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Fibrous Material Microstructure Design for Optimal Structural Damping

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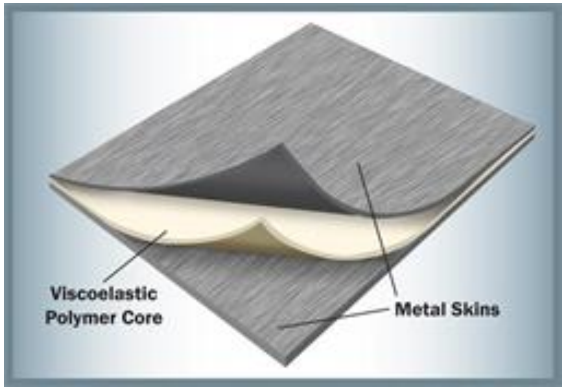
Presentation available at Herrick E-Pubs: http://docs.lib.purdue.edu/herrick/

INTRODUCTION

- **Traditional Damping Treatments – Visco-elastic Core with Metal Skins**

Traditional Damping Material[1]



Structure of a Traditional Damper[2]

- **Fibrous Damping Treatments – Target Material of this Study**

Fibrous Damping Material[3]

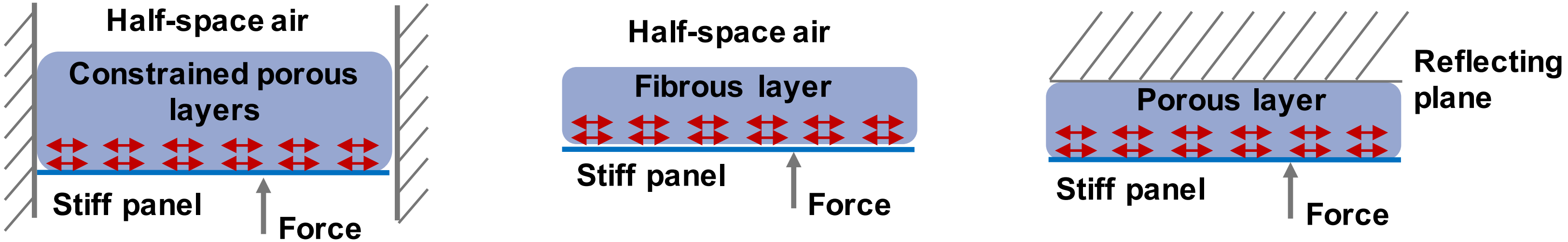
Test on Fibrous Dampers[4]

INTRODUCTION

- **Literature Review**

- **Bruer & Bolton, *AIAA* 1987**[5] – Analysis of different waves propagating in the layered damping structures
- **Wahl & Bolton, *JASA* 1992**[6] – Analysis by Inverse Discrete Fourier Transform (IDFT) on the spatial / temporal response of the layered damping system under line driving force
- **Lai & Bolton, *Noise-Con* 1998**[7] – Modeling to prove reasonable structural damping effect from the light fibrous materials through dissipating nearfield energy
- **Gerdes *et al.*, *Noise-Con* 1998**[8] – Numerical modeling of the structural damping effect from the light fibrous materials by evaluating the in-plane direction particle velocity
- **Nadeau *et al.*, *Journal of Aircraft* 1999**[9] – Tests of aircraft fuselage damping treatment by sound-absorbing blankets and related layered structures
- **Gerdes *et al.*, *Noise-Con* 2001**[4] – Numerical modeling of the structural damping effect from three different visco-elastic dampers compared with fibrous dampers
- **Kim *et al.*, *Noise-Con* 2015**[10] – Bulk property (thickness) design for fibrous materials' structural damping
- **Xue *et al.*, *Applied Acoustics* 2018**[11] – Fibrous material airflow resistivity prediction based on verified microstructure

- **Layered Structures Shown in the Literature**



- The panel damping mostly arises because of the viscous interaction of the fibrous medium and the evanescent near-field of the panel associated with subsonic panel motion

INTRODUCTION

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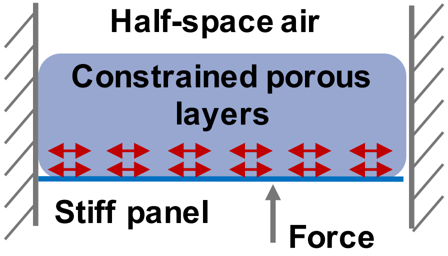
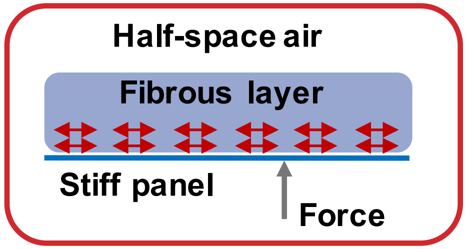
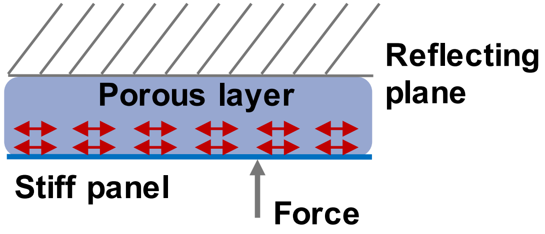

Half-space air
Constrained porous layers
Stiff panel
Force

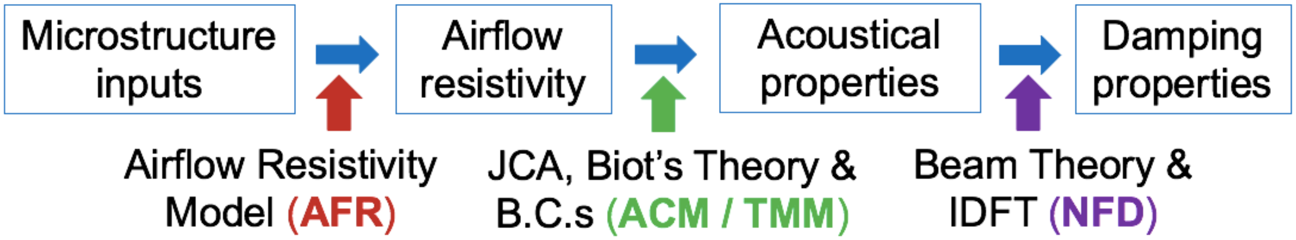

Half-space air
Fibrous layer
Stiff panel
Force


Reflecting plane
Porous layer
Stiff panel
Force

Target Structure of this Study

GENERAL APPROACH

- **Acoustical / Damping Performance Prediction Process**



Microstructure inputs → Airflow resistivity → Acoustical properties → Damping properties

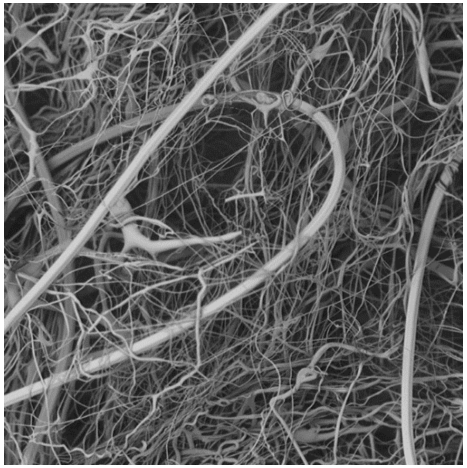
Airflow Resistivity Model **(AFR)**

JCA, Biot's Theory & B.C.s **(ACM / TMM)**

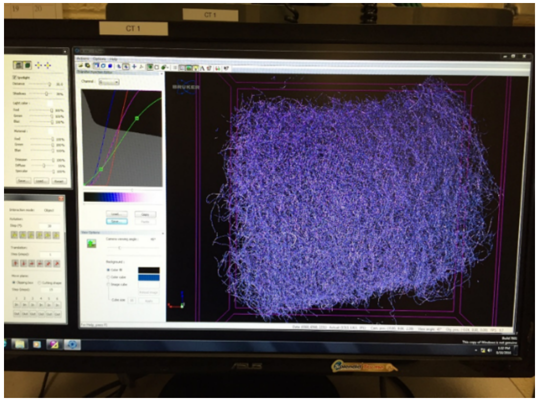
Beam Theory & IDFT **(NFD)**

AFR

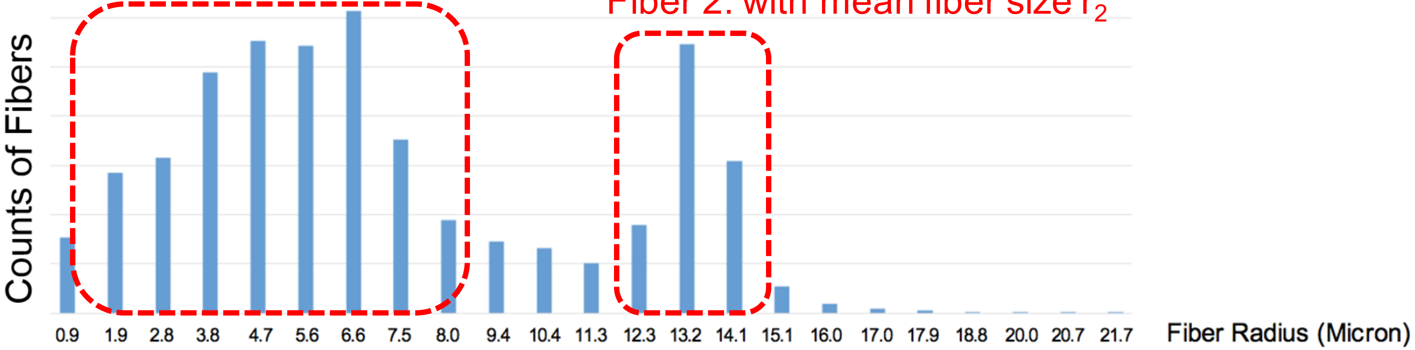
- **Fibrous Medium Airflow Resistivity Prediction**[11]



SEM of the target fibrous medium



Fibrous medium micro-CT scanning



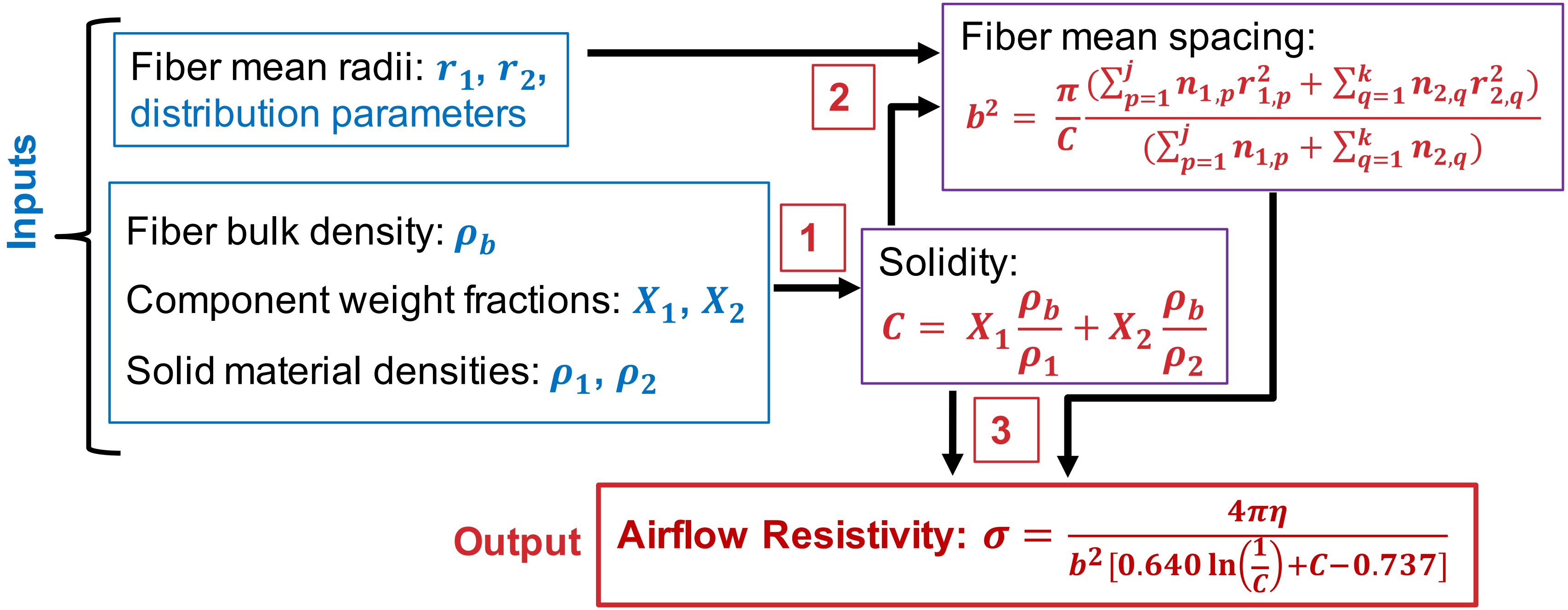
Fiber 1: main AFR contributor, with mean fiber size r_1

Fiber 2: with mean fiber size r_2

Micro-CT scanned fiber radii distribution of the fibrous medium

AFR

- **Fibrous Medium Airflow Resistivity Prediction**[11]



Inputs

Fiber mean radii: r_1, r_2, distribution parameters

Fiber bulk density: ρ_b

Component weight fractions: X_1, X_2

Solid material densities: ρ_1, ρ_2

1

Solidity:

$$C = X_1 \frac{\rho_b}{\rho_1} + X_2 \frac{\rho_b}{\rho_2}$$

2

Fiber mean spacing:

$$b^2 = \frac{\pi}{C} \frac{\left(\sum_{p=1}^{j} n_{1,p} r_{1,p}^2 + \sum_{q=1}^{k} n_{2,q} r_{2,q}^2\right)}{\left(\sum_{p=1}^{j} n_{1,p} + \sum_{q=1}^{k} n_{2,q}\right)}$$

3

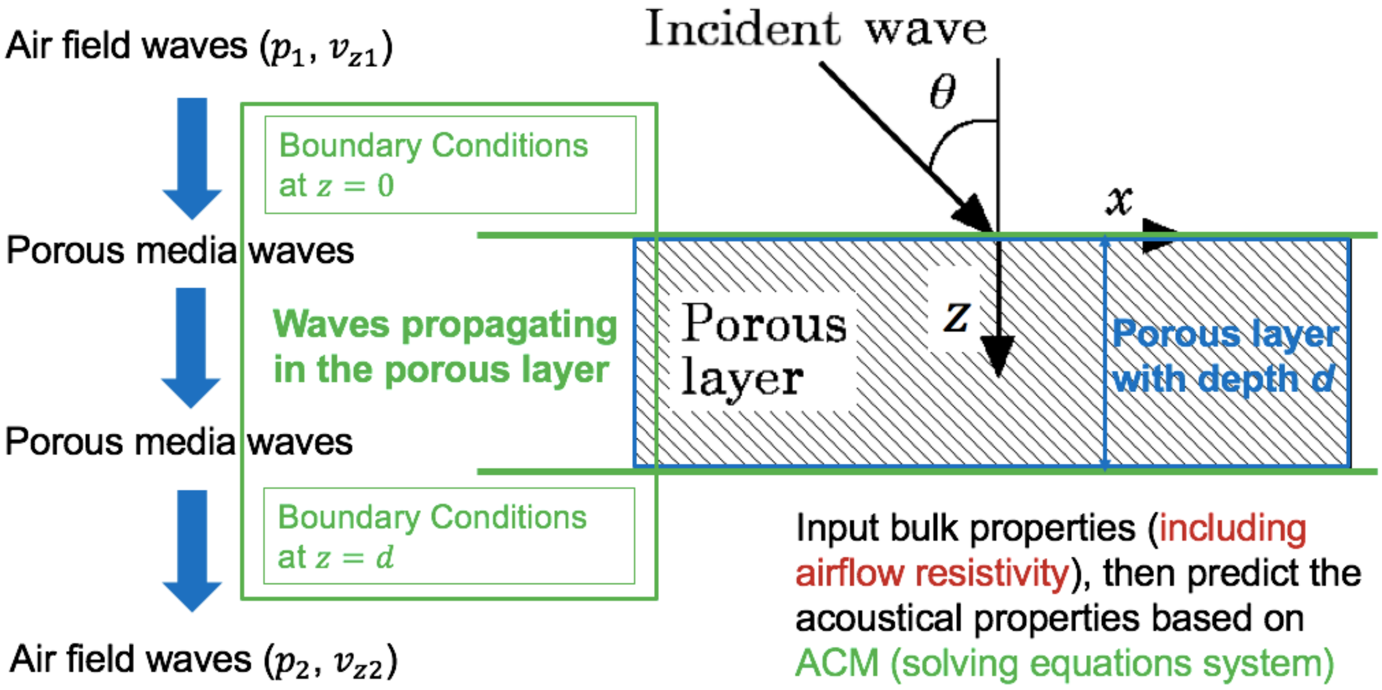
Output

Airflow Resistivity: $\sigma = \dfrac{4\pi\eta}{b^2\left[0.640 \ln\left(\frac{1}{C}\right) + C - 0.737\right]}$

- **Step 1**: C calculation based on ρ_b, X_1, X_2, ρ_1, ρ_2
- **Step 2**: b^2 calculation based on r_1, r_2, distribution parameters and C
- **Step 3**: σ calculation base on C and b^2

ACM / TMM

- **After having bulk moduli and wavenumbers of elastic fibers[13], [14]**



Input bulk properties (including airflow resistivity), then predict the acoustical properties based on ACM (solving equations system) or TMM (matrix operations)

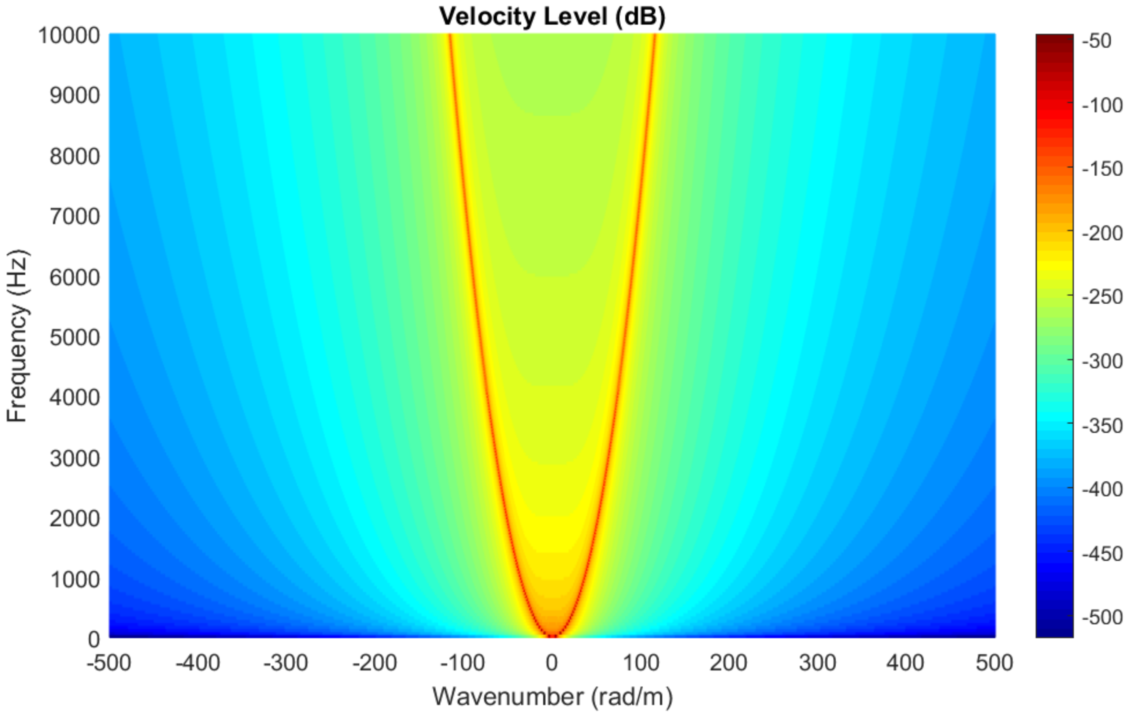
- **ACM: incorporate B.C.s into equations system and solve for acoustical properties**
- **TMM: reduce higher order matrices ([6x6] or [4x4]) to [2x2] by SVD + QR + B.C.s, then combine them with other [2x2] element matrices to solve for acoustical properties**

NFD

- **Choice of IDFT sampling rate γ_s and sampling points number N[12]**

➢ **Target of the NFD model**: calculate spatial responses for wide frequency range

➢ **Key point**: for each frequency input, choosing proper γ_s and N to ensure **accurate IDFT results over a large enough spatial span** for observation

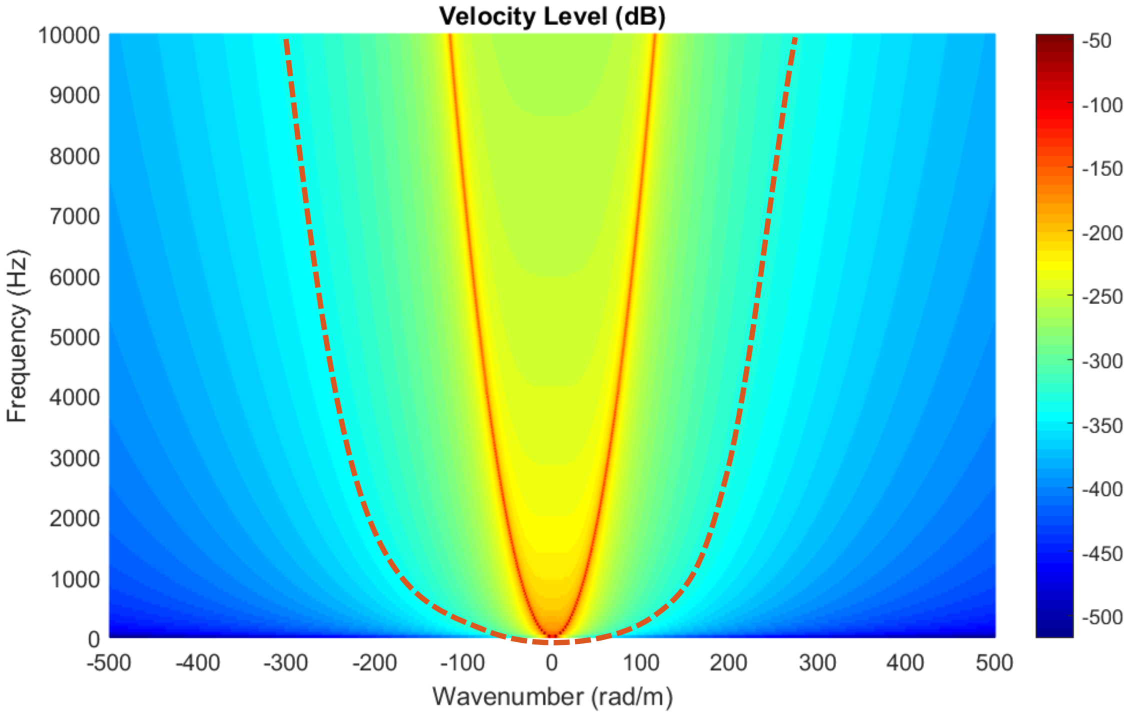


- Step 1: evaluate the wave number domain response of the panel

NFD

- **Choice of IDFT sampling rate γ_s and sampling points number N[12]**

- **Target of the NFD model**: calculate spatial responses for wide frequency range
- **Key point**: for each frequency input, choosing proper γ_s and N to ensure **accurate IDFT results over a large enough spatial span** for observation



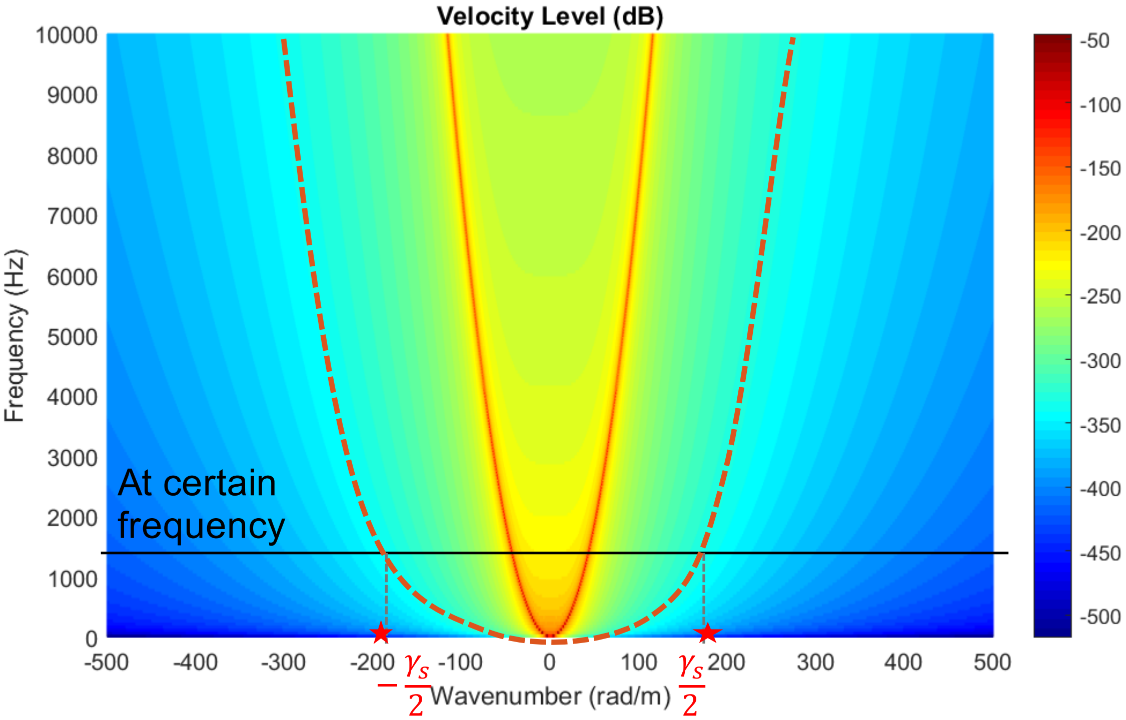
- Step 2: decide a proper cutoff level to avoid windowing/truncation effect

NFD

- **Choice of IDFT sampling rate γ_s and sampling points number N[12]**

➢ **Target of the NFD model**: calculate spatial responses for wide frequency range

➢ **Key point**: for each frequency input, choosing proper γ_s and N to ensure **accurate IDFT results over a large enough spatial span** for observation



- N should be large enough to avoid bias

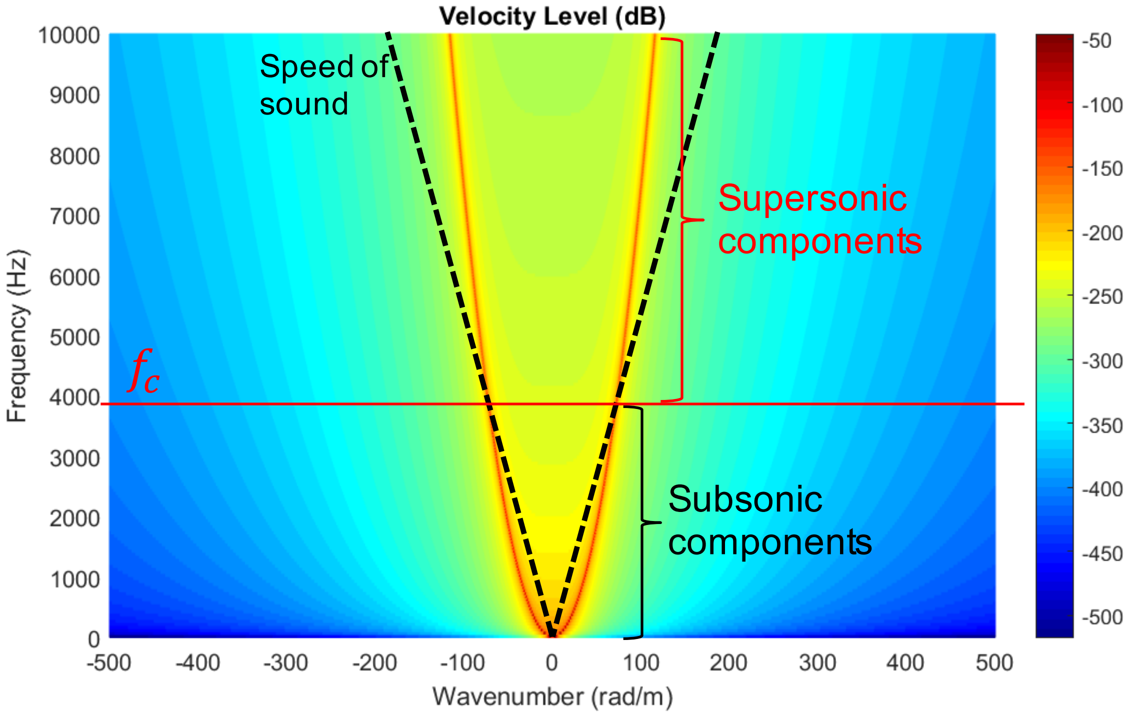
- Step 3: find the proper sampling rate γ_s for each input frequency

NFD

- **Choice of IDFT sampling rate γ_s and sampling points number N[12]**

➢ **Target of the NFD model**: calculate spatial responses for wide frequency range

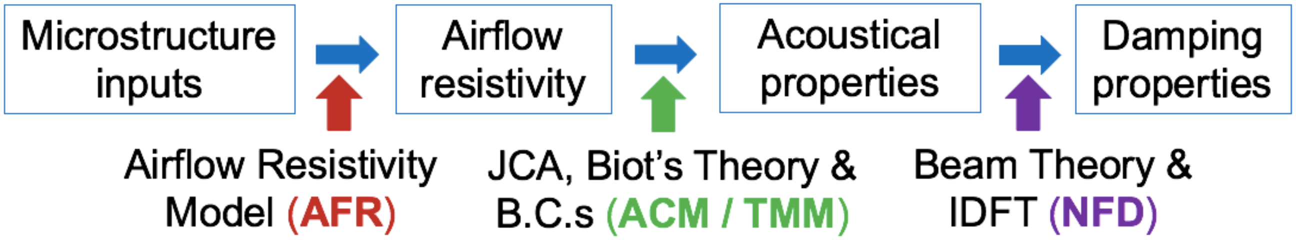
➢ **Key point**: for each frequency input, choosing proper γ_s and N to ensure **accurate IDFT results over a large enough spatial span** for observation



- Step 4: identify the critical frequency f_c

GENERAL APPROACH

- **Acoustical / Damping Performance Prediction Process**



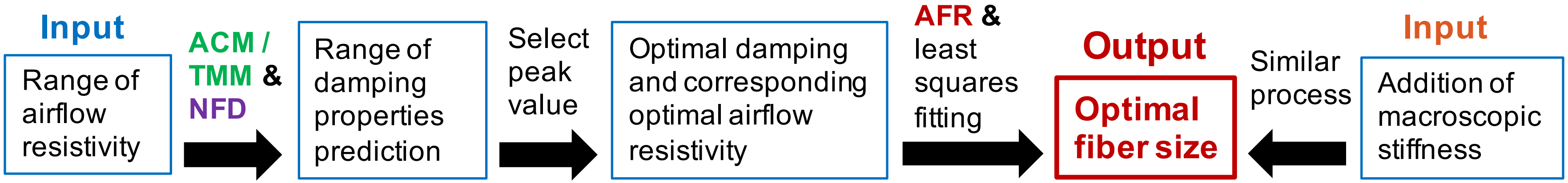
Microstructure inputs → Airflow resistivity → Acoustical properties → Damping properties

Airflow Resistivity Model (**AFR**)

JCA, Biot's Theory & B.C.s (**ACM / TMM**)

Beam Theory & IDFT (**NFD**)

- **Materials Microstructure Design Process**



Input

Range of airflow resistivity

ACM / TMM & **NFD**

Range of damping properties prediction

Select peak value

Optimal damping and corresponding optimal airflow resistivity

AFR & least squares fitting

Output

Optimal fiber size

Similar process

Input

Addition of macroscopic stiffness

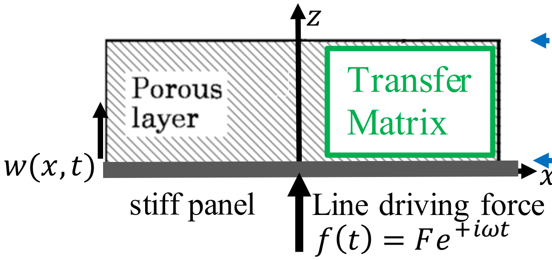
- **Objectives of this Study**
  - **Identify the airflow resistivity providing optimal damping performance given panel structure and frequency range of interest**
  - **Translate the optimal airflow resistivity into optimal fiber sizes for fibrous material microstructure design**
  - **Demonstrate effect of macroscopic stiffness**

MODELING

- **Modeling Process**[12]

Porous medium: thickness ($\boldsymbol{d}$), AFR ($\boldsymbol{\sigma}$), bulk density ($\boldsymbol{\rho_b}$), porosity ($\boldsymbol{\phi}$), tortuosity ($\boldsymbol{\alpha_\infty}$), Young's modulus ($\boldsymbol{E}$), Poisson's ratio ($\boldsymbol{\nu}$), loss factor ($\boldsymbol{\eta}$)

Panel: basis weight ($\boldsymbol{m_s}$), flexural stiffness per unit width ($\boldsymbol{D}$), longitudinal stiffness per unit width ($\boldsymbol{D_p}$)



Far-field pressure, $\boldsymbol{p_2}$, and normal velocity, $\boldsymbol{v_{z2}}$, at $\boldsymbol{z = d}$

Near-field pressure, $\boldsymbol{p_1}$, and normal velocity, $\boldsymbol{v_{z1}}$, at $\boldsymbol{z = 0}$

Euler-Bernoulli beam theory (neglect rotary inertia & shear deformation)

Governing Equation (GE)

$$D\frac{\partial^4 w(x,t)}{\partial x^4} + m_s\frac{\partial^2 w(x,t)}{\partial t^2} = -p_1(x,t) + f(t)\delta(x)$$

Governing Equation Fourier Transform (GEFT)

$$(Dk_x^4 - \omega^2 m_s)W(k_x,\omega) = -P_1(k_x,t) + F$$

Input $\boldsymbol{N}$, $\boldsymbol{\gamma_s}$ for IDFT

Panel normal velocity response

$$v_{z1}(k_x,\omega) = F/[Z_{a1}(k_x,\omega) + Z_m(k_x,\omega)]$$

Inverse Fourier Transform (IFT) for spatial response

$$v_{z1}(x,\omega) = \frac{1}{2\pi}\int_{-\infty}^{\infty} v_{z1}(k_x,\omega)e^{-ik_x x}\,dk_x$$

Use IDFT to approximate the numerical IFT results

$$v_{z1}(k\Delta x,\omega) = \frac{1}{N\Delta x}\sum_{l=0}^{N-1} v_{z1}(l\Delta k_x,\omega)e^{\frac{-i2\pi kl}{N}}$$

Integration

Power radiation into the porous layer

$$P_1 = \frac{1}{2}\mathrm{Re}\left\{\int_{-\infty}^{\infty} p_1 v_{z1}^*\,dx\right\} = \frac{1}{4\pi}\mathrm{Re}\left\{\int_{-\frac{\gamma_s}{2}}^{\frac{\gamma_s}{2}} Z_{a1}|v_{z1}(k_x,\omega)|^2\,dk_x\right\}$$

Power radiation into the air

$$P_2 = \frac{1}{2}\mathrm{Re}\left\{\int_{-\infty}^{\infty} p_2 v_{z2}^*\,dx\right\} = \frac{1}{4\pi}\mathrm{Re}\left\{\int_{-\frac{\gamma_s}{2}}^{\frac{\gamma_s}{2}} Z_{a2}|v_{z2}(k_x,\omega)|^2\,dk_x\right\}$$

Power dissipation in the porous layer

$$\bar{P}_d = \bar{P}_1 - \bar{P}_2$$

Find maximum $\bar{P}_d$ (optimal damping) and corresponding optimal σ for frequency of interest

AFR Model combined with least square optimization returns optimal porous material microstructure details (e.g. fiber sizes)

Input porous medium bulk properties ($\boldsymbol{\sigma}$, $\boldsymbol{\rho_b}$, $\boldsymbol{\phi}$, $\boldsymbol{\alpha_\infty}$, $\boldsymbol{E}$, $\boldsymbol{\nu}$, $\boldsymbol{\eta}$)

JCA & Biot's Theory

Complex wavenumbers ($\boldsymbol{k_i}$, $\boldsymbol{i = 1}$ for limp or rigid frame, $\boldsymbol{i = 3}$ for elastic frame)

Input $\boldsymbol{d}$, $\boldsymbol{k_x}$, $\boldsymbol{D}$, $\boldsymbol{D_p}$

$$k_{zi} = \sqrt{k_i^2 - k_x^2}$$

Equation system based on propagating waves and B.C.s

Transfer Matrix Method / Arbitrary Coefficient Method

Near-field-far-field relation

$$\begin{bmatrix} p_1 \\ v_{z1} \end{bmatrix}_{z=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix}\begin{bmatrix} p_2 \\ v_{z2} \end{bmatrix}_{z=d}$$

Combined with GEFT

Panel mechanical impedance: $Z_m = i[(D/\omega)k_x^4 - \omega m_s]$

Near-field acoustic impedance: $Z_{a1} = \frac{T_{11}Z_{a2}+T_{12}}{T_{21}Z_{a2}+T_{22}}$

Far-field acoustic impedance: $Z_{a2} = (\omega\rho_{air})/k_{zair}$

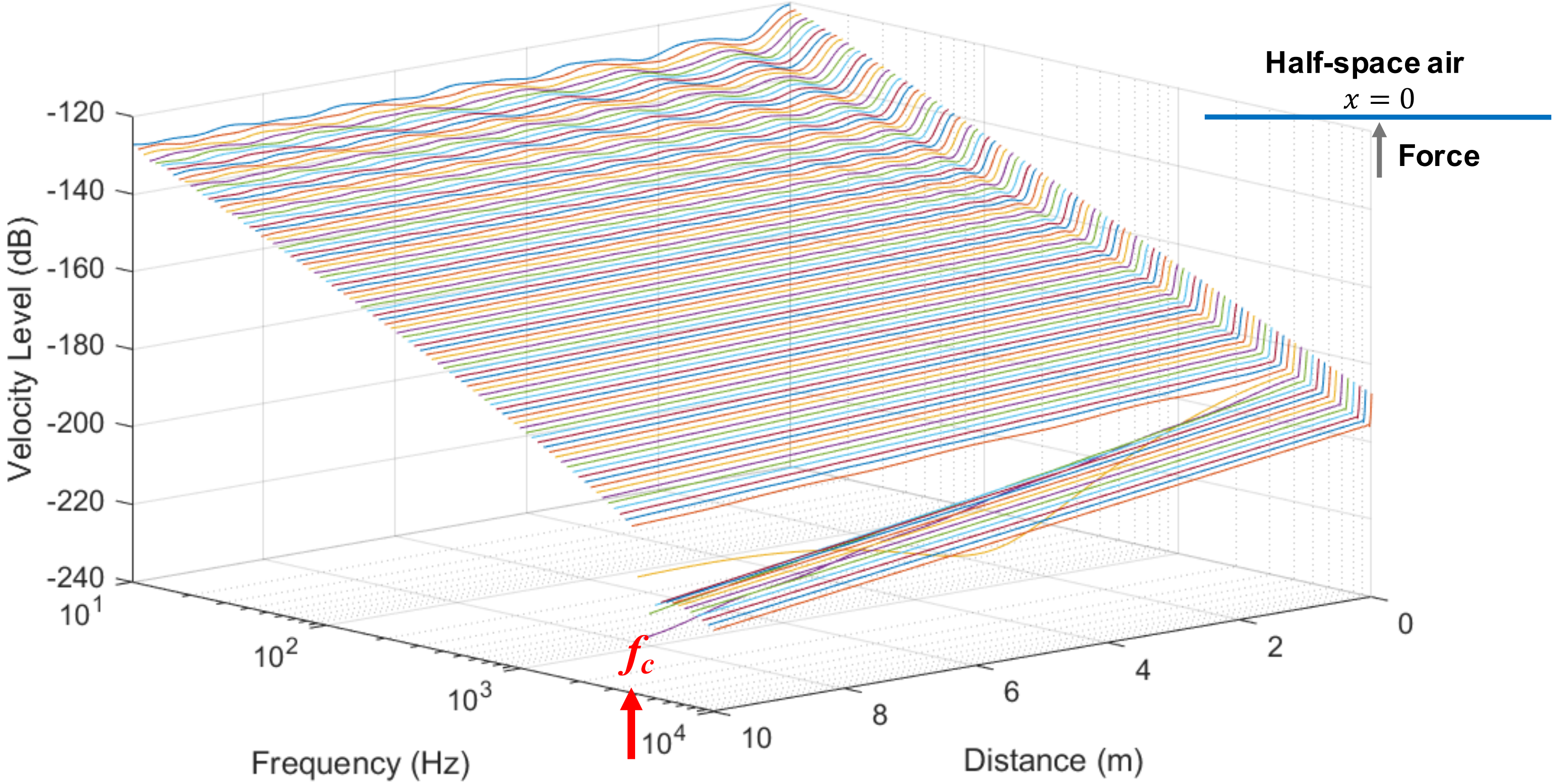
Pressure-velocity relation: $p_i = Z_{ai}v_{zi}$ (i = 1,2)

RESULTS – BARE PANEL

- **Spatial Velocity Level (dB)**

Total points N = 16384. Wave# sampling rate γ_s = 66-383rad/m. Frequency range = 10-10000Hz

Panel Thickness = 3 mm. Panel Loss Factor = 0.003. Air Loss Factor = 0.0005

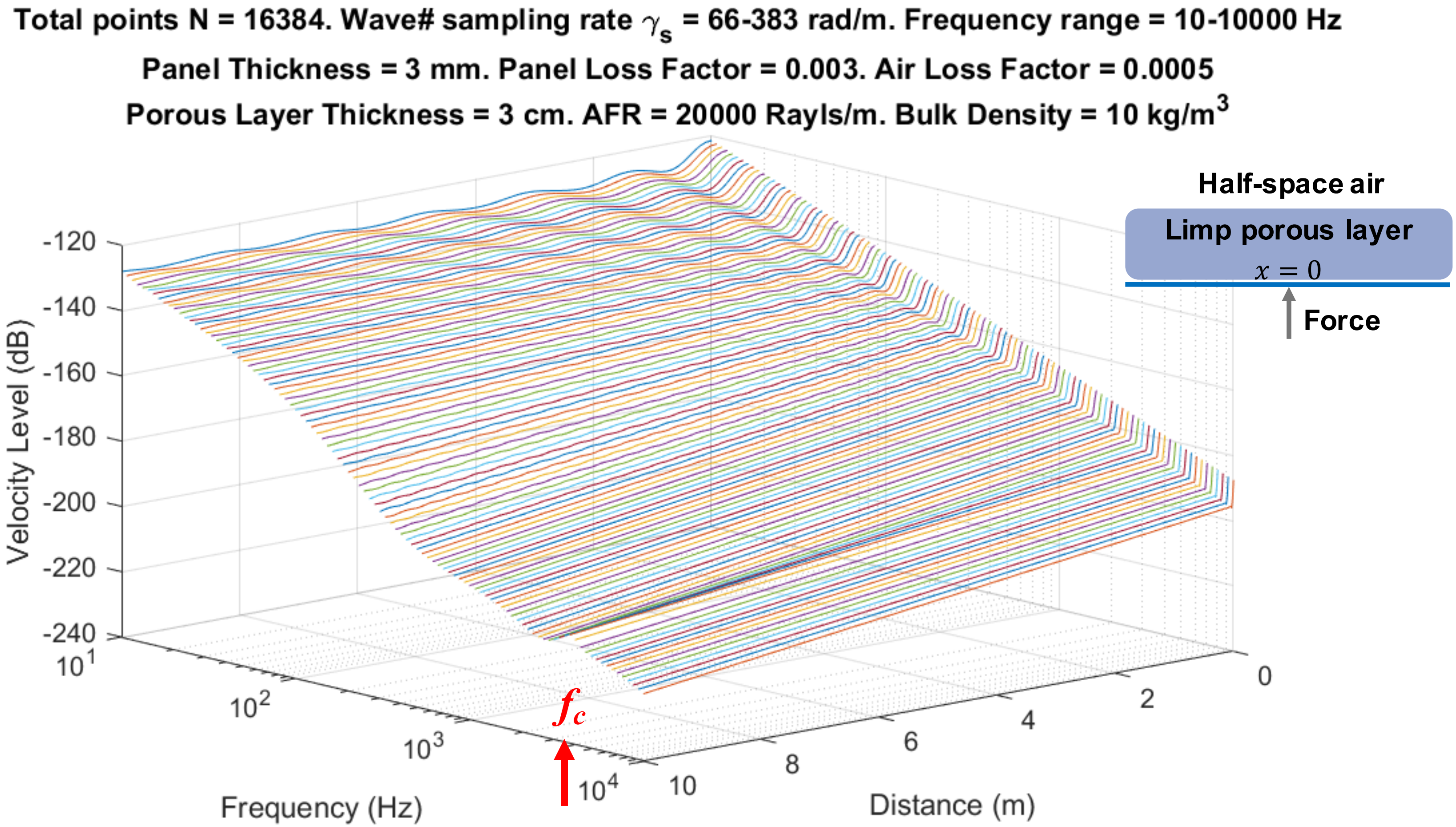


RESULTS – LIMP LAYER

PURDUE MECHANICAL ENGINEERING

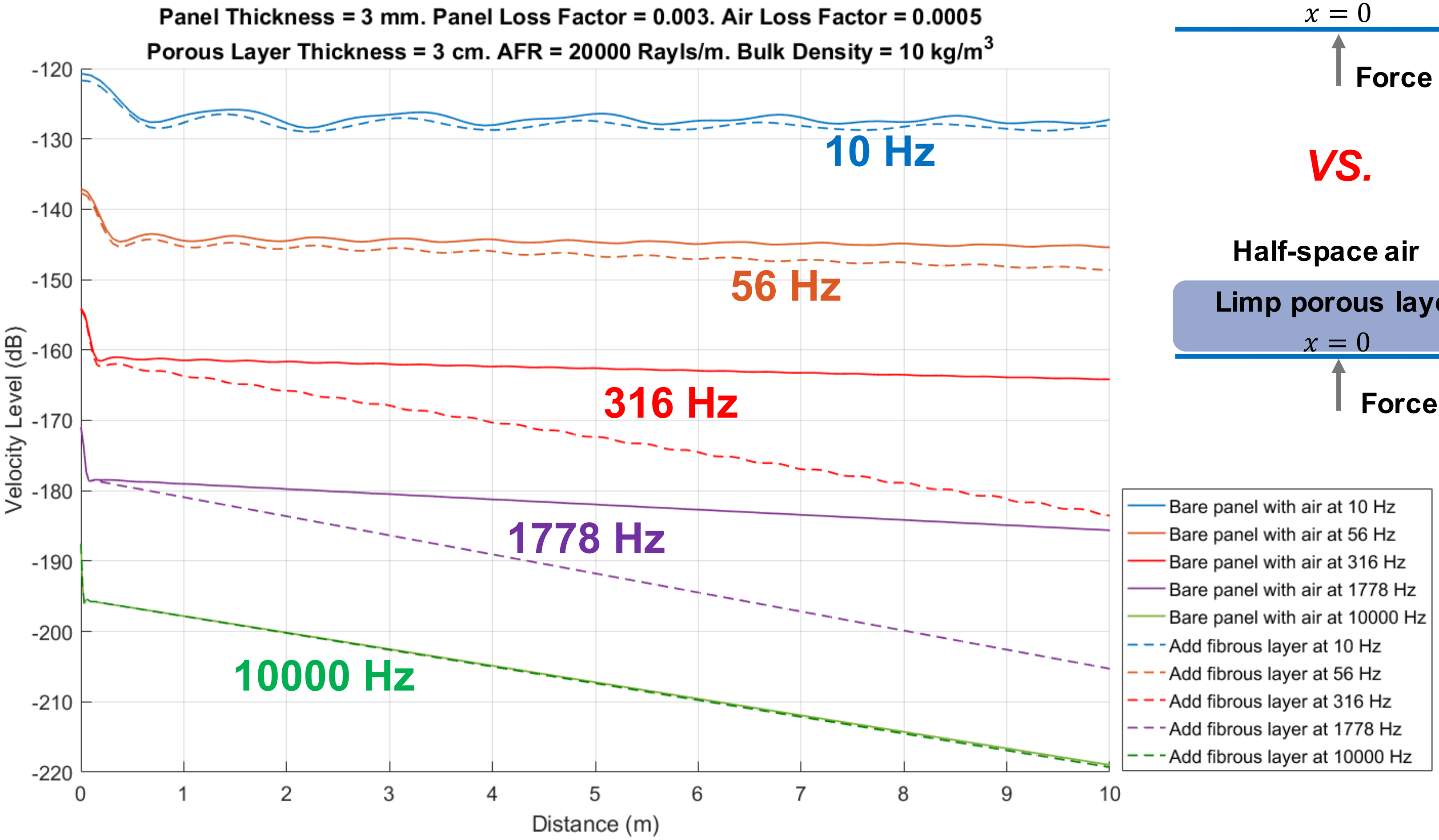
RAY W. HERRICK LABORATORIES

- **Spatial Velocity Level (dB)**



RESULTS – COMPARISON

- **Spatial Velocity Level (dB)**



Panel Thickness = 3 mm. Panel Loss Factor = 0.003. Air Loss Factor = 0.0005

Porous Layer Thickness = 3 cm. AFR = 20000 Rayls/m. Bulk Density = 10 kg/m^3

Half-space air

$x = 0$

Force

VS.

Half-space air

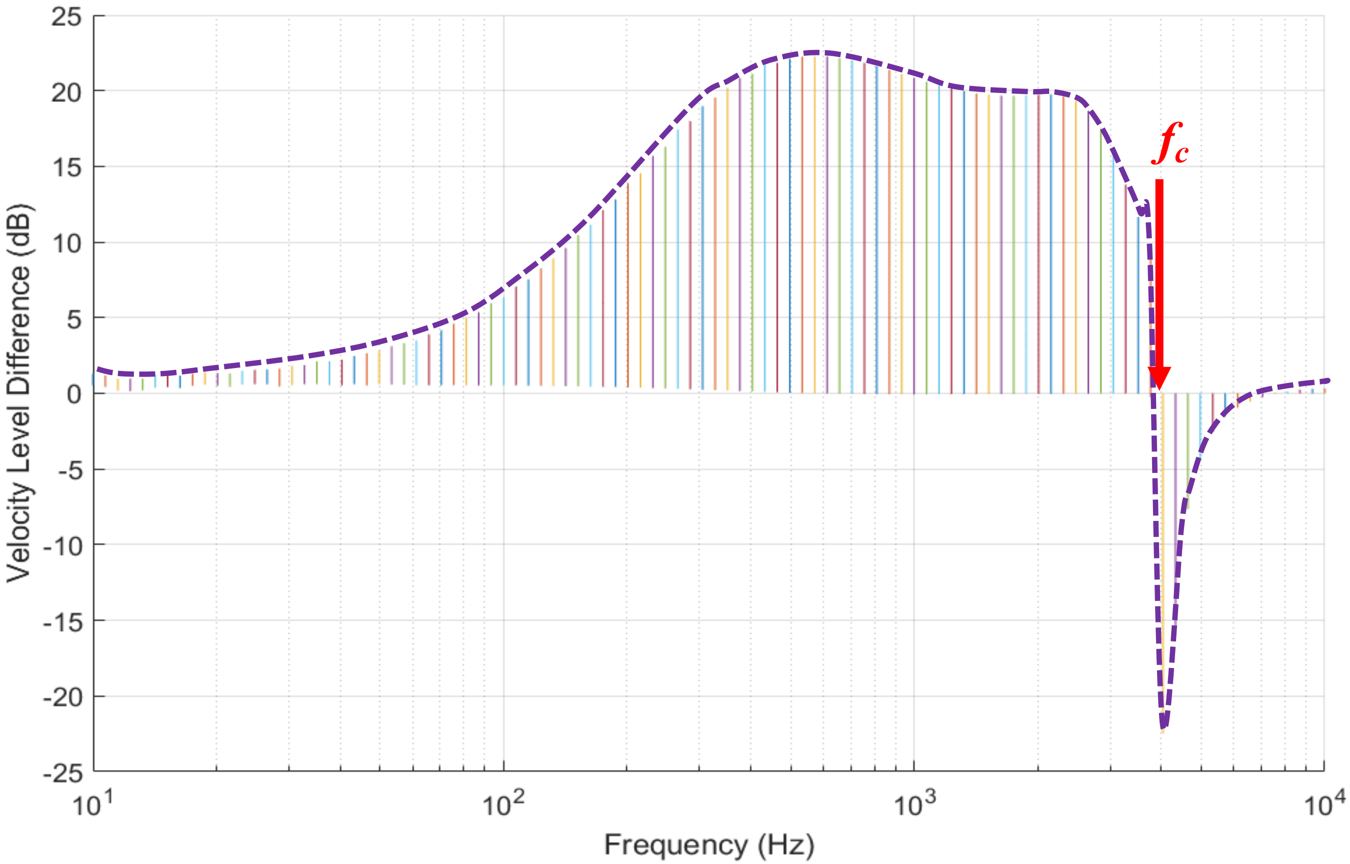
Limp porous layer

$x = 0$

Force

RESULTS – COMPARISON

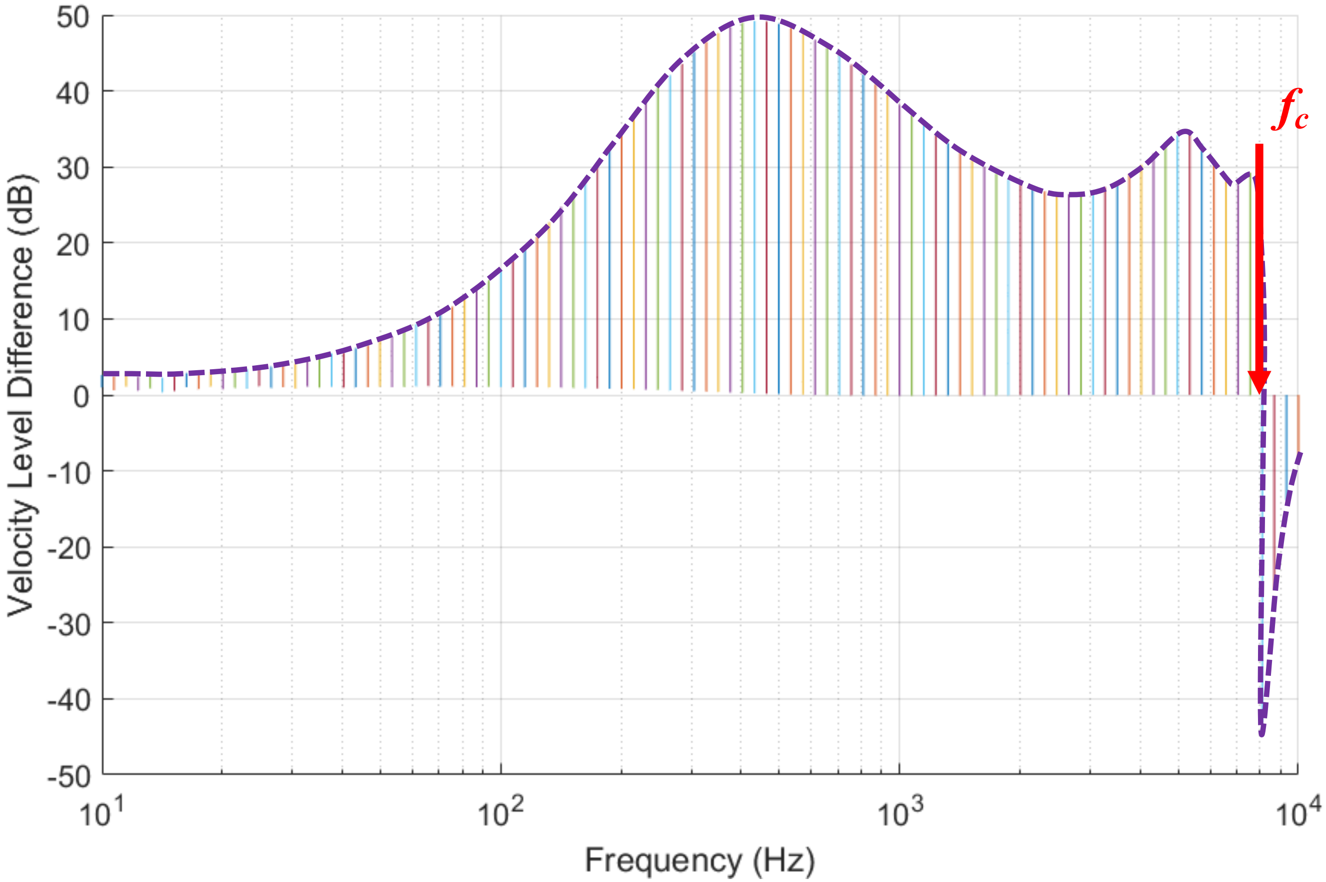
- **Spatial Velocity Level (dB)**
  - **Difference between two cases for a 3 mm thick aluminum**
  - **Significant attenuation in subsonic region below critical frequency**



ASA May 2018, Minneapolis, MN

RESULTS – COMPARISON

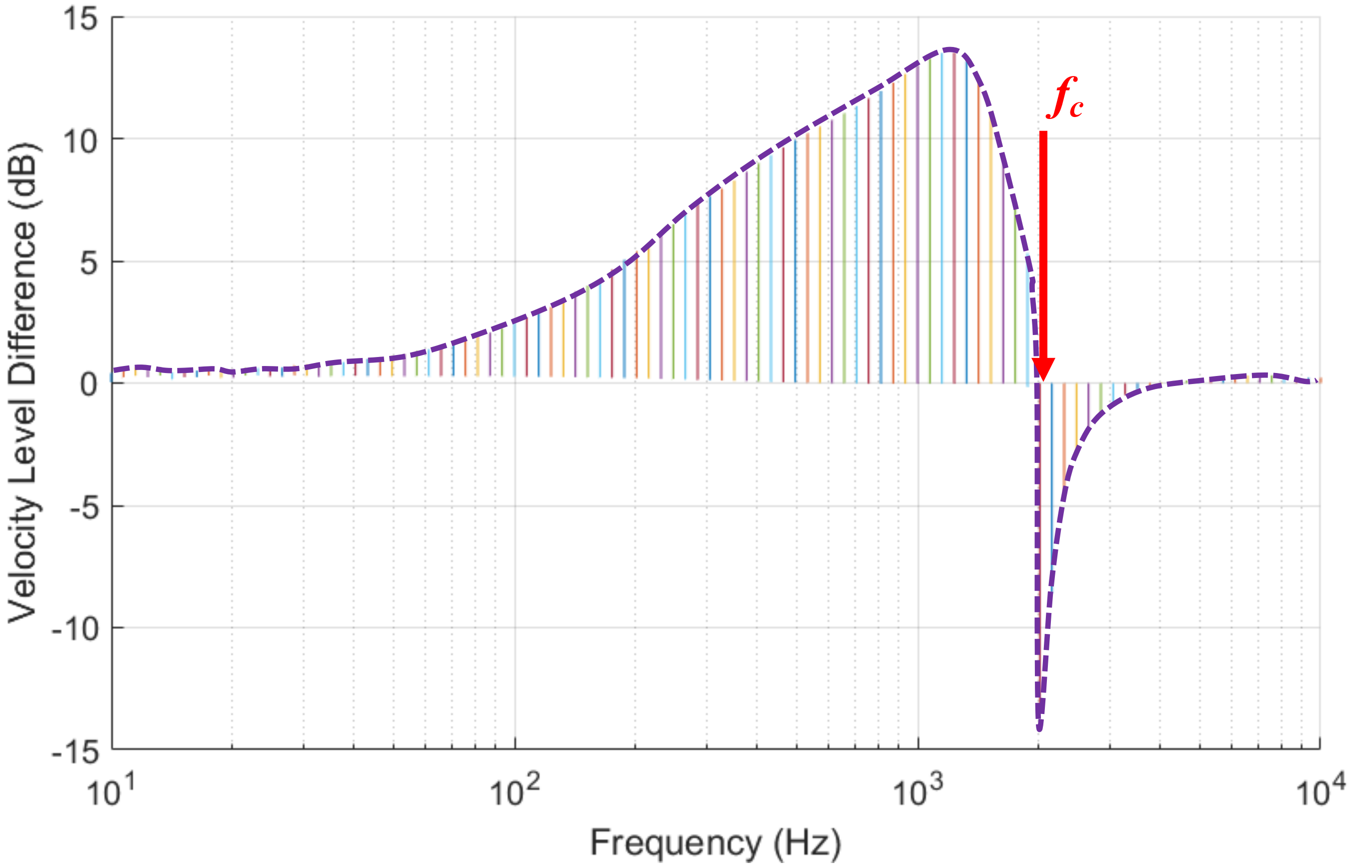
- **Spatial Velocity Level (dB)**
  - **Difference between two cases for a 1.5 mm thick aluminum**
  - **Compare to 3 mm panel: higher critical frequency and stronger attenuation**



ASA May 2018, Minneapolis, MN

RESULTS – COMPARISON

- **Spatial Velocity Level (dB)**
  - **Difference between two cases for a 6 mm thick aluminum**
  - **Compare to 3 mm panel: lower critical frequency and smaller attenuation**



RESULTS – BARE PANEL

PURDUE MECHANICAL ENGINEERING

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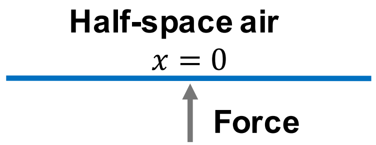
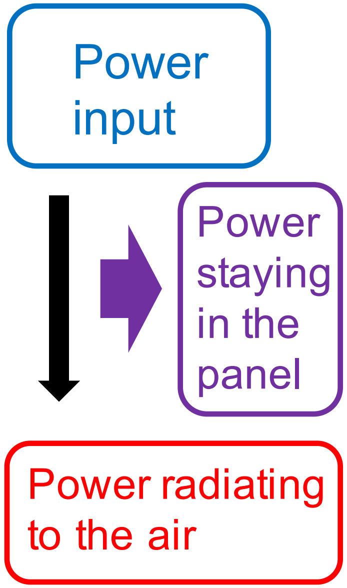
- **Power Distribution**



Half-space air

$x = 0$

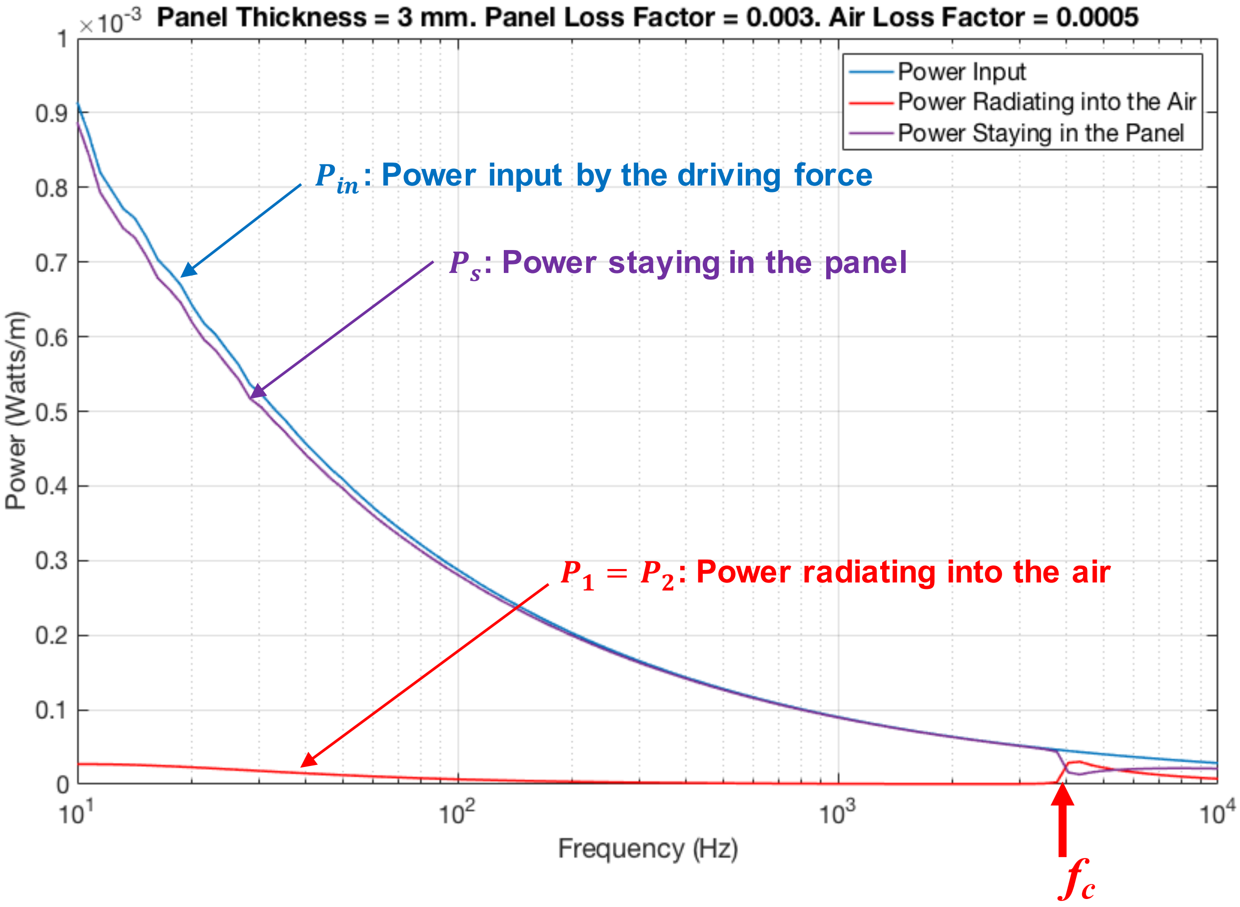
Force



Power input

Power staying in the panel

Power radiating to the air



P_{in}: Power input by the driving force

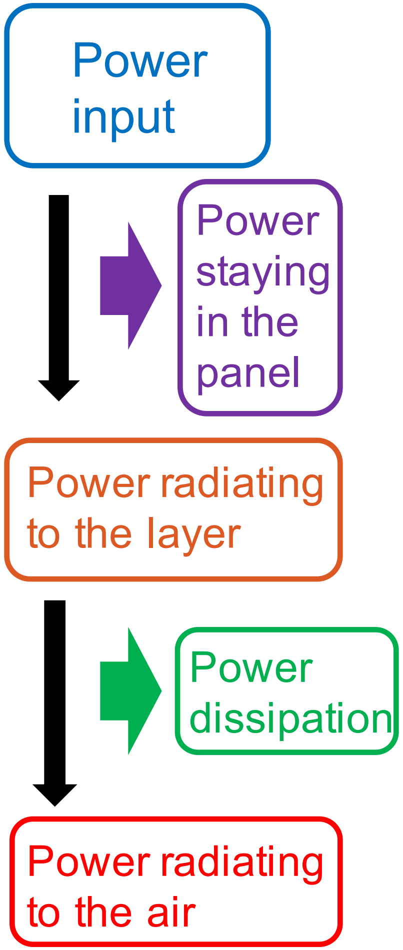
P_s: Power staying in the panel

$P_1 = P_2$: Power radiating into the air

f_c

RESULTS – LIMP LAYER

- **Power Distribution**



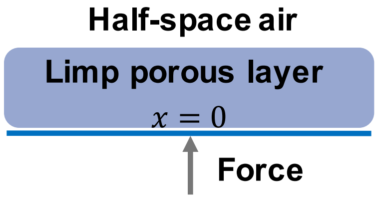
Power input

Power staying in the panel

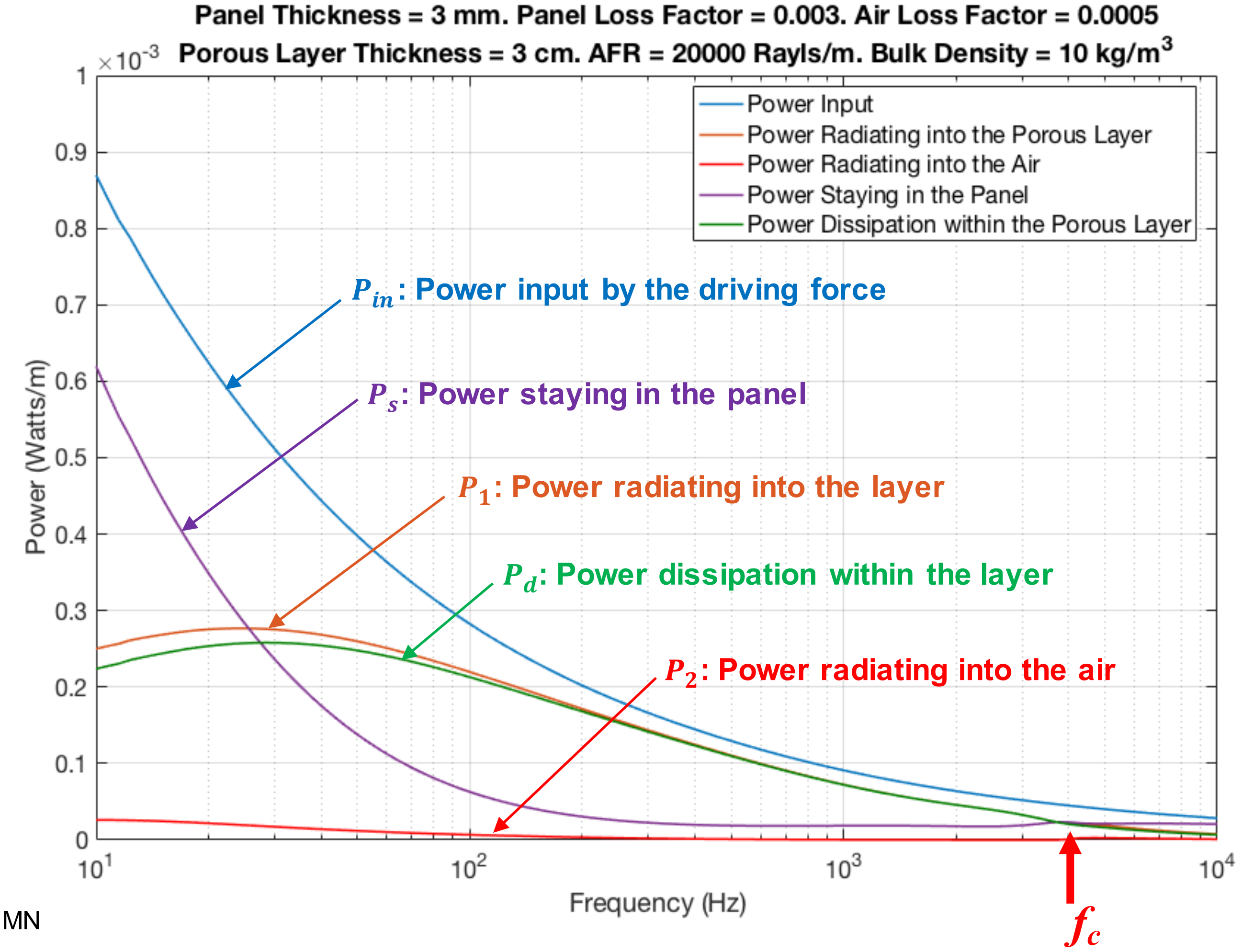
Power radiating to the layer

Power dissipation

Power radiating to the air



➢ **Subsonic region attenuation due to power dissipation within the layer**



Panel Thickness = 3 mm. Panel Loss Factor = 0.003. Air Loss Factor = 0.0005

Porous Layer Thickness = 3 cm. AFR = 20000 Rayls/m. Bulk Density = 10 kg/m^3

P_{in}: Power input by the driving force

P_s: Power staying in the panel

P_1: Power radiating into the layer

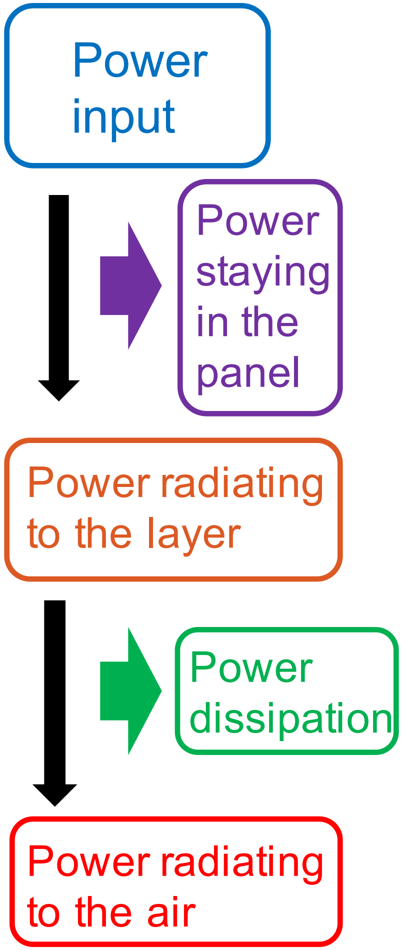
P_d: Power dissipation within the layer

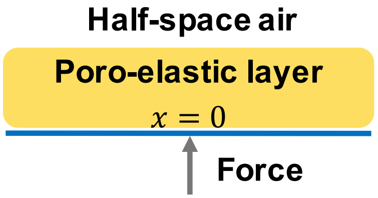
P_2: Power radiating into the air

f_c

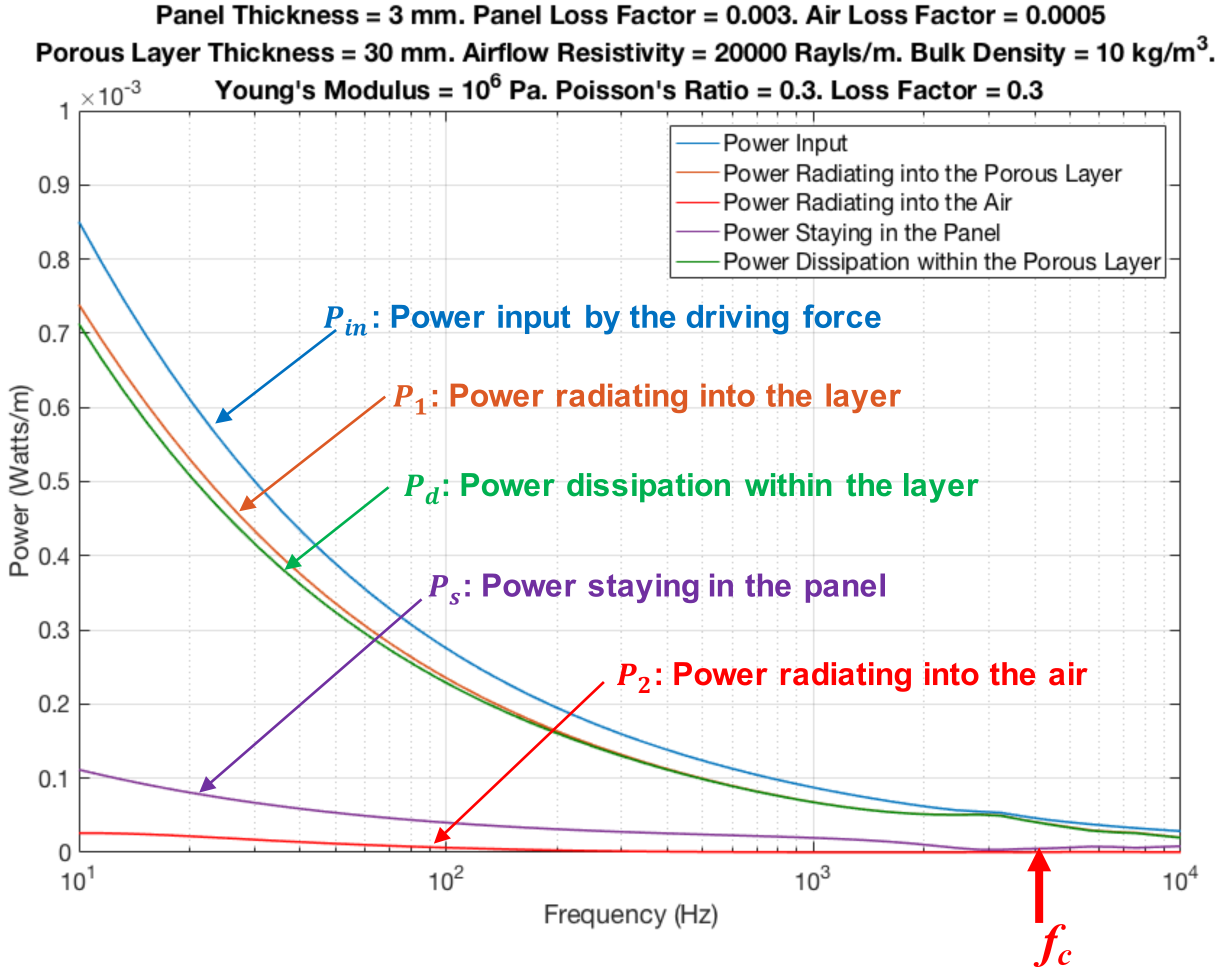
RESULTS – ELASTIC LAYER

- **Power Distribution**



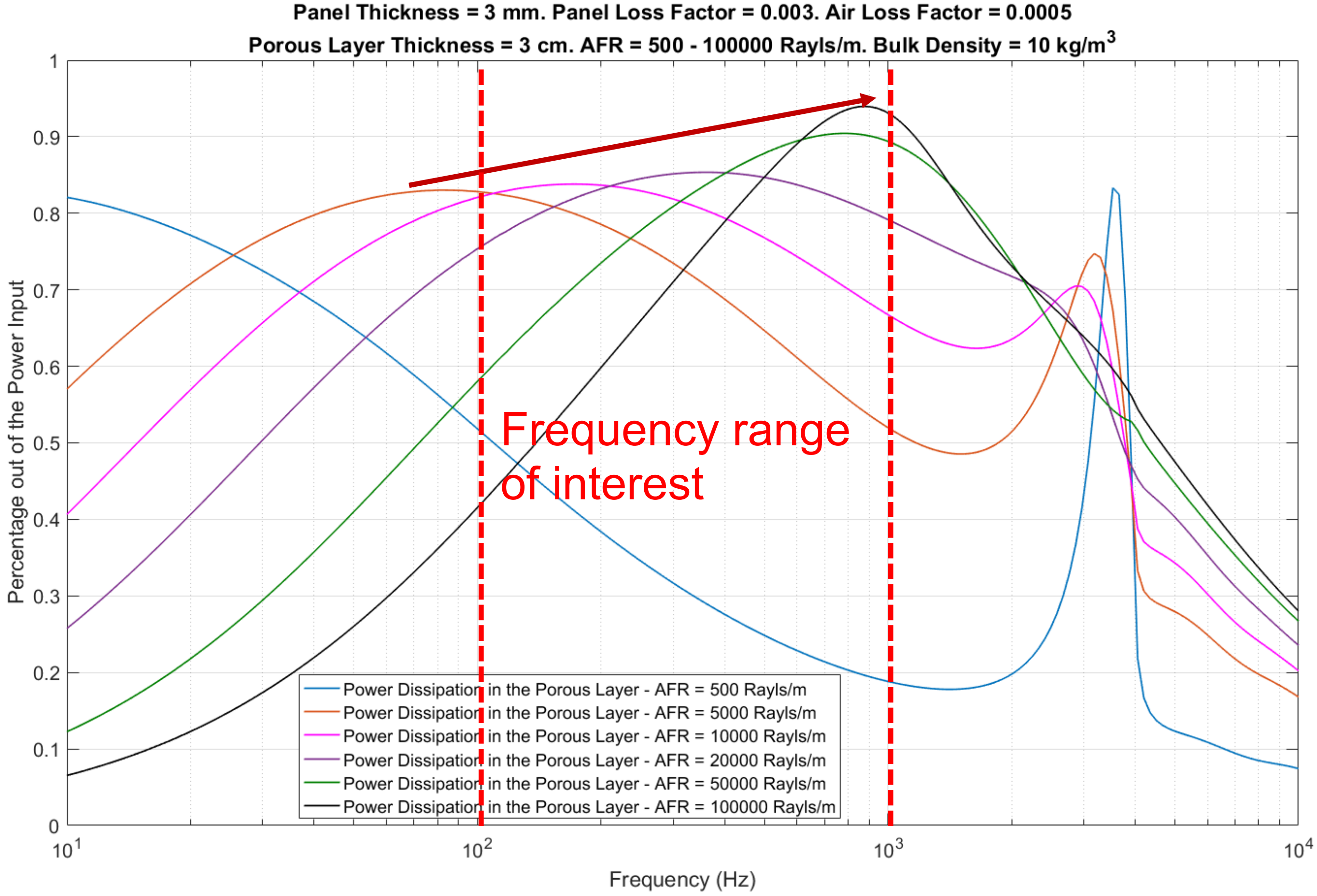


➢ **Stronger attenuation achieved by adding macroscopic stiffness to the layer**



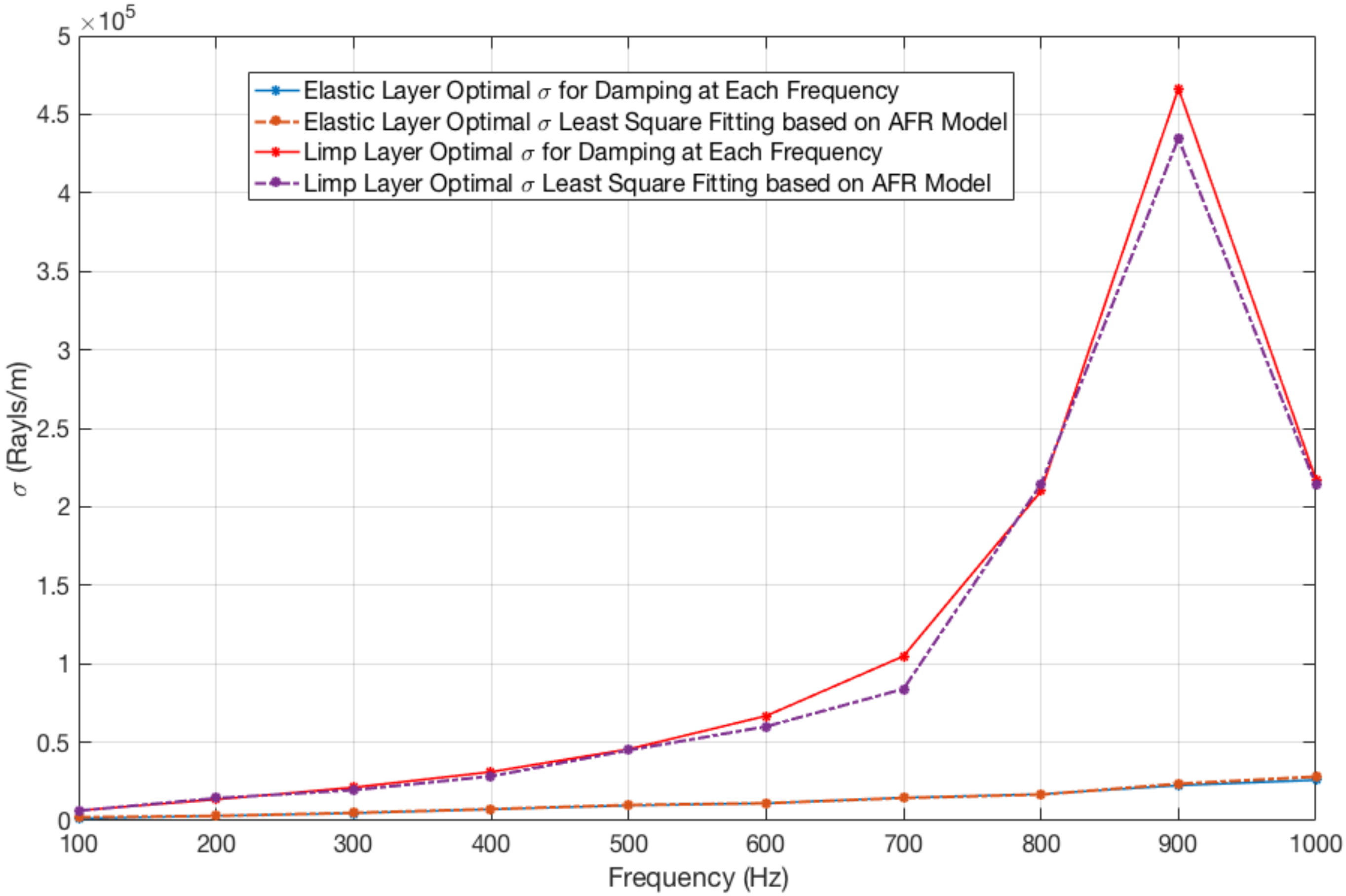
RESULTS

- **Airflow Resistivity Effect on Power Dissipation**
  - **Optimal damping and corresponding optimal AFRs at different frequencies**



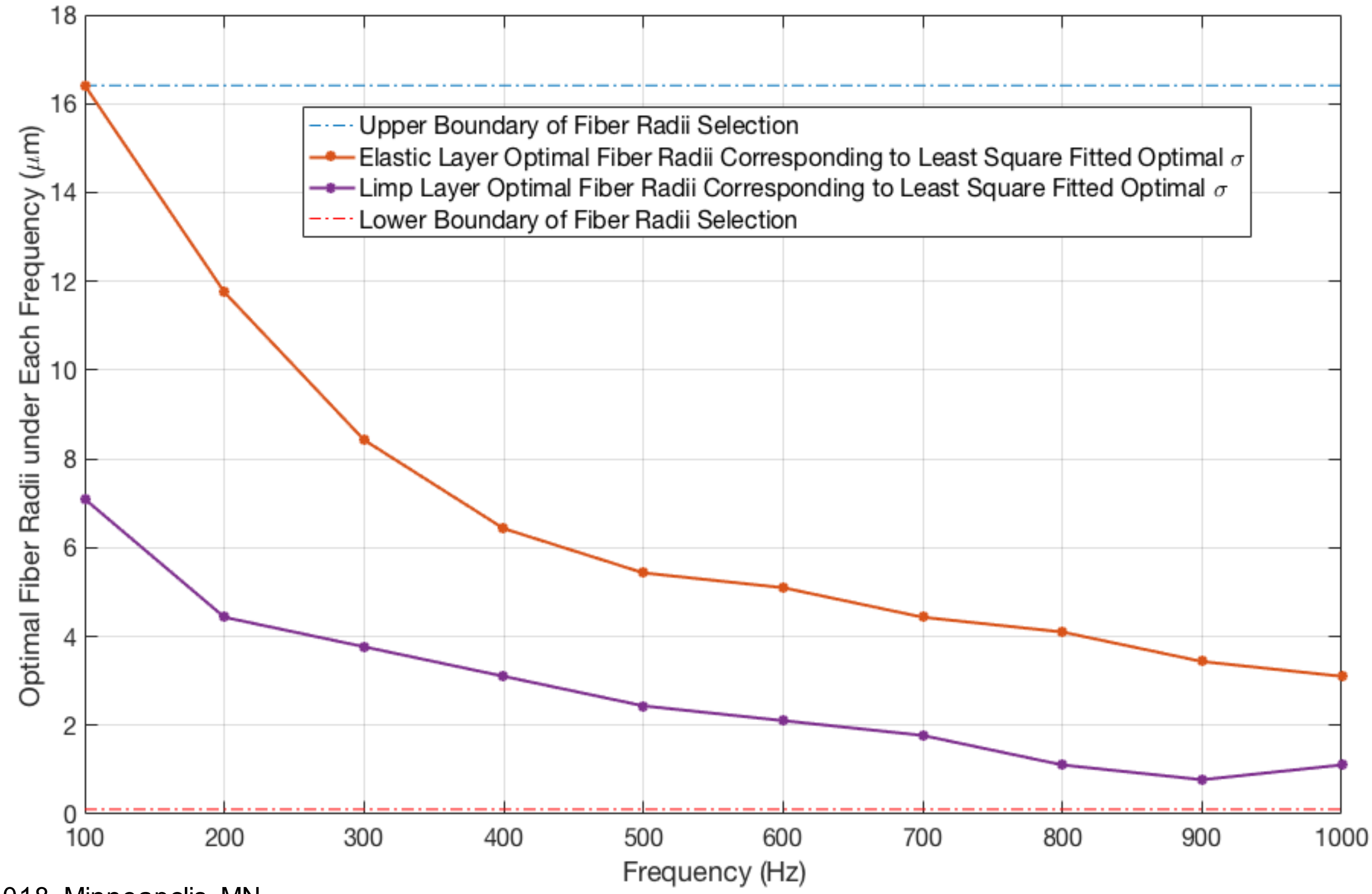
RESULTS – FIBER DESIGN

- **Finding Optimal Fiber Size for Optimal Damping – least square fitting σ's**
  - ➢ Aluminum panel thickness = 3 mm; Loss factor = 0.003; Air loss factor = 0.0005
  - ➢ Polymer fibrous layer thickness = 3 cm; Bulk density= 10 kg/m^3; Tortuosity = 1.2; Porosity = 99%
  - ➢ Fiber inputs: $\boldsymbol{\rho_1 = 910\ \mathrm{kg/m^3}}$; $\boldsymbol{\rho_2 = 1380\ \mathrm{kg/m^3}}$; $\boldsymbol{X_1 = X_2 = 50\%}$; $\boldsymbol{r_2 = 13\ \mu\mathrm{m}}$; $\boldsymbol{r_1}$ → design target



RESULTS – FIBER DESIGN

- **Finding Optimal Fiber Size for Optimal Damping – translating into optimal fiber sizes**
  - Aluminum panel thickness = 3 mm; Loss factor = 0.003; Air loss factor = 0.0005
  - Polymer fibrous layer thickness = 3 cm; Bulk density= 10 kg/m^3; Tortuosity = 1.2; Porosity = 99%
  - Fiber inputs: $\rho_1 = 910\ \mathrm{kg/m^3}$; $\rho_2 = 1380\ \mathrm{kg/m^3}$; $X_1 = X_2 = 50\%$; $r_2 = 13\ \mu\mathrm{m}$; r_1 → design target



CONCLUSIONS

- An optimal airflow resistivity can be found to provide optimal damping (power dissipation within the fibrous layer) at each frequency based on ACM / TMM and NFD
- Corresponding to the optimal airflow resistivity, an optimal fiber size then can be found at each frequency based on AFR and numerical optimization method
- Fibrous dampers are effective at reducing subsonic panel vibrations while absorbing the radiating sound from the panel at the supersonic region
- Fibrous dampers are more effective on thinner structures
- Adding macroscopic stiffness to the fibers helps to improve damping performance
- Relatively large fibers are effective at damping low frequency vibration

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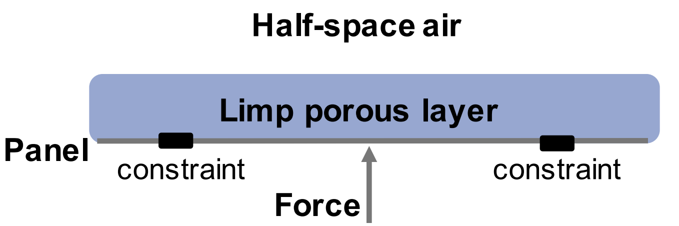
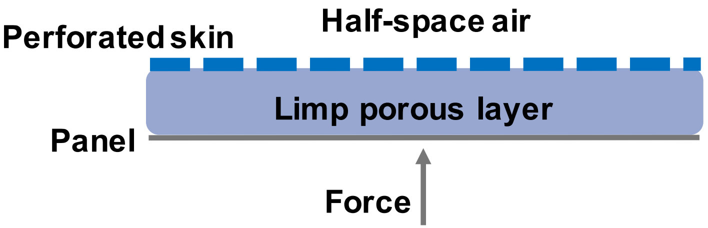
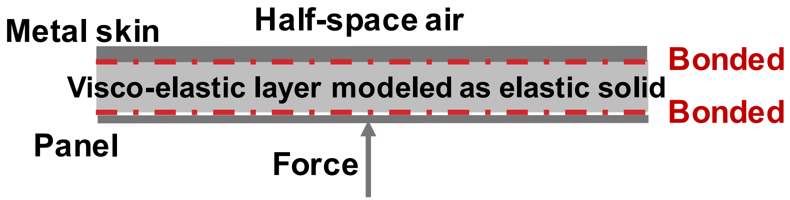
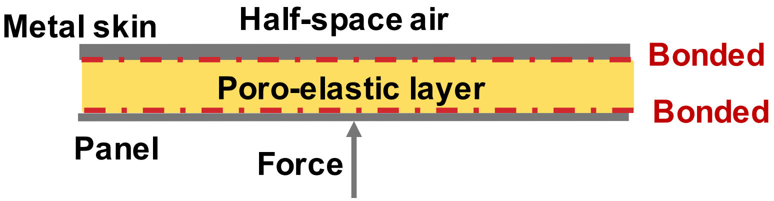
EXTENSION

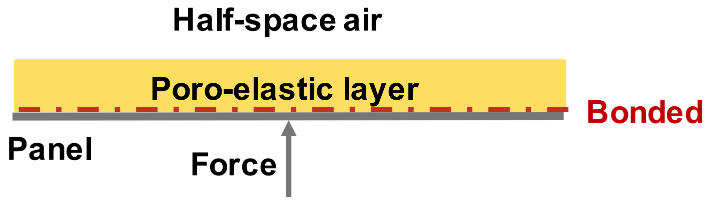
- **Developed cases for the "TMM + NFD + AFR" structural damping model**

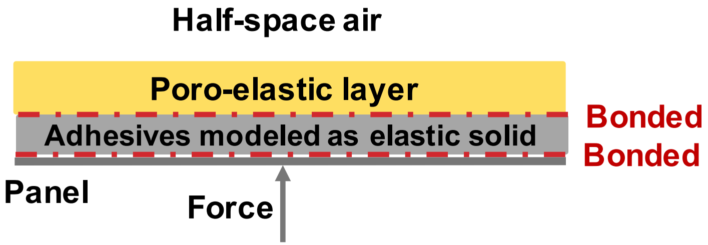

Half-space air
Limp porous layer
Panel
constraint
constraint
Force


Perforated skin
Half-space air
Limp porous layer
Panel
Force


Metal skin
Half-space air
Bonded
Visco-elastic layer modeled as elastic solid
Bonded
Panel
Force

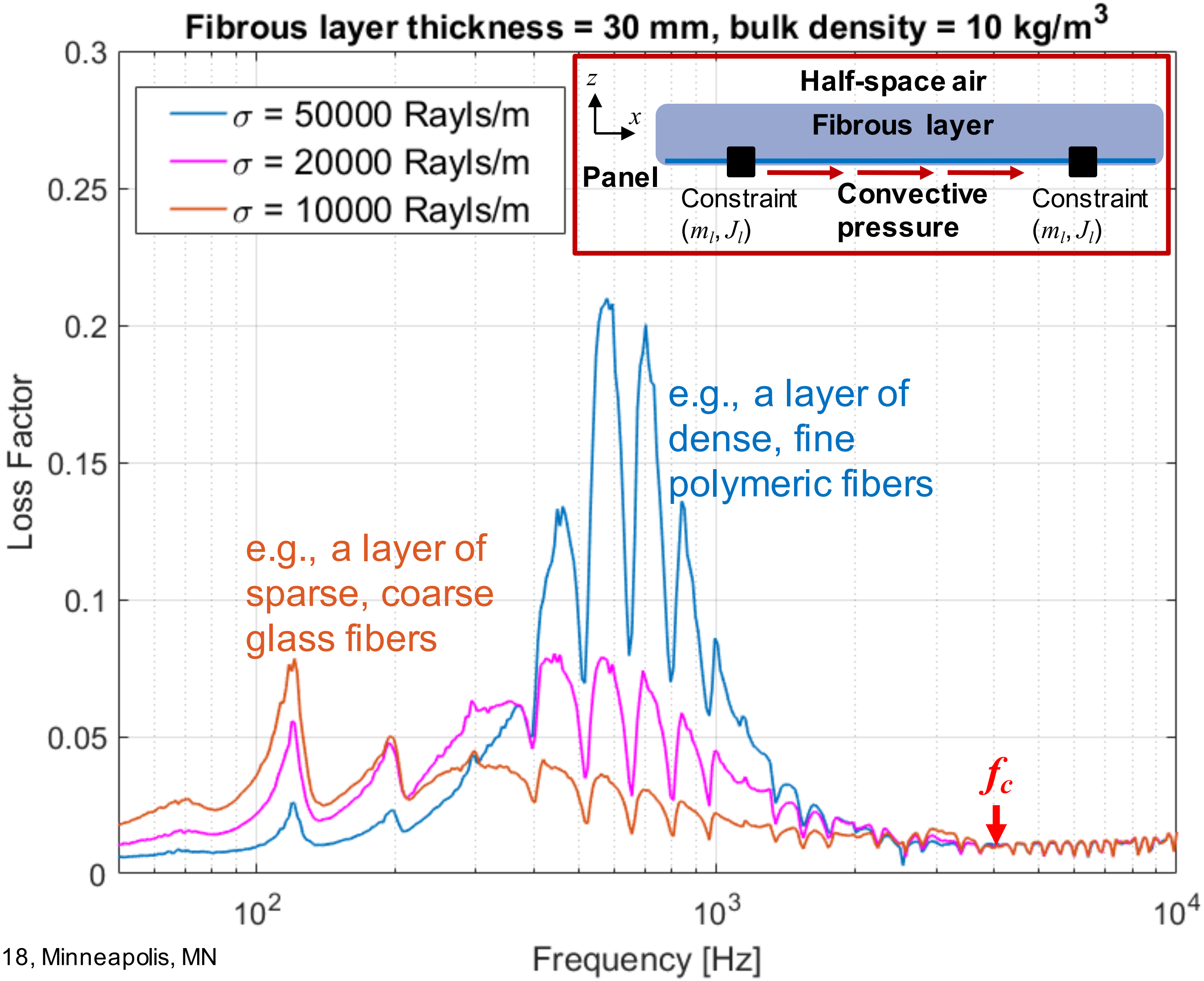

Metal skin
Half-space air
Bonded
Poro-elastic layer
Bonded
Panel
Force


Half-space air
Poro-elastic layer
Bonded
Panel
Force


Half-space air
Poro-elastic layer
Bonded
Adhesives modeled as elastic solid
Bonded
Panel
Force

EXTENSION

- **Inter-Noise 2018 at Chicago, IL**



Fibrous layer thickness = 30 mm, bulk density = 10 kg/m^3

e.g., a layer of dense, fine polymeric fibers

e.g., a layer of sparse, coarse glass fibers

f_c