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# An Active Damping at Blade Resonances Using Piezoelectric Transducers

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June 2008

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## Abstract

The NASA Glenn Research Center (GRC) is developing an active damping at blade resonances using piezoelectric structure to reduce excessive vibratory stresses that lead to high cycle fatigue (HCF) failures in aircraft engine turbomachinery. Conventional passive damping work was shown first on a nonrotating beam made by Ti-6A1-4V with a pair of identical piezoelectric patches, and then active feedback control law was derived in terms of inductor, resister, and capacitor to control resonant frequency only. Passive electronic circuit components and adaptive feature could be easily programmable into control algorithm. Experimental active damping was demonstrated on two test specimens achieving significant damping on tip displacement and patch location. Also a multimode control technique was shown to control several modes.

**Objective**

To investigate possibility of using an active resonance controller for turbomachinery blade with piezoelectric patches.

**Outline**

I. Introduction
II. Passive shunt damping
III. Active feedback controller design and analysis
IV. Experimental results
V. Summary

## I. Introduction

**Previous activities at GRC**

- Developed new damping technologies to reduce excessive vibratory stresses that lead to high cycle fatigue (HCF) failures in aircraft engine turbomachinery.
- Investigated several damping methods such as viscoelastic damping (O. Mehmed with J. Kosmatka, UC San Diego), passive impact damper, plasma sprayed damping coating, and HTSMA (K. Duffy).

**Current effort at GRC**

To develop a damping technology for fan blade incorporating smart structure vs. material such as piezoelectric (PE) materials or shape memory alloy (SMA).

- Shape memory alloy (SMA): blade stiffness change or shape change by electrical heating. Slow response to rotating/moving parts. SMA damping is also considered.
- Piezoelectric (PE) devices: change in peak amplitude at blade resonance by oscillating electric signal. Selected due to fast response to voltage/current signal from controller.

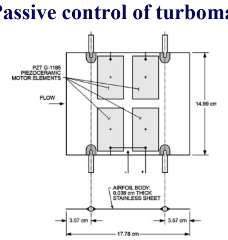
## Two conventional control approaches for PE blade damping

- ❖ Passive damping (or shunt damping): PE transducer is shunted by a **passive electric circuit** that acts as a medium for dissipating mechanical energy of the base structure – (Hagood and V. Flotow, 1990).
- ❖ Typical active control: PE transducers are being used as actuators and sensors for vibration control of flexible structures. These materials strain when exposed to a voltage and conversely produce a voltage when strained. Thus, one can minimize unwanted vibrations to the base structure by applying a **180° out-of-phase** voltage to the PE actuator (numerous references available).

## Literature survey for recent advances

### 1. Shunted piezoelectric circuit for turbomachinery blades

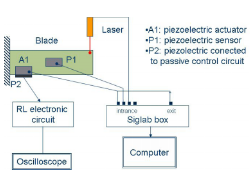
- **Passive control of turbomachine blading flow-induced vibrations (C. Cross, 2002).**




**Flat plate airfoil with PE elements**

  - ➢ Passage perforated plates on the rotor was used to generate wake ➔ Then the PE stators were excited in a chordwise bending mode.
  - ➢ A synthetic inductor was produced (Chen, 1986) to replace $L$ = 342 $H$ to control the first bending mode at 64.6 Hz.

- **Passive shunt circuit was tested for piezoblade damping (S. Livet, 2007).**




**Passive control diagram**

  - ➢ A virtual inductor (or gyrator) that consists of op amps, external power supply, resistors and capacitors was used to replace $L$ = 7.9 $H$ for a compressor disk blade.

**Literature survey** ***(continued)***

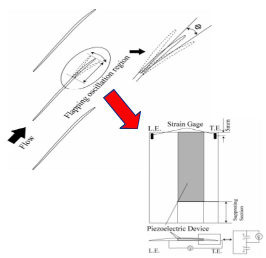
- **Numerous papers published for passive control for rotorcraft vibration.**

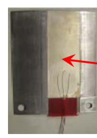

Typical example of experimental setup

- Demonstrated a variety of passive techniques using PE transducer at the test lab. To authors' best knowledge, none of them showed the actual demonstration on the rotating blades.

## 2. Active control of PE actuator for turbomachinery blades

- **Cascade flutter control using PE device in subsonic flow. (T. Watanabe, 2005, under NASA Quiet Engine Program)**



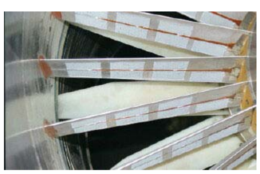
- Passage shock waive was generated at the trail edge of airfoil ➔ Induced unsteady aerodynamic work, causing instability.
- **Principle**: If the passage shock movement is controlled, blade vibration stability can be changed.

To control the passage shock movement, trailing edge of airfoil was oscillated by P.E.

**Literature survey** ***(continued)***

## Active control of PE actuator for turbomachinery blades *(continued)*

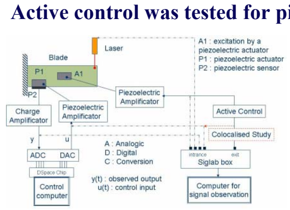
- **Low-speed fan noise control using PE actuators mounted on stator vanes: (P. Remington, 2003)**



**Actuators installed in the Stator Vanes.**

- 210 vane actuators in the stator vanes of the Active Noise Control Fan (ANCF) test rig were tested to control fan-stator interaction noise. Good noise reductions achieved.

- **Active control was tested for piezoblade damping (S. Livet, 2007).**




**Experimental active control protocol**

- Used a typical active control law (PD control) and achieved $\zeta = 0.9\%$.

**I. Introduction** *(continued)*

## Summary of literature survey

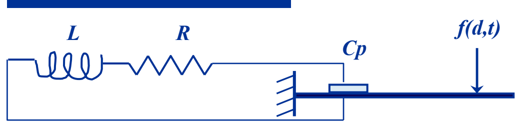
- Room temperature PE patches were used for the **stationary** blades, not for the rotating blades yet.
- No pure passive circuit was used because of huge inductor size. Instead semi-passive circuits were used to simulate physical inductors.
- Conventional active control laws were used, which cannot make feedback effective only at resonant frequencies.
- Wider and thicker patches were used, possibly resulting in aerodynamic performance penalty.

## Our unique approach

- Extend to 1) rotating blades and 2) high temperatures. Need adaptive capability to change in eigen frequencies and material properties.
- Implement pure passive and semi-passive circuits with $L_i \approx 1H$ on the rotating fan blade.
- In this presentation, we will demonstrate an active feedback architecture to control **resonant frequency only**. Passive circuits and adaptive feature can be easily programmable into control code.

# II. Passive Shunt Damping

## Resistive/Inductive shunt




The electrical resonance frequency for $i_{th}$ mode is

$$\omega_i = \frac{1}{\sqrt{L_i C}}$$

| Ti 6A1-4V beam | PE |
|---|---|
| $E$ = 15.2*1e6; 15.2 Mpsi, | $E_p$ = 1.03e7 psi |
| $t_b$ = 3/32 in, thickness | $t_a$ = 3/32 in, thickness |
| $L$ = 8 in, length | $L_p$ = 2 in, length |
| $\rho$ = 0.16 lb/in$^3$, linear mass density | $d_{31}$ = -60e-12, electric charge constant of PE |
| $w_b$ = 0.75 in, width | $w_p$ = 0.75 in, width |
| | $k_{31}$ = 0.30, electro-mechanical coupling factor |
| | $g_{31}$ = -15e-3, voltage constant |
| | $c_{cap}$ = 32.9e-9, nF |

Material properties and dimension of test specimen.

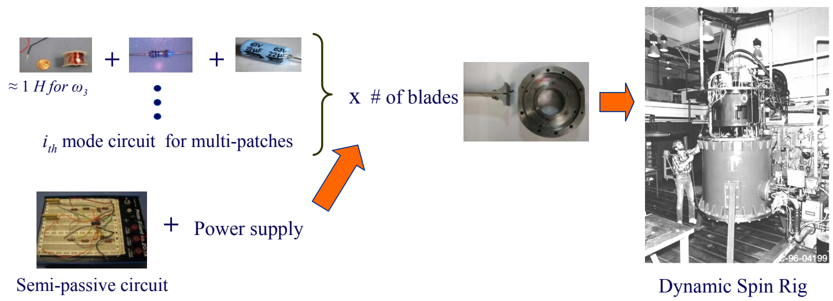
| Bending resonance | $L_i$ |
|---|---|
| (1st ) 78 Hz | 126.6 H |
| (2nd ) 402 Hz | 4.76 H |
| (3rd ) 801 Hz | 1.2 H |
| (4th ) 619 Hz | 0.3 H |
| (5th ) 2989 Hz | 0.086 H |

Target mode

Serial shunt circuit inductor size for resonant damping.

## II. Passive shunt damping *(continued)*

### Passive controller implementation issues




- Required **huge inductor size** mass to get a well tuned damping circuit.
- Semi-passive circuit also required constant power supply.
- Rotor imbalance, electronic parts at high centrifugal loads/g-loads, space problem for multi-patches situation.
- Seems to be not practical in reality for the rotating blade.

## III. Feedback Controller Design and Analysis

Investigated similarities between the shunt damping systems and collocated active vibration controllers (S. Moheimani, 2003).

### Digital control approach

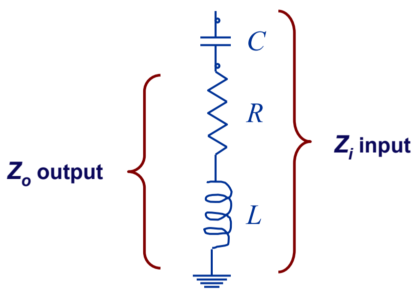
- Passive circuits can be easily **programmable** showing the shunted PE transducers can be viewed as a feedback control problem.
- Like a shunt circuit, the feedback is **effective only** at resonant frequencies.
- Also the active controller uses only one actuator to damp **several** of the blade's resonant modes (bending modes only at this study).

### Digital controller implementation issues

- Operational overhead of transducing high voltage power to blade across slip ring.
- Potential cross talk between high voltage control signals to on blade sensors.
- Might encounter other unexpected problems.

**III. Feedback Controller Design/Analysis** *(continued)*

**Transfer function of *RLC* circuit**




**General feedback control *RLC* network.**

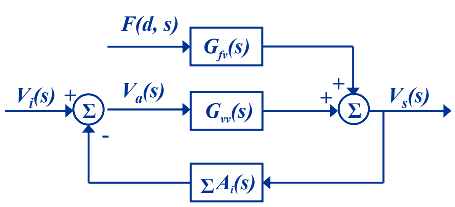
$$Z_i = R + i\omega L - i/(\omega C)$$

$$Z_o = R + i\omega L$$

$$\frac{V_o}{V_i} = \frac{Z_o}{Z_i} = \frac{R + i\omega L}{R + i\omega L - i/(\omega C)} = \frac{Cs(R+Ls)}{LCs^2 + CRs + 1}$$

➔ The controller is expressed in terms of passive circuit components.

**III. Feedback Controller Design/Analysis** *(continued)*




Feedback control block diagram for blade structure with PE.

**The actuator voltage $V_a(s)$ is**

$$V_a(s) = -A_i(s)V_s(s) + V_i(s)$$

**where $A_i(s)$ is**

$$A_i(s) = \frac{Cs(R + L_i s)}{L_i Cs^2 + CR_i s + 1}$$

**The closed-loop transfer functions**

$$V_s(s) = \frac{G_{fv}(s)F(d,s)}{1 + A(s)G_{vv}(s)} + \frac{G_{vv}(s)V_i(s)}{1 + A(s)G_{vv}(s)}$$

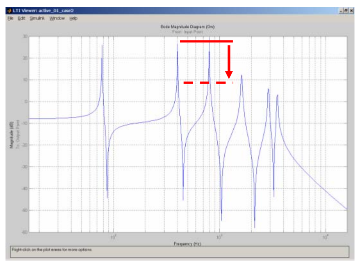
$$Y(r,s) = \frac{G_{fy}(r,s)F(d,s)}{1 + A(s)G_{vv}(s)} + \frac{G_{vy}(r,s)V_i(s)}{1 + A(s)G_{vv}(s)}$$

**where**

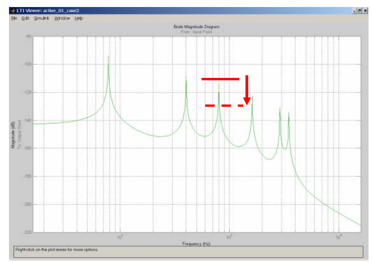
$$G_{fv}(s) = \frac{V_a(s)}{F(d,s)}, G_{vv}(s) = \frac{V_s(s)}{V_a(s)}$$

$$G_{fy}(r,s) = \frac{Y(r,s)}{F(d,s)}, G_{vy}(s) = \frac{Y(r,s)}{V_a(s)}$$

## III. Feedback Controller Design/Analysis *(continued)*



**Analytical frequency response of $G_{vv}$**

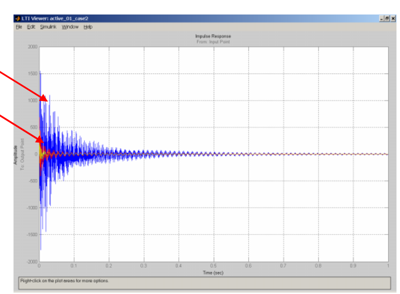


**Analytical frequency response of $G_{vy}$**

- ❖ Controller has to push peaks down to meet the design specification required for blade damping at resonances.
- ❖ Similarly, the transfer functions $G_{fv}$ and $G_{fy}$ can be obtained to complete a theoretical model of the laminated beam.

## III. Feedback Controller Design/Analysis *(continued)*


**Analytical open- and closed-loop frequency responses for multi-resonant damping.**



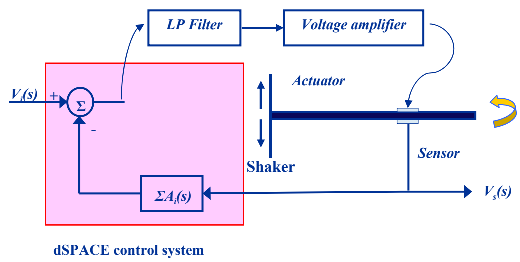
**Impulse responses for open- and closed-loop system.**

- ❖ This method can damp multiple resonance modes using one sensor-actuator pair.
- ❖ However, the control effort is very high because the PE patch was located very close to the root side (worst location) and the 1st bending mode included.
- ❖ Optimization process might be needed for an optimal control effort given performance requirement at each mode and existing control hardware capacity.

## IV. Experimental Results

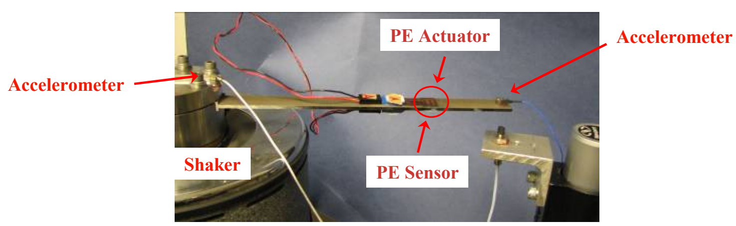
### Active feedback controller implementation using dSPACE system

- A very small and thin patch (0.685″x0.5″x0.015″) was bonded at the optimal place for the third bending mode (target mode) to demonstrate an active controller performance to see how peak amplitude can be reduced at the resonance.
- After fine-tuning the controller to the experimental target resonance, download the control algorithm to the dSPACE control system.




## IV. Experimental Results (*continued*)

### 1. Experimental test setup

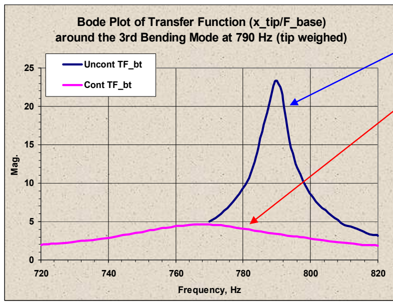



Beam: 8″×.75″×.095″, PE wafer: .5″×.25″×.01″

- A pair of identical PE patches (sensor and actuator) are bonded at the optimal location for the target resonance – 3rd bending mode at this demo.
- To measure an experimental transfer function of the beam from the shaker (excitation source) to the tip displacement, two accelerometers are attached accordingly.
- HP Analyzer generates swept sine signal to send to the shaker and reads all signals from accelerometers, PE sensor and actuator, and controller voltage from the dSPACE control system.
- Analyze closed-loop and open-loop transfer functions to investigate achieved damping ratios for the target mode.

**IV. Experimental Results** *(continued)*

## Bode plot of transfer functions around the target mode




| | Uncontrolled peak at tip | Controlled peak at tip |
|---|---|---|
| ζ | 0.00527 | 0.03 |
| Q | 94.9 | 16.7 |

| | Uncont. peak at PE | Cont. peak at PE |
|---|---|---|
| ζ | 0.00531 | 0.029 |
| Q | 94.2 | 17.24 |

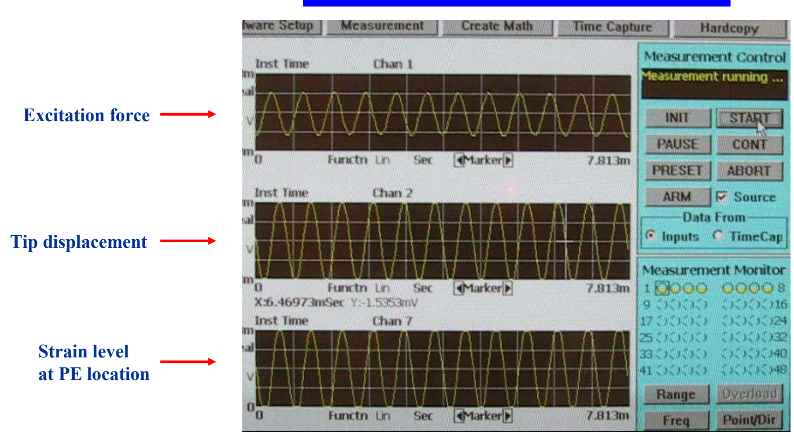
Linear scale of transfer functions $Tf\,|x_{tip}/f_{base}|$ around the 3rd bending mode at 790 Hz.

- Achieved about **83%** damping performance at tip displacement, changing damping ratio from 0.00527 to 0.03. Also strain level at the PE location was reduced by about **81%.**
- The damped natural frequency is lowered by about 2.5%.
- Passive ***LR*** circuit achieved about 5%, changing damping ratio from 0.00113 to 0.00119.

**IV. Experimental Results** *(continued)*

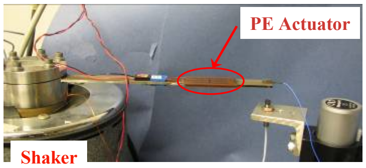
This movie shows the time history of controlled and uncontrolled tip displacement and strain level at the PZT location when the excitation force of 5mV*sin(790 Hz*t) was injected.

**➔ Please click the screen for the demo video.**

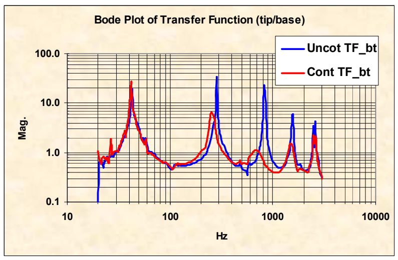


IV. Experimental Results *(continued)*

## 2. 2nd demo using a larger patch




Beam: 8″×.75″×.095″, PE wafer: 1.8″×.8″×.01″




Exp transfer functions $Tf\,|x_{tip}/f_{base}|$ from 20 Hz through 3k Hz.

In this demo, a larger patch was used to increase dynamic ratio of signals.

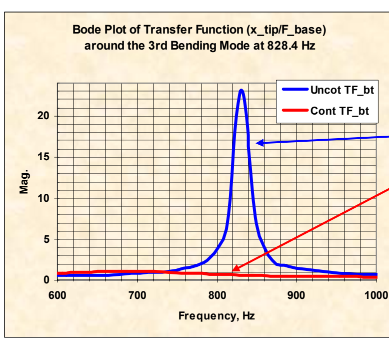
➔Unlike the previous demo, a set of control laws with **parallel connection** to combine control efforts for several modes were used in order to investigate active controller performance at all bending modes up to 3k Hz.

Reduced the tip displacement of the 3rd bending mode (target) at 824.4 Hz by more than **98%.** Also reduced neighboring peaks (2nd bending and 4th bending modes) by more than **83%.**

➔ Notice that 5th and 6th modes were affected significantly. However, the first mode was not affected at all because it's not located for controlling 1st bending mode.

IV. Experimental Results *(continued)*

## Bode plot of transfer function at the 3rd bending mode




Uncontrolled Peak

Controlled Peak

| | Uncontrolled Peak | Controlled Peak |
|---|---|---|
| ζ | 0.00371 | 0.141 |
| Q | 134 | 3.54 |

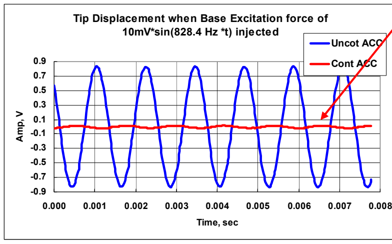
Linear scale of transfer functions $Tf\,|x_{tip}/f_{base}|$ at 828.4 Hz.

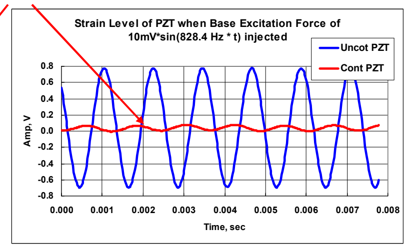
- Achieved about **98%** damping performance at tip displacement, changing damping ratio from 0.00371 to 0.141. Also strain level at the PE location was reduced by about **90%.**
- The damped natural frequency is lowered by about 12%.

**IV. Experimental Results** *(continued)*

## Time history when base excitation force of 10mV*sin(828.4 Hz*t) injected

**Actively Controlled Peaks**




Tip displacement and strain level of PZT when the base excitation force of sinusoidal signal at the 3rd bending frequency was injected.

- Showed time history of controlled and uncontrolled tip displacement and strain level at the PE location when excitation force with 828.4 Hz was applied.
- As anticipated, about **98%** reduction at the tip displacement and **90%** reduction in strain level on the PE location was achieved.

**IV. Experimental Results** *(continued)*

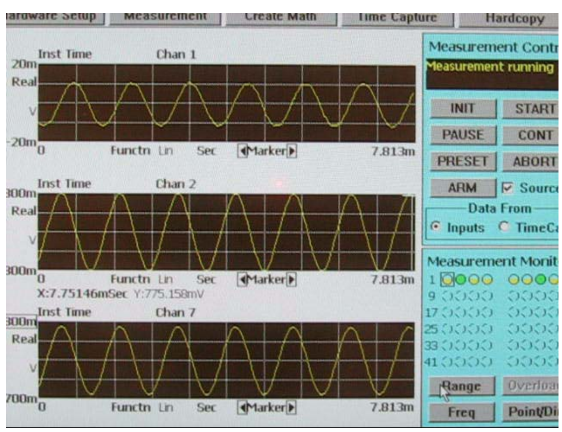
This movie shows the time history of controlled and uncontrolled tip displacement and strain level at the PZT location when the excitation force of 10mV*sin(828.4 Hz*t) was injected.

**➔ Please click the screen for the demo video.**

**Excitation force**

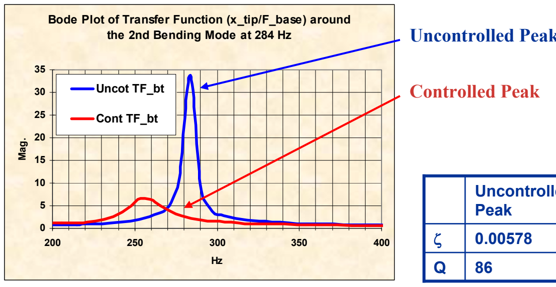
**Tip displacement**

**Strain level at PE location**



### IV. Experimental Results (*continued*)

## Bode plot of transfer function at the 2nd bending mode



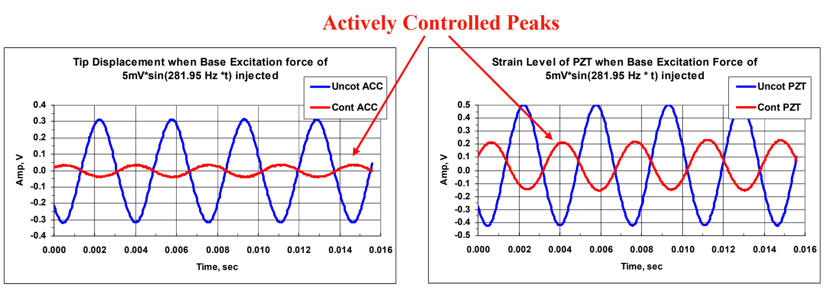

| | Uncontrolled Peak | Controlled Peak |
|---|---|---|
| ζ | 0.00578 | 0.03461 |
| Q | 86 | 14.45 |

Linear scale of transfer functions $Tf\,|x_{tip}/f_{base}|$ at 284 Hz.

- Achieved about **84%** damping performance at tip displacement, changing damping ratio from 0.00578 to 0.03461. Also strain level at the PE location was reduced by about **62%.**
- The damped natural frequency is lowered by about 8.5%.

### IV. Experimental Results (*continued*)

## Time history when base excitation force of 5mV*sin(281.95 Hz*t) injected

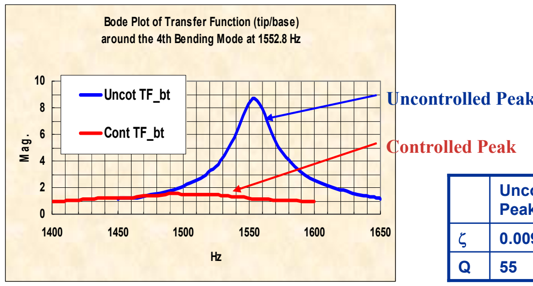



Tip displacement and strain level of PZT when the base excitation force of sinusoidal signal at the 2nd bending frequency was injected.

- Showed time history of controlled and uncontrolled tip displacement and strain level at the PE location when excitation force with 281.95 Hz was applied.
- As anticipated, about **84%** reduction at the tip displacement and **62%** reduction in strain level on the PE location was achieved.

IV. Experimental Results (*continued*)

## Bode plot of transfer function at the 4th bending mode



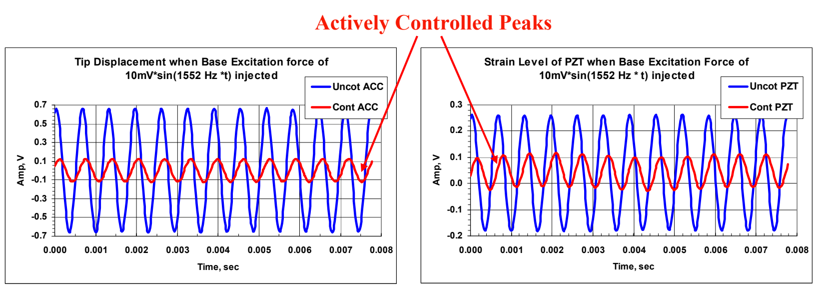

| | Uncontrolled Peak | Controlled Peak |
|---|---|---|
| ζ | 0.009 | 0.052 |
| Q | 55 | 9.62 |

Linear scale of transfer functions $Tf\,|x_{tip}/f_{base}|$ at 1552.8 Hz.

- Achieved about **83%** damping performance at tip displacement, changing damping ratio from 0.009 to 0.052. Also strain level at the PE location was reduced by about **78%.**
- The damped natural frequency is lowered by about 3.2%.

IV. Experimental Results (*continued*)

## Time history when base excitation force of 10mV*sin(1552 Hz*t) injected

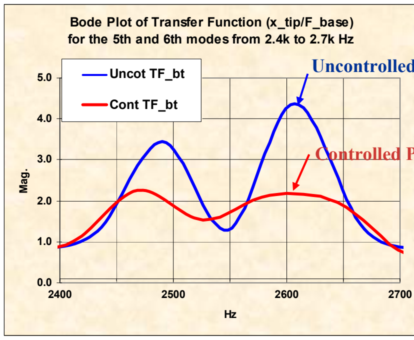



Tip displacement and strain level of PZT when the base excitation force of sinusoidal signal at the 4th bending frequency was injected.

- Showed time history of controlled and uncontrolled tip displacement and strain level at the PE location when excitation force with 1552 Hz was applied.
- As anticipated, about **83%** reduction at the tip displacement and **78%** reduction in strain level on the PE location was achieved.

**IV. Experimental Results** *(continued)*

## Bode plot of transfer function at the 5th and 6th bending modes




**5th Mode**

| | Uncontrolled Peak | Controlled Peak |
|---|---|---|
| ζ | 0.0094 | 0.013 |
| Q | 53.2 | 38.5 |

**6th Mode**

| | Uncontrolled Peak | Controlled Peak |
|---|---|---|
| ζ | 0.0076 | 0.021 |
| Q | 65.8 | 23.8 |

Linear scale of transfer functions $Tf\,|x_{tip}/f_{base}|$ at the 5th (2490 Hz) and 6th modes (2610 Hz).

- Achieved about **28%** damping performance at tip displacement at the 5th mode and **64%** damping performance at the 6th mode.

## Summary

- Suppression effect of the present active control with piezoelectric device was demonstrated to reduce **resonance peaks only** at the bending modes.
- A **single** patched beam could reduce the target resonant peak (3rd bending mode at this study) as well as neighboring modes - **multi-mode control**.
- This demo showed that this approach would reduce a number of patches for multi-mode control for rotating plate blade.
- Further comprehensive research including a trade-off study must be done to demonstrate a viable means of using this approach for rotating blades through GRC’s Dynamic Spin Rig test.

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**13. SUPPLEMENTARY NOTES**

**14. ABSTRACT**
The NASA Glenn Research Center (GRC) is developing an active damping at blade resonances using piezoelectric structure to reduce excessive vibratory stresses that lead to high cycle fatigue (HCF) failures in aircraft engine turbomachinery. Conventional passive damping work was shown first on a nonrotating beam made by Ti-6A1-4V with a pair of identical piezoelectric patches, and then active feedback control law was derived in terms of inductor, resister, and capacitor to control resonant frequency only. Passive electronic circuit components and adaptive feature could be easily programmable into control algorithm. Experimental active damping was demonstrated on two test specimens achieving significant damping on tip displacement and patch location. Also a multimode control technique was shown to control several modes.

**15. SUBJECT TERMS**
Piezoelectric transducer; Feedback control; Damping tests; Fan blades; Vibration damping

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