ADAPTIVE STRUCTURES AND NOISE CONTROL

Faculty Members

• George Lesieutre
• Mary Frecker
• Reginald Hamilton
• Zoubeida Ounaies
• Chris Rahn
• Kenji Uchino
• Gordon Warn
ADAPTIVE STRUCTURES AND NOISE CONTROL

- Zoubeida Ounaies
  - Active Fiber Composites
- Mary Frecker
  - Multi-field Responsive Origami Structures
- Chris Rahn
  - Piezo Energy Harvesting from Human Motion
- George Lesieutre
  - Multilayered Radial Isolator for Helo Noise Reduction
- Gordon Warn
  - Optimal Topology of Column Bearings to Reduce Floor Accel in Multi-story Base-isolated Buildings
Active Fiber Composites

Electroactive Materials Characterization Lab

Zoubeida Ounaies and Group
Mechanical and Nuclear Engineering,
Penn State University, University Park, PA
What we do...

*Develop, synthesize, process, and characterize new electro-active, possibly nano-enhanced materials*

**Capabilities for synthesis and fabrication**

**Structure-property relationship**

**Sensing-Actuation-Storage Applications**
Active Fiber Composites for High Strain Apps

**Active fiber composites (AFC) advantages**
- Flexibility
- Different shapes inexpensively
- Light weight
- Embedded easily in laminate composites
- Self powered

**AFCs limitations**
- Coupled electro-mechanical stimuli and hostile environments cause AFC constituents to experience nonlinear and inelastic behaviors, leading to complex failure modes
- More than 50% vol polymer, which makes behavior time- and temperature- dependent

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**Tensile tests**

**Hysteresis loops**

**Electro-mechanical loops**
Schematic of an AFC

- Positive electrodes
- Negative electrodes
- Epoxy matrix
- Kapton tape
- PZT fibers

SEM of AFC cross section
Motivation

- Non-linearity in the piezoelectric material
- Non-uniformity of the electric field inside the AFC
Our Focus

• Use a combination of experimental and numerical approaches to examine overall behavior of AFCs to:
  – Quantify the impact of constituent properties (polymer matrix and PZT fiber) on coupled response
  – Build a model that considers non-uniform electric field behavior and time- and temperature-dependent properties
  – Conduct an exhaustive parametric study of AFC design

One outcome -> re-design an improved AFC device with optimized electro-mechanical coupling by taking advantage of technology advances in manufacturing and electronics
Multi-field responsive Origami Structures

Mary Frecker, Ph.D.
Professor of Mechanical Engineering & Bioengineering
Director, the Learning Factory

Sponsored by NSF EFRI 1240459
The research team includes faculty in design, active materials, origami math, and art.

- Jyh-Ming Lien
  - GMU
  - Origami math

- Paris von Lockette
  - Rowan
  - Active materials & modeling

- Mary Frecker
  - PSU
  - Compliant mechanisms

- Zoubeida Ounaies
  - PSU
  - Active materials

- Rebecca Strzelec
  - PSU Altoona
  - Art

- Tim Simpson
  - PSU
  - Design theory
The research vision is multi-field origami structures for active folding and unfolding into complex 3D shapes

- Initial flat sheet $S_i$
- Folded shape $S_1$ due to magnetic field
- Folded shape $S_2$ due to electric field
- Folded shape $S_3$ due to thermal field
- Unfold back to flat sheet $S_i$ (M. Shlian)
We are developing dielectric elastomers to achieve active folding
We are also developing magneto-active elastomers capable of active folding and unfolding.
We have demonstrated 3D folding and locomotion with the MAE composite material
A multi-field responsive DE / MAE sheet bends in orthogonal directions due to electric / magnetic fields.
Piezoelectric Energy Harvesting from Human Motion

Xiaokun Ma
Christopher D. Rahn
Department of Mechanical and Nuclear Engineering
Penn State University
Piezoelectric EH Devices

Bimorph energy harvester with a tip proof mass
Device dimension: $53.0 \times 31.7 \times 0.675 \text{ mm}^3$
(Kim, Smart Materials and Structures, 2010)

Impulse-excited energy harvester
Cantilever dimension: $72 \times 5 \times 0.5 \text{ mm}^3$
(Pillatsch, Smart Materials and Structures, 2012)

Parametric frequency increased generator
Inertial mass: 9.3 g, Total volume: 2.8 $\text{cm}^3$
(Galchev, J. Microelectromechanical Systems, 2012)
# Piezoelectric EH Benchmarks

<table>
<thead>
<tr>
<th>Reference</th>
<th>Material</th>
<th>Size</th>
<th>Power (µW)</th>
<th>Power/Area*Accel (µW/cm²g)</th>
<th>Frequency (Hz)</th>
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</table>

**ASSIST Goals:**
- Small devices
- Low frequency excitation
- High µW/cm²g

**Promising Approaches:**
- Bistable devices
- Compliant mechanisms
- High performance materials and flexible substrates
- Strain harvesting

**Size (area in cm²):** 0-.001 (n), .001-.01 (µ), .01-.1 (m), .1-1 (M), and >1 (G)

**Note:** Mix of average/peak power and device/material area
Piezo Unimorph Cantilever Model for EH Analysis / Design

Euler-Bernoulli beam model:

Beam mechanics:

\[
YI \frac{\partial^4 \omega_{rel}(x, t)}{\partial x^4} + c_s I \frac{\partial^5 \omega_{rel}(x, t)}{\partial x^4 \partial t} + c_a \frac{\partial \omega_{rel}(x, t)}{\partial t} + m \frac{\partial^2 \omega_{rel}(x, t)}{\partial x^2} = -m \frac{\partial^2 \omega_b(x, t)}{\partial t^2}
\]

Coupled electrical circuit equation:

\[
\frac{\varepsilon^{s33} b L \frac{dv(t)}{dt}}{h_p} + \frac{v(t)}{R_l} = - \int_{x=0}^{L} d_{31} Y_p h_{pc} b \frac{\partial^3 \omega_{rel}(x, t)}{\partial x^2 \partial t} dx
\]

Boundary conditions:
- Cantilevered at \( x = 0 \).
- Proof mass at \( x = L \).

Model parameters:
- \( c_s \): internal damping term (strain rate)
- \( d_{31 \, \text{piezoelectric}} \)
- \( Y, Y_p \): beam, piezo Young's modulus
- \( m \): mass per unit length of the beam
- \( c_a \): external viscous damping term (air)
- \( \varepsilon^{s33} \): permittivity
- \( I \): moment of inertia
- \( R_l \): load resistance
- \( h_{pc} \): distance from PZT center to neutral axis

Model input: Base excitation
Model outputs: Beam voltage \( v(t) \), current, power and tip displacement
Energy Harvester Model Results

![Diagram showing power and tip displacement graphs with different resistances.](image)

- **Power**
  
  - Yellow: $R = 1 \, \text{M}\Omega$
  - Dashed: $R = 100 \, \text{k}\Omega$
  - Green: $R = 10 \, \text{k}\Omega$
  - Red: $R = 1 \, \text{k}\Omega$
  - Blue: $R = 100 \, \text{\Omega}$

- **Tip Displacement**

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$\frac{\langle \omega^2 Y_0 \rangle^2}{\text{W.s}^2/\text{m}^2}$

Frequency (Hz)
Next Steps: Model Based Design of ASSIST Piezoelectric Harvesters

Cantilevered beam design:

![Cantilevered beam design diagram]

Circular membrane design:

![Circular membrane design diagram]

New model under development...(stay tuned!)
Multilayered Radial Isolator for Helicopter Interior Noise Reduction

Pauline Autran
David J. Materkowski
George A. Lesieutre
Isolation mounts can reduce vibration transmitted to helo cabin

- Interior noise: safety and comfort
- Need to attenuate vibration transmission
- Concept: multi-layered radial isolators at gearbox
  - Statically stiff, dynamically soft?

- Models
  - Finite element
  - Augmented assumed modes
- Transmissibility exhibits “stop bands”
- Validated experimentally
Multilayered axial isolator exhibits stop band over key freq range

- Stop band freqs bounded by high global mode and low local mode
- Objective: reduce vibration in the frequency range [500Hz; 2000Hz]
Can a radial isolator offer similar performance?

Conceive, model and design layered radial isolator
Models can indicate potential isolation performance

- Mode shapes and frequencies
- Transmissibility

**Finite-element**

all modes of interest

**Augmented assumed-modes**

- Assumed-modes global
- Analytical local
Mode shapes and frequencies of a five-layered isolator agree well

- Pure rotational mode: $f = 71.5$ Hz
- Translation with in-phase motion of the metal layers: $f = 105.2$ Hz
- Translation with out-of-phase motion of the metal layers: $f = 179.5$ Hz
- Bending of the outermost layer: $f = 361.8$ Hz
Better performance achieved with a five-layered isolator

Characteristics of five-layered isolator

<table>
<thead>
<tr>
<th></th>
<th>metal</th>
<th>elastomer</th>
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<tbody>
<tr>
<td>$E$ (Pa)</td>
<td>2.10E+11</td>
<td>1.20E+06</td>
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<tr>
<td>$ρ$ (kg/m$^3$)</td>
<td>7850</td>
<td>1000</td>
</tr>
<tr>
<td>Poisson's ratio</td>
<td>0.3</td>
<td>0.499</td>
</tr>
<tr>
<td>thickness (m)</td>
<td>0.008</td>
<td>0.005</td>
</tr>
<tr>
<td>loss factor</td>
<td>0.0005</td>
<td>0.01</td>
</tr>
<tr>
<td>Inner radius (m)</td>
<td>0.021</td>
<td></td>
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</table>

- Stop band: 500 to 2011 Hz
- Transmissibility: 0.0009

The two models agree very well in the target frequency range
Experiments were performed to validate the computational models

Accelerance data recorded at the innermost and outermost rings over a 5000 Hz range. Gaussian noise and linear sine sweep forcing signals were used for experimental verification.
Five-layer assumed-modes model and measurement agree pretty well.

Stop band: 800 / 880 Hz – 2470 Hz

Transmissibility: 0.10 vs. 0.21 (50%)

Model does not include frequency-dependence of elastomer behavior.
Multilayered radial isolators reduce transmitted vibration

- Models (3 & 5 lyr)
  - Finite element
  - Augmented assumed modes (2000 X faster)

Helo interior noise: safety and comfort

Need to attenuate vibration, 500-2000 Hz

- Transmissibility shows predictable “stop bands”

- Multilayered radial isolator at gearbox
  - Designs can

- Validated experimentally
  - Transmissibility <20% (900-2500)
Optimal Topology of Column Bearings for Reducing Vertical Floor Acceleration in Multi-story Base-isolated Buildings

Gordon Warn¹
Mehmet Unal²

Department of Civil and Eng. Engineering
1. Assistant Professor
2. PhD Candidate
Shape Memory Alloys: Material Design

Reginald F. Hamilton, PhD
Assistant Professor of Engineering Science and Mechanics
Optimal Topology of Column Bearings for Reducing Vertical Floor Acceleration Demands in Multi-story Base Isolated Buildings

Gordon Warn\textsuperscript{1}
Mehmet Unal\textsuperscript{2}

Department of Civil and Eng. Engineering
1. Assistant Professor
2. PhD Candidate
Background – Base isolation

Fixed Base

Isolated

ACCELERATION

PERIOD

DISPLACEMENT

PERIOD

PERIOD SHIFT

PERIOD SHIFT
Background – Base isolation hardware

Elastomeric
- Low-damping natural rubber
- High-damping rubber
- Lead-rubber

Sliding
- Friction Pendulum™
- Triple Pendulum™
- Others
Background – Elastomeric bearings

Shape factor \((S)\):

\[
S = \frac{\text{Loaded area}}{\text{Area free to bulge}}
\]

<table>
<thead>
<tr>
<th>Country</th>
<th>Application</th>
<th>Typical range of Shape factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>U.S.</td>
<td>Non-seismic bridge</td>
<td>4 – 6</td>
</tr>
<tr>
<td>U.S.</td>
<td>Seismic Isolation</td>
<td>15 – 25</td>
</tr>
<tr>
<td>Japan</td>
<td>Seismic Isolation</td>
<td>&gt; 30</td>
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</tbody>
</table>

Bonded rubber diameter

Thickness of individual rubber layer

Area free to bulge

Loaded area
Earthquake hazard & Base isolated periods

Elastic earthquake response spectra
- 6.5 – 8 Magnitude
- R < 10 km
- Stiff soil / Rock
- Damping - 5% of critical

Base isolated periods
- Horizontal: 2.5 s – 4 s
- Vertical: 0.05 s – 0.2 s
**Problem statement**

1. Base isolation does not protect against vertical ground motion
2. Vertical period aligns with dominate frequency content
3. Large vertical acceleration demands can lead to:
   - Ceiling system failure
   - Piping system failure
   - Content disruption
Vertically distributed flexibility (VDF) concept

Base isolated

Base isolated with VDF

Base Isolators
Column Bearings

S=4
S=5
S=7
S=30
VDF model

\[
\begin{align*}
\begin{bmatrix} M \end{bmatrix} \{ \ddot{U} \} + \begin{bmatrix} C \end{bmatrix} \{ \dot{U} \} + \begin{bmatrix} K \end{bmatrix} \{ U \} &= \begin{bmatrix} M \end{bmatrix} \{ I \} \{ \ddot{U}_g \} \\
\text{Influence vector}
\end{align*}
\]
Proof of VDF concept using 9 story frame

1. VDF effectively reduced vertical acceleration demands
2. VDF does not alter horizontal response
3. Location of column bearings arbitrarily chosen
Multi-objective optimization

Objectives:
1. Minimize cost of VDF configuration
2. Minimize $\max_i q_i$
   
   $q_i$: median vertical acceleration of $i$th floor

Evolutionary Algorithm
- Nondominated sorting genetic algorithm (NSGA II)
- Fast and Elitist Multiobjective (MOEA)
- Deb et al. (2000)
Multi-objective optimization – Three story

Generations of pareto front surface

- Generation 1
- Generation 4
- Generation 10

Cost in USD (millions)

Max $|q_i| (g)$
Multi-objective optimization – Three story

Generation 10

Cost in USD (millions)

Max $|q_i| (g)$

1 level of VDF

3 levels of VDF

Base isolated

1 level of VDF

3 levels of VDF

Peak vert. abs. acc. (g)

Floor
Multi-objective optimization – Nine story

Generations of pareto front surface

![Graphs showing generations of the pareto front surface with cost in USD (millions) on the x-axis and Max |q_i| (g) on the y-axis.]
Multi-objective optimization – Nine story

- Base isolated
- 2 levels of VDF
- 4 levels of VDF

Cost in USD (millions)

Max $|q_i| (g)$

Generation 10

- 2 levels of VDF
- 4 levels of VDF

Peak vert. abs. acc. (g)

Floor

2013 CAV Workshop
Multi-objective optimization – Twenty story

Generations of pareto front surface

- Generation 1
- Generation 4
- Generation 10

Cost in USD (millions)
Multi-objective optimization – Twenty story

Generation 10

4 levels of VDF

9 levels of VDF

Max $|q_i| (g)$

Cost in USD (millions)

Base isolated

4 levels of VDF

9 levels of VDF

Floor 14
Floor 9
Floor 6
Floor 2

Floor 12
Floor 11
Floor 7
Floor 6
Floor 4
Floor 3
Floor 2

Floor 7
Floor 9
Floor 6
Floor 2

Floor 3

2013 CAV Workshop
Summary

1. VDF Concept effectively reduced vertical acceleration demands

1. Multi-objective evolutionary algorithms useful tool to quantify trade-off

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VDF Concept effectively reduced vertical acceleration demands. Multi-objective evolutionary algorithms are a useful tool to quantify the trade-off. VDF is most beneficial in taller multi-story buildings.
Summary

1. VDF Concept effectively reduced vertical acceleration demands

1. Multi-objective evolutionary algorithms useful tool to quantify trade-off

1. VDF most beneficial in taller multi-story buildings