Fibrous Material Microstructure Design for Optimal Structural Damping

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FIBROUS MATERIAL MICROSTRUCTURE DESIGN FOR OPTIMAL STRUCTURAL DAMPING

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INTRODUCTION

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

![](image1.png)

Traditional Damping Material[1]

![](image2.png)

Structure of a Traditional Damper[2]

• Fibrous Damping Treatments – Target Material of this Study

![](image3.png)

Fibrous Damping Material[3]

![](image4.png)

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

• Literature Review
  - Bruer & Bolton, AIAA 1987\cite{5} – Analysis of different waves propagating in the layered damping structures
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  - Lai & Bolton, Noise-Con 1998\cite{7} – Modeling to prove reasonable structural damping effect from the light fibrous materials through dissipating nearfield energy
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• Layered Structures Shown in the Literature

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)
- **Fibrous Medium Airflow Resistivity Prediction**\[^{11}\]

  ![](image1.png)

  **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 $r_2$

  ![](image2.png)

  **Micro-CT scanned fiber radii distribution of the fibrous medium**
Fibrous Medium Airflow Resistivity Prediction\cite{11}

**Inputs**
- Fiber mean radii: $r_1, r_2$, distribution parameters
- Fiber bulk density: $\rho_b$
- Component weight fractions: $X_1, X_2$
- Solid material densities: $\rho_1, \rho_2$

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

**Step 1:** $C$ calculation based on $\rho_b, X_1, X_2, \rho_1, \rho_2$

**Step 2:** $b^2$ calculation based on $r_1, r_2$, distribution parameters and $C$

**Step 3:** $\sigma$ calculation based on $C$ and $b^2$
After having bulk moduli and wavenumbers of elastic fibers\cite{13, 14},

- **ACM / TMM**
  - **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

Air field waves \((p_1, v_{z1})\):
- Porous media waves
- Waves propagating in the porous layer
- Air field waves \((p_2, v_{z2})\)

Incident wave:
- Boundary Conditions at \(z = 0\)
- Boundary Conditions at \(z = d\)

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

• Choice of IDFT sampling rate $\gamma_s$ 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 $\gamma_s$ 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 $\gamma_s$ 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 $\gamma_s$ 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 $\gamma_s$ and sampling points number $N$\textsuperscript{[12]}
- Target of the NFD model: calculate spatial responses for wide frequency range
- Key point: for each frequency input, choosing proper $\gamma_s$ and $N$ to ensure accurate IDFT results over a large enough spatial span for observation

- Step 3: find the proper sampling rate $\gamma_s$ for each input frequency

$N$ should be large enough to avoid bias
NFD

- Choice of IDFT sampling rate $\gamma_s$ and sampling points number $N$\cite{12}
- Target of the NFD model: calculate spatial responses for wide frequency range
- Key point: for each frequency input, choosing proper $\gamma_s$ and $N$ to ensure accurate IDFT results over a large enough spatial span for observation

- Step 4: identify the critical frequency $f_c$

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**GENERAL APPROACH**

- **Acoustical / Damping Performance Prediction Process**
  - Input: 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**
  
  - **Output**
    - Optimal fiber size
  
  - **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

- **Porous medium:** thickness ($d$), AFR ($\sigma$), bulk density ($\rho_b$), porosity ($\phi$), tortuosity ($\alpha_\infty$), Young’s modulus ($E$), Poisson’s ratio ($\nu$), loss factor ($\eta$)
- **Panel:** basis weight ($m_s$), flexural stiffness per unit width ($D_p$)

**Power radiation into the porous layer:**

$$P_1 = \frac{1}{2\pi} \text{Re} \left\{ \int_{-\infty}^{\infty} p_1 v_{z1}^2 \, dx \right\}$$

**Power radiation into the air:**

$$P_2 = \frac{1}{2\pi} \text{Re} \left\{ \int_{-\infty}^{\infty} p_2 v_{z2}^2 \, dx \right\}$$

**Power dissipation in the porous layer**:

$$P_d^{\text{air}} = P_1 - P_2$$

- **Find maximum $P_d$ (optimal damping) and corresponding optimal $\sigma$ for frequency of interest**
- **AFR Model combined with least square optimization returns optimal porous material microstructure details (e.g. fiber sizes)**

**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:

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

**Panel mechanical impedance**:

$$Z_m = i [(D/\omega)k_x^2 - \omega m_s]$$

**Near-field acoustic impedance**:

$$Z_{a1} = \frac{\tau_{11}Z_{a2} + \tau_{12}}{\tau_{21}Z_{a2} + \tau_{22}}$$

**Far-field acoustic impedance**:

$$Z_{aF} = (\omega \rho_{air})/k_{zair}$$

**Pressure-velocity relation**:

$$p_i = Z_{a1}v_{z1} (i = 1, 2)$$

**Transfer Matrix Method / Arbitrary Coefficient Method**

**Governing Equation Fourier Transform (GEFT)**

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

**Near-field-far-field relation**

$$\begin{bmatrix} P_1 \\ v_{z1} \end{bmatrix}_{z=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} P_2 \\ v_{z2} \end{bmatrix}_{z=d}$$

**Complex wave numbers**

For limp or rigid frame, $i = 1$ for limp or rigid frame, $i = 3$ for elastic frame

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

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

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RESULTS – BARE PANEL

- Spatial Velocity Level (dB)

Total points N = 16384. Wave# sampling rate $\gamma_s = 66-383\text{rad/m}$. Frequency range = 10-10000Hz

Panel Thickness = 3 mm. Panel Loss Factor = 0.003. Air Loss Factor = 0.0005
RESULTS – LIMP LAYER

- Spatial Velocity Level (dB)

Total points N = 16384. Wave# sampling rate $\gamma_s = 66-383$ rad/m. Frequency range = 10-10000 Hz
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$^3$

![Graph showing velocity level vs. frequency and distance]
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/m³

Half-space air

Force

x = 0

VS.

Limp porous layer

Force

x = 0

10 Hz

56 Hz

316 Hz

1778 Hz

10000 Hz

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RESULTS – COMPARISON

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

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

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

• Power Distribution

Half-space air

\[ x = 0 \]

\[ \text{Force} \]

\[ P_{in} : \text{Power input by the driving force} \]

\[ P_s : \text{Power staying in the panel} \]

\[ P_1 = P_2 : \text{Power radiating into the air} \]

\[ f_c \]

\[ \times 10^{-3} \text{ Panel Thickness = 3 mm. Panel Loss Factor = 0.003, Air Loss Factor = 0.0005} \]
RESULTS – LIMP LAYER

• Power Distribution

Half-space air
Limp porous layer
\[ x = 0 \]

Subsonic region attenuation due to power dissipation within the layer

\[ P_i \]: 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

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/m³
• Power Distribution

Panel Thickness = 3 mm. Panel Loss Factor = 0.003. Air Loss Factor = 0.0005
Porous Layer Thickness = 30 mm. Airflow Resistivity = 20000 Rayls/m. Bulk Density = 10 kg/m³.
Young's Modulus = $10^6$ Pa. Poisson's Ratio = 0.3. Loss Factor = 0.3

$P_{\text{in}}$: Power input by the driving force
$P_1$: Power radiating into the layer
$P_d$: Power dissipation within the layer
$P_s$: Power staying in the panel
$P_2$: Power radiating into the air

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
• Finding Optimal Fiber Size for Optimal Damping – least square fitting $\sigma$’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: $\rho_1 = 910$ kg/m$^3$; $\rho_2 = 1380$ kg/m$^3$; $X_1 = X_2 = 50\%$; $r_2 = 13$ $\mu$m; $r_1 \rightarrow$ 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: \( \rho_1 = 910 \text{ kg/m}^3; \rho_2 = 1380 \text{ kg/m}^3; X_1 = X_2 = 50\%; r_2 = 13 \text{ \mu m}; r_1 \to \text{design target} \)
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.
ACKNOWLEDGEMENTS

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REFERENCES

• Developed cases for the “TMM + NFD + AFR” structural damping model
Fibrous layer thickness = 30 mm, bulk density = 10 kg/m$^3$

- Inter-Noise 2018 at Chicago, IL

- e.g., a layer of sparse, coarse glass fibers
- e.g., a layer of dense, fine polymeric fibers

$\sigma = 50000$ Rayls/m
$\sigma = 20000$ Rayls/m
$\sigma = 10000$ Rayls/m

Convective pressure

Constraint ($m_i, J_i$)

Half-space air

Fibrous layer

Panel

$z$
$x$