Recent Advances in Structural Dynamics and Controls

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Structural Dynamics and Controls Research Lab

The Penn State Structural Dynamics and Controls Research Lab (SDCL) emphasizes the study of vibration, stability, and control of mechanical structures. Programs supported by various government and industrial sponsors.

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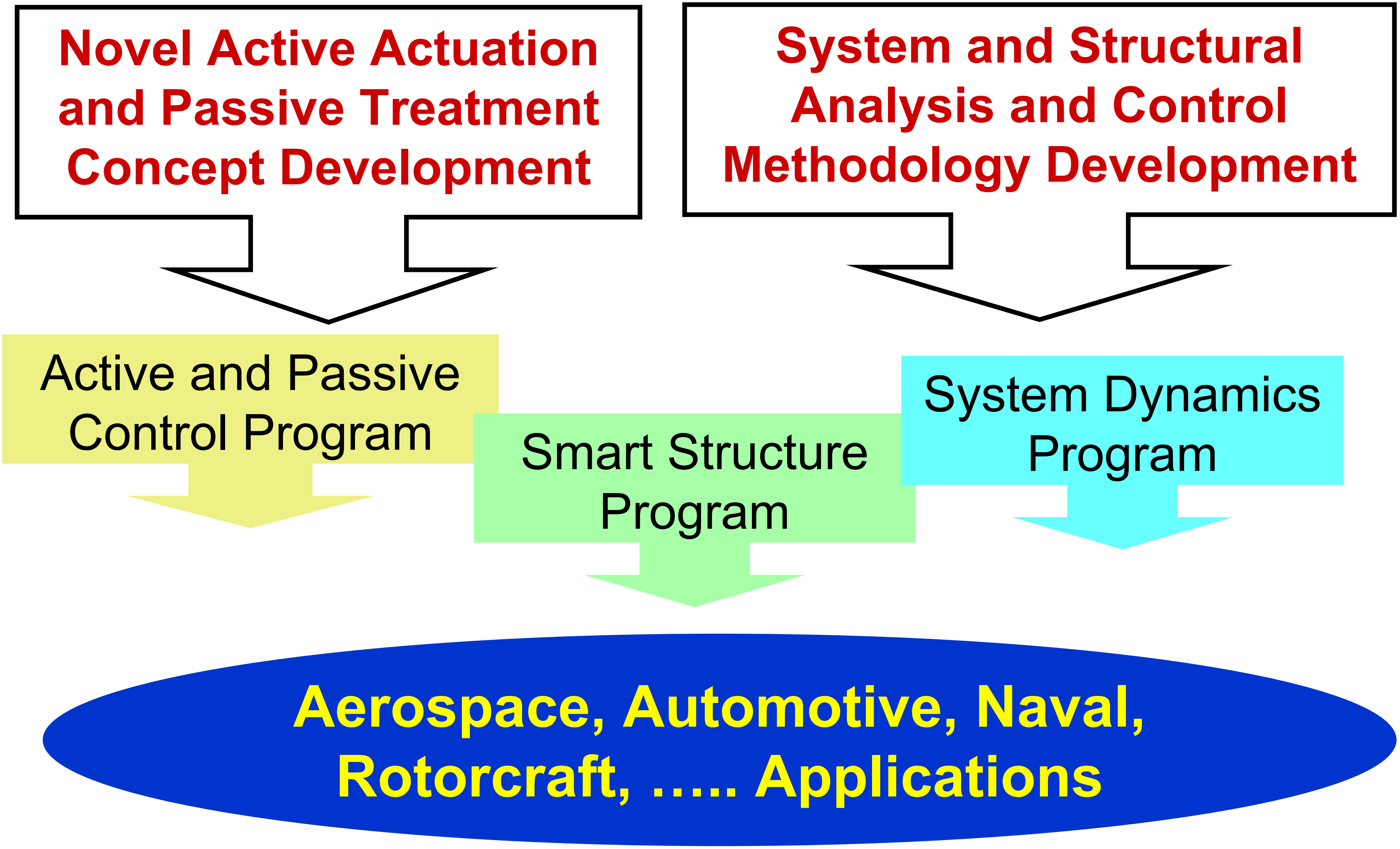
Structural Dynamics and Controls Research Lab



Novel Active Actuation and Passive Treatment Concept Development

System and Structural Analysis and Control Methodology Development

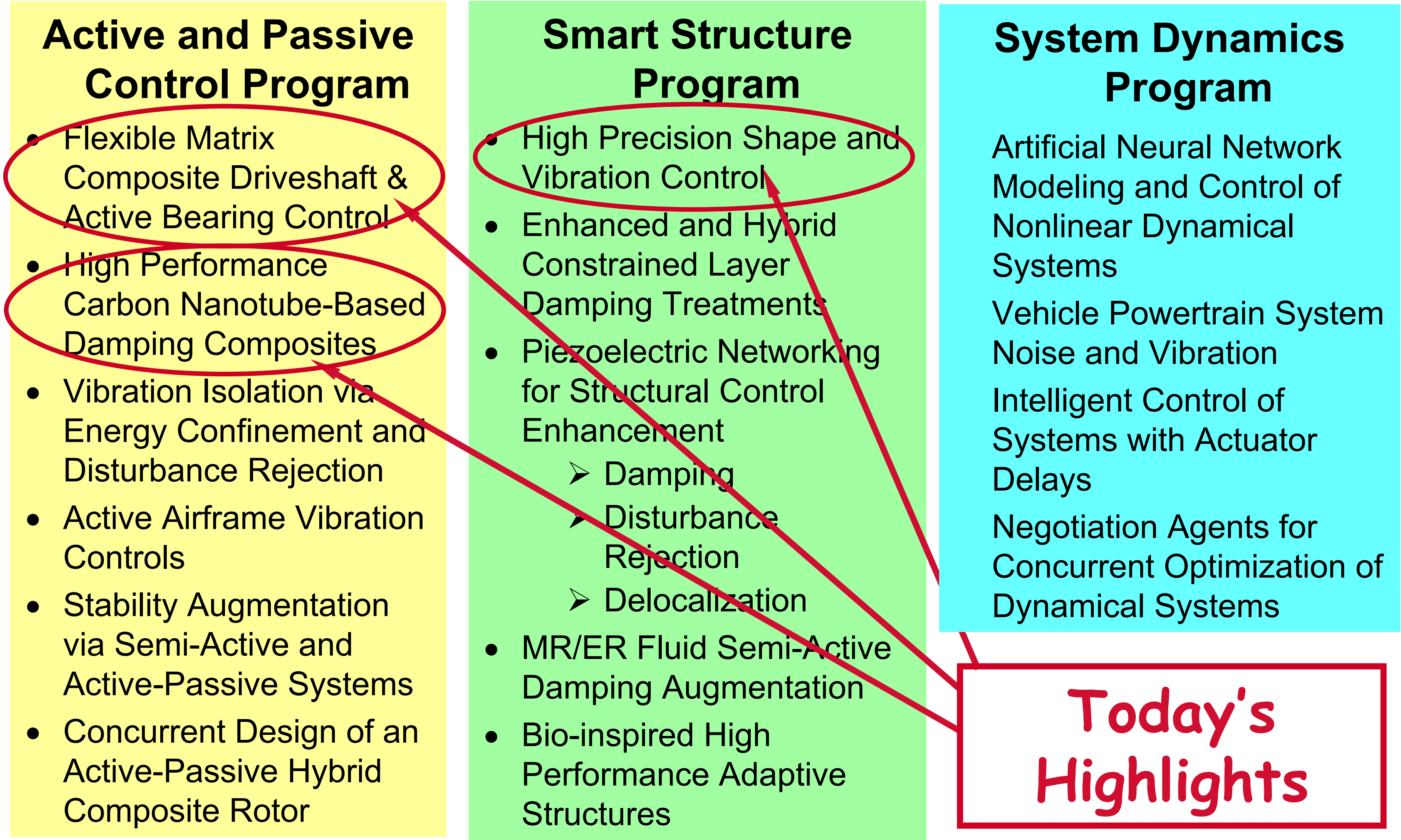
Active and Passive Control Program

Smart Structure Program

System Dynamics Program

Aerospace, Automotive, Naval, Rotorcraft, ….. Applications

Recent Research Projects



Active and Passive Control Program

- Flexible Matrix Composite Driveshaft & Active Bearing Control
- High Performance Carbon Nanotube-Based Damping Composites
- Vibration Isolation via Energy Confinement and Disturbance Rejection
- Active Airframe Vibration Controls
- Stability Augmentation via Semi-Active and Active-Passive Systems
- Concurrent Design of an Active-Passive Hybrid Composite Rotor

Smart Structure Program

- High Precision Shape and Vibration Control
- Enhanced and Hybrid Constrained Layer Damping Treatments
- Piezoelectric Networking for Structural Control Enhancement
  - ➢ Damping
  - ➢ Disturbance Rejection
  - ➢ Delocalization
- MR/ER Fluid Semi-Active Damping Augmentation
- Bio-inspired High Performance Adaptive Structures

System Dynamics Program

Artificial Neural Network Modeling and Control of Nonlinear Dynamical Systems

Vehicle Powertrain System Noise and Vibration

Intelligent Control of Systems with Actuator Delays

Negotiation Agents for Concurrent Optimization of Dynamical Systems

Today's Highlights

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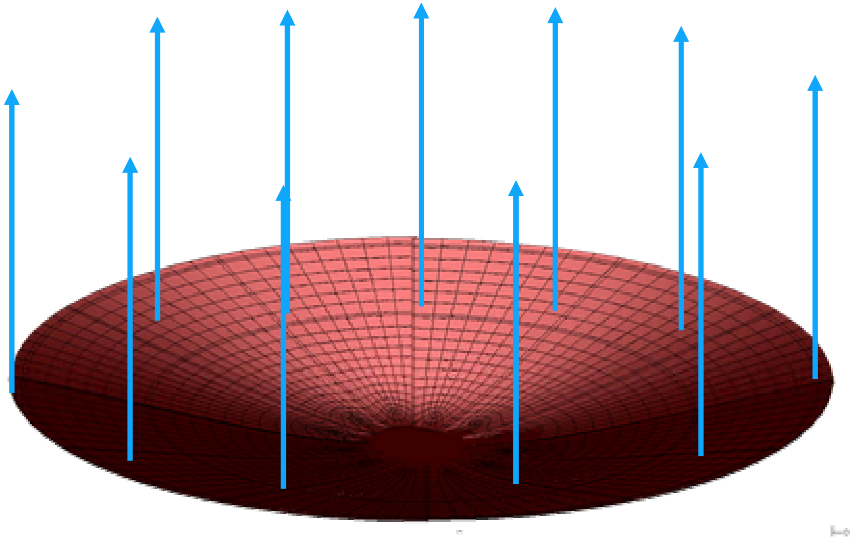
Today's Highlights

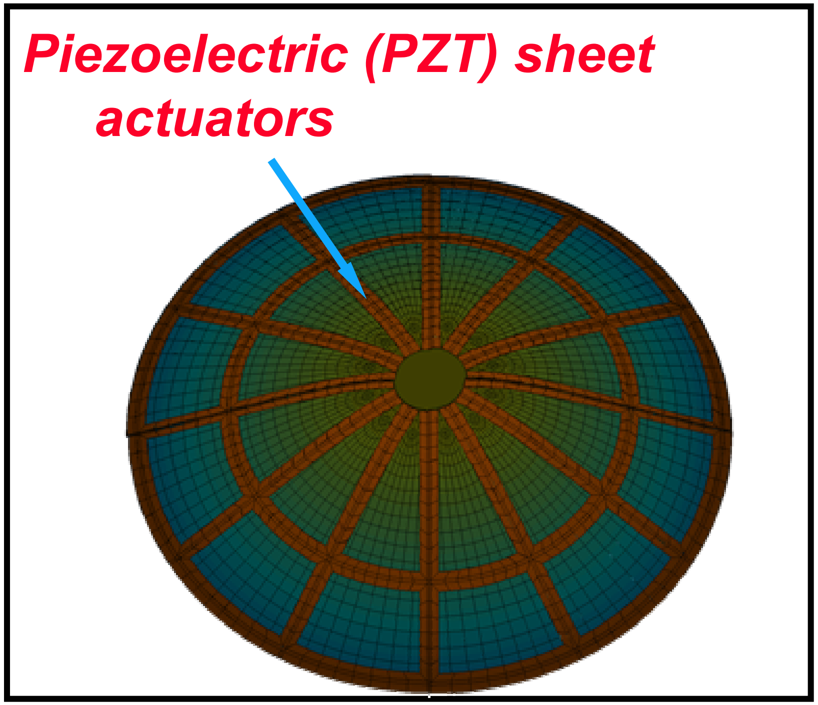
- **Directional Decoupling Piezoelectric Actuators for Shape and Vibration Control of Plate Structures**
- **Robust Adaptive Vibration and Stability Control of Flexible Driveshafts via Magnetic Bearings**
- **Carbon Nanotube-Based Damping Composites**

Introduction - Overall Goal

High-precision shape and vibration control of a light-weight face sheet mirror to compensate for manufacturing defects, thermal loads, mechanical vibration, and orbital corrections

- *Previous investigators have shown promising results*
- *But limited by the two-dimensional effect of PZT sheets*





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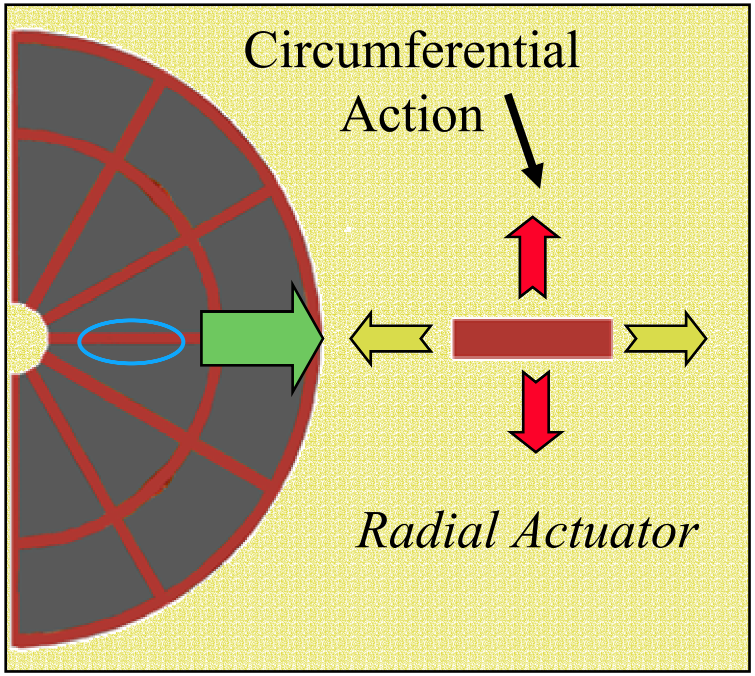
Effect of Circumferential Action of Radially Placed Actuators

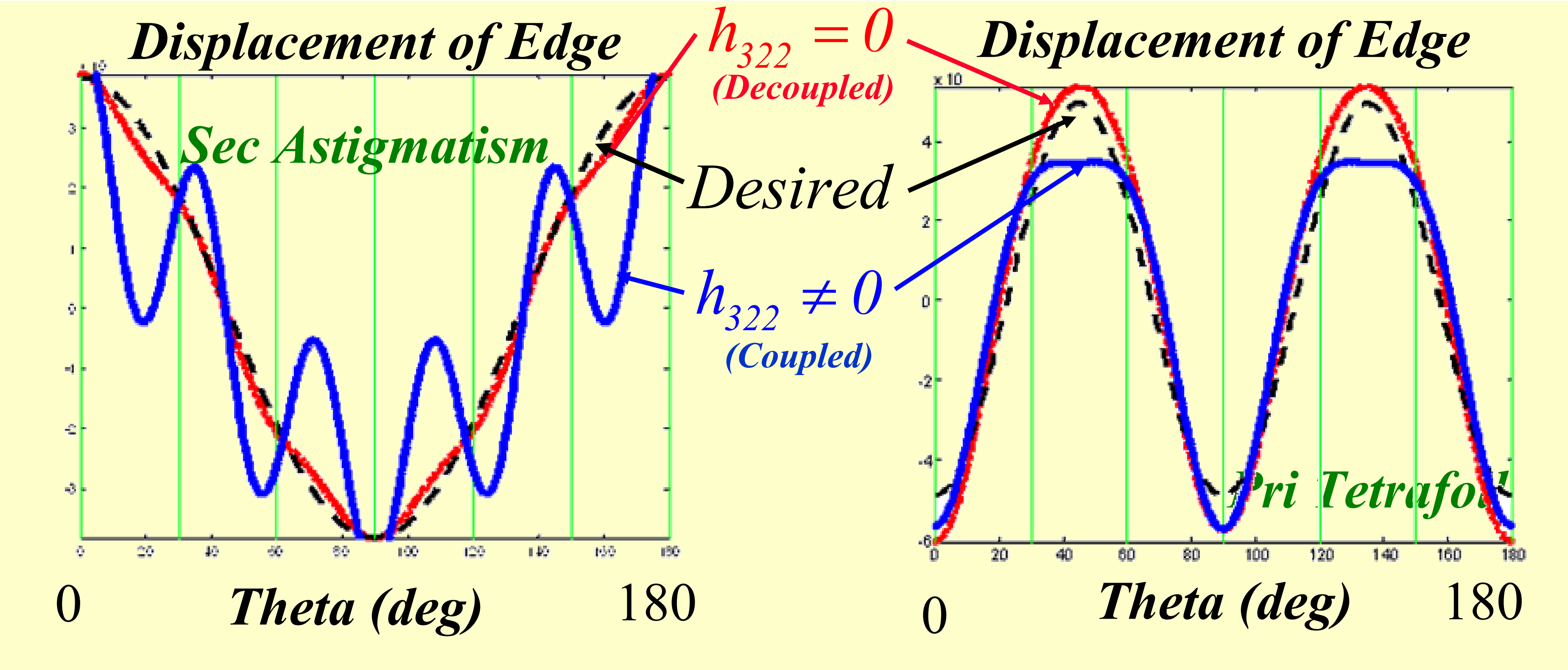
- Electromechanical coupling equal in both directions
- Effect of circumferential action of thin radial actuators
  - **Yields negligible control authority in the lower order circumferential modes**
  - **But can cause unwanted localized deformation in the circumferential direction**



Eliminating the circumferential (radial) actions of the radial (circumferential) actuators will reduce the possibility of 'exciting' the higher order modes

Effect of Directional Decoupling

- Circumferential expansion of radial actuators can “excite” the higher order modes
- Decoupling of actuator can improve the surface smoothness



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Directional Decoupling Ideas

Actuator Issues

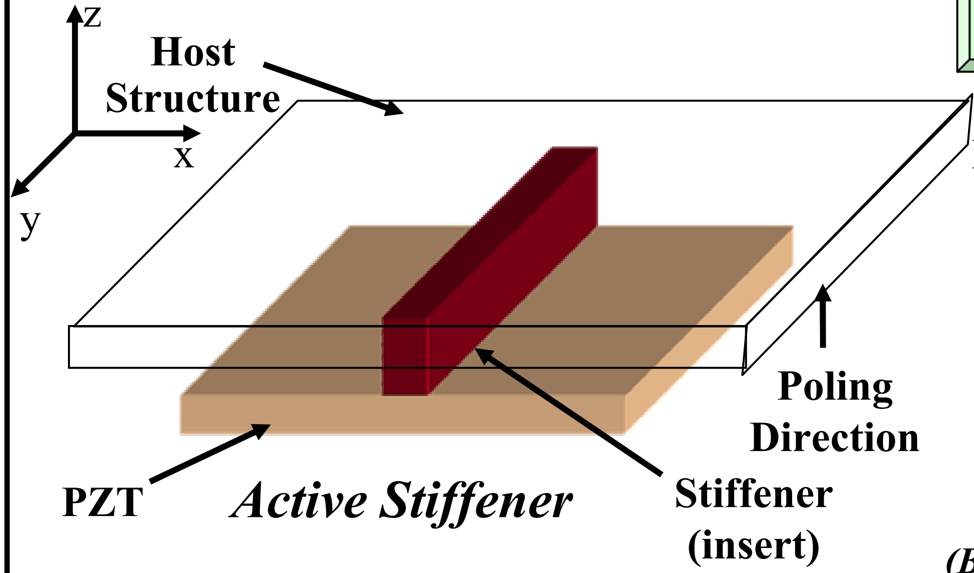
- Two-dimensional action of actuator increases the difficulty in achieving the high-precision requirement → Decoupling action will be helpful
- Decoupling requires a **mechanism** to reduce the transmitted actuation in one of the two directions

Key Properties

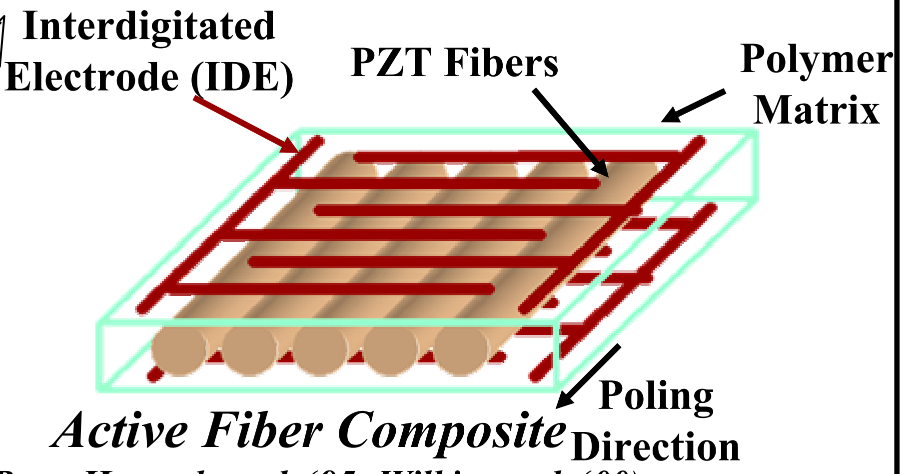
- **Reduces authority in decoupled direction**
- **Stiffens the structure**
- **Simple and effective design**

Directional Decoupling Methods

- ➢ Active Stiffener (AS)
- ➢ Active Fiber Composite (AFC)



Active Stiffener



Active Fiber Composite

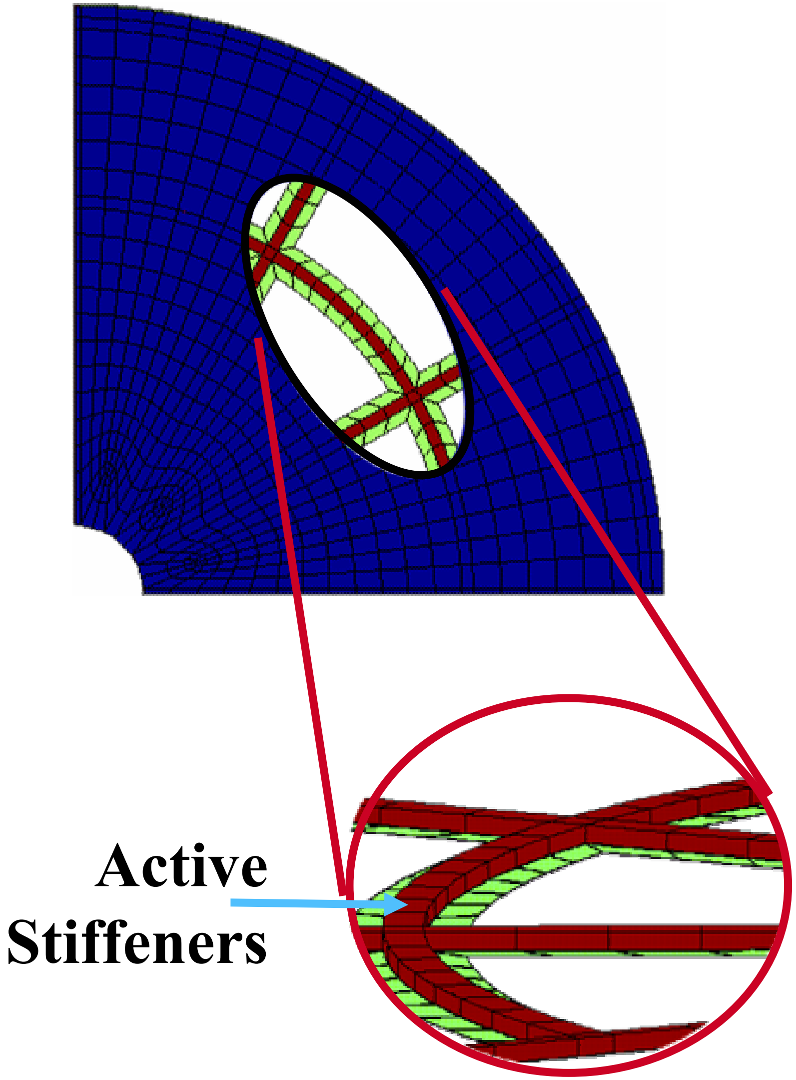
(Bent, Hagood, et al. '95; Wilkie, et al. '00)

Technical Objectives

- Investigate the AS and AFC concepts for directional decoupling
- Evaluate the performance of AFC and AS for shape and vibration control

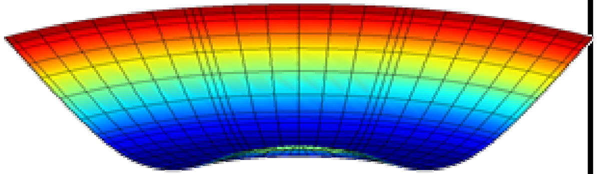
Finite Element Model of Large Circular Plate

- **3-D finite element model of system**
  - 20 node brick elements
  - Symmetric boundary conditions
- **System Properties**
  - Mirror - ULE glass
  - Diameter 2.54 m (100 in)
  - Surface mass < 10 kg/m^2 (50 kg)
- **144 actuators**
  - 72 radial actuators
  - 72 circumferential actuators



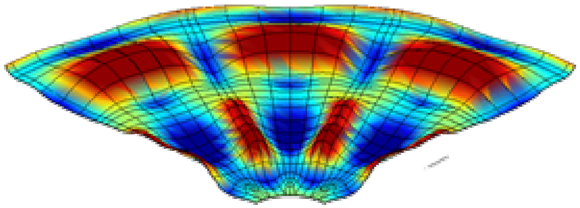
Surface Error for Primary Spherical

- The RMS surface error percent reductions for DA, AFC, and AS are 97.5%, 98.8%, and 99.2%, respectively
- AFC and AS both reduce wrinkling effect → Maximum local errors for AS and AFC are much smaller than DA



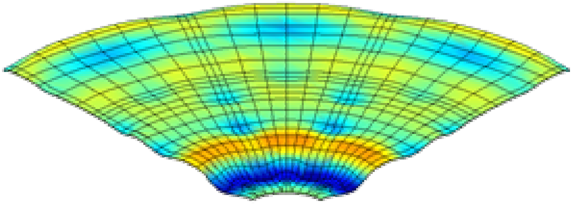
Max Error = 10 waves (one vacuum wavelength of a HeNe laser = 633 nm)

DA



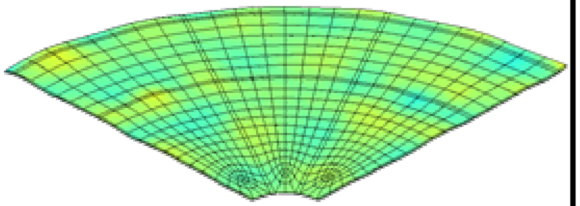
Max Error = 1.99 waves

AFC



Max Error = 0.985 waves

AS



Max Error = 0.456 waves

For all (18) modes considered, AFC and AS both outperform DA

- *AFC outperforms AS in 6 modes*
- *AS outperforms AFC in 12 modes*

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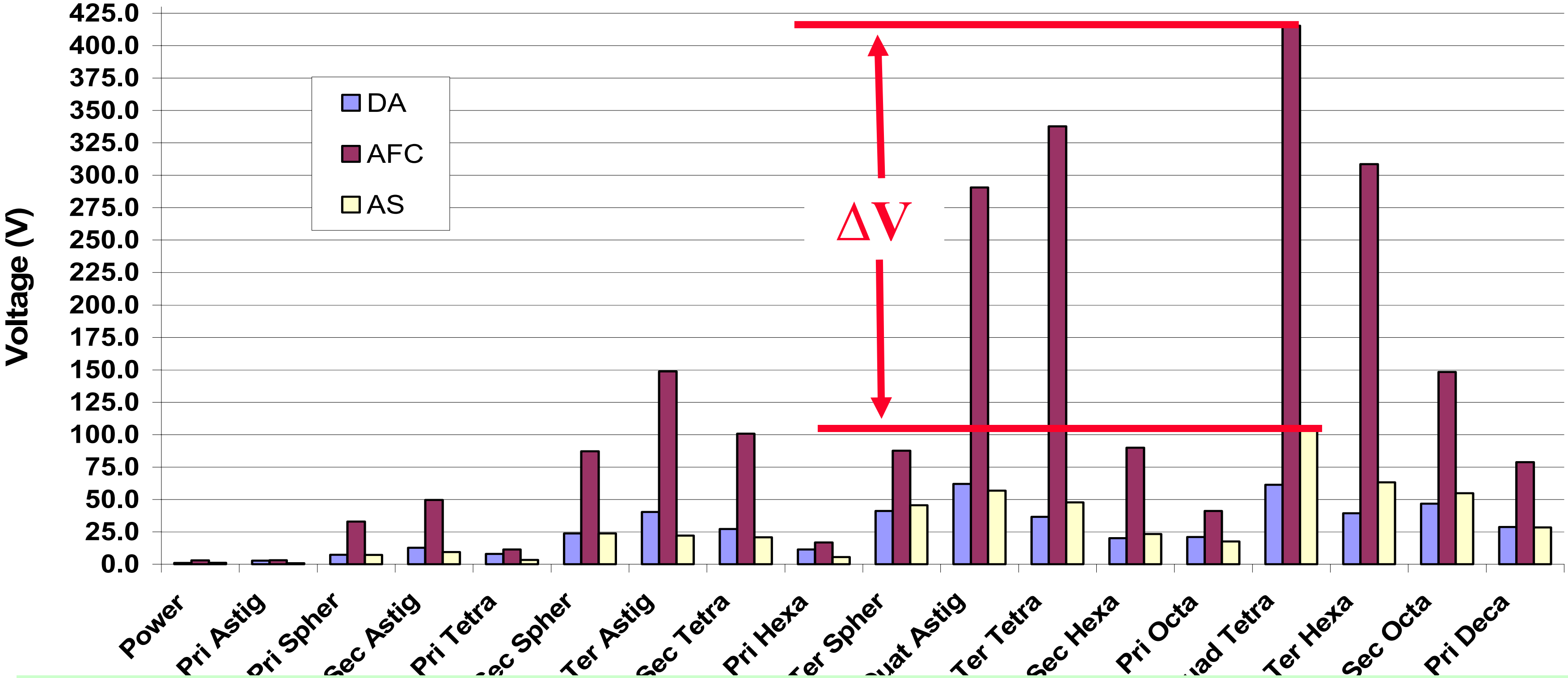
Maximum Voltage

- **Maximum voltage across any actuator when correcting for error (max *10 waves*) in each of the modes**



Voltages for the AFCs are much higher than the DA and AS due to the poling of the actuator and the pitch of the electrodes – could limit the use of the actuator for certain applications

Vibration Control

- Due to low damping of large flexible structures in space, *controller* and *observer* spillover effects present problems in active vibration controls
- It is observed that in the shape control analysis, the AFC and AS are less likely to ‘excite’ higher order modes (analogous to the controller spillover phenomenon in vibration control)
- ***Question:*** From the above observation, can the AFC and AS reduce the controller spillover effect and how do they compare with one another?

Controller/Observer Spillover

- Controller spillover – Spillover of control energy into uncontrolled (residual) modes, x_r → excite overall response
- Observation spillover – Spillover of the residual modes (x_r) into the estimation of the controlled modes
- Combining controller and observer spillovers → shift system eigenvalues → could induce instability

Observer Spillover

Idea: ***AS or AFC to reduce B_r effect → reduce control spillover***

$$\begin{Bmatrix} \dot{x}_c \\ \dot{e}_c \\ \dot{x}_r \end{Bmatrix} = \begin{bmatrix} A_c - B_c K_c & -B_c K_c & 0 \\ 0 & A_c - L_c C_c & L_c C_r \\ -B_r K_c & -B_r K_c & A_r \end{bmatrix} \begin{Bmatrix} x_c \\ e_c \\ x_r \end{Bmatrix}$$

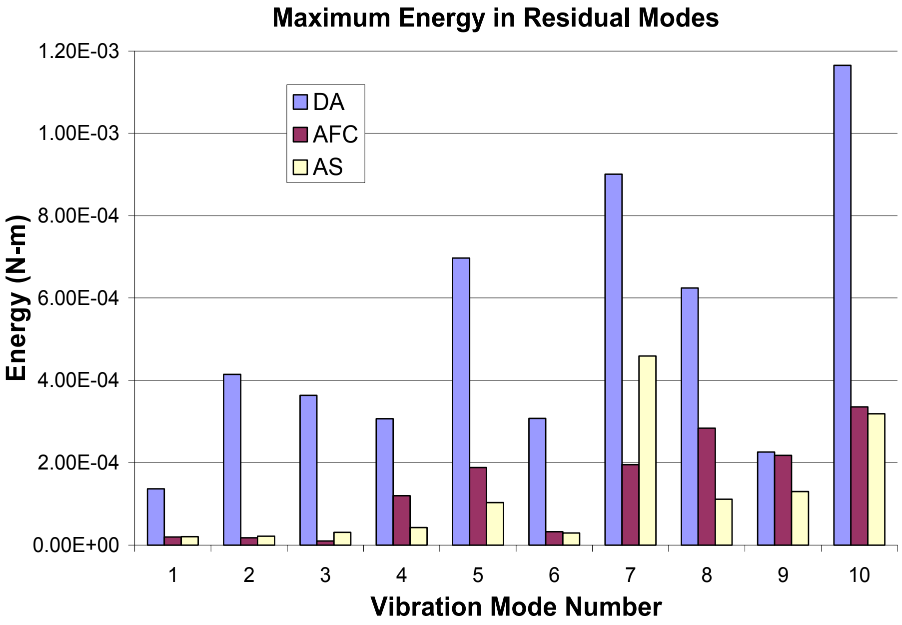
x_c : Controlled modes
x_r : Residual modes
e_c : Estimator error

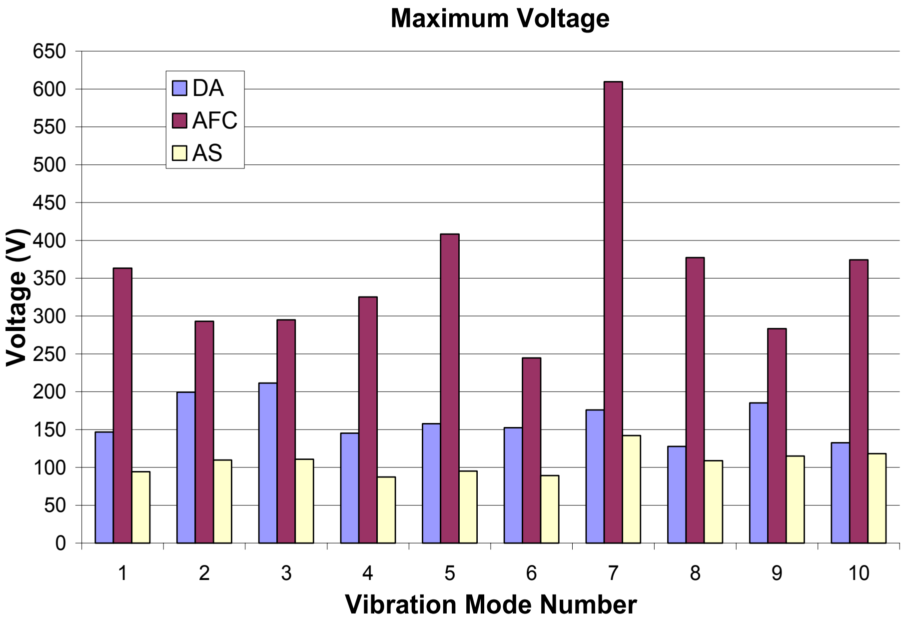
K_c : Controller gain
L_c : Observer gain
A_c : Controlled system

B_c : Controlled forcing input
B_r : Residual forcing input
A_r : Residual system

Controller Spillover – Maximum Energy in Residual Modes

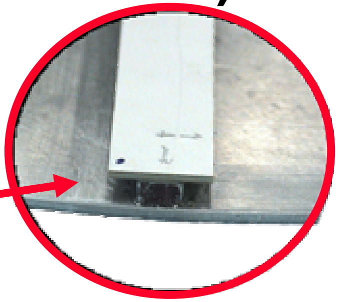
- ***AS and AFC can significantly reduce the vibration spillover energy shown in systems with DA treatments***
- ***Energy in the residual modes for AS is less than AFC for 6 of the 10 modes, and the maximum voltage is significantly less than AFC for all 10 modes***



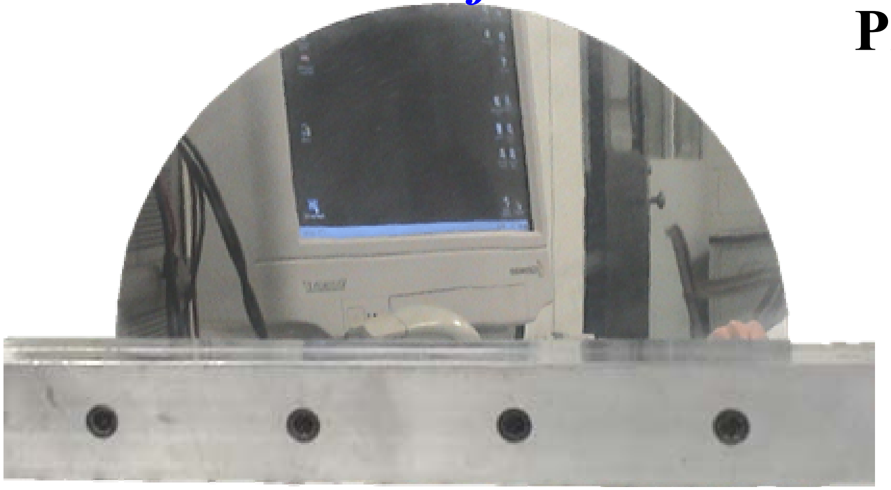


Experiment

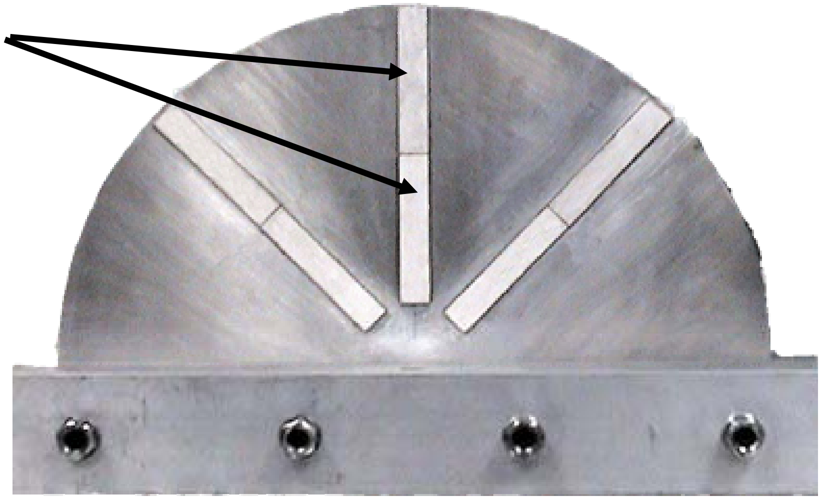
- **12” diameter 1/16” cantilevered aluminum circular plate with reflective surface**
- **6 piezoelectric sheet actuators (2.5” x 0.5” x 0.04”)**
- **Two systems**
  - Directly attached actuators
  - 5” x 0.2” x 1/8” (L,W,H) Al stiffeners inserted between plate and PZTs



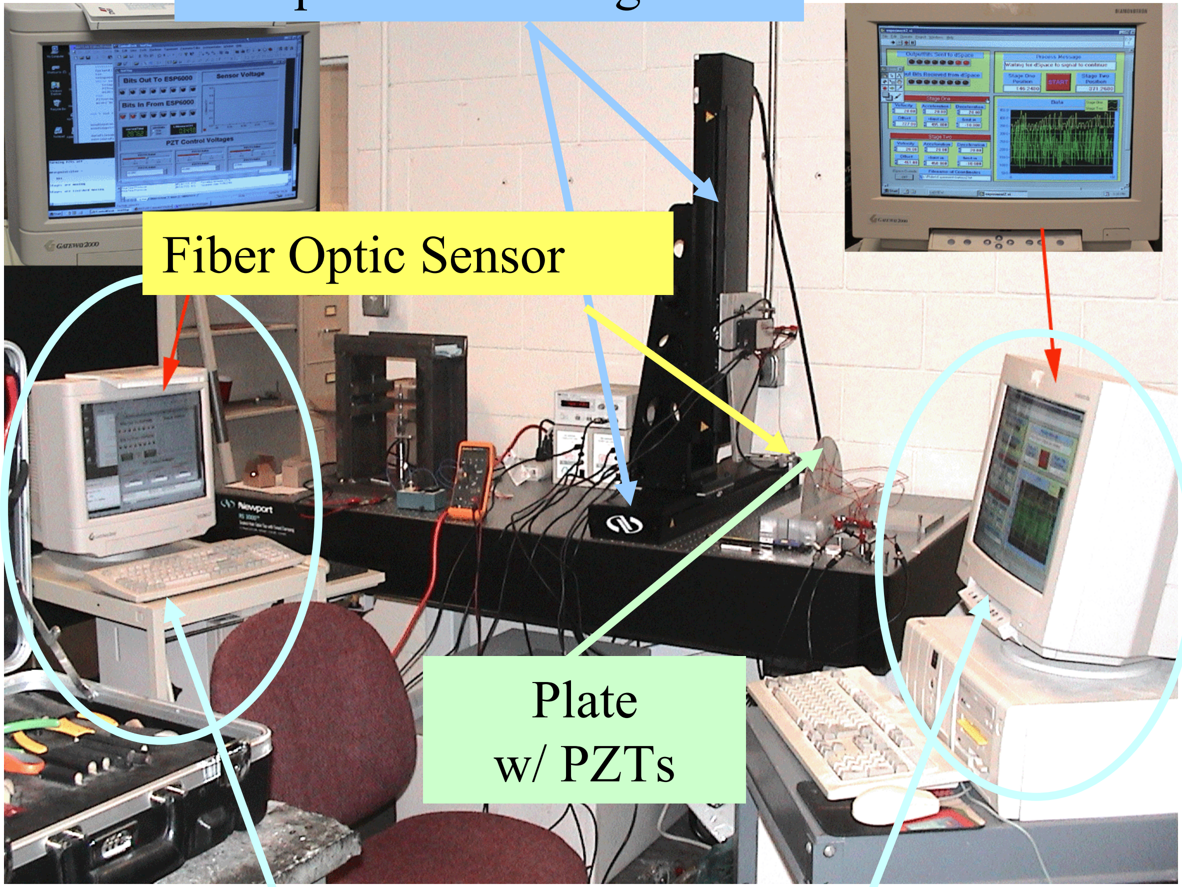
Front of Plate



Back of Plate



Shape Control Experimental Setup



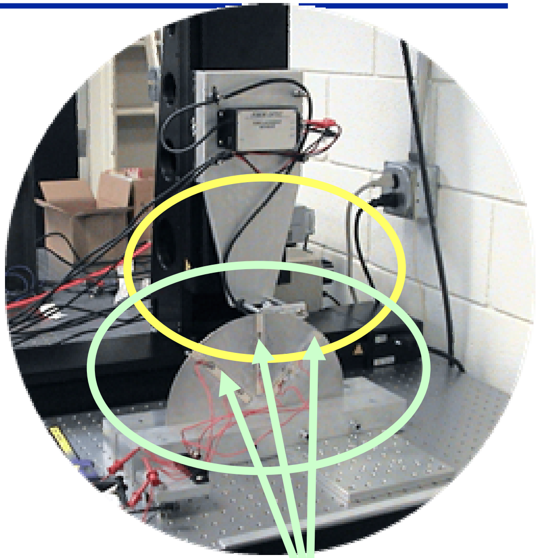
Newport Linear Stages

Fiber Optic Sensor

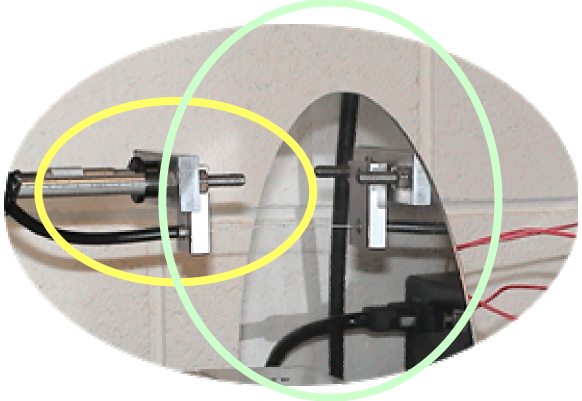
Plate
w/ PZTs

dSpace System
- Records displacement
- Actuates the PZTs

Labview w/ Controller Card
- Positions sensor



PZTs



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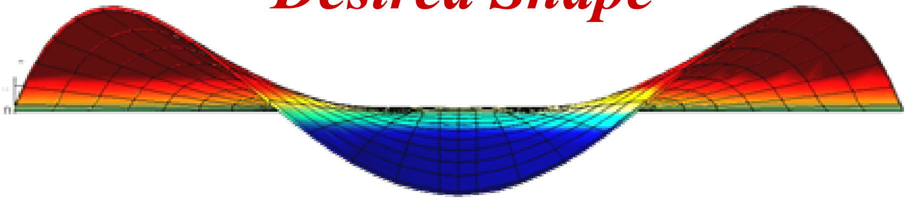
Shape Control Experimental Results

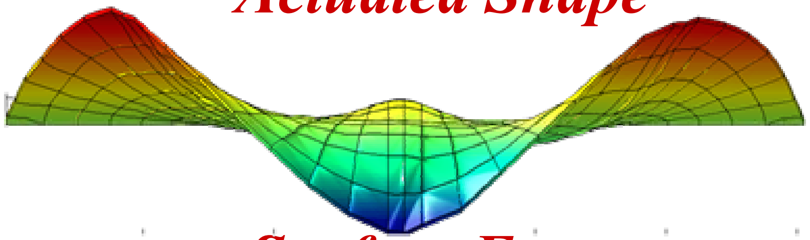
Desired Shape



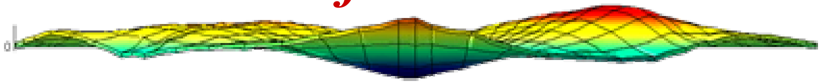
Max Disp = 10 μm

Direct Attached

Actuated Shape



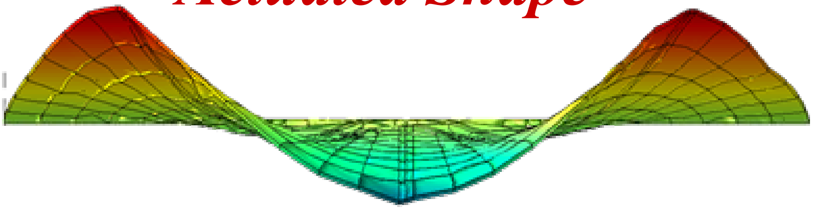
Surface Error



Max Error = 3.6 μm

Active Stiffener

Actuated Shape



Surface Error

Max Error = 2.5 μm

Active stiffener reduces wrinkling effect, and therefore outperforms directly attached actuator for shown shape

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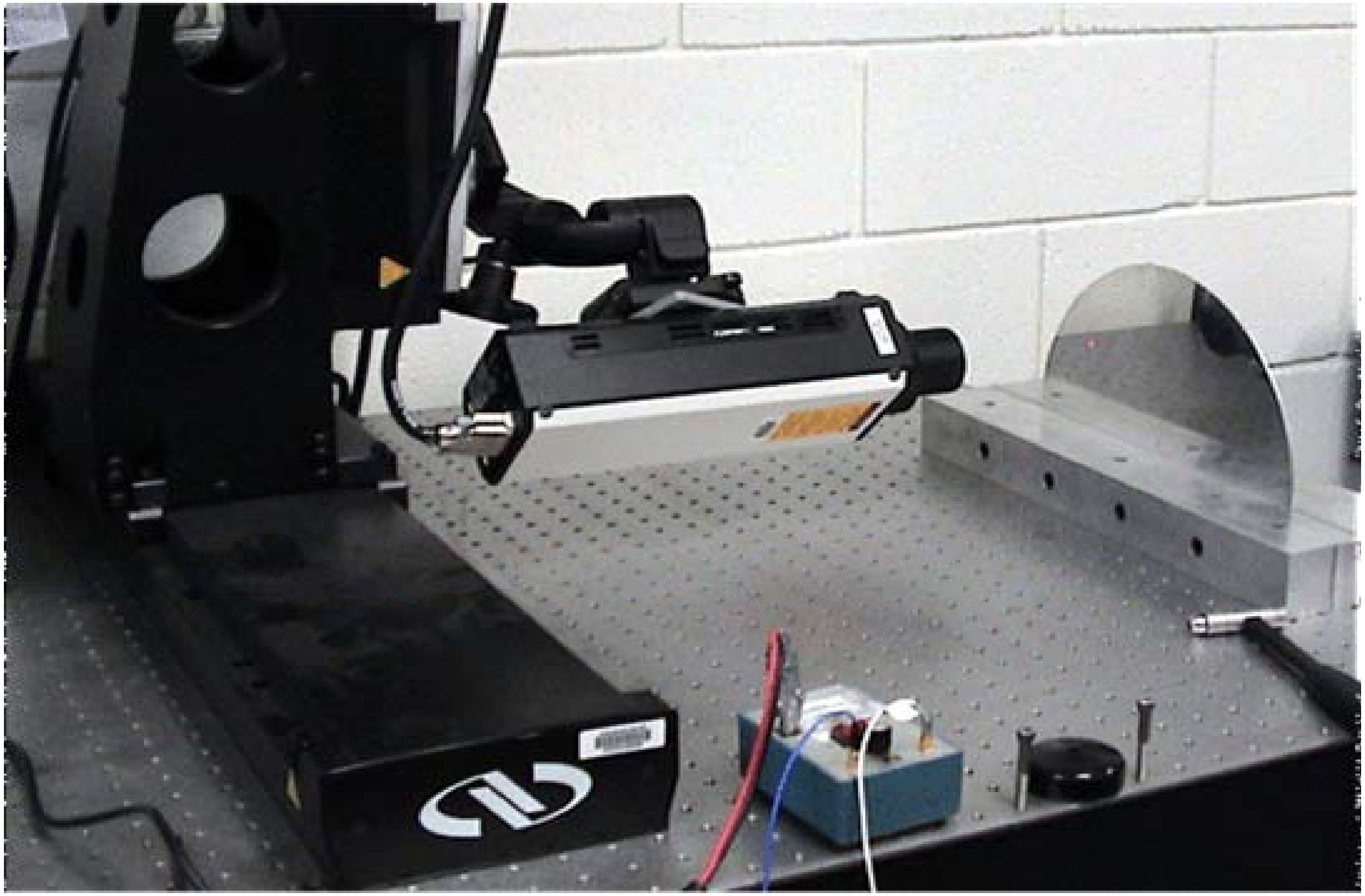
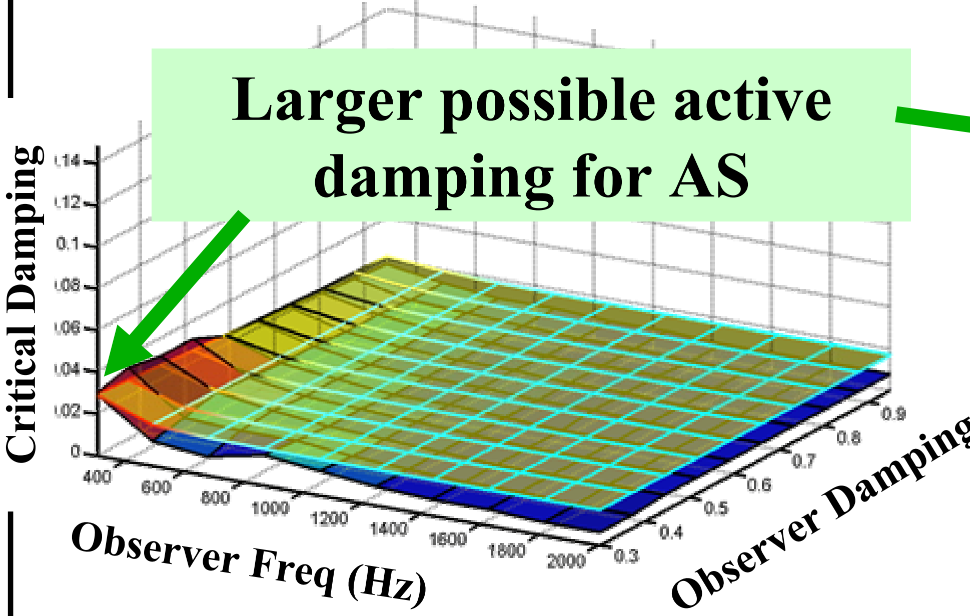
Vibration Control Experiment



Vibration Control Stability Region

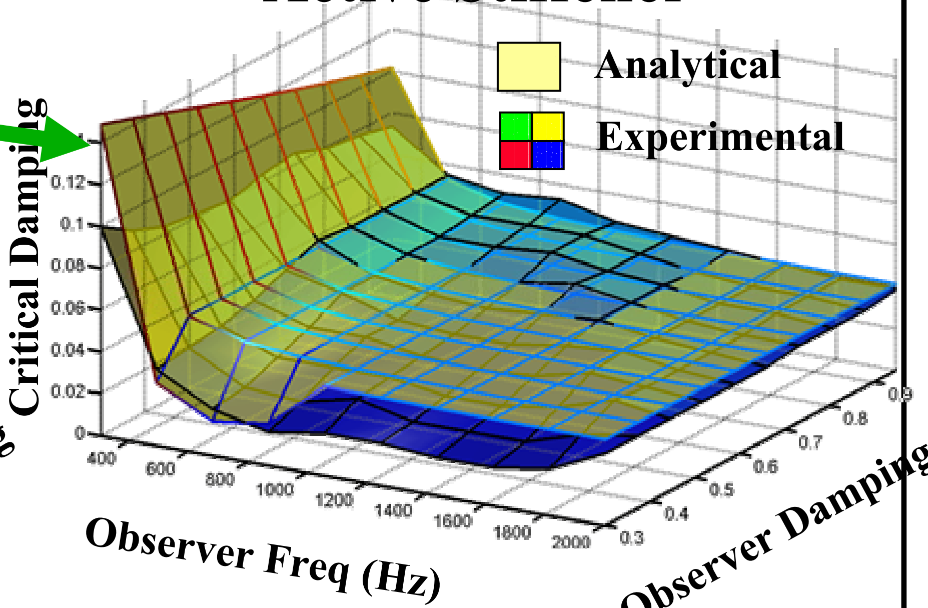
- Response measured for each observer pole placement (ω_n, ζ), the designed active damping is increased until the system goes unstable
- Close agreement between analytical and experimental results for both DA and AS (model validation)

AS has greater stability region than DA



Direct Attached

Larger possible active damping for AS



Active Stiffener

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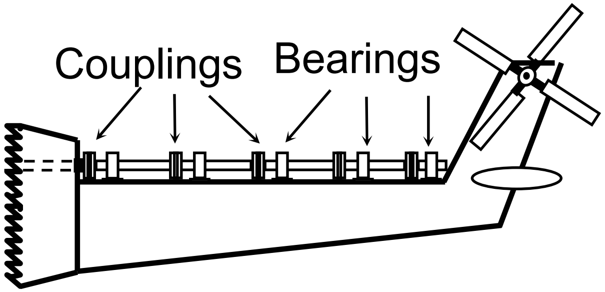
Today's Highlights

- Directional Decoupling Piezoelectric Actuators for Shape and Vibration Control of Plate Structures
- Robust Adaptive Vibration and Stability Control of Flexible Driveshafts via Magnetic Bearings
- Carbon Nanotube-based Damping Composites

Issues of Current Helicopter Driveline Systems

- **Current Drivelines**
  - Segmented Shafting with Significant # of Flex Couplings/Bearings for Misalignment Compensation



  - Passive Dampers needed for Supercritical Speed Shafts
- **High Maintenance and Cost**
  - Component (Bearings, Couplings, Dampers) Wear
  - Shaft Balancing and Alignment
  - Strict Shaft Eccentricity Tolerances
- **Passive Vibration Reduction Difficult**
  - Not Effective for Isolation (Force Transmissibility)
  - Not Effective for Off-Resonance Reduction
  - Cannot Compensate for System Variations

Program Goal and Ideas

To address the issues with current systems and overcome the technical barriers for achieving a simple, high performance, low vibration, low cost, and low maintenance driveline of rotary-wing aircraft

- **Reduce number of mechanical contact components**
- **Reduce maintenance need**
- **Suppress vibration and ensure stability**

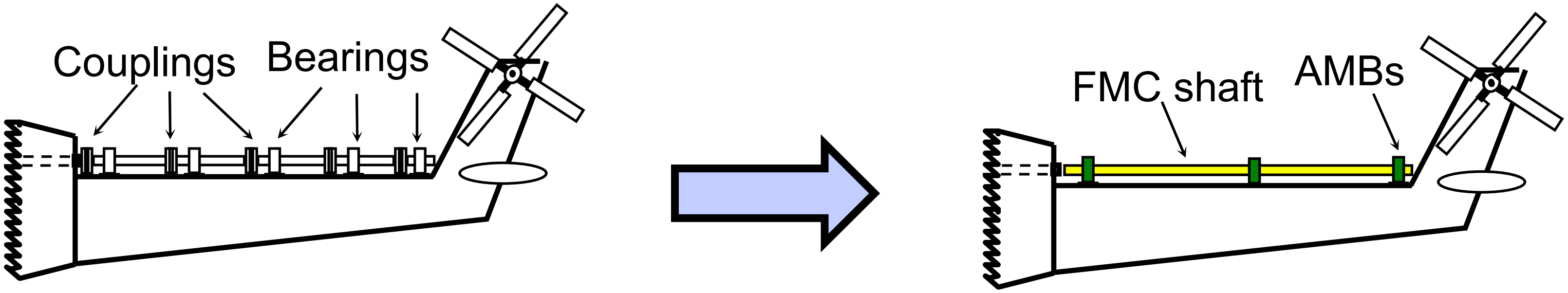
Develop and utilize newly emerging materials and active control technologies -- a combination of

- ***Flexible matrix composite (FMC)*** materials and
- ***Active magnetic bearings (AMB)***

Ideas

Flexible matrix composite (FMC) materials with tailored ply orientations for shafting

- **Soft in flexure and stiff in torsion – to accommodate for large misalignment and effectively transmit power**
- **Without multi-segment shafting and large # of bearings/couplings -- reduce cost and maintenance need**



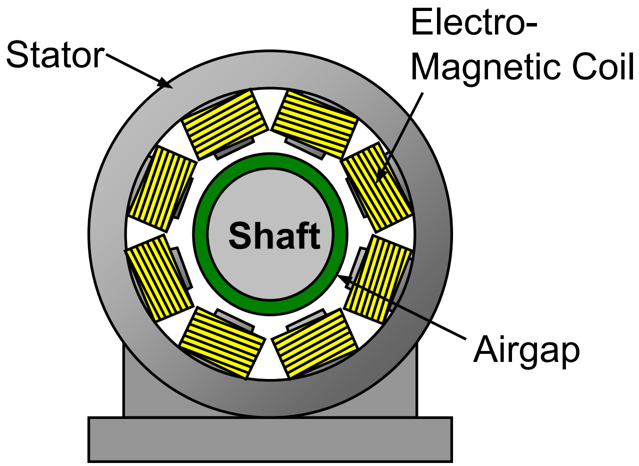
Current **New**

Ideas (cont.)

Active magnetic bearings for vibration an tability control

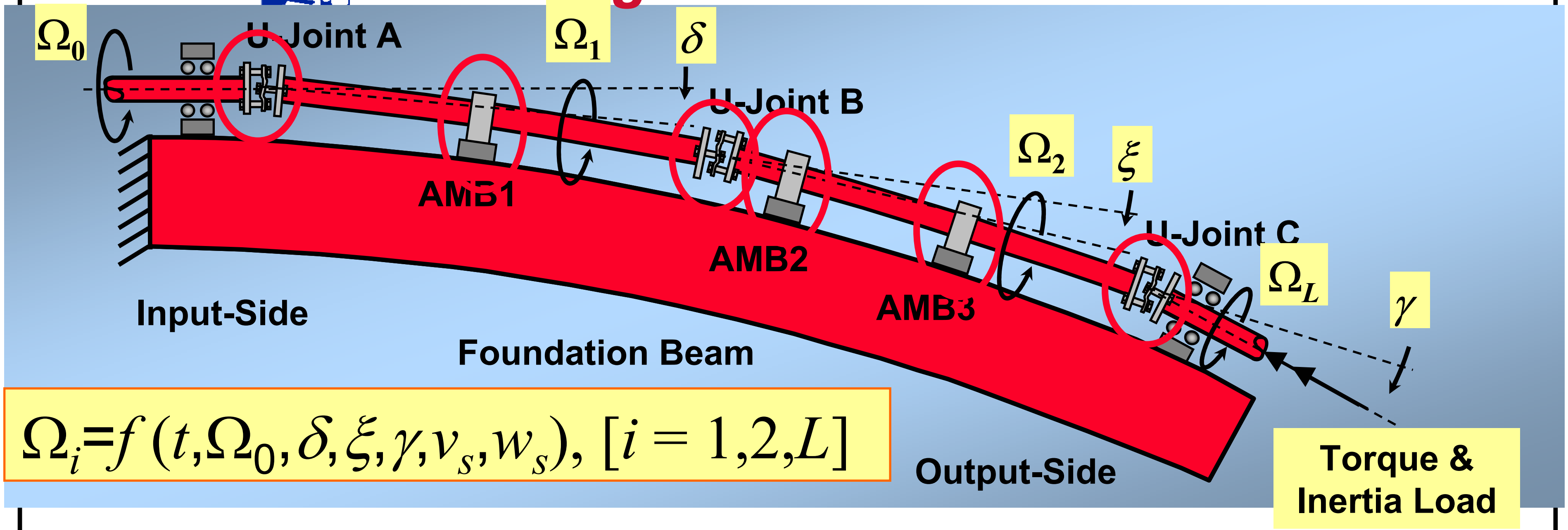
- ➢While highly flexib comp haft systems have many advantages, their avior could be issues that nee lizing the idea
- ➢Feasibility trol of a tailrotor-drivetrain struc c bearings (AMB) → by proper controller e AMB actuator could be a good candidate for helic pter driveline control (size, weight, power)

Today's highlight – AMB control of current segmented driveline shafting



- » **Non-contact -- no frictional wear and permits high operating RPM**
- » **Large frequency range -- ideal for active vibration control in rotorcraft setting**
- » **Light backup roller bearings (only contact with active failure) for fail-safe purpose**

Multi-Segment Flexible Driveline Model



$$\Omega_i = f\,(t, \Omega_0, \delta, \xi, \gamma, v_s, w_s),\ [i = 1,2,L]$$

Model Features

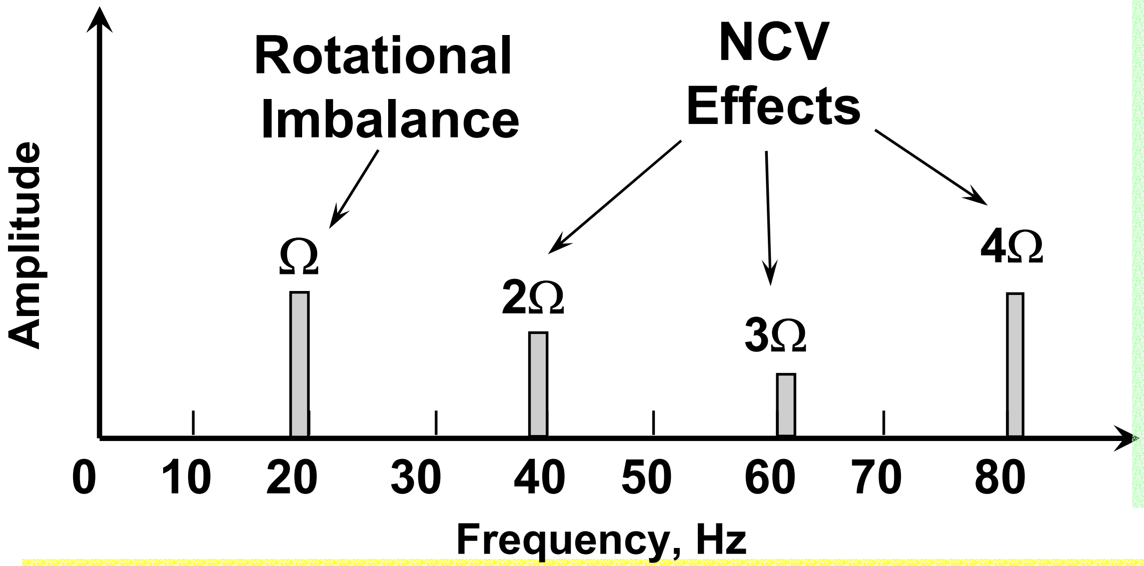
- Two Flexible Shaft Segments, (Bending and Torsion Flexibility)
- Segments Connected By NCV Couplings (U-joints at A, B & C)
- Bearings & Dampers (Conventional Config. or Active Magnetic Bearings)
- Driveline Mounted on Flexible Foundation Beam

Operating Conditions

- Nominal Misalignments δ, ξ and γ due to Static Foundation Deflections
- Output-Shaft Drives Torsional Inertia & Load Torque
- Input-Shaft Driven at Constant Supercritical Speed, Ω_0

Vibration and Instability Sources

Typical Shaft Vibration Spectrum



Research Challenge

- Control of vibration and stability for both
  - **Rotating unbalance**
  - **NCV effects (misalignment and load torque)**

- **Excitation Sources (Multi-Frequency Harmonic)**
  - Imbalance excites synchronous vibration at Shaft Speed, Ω
  - Misalignment and Torque excite super-synchronous vibration at Frequencies *N*Ω (*NCV Effects*)
- **Instability Sources**
  - Rotating-Frame Damping (Whirl Instability)
  - NCV Effects (Parametric Instability)

Rarely addressed in previous studies

Equations of Motion

Driveline Equations of Motion

$$\left[\mathrm{M}+\mathrm{M}_{\mathrm{NCV}}(\boldsymbol{t})\right]\ddot{\eta}+\left[\mathrm{C}_{\mathrm{sd}}+\mathrm{G}+\mathrm{C}_{\mathrm{NCV}}(\boldsymbol{t})\right]\dot{\eta}+\left[\mathrm{K}+\mathrm{K}_{\mathrm{rd}}+\mathrm{K}_{\mathrm{NCV}}(\boldsymbol{t})\right]\eta=\mathrm{F}_{\mathrm{Imb}}(\boldsymbol{t})+\mathrm{F}_{\mathrm{NCV}}(\boldsymbol{t})$$

Linear Periodically Time Varying System

- Nominal LTIV Rotordynamic system (gyroscopic, rotating-frame damping and shaft Imbalance)
- Misalignment and Load-Torque Gives Rise to Periodic Parametric and Forcing terms (NCV Effects)
- NCV Terms have variation frequency = $2\Omega_0$ and $4\Omega_0$

Equations of Motion

Recast System Equations Into First-Order State-Space Form

AMB-Driveline State-Space System

$$[\boldsymbol{I} + \Delta \boldsymbol{E}(\boldsymbol{t})]\dot{\boldsymbol{x}} = \left[\boldsymbol{A}_n + \Delta \boldsymbol{A}(\boldsymbol{t})\right]\boldsymbol{x} + \boldsymbol{B}_d \boldsymbol{d}(\boldsymbol{t}) + \boldsymbol{B}_u \boldsymbol{u}(\boldsymbol{t})$$

$$\boldsymbol{y} = \boldsymbol{C}_y \boldsymbol{x}$$

LPTV Descriptor System

Function of misalignment, load torque, shaft speed

Function of misalignment, load torque, shaft speed, and unbalance

System Matrices

A_n Nominal System

$\Delta A(t)$, $\Delta E(t)$ Periodic Matrices (NCV Effects)

Multi-Harmonic Disturbance Input

$B_d d(t)$ Imbalance & NCV Forcing Terms

With $d(t) = \mathbf{d}_0 + \sum_{n=1}^{4}\left[\mathbf{d}_{sn}\sin(n\Omega_0 t) + \mathbf{d}_{cn}\cos(n\Omega_0 t)\right]$

AMB Inputs/Outputs

$u(t)$, AMB Current Inputs

$y(t)$, AMB Displacement Sensor Outputs

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Harmonic Fourier Coefficient Representation

Driveline System Matrices

$$E(t) = I + \sum_{n=1}^{4}\left[E_{sn}\sin(n\Omega_0 t) + E_{cn}\cos(n\Omega_0 t)\right]$$

$$A(t) = A_n + \sum_{n=1}^{4}\left[A_{sn}\sin(n\Omega_0 t) + A_{cn}\cos(n\Omega_0 t)\right]$$

Disturbance Input

$$d(t) = \mathbf{d}_0 + \sum_{n=1}^{4}\left[\mathbf{d}_{sn}\sin(n\Omega_0 t) + \mathbf{d}_{cn}\cos(n\Omega_0 t)\right]$$

Control Input

$$u(t) = \mathbf{u}_0 + \sum_{n=1}^{4}\left[\mathbf{u}_{sn}\sin(n\Omega_0 t) + \mathbf{u}_{cn}\cos(n\Omega_0 t)\right]$$

Response Output

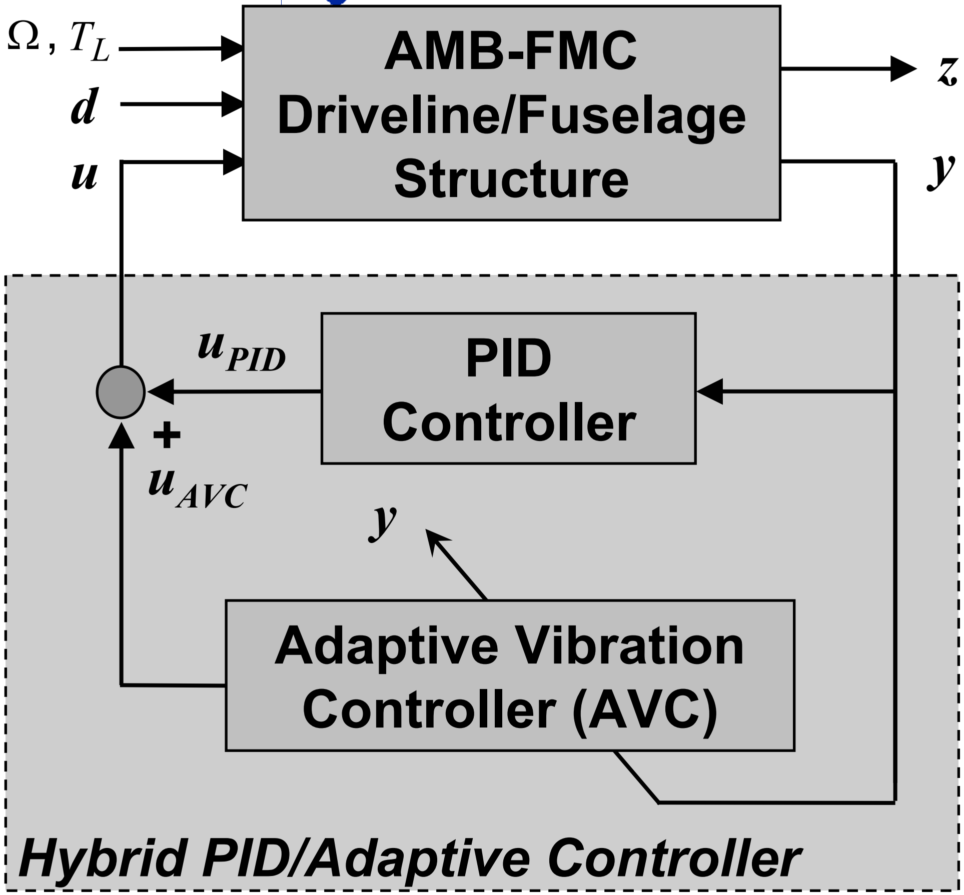
$$y(t) = \mathbf{y}_0 + \sum_{n=1}^{4}\left[\mathbf{y}_{sn}\sin(n\Omega_0 t) + \mathbf{y}_{cn}\cos(n\Omega_0 t)\right]$$

- **Apply Harmonic Balance Method to obtain Multi-Harmonic Transfer function relations analogous to Linear Time-Invariant System**

Steady-State Input/Output Relations for Driveline System

$$Y = T_{yu}U + T_{yd}D$$

Closed-Loop System -- Hybrid PID/Adaptive Control



PID Feedback Input

$$u_{PID} = -k_p y - k_d \dot{y} - k_I \int y \, dt$$

Adaptive Control Input

$$u_{AVC}(t) = \sum_{n=1}^{N_h} [\mathbf{u}_{sn} \sin(n\Omega t) + \mathbf{u}_{cn} \cos(n\Omega t)]$$

PID/Adaptive Control Input

$$u(t) = u_{PID}(t) + u_{AVC}(t)$$

Hybrid Control Strategy

- PID feedback designed for *stability* via Floquet
- Adaptive control designed to *suppress vibration* through minimizing vibration cost function
- Slow adaptation rate -- adaptive control *does not affect stability*

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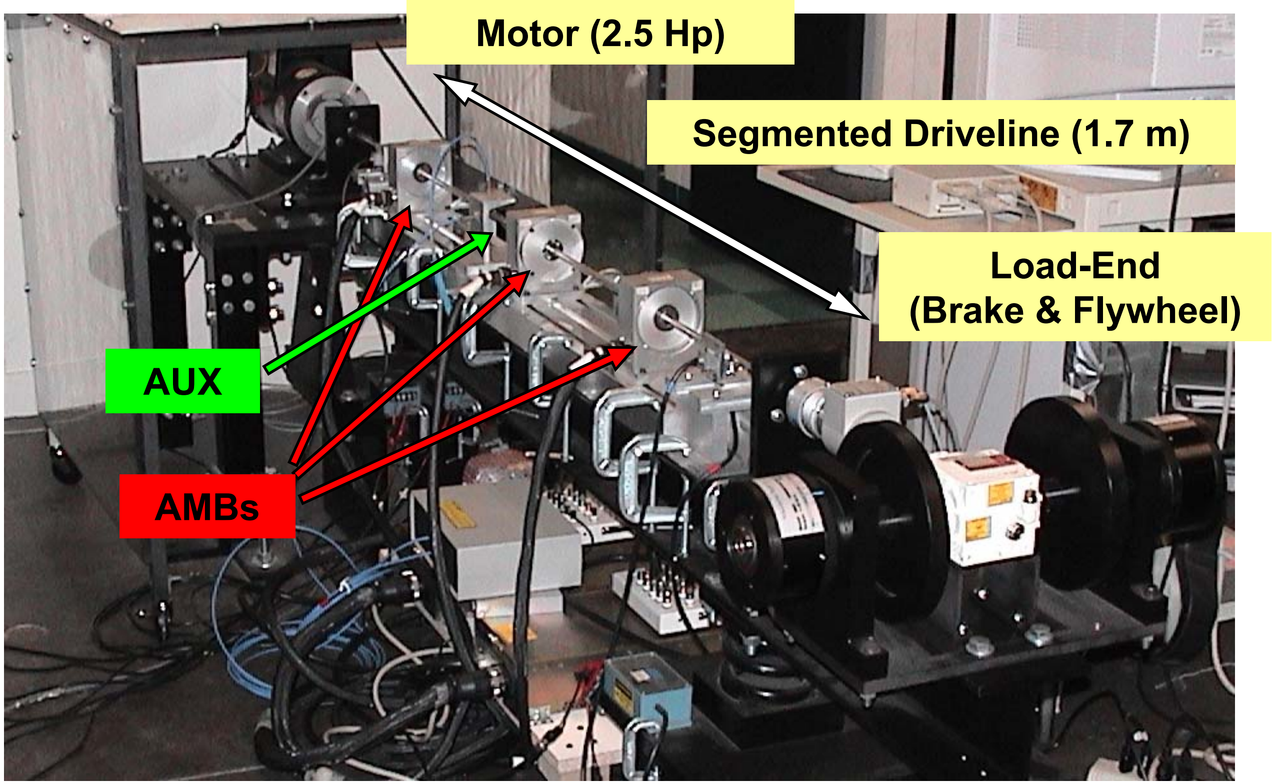
Frequency Scaled Tailrotor-Driveline Testrig

- **Frequency Scaled Model of Helicopter (AH64) Tailrotor Driveline**
  - Shaft supported by AMBs, mounted on flexible foundation
  - Instrumented with accelerometers and displacement probes
  - Adjustable load-inertia, load-torque and driveline misalignment



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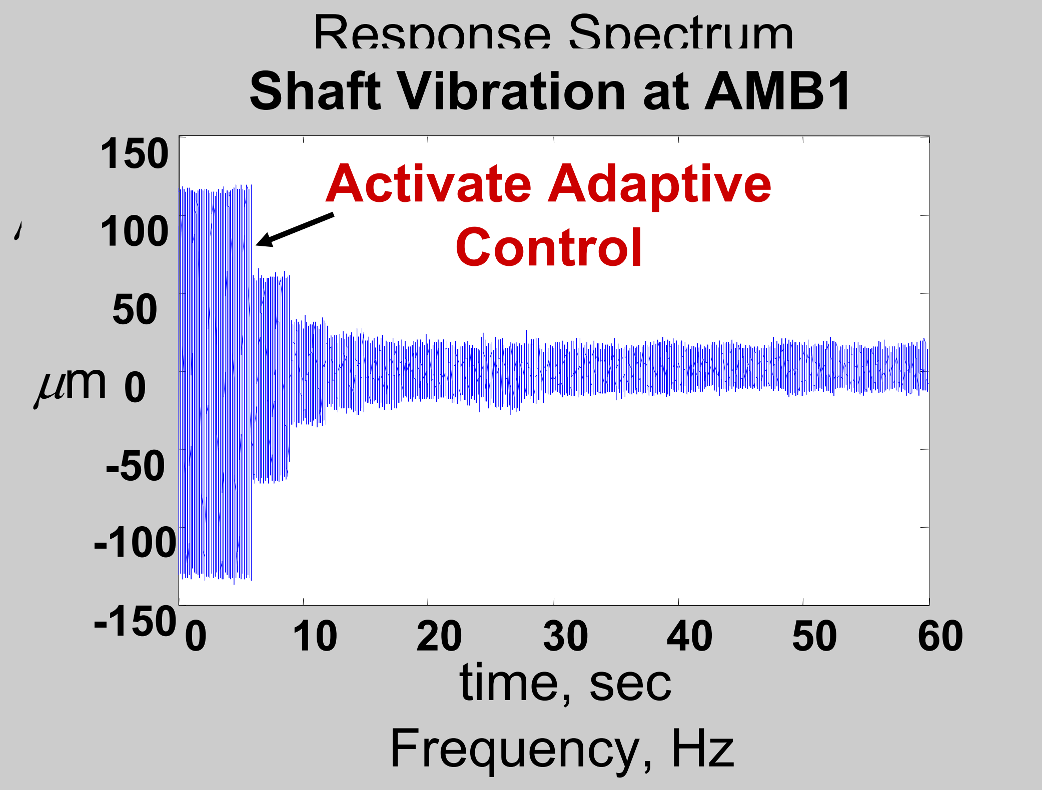
Experimental Results: Closed-Loop AMB Vibration Control

- **Control design based on zero shaft speed, misalignment, and load**
- **No knowledge of Imbalance**

Operating Conditions:

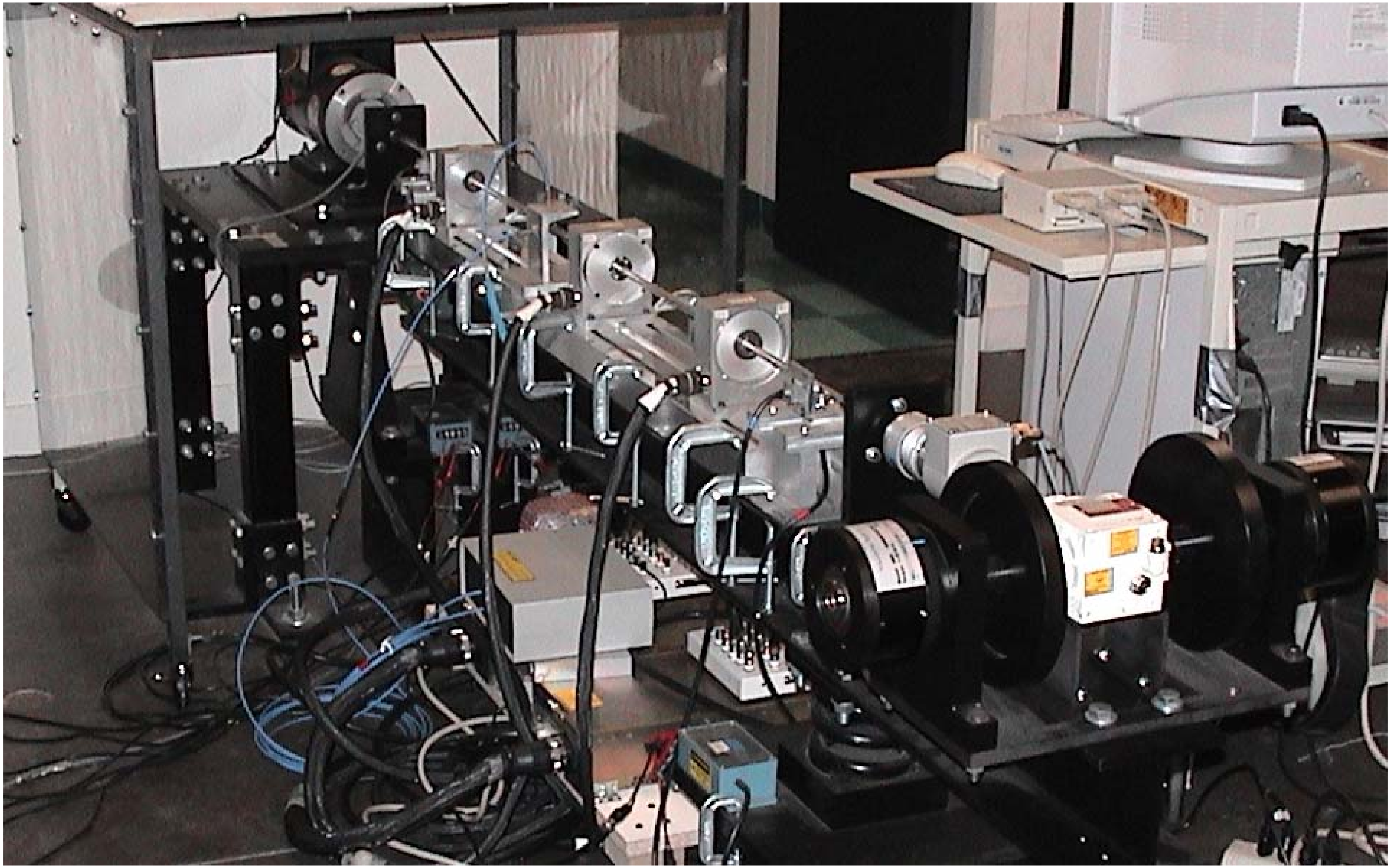
$[\Omega = 1350 \text{ RPM}, \quad \delta = 3.0°, \quad T_L = 5.1 \text{ Nm}]$

With rotor imbalance



- Control achieves *significant vibration suppression*
- Vibration suppressed with *No knowledge* of operating conditions or disturbance
- Tests conducted at multiple operating conditions with similar results -- *Robust Performance*

AMB Control Experiment



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Today's Highlights

- **Directional Decoupling Piezoelectric Actuators for Shape and Vibration Control of Plate Structures**
- **Robust Adaptive Vibration and Stability Control of Flexible Driveshafts via Magnetic Bearings**
- **Carbon Nanotube-Based Damping Composites**

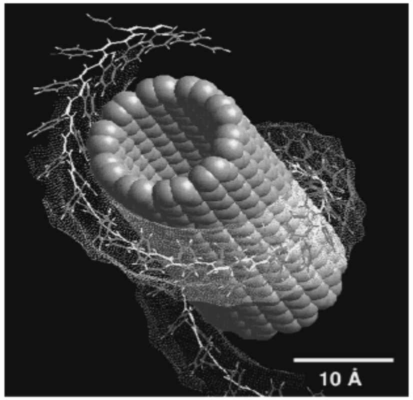
Nanotube Enhanced Damping

Nanotube Features

- Low density high-modulus/strength fibers
  - 1 TPa in Young's modulus (diamond)
  - 100 GPa in tensile strength (10-100 times higher than steel)
- Large aspect ratio
- Large surface areas

→ Increase interaction between the different materials and enhance damping

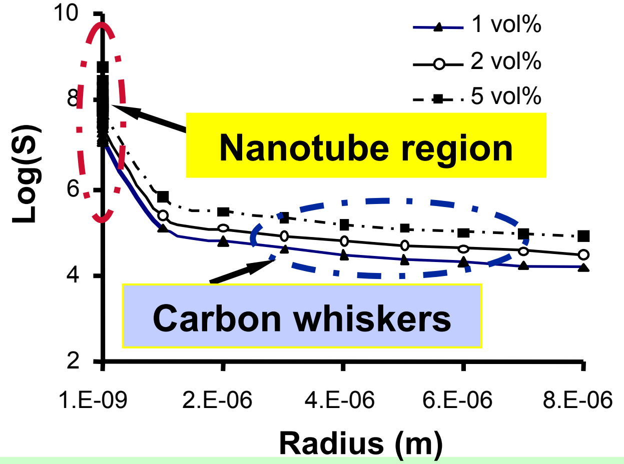
Idea -- Develop ***high damping*** composites via nanotube distributions utilizing the large interfacial interaction force



Tasks

- Nanotube dispersion and composite fabrication
- Damping model
- Damping characterization

Modeling Damping Mechanism In Nano-Composites

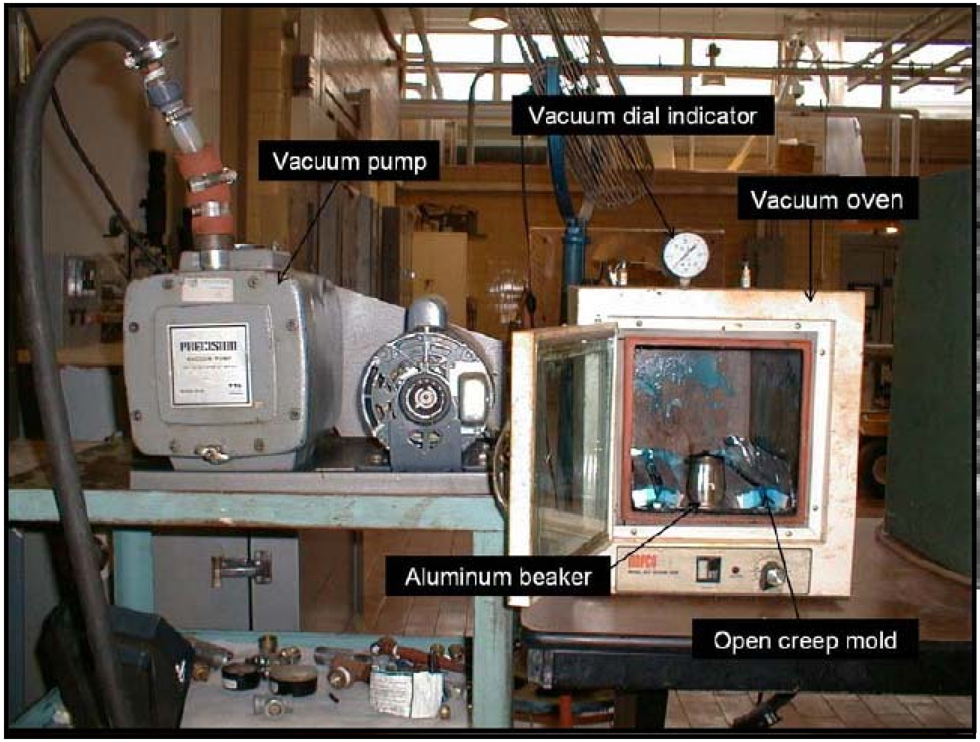


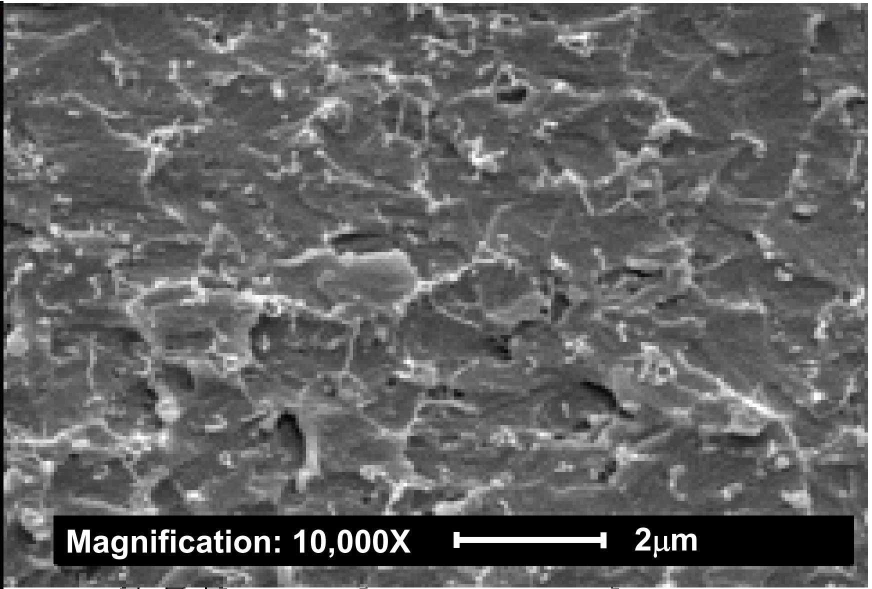
Possible Damping Mechanisms

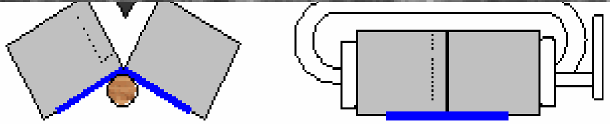
- **Nanotubes - low**
- **Polymer matrix - moderate**
- **Energy dissipated through "Stick- Slip" behavior between polymer surfaces & nano-fillers**
  - Maybe most significant
  - Nano tubes have large interfacial surface areas ($S_{interface}$)
  - Interfacial actions of nano-composites are critical for both load transfers and damping effects

Experiment: Specimen Preparation

- **Resin system:** Epon 9405/Epodil 749/Ancamine 9470
- **SWNT:** “as-prepared” grade from Carbolex®
- **Surfactants:** polyoxyethylene 8 lauryl ether
- **Procedure:** mixing, ultrasonic agitation,and casting/degassing

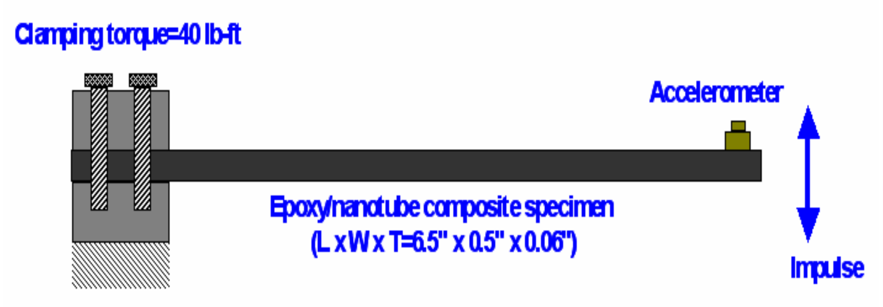


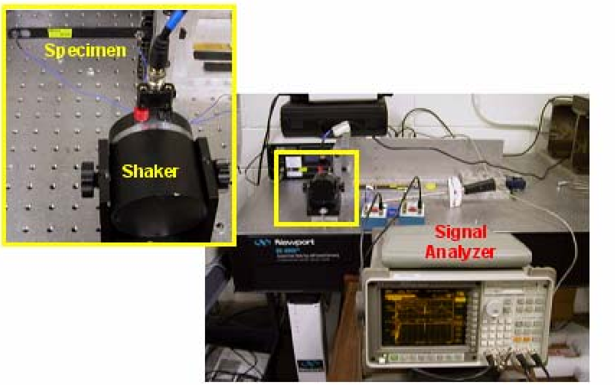




Damping Experiment: Vibration Measurement

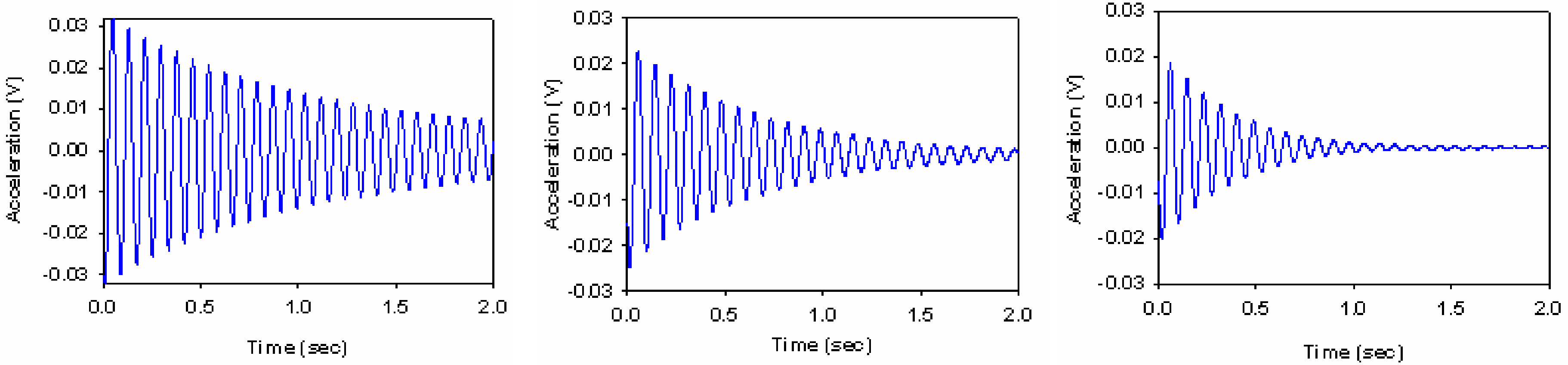
- Cantilever beam specimen





- Logarithmic decrement method to obtain damping ratio

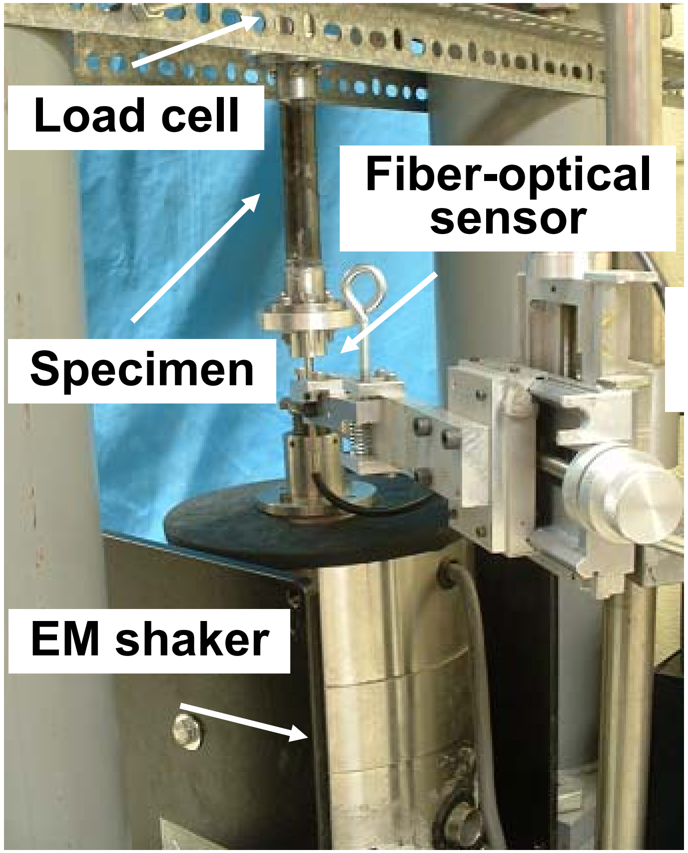
0% wt. specimen **0.5% wt. specimen** **1.0% wt. specimen**

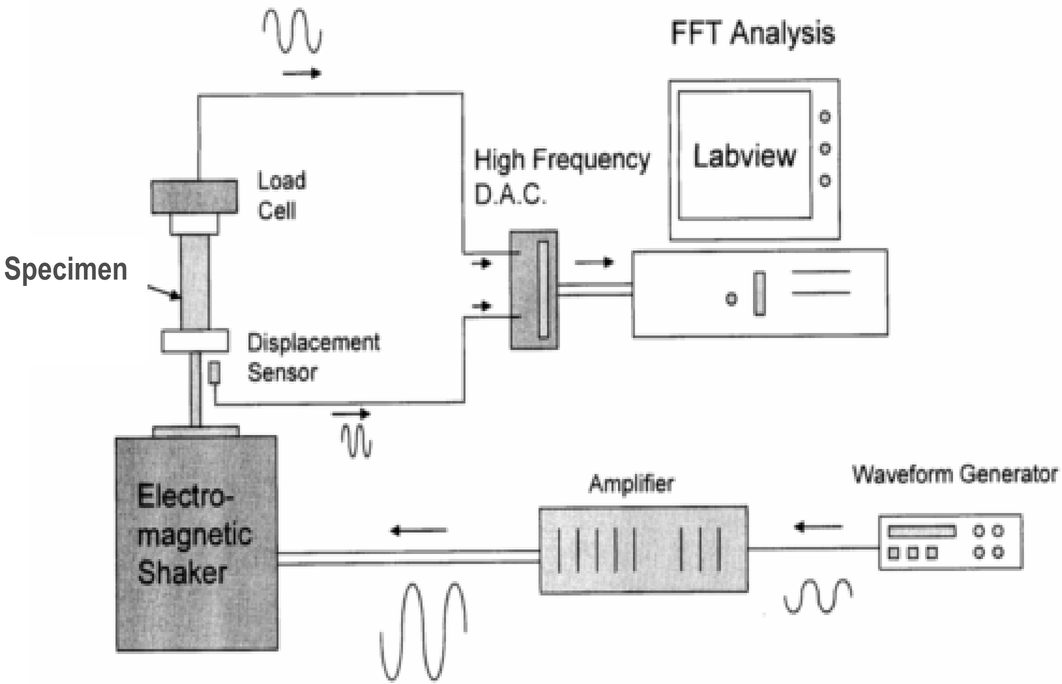


Adding CNTs increases system damping

Damping Experiment: Uniaxial Testing

- Uniaxial testing → direct measurement of stress and strain phase lag
- Loss factor = tan(δ)





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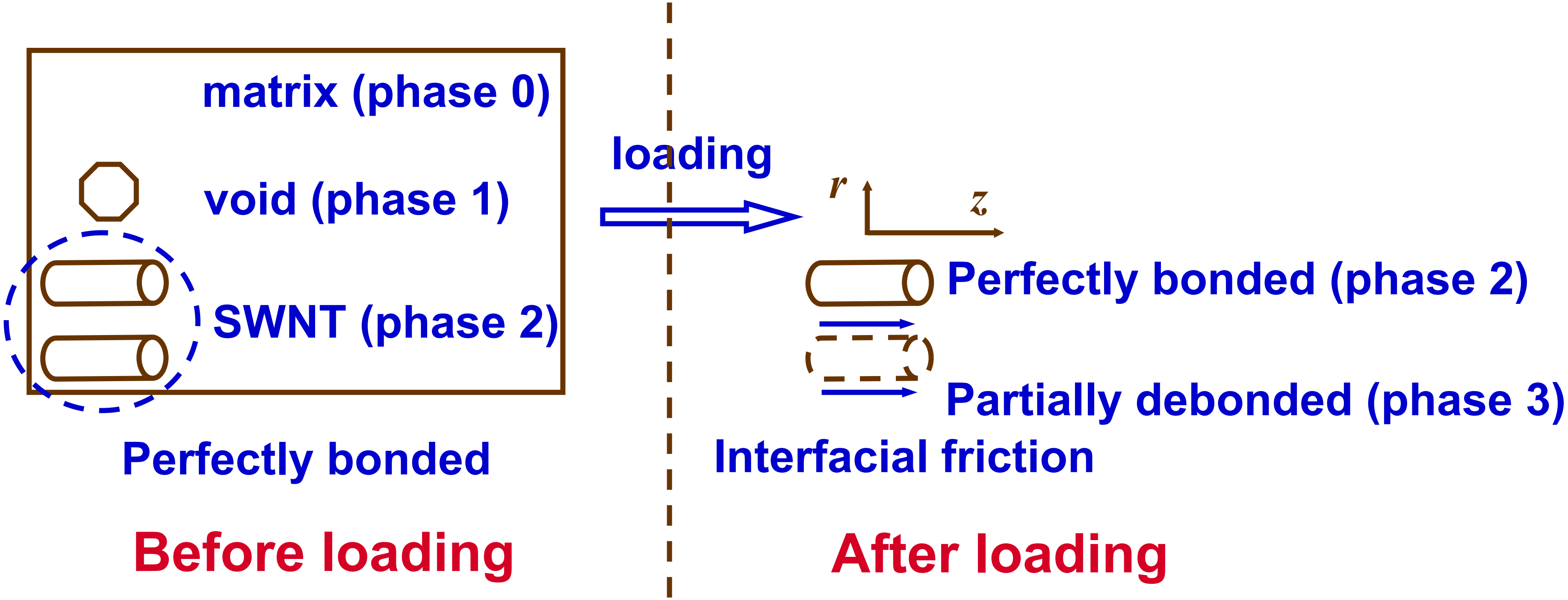
Modeling: Single-Walled Nanotube (SWNT) Debonding

- Evolutionary modeling



matrix (phase 0)

void (phase 1)

SWNT (phase 2)

Perfectly bonded

loading

r z

Perfectly bonded (phase 2)

Partially debonded (phase 3)

Interfacial friction

Before loading

After loading

Modeling: SWNT Debonding

- **SWNT with interfacial debonding**
  - Radial/circum stress vanish
  - Critical surface shear stress (τ^c)

$$\sigma_{db}^{nt} = \left[\underbrace{0 \;\; 0}_{\text{Radial/circum}} \;\; \bar{\sigma}_z^c \;\; \underbrace{\tau^c \;\; \tau^c \;\; \tau^c}_{\text{Critical shear}} \right]$$

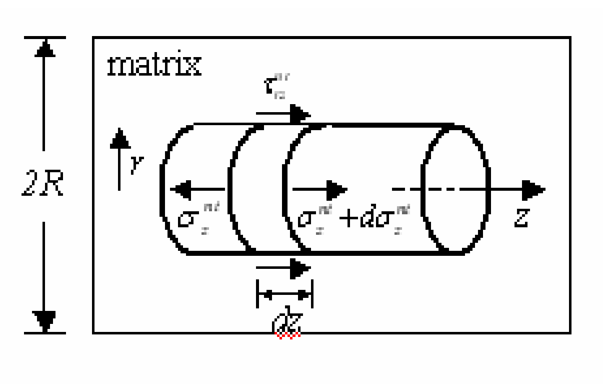
- **Equivalent modeling of debonded SWNTs**
  - Replaced by perfectly bonded SWNTs with degraded stiffness → overcome modeling difficulties

$$\begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \bar{\sigma}_z^c / \varepsilon_z & 0 & 0 & 0 \\ 0 & 0 & 0 & \tau^c / \gamma_{z\theta} & 0 & 0 \\ 0 & 0 & 0 & 0 & \tau^c / \gamma_{rz} & 0 \\ 0 & 0 & 0 & 0 & 0 & \tau^c / \gamma_{r\theta} \end{bmatrix} \longrightarrow$$

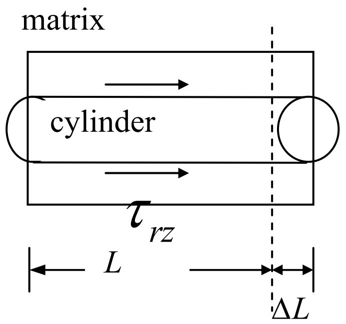
Stiffness depends on material deformation

Modeling: Stick-Slip Analysis and Loss Factor

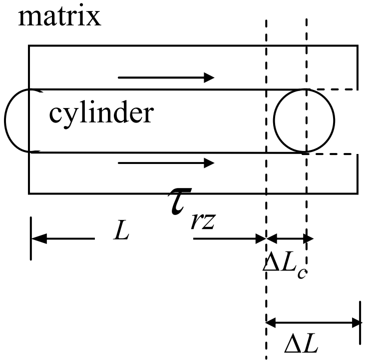
- **Stick-slip frictional behavior with critical shear interfacial strength (well dispersed SWNTs)**



Interfacial shear stress < critical shear strength



Interfacial shear stress > critical shear strength



- **Shear lag analysis (σ_z^{nt} τ_{rz}^{nt})**

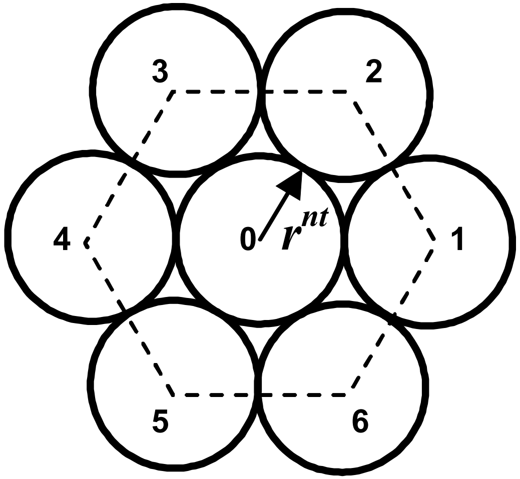
- **Energy dissipation and loss factor**

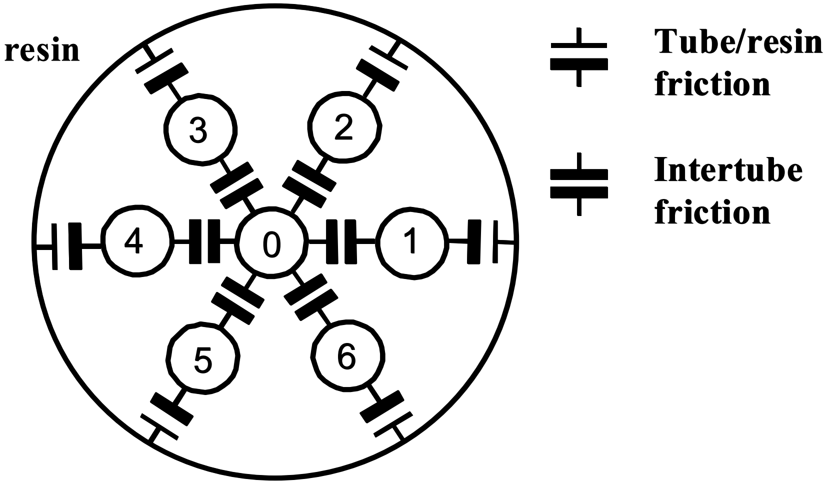
$$\Delta W = \int_S \tau^c \left(u_z^m - u_z^{nt} \right) dS \qquad \eta = \sin^{-1}\left(\frac{1}{2\pi} \frac{\Delta W}{W_{st}} \right)$$

- **Randomization**

Modeling: Load Transfer in CNT Lattice

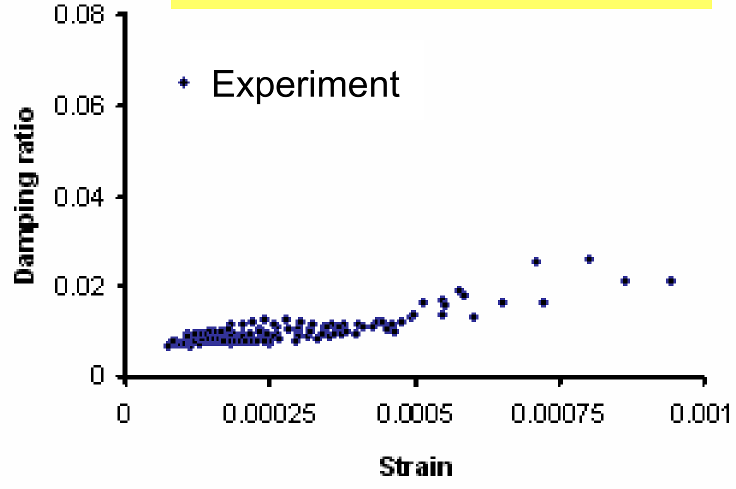
- **Composites containing SWNTs in hexagonal array**
  - Inner SWNT (0); Surrounding SWNTs (1~6)
  - Only longitudinal loading considered
- **Load transfer in CNT rope lattice**
  - Two critical shear stresses: inter-tube τ_{t-t}^{c} and SWNT/resin τ_{t-r}^{c}
  - Inter-tube sliding occurs first, followed by SWNT/resin sliding



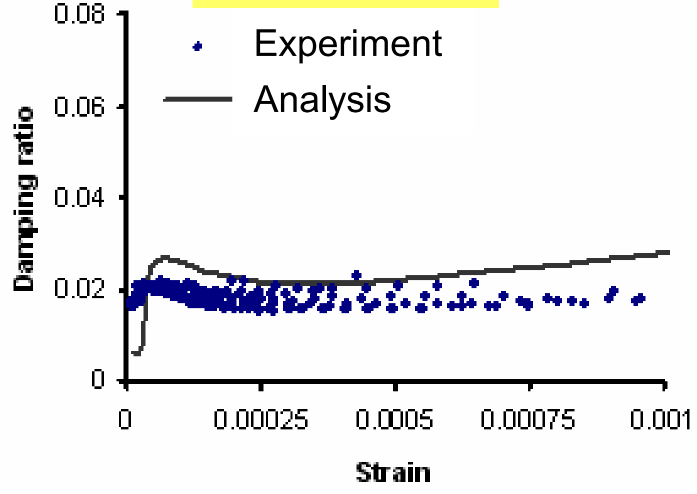


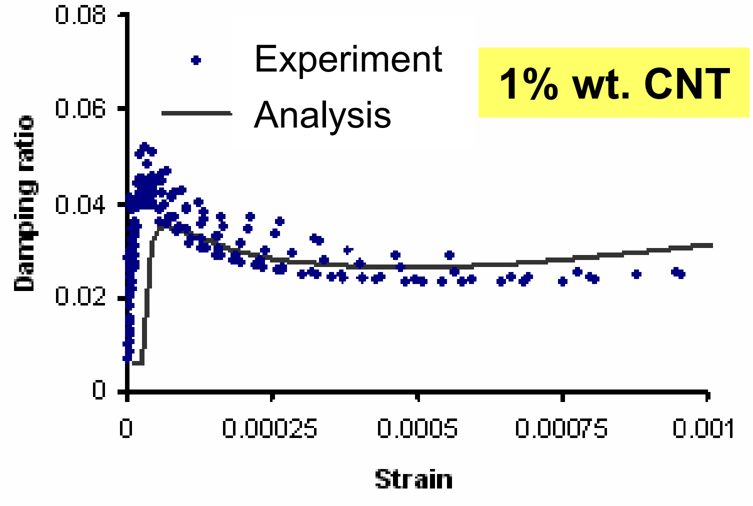
Results and Comparison

Neat specimen (0% CNT)



0.5% wt. CNT





1% wt. CNT

- Damping trend (stick-slip → peak) is well observed in tests and captured by the model
- Adding CNTs could affect damping characteristics significantly
- Increase CNT % → increase peak damping

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Recent Research Projects

Active and Passive Control Program

- Flexible Matrix Composite Driveshaft & Active Bearing Control
- High Performance Carbon Nanotube-Based Damping Composites
- Vibration Isolation via Energy Confinement and Disturbance Rejection
- Active Airframe Vibration Controls
- Stability Augmentation via Semi-Active and Active-Passive Systems
- Concurrent Design of an Active-Passive Hybrid Composite Rotor

Smart Structure Program

- High Precision Shape and Vibration Control
- Enhanced and Hybrid Constrained Layer Damping Treatments
- Piezoelectric Networking for Structural Control Enhancement
    - Damping
    - Disturbance Rejection
    - Delocalization
- MR/ER Fluid Semi-Active Damping Augmentation
- Bio-inspired High Performance Adaptive Structures

System Dynamics Program

- Artificial Neural Network Modeling and Control of Nonlinear Dynamical Systems
- Vehicle Powertrain System Noise and Vibration
- Intelligent Control of Systems with Actuator Delays
- Negotiation Agents for Concurrent Optimization of Dynamical Systems

Tomorrow's Topic

The End

Questions?