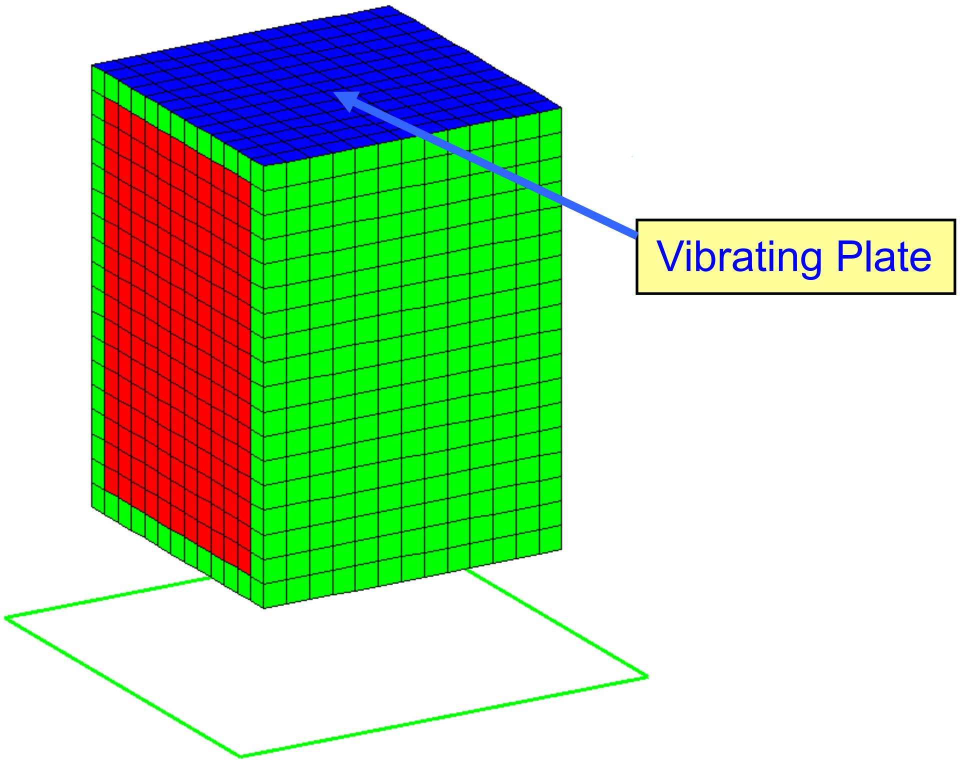
Modeling Approach Panel Absorption



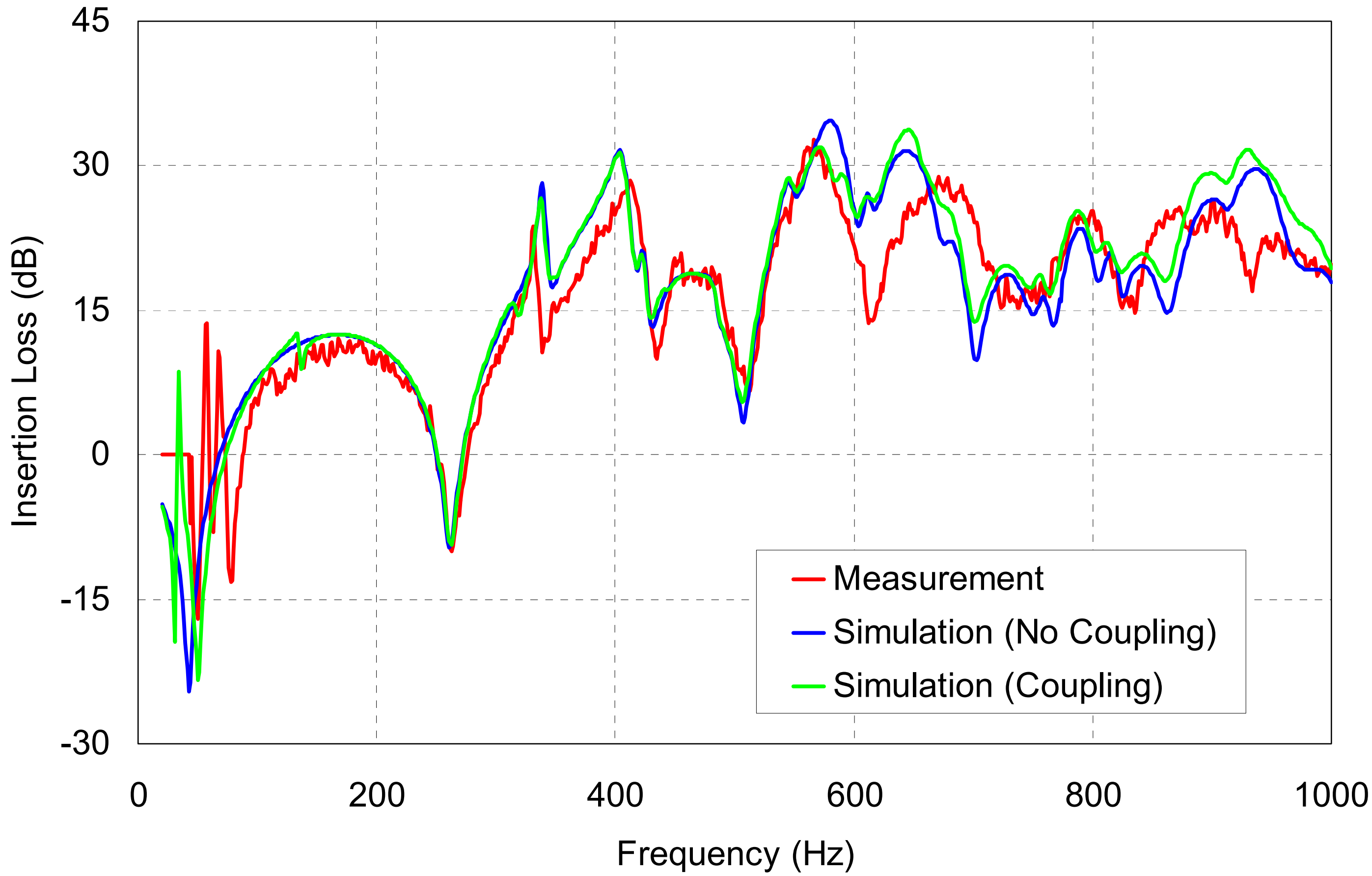
Modeling Approach Panel Absorption



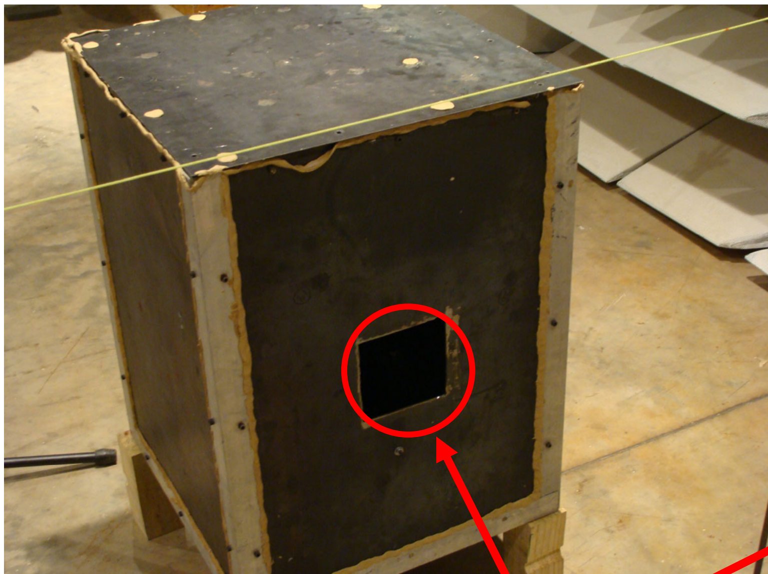
Modeling Approach Coupling

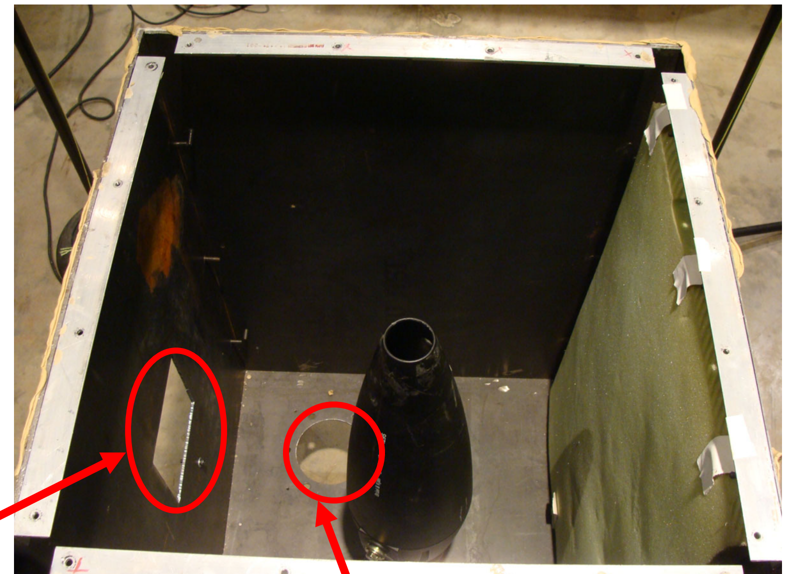


Modeling Approach Coupling



Validation Test Two Openings

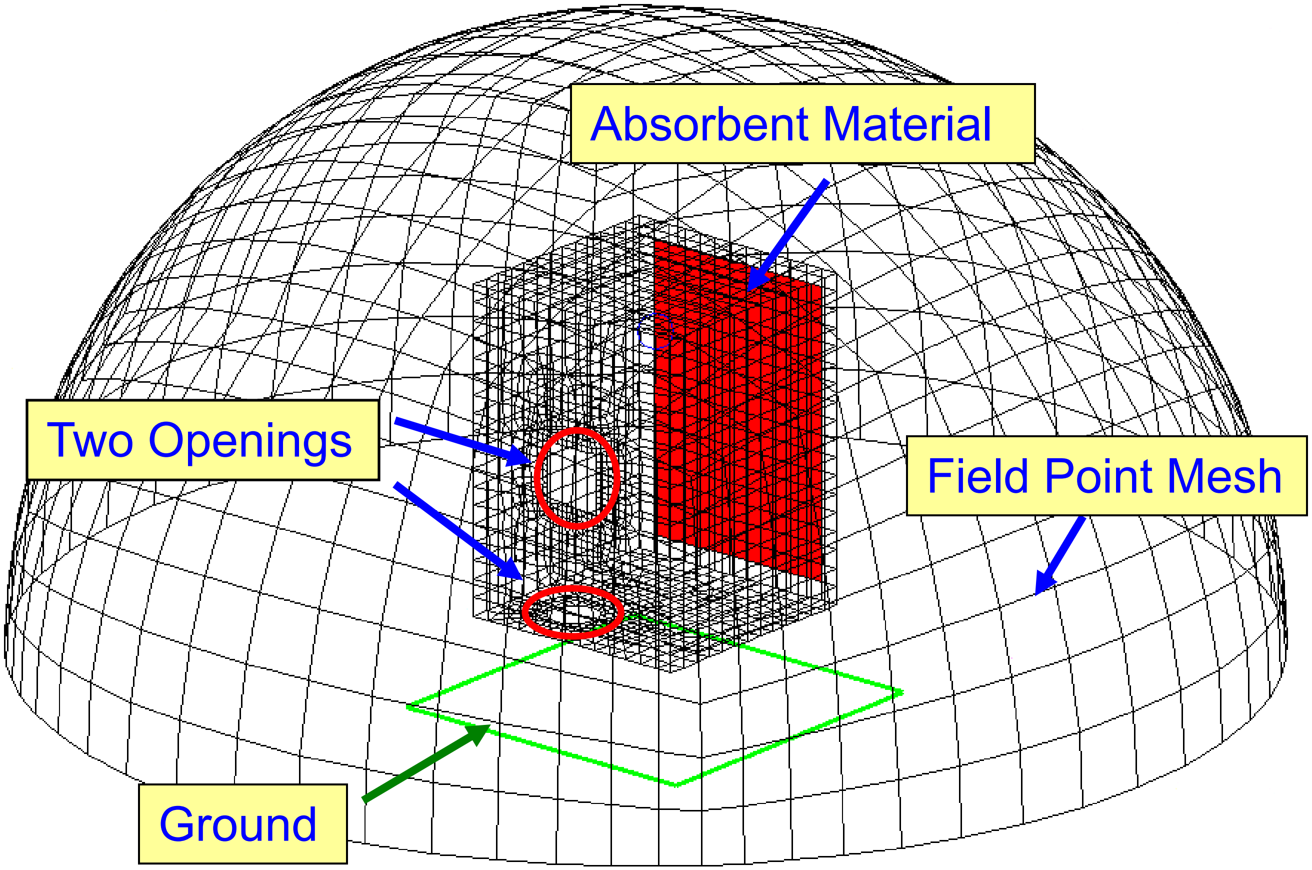




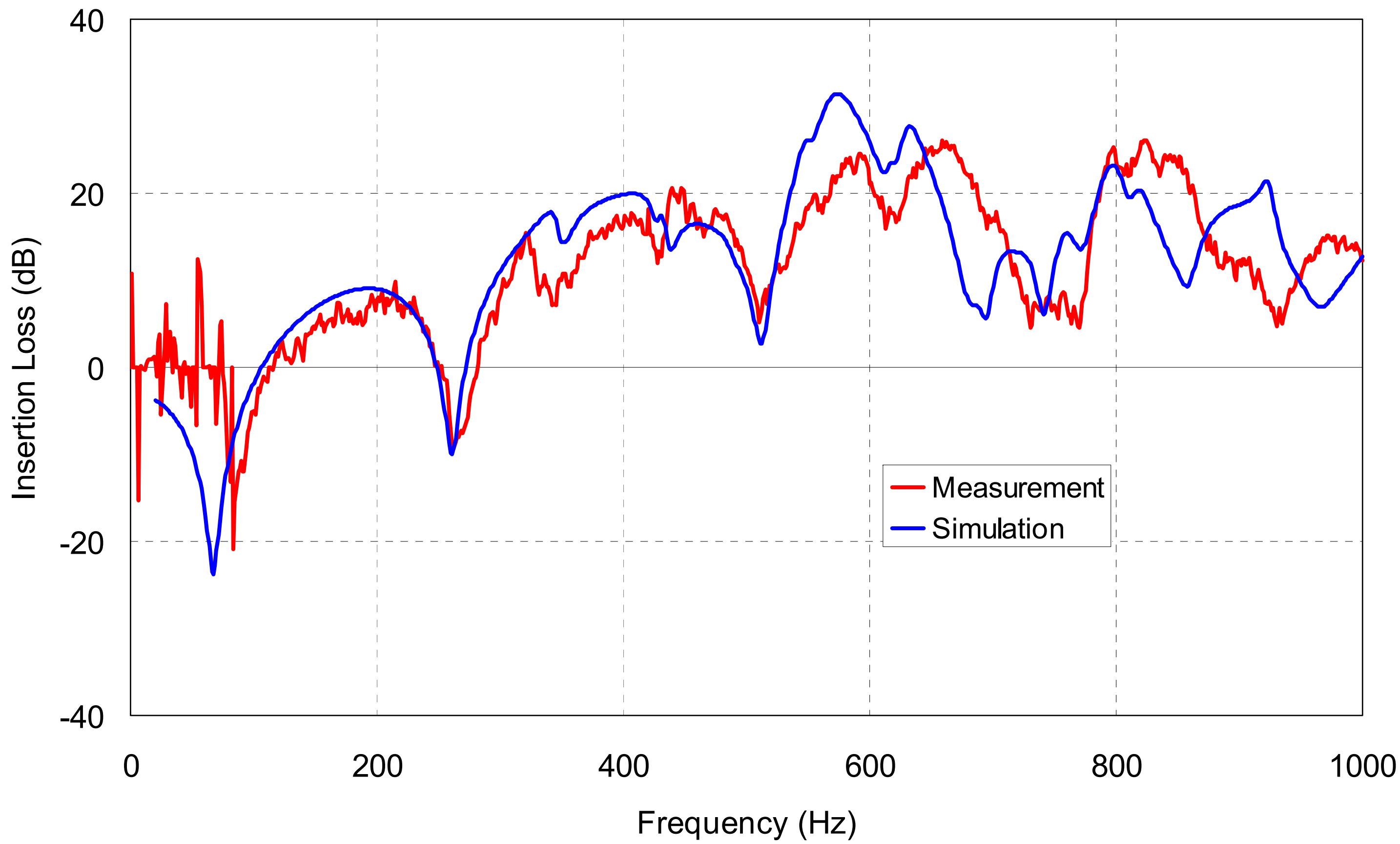
Additional Opening

Original Opening

Validation Test Two Openings



Validation Test Increased Open Area



References

1. D. A. Bies, C. H. Hansen, and C. Q. Howard (2018). *Engineering Noise Control.* CRC Press, Boca Raton, FL*.*
2. Ver, I. L., and Beranek, L. L. (2005). *Noise and Vibration Control Engineering: Principles and Applications*. John Wiley and Sons.
3. Sharp, B. H. (1973). *A study of techniques to increase the sound insulation of building elements*. U.S. Department of Commerce, National Technical Information Service (NTIS).
4. Zhou, L., Carter, A. E., Herrin, D. W., Shi, J., and Copley, D. C. (2011). "Airborne Path Attenuation of Partial Enclosures: Simulation and Sensitivity Study," Applied Acoustics, Vol. 72, pp. 380-386.