Today’s topics

• Quiet rotorcraft roof panels
  • Dr. Steve Hambric

• Vibration and high cycle fatigue of turbine blades
  • Alok Sinha

• Flow-excited ribbed panel optimization
  • Micah Shepherd, ARL and PhD student, Acoustics
    • Winner of INCE-USA Beranek Student Medal, 2014

• 2015 Noise and Vibration Conference

• Acoustic black holes
  • Dr. Stephen Conlon
  • Phil Feurtado, PhD student, Acoustics
2015 Conference

- structural vibration
- vibro-acoustics
- flow-induced noise & vibration
- noise and vibration control

Noise and Vibration Emerging Technologies

Dubrovnik, Croatia
April 13-15, 2015

novem2015.sciencesconf.org
novem2015@sciencesconf.org
Quiet Rotorcraft Roof Panels

Principal Investigators: Dr. S.A. Hambric, Dr. K.L. Koudela, M.R. Shepherd (PhD, Acoustics), and D.B. Wess

Sponsor: NASA

Collaborators:
• Strong transmission gear meshing tones excite roof panel
Baseline Panel

- Manufactured at Bell Helicopter (Textron)
Honeycomb core sandwich panel

- **Stiff and lightweight**
  - Carbon fiber face sheets
  - Nomex core
Baseline Panel – FE/BE modeling

- All solid quadratic elements, smeared face sheet properties
- BE model of surrounding air
Transmission Loss

Simulations within 3 dB of NASA SALT measurements
Damping of face sheets

- **3M VHB 9469**
  - Very thin (0.005”) adhesive with high material losses

![Graph showing Young's Modulus and Loss Factor vs Frequency](image)

![Diagram of composite structure](image)
Damping of face sheets – coupon tests and FE simulations

FE within 4% of measurements

Measured frequency (Hz)

FE frequency (Hz)
Damping of face sheets – coupon tests and FE simulations
Damping of face sheets – Projected performance

Transmission Loss (dB) vs Frequency (Hz)

- Simulated - \( f = 0.05 \)
- Simulated - \( f = 0.01 \)
Vibration and High Cycle Fatigue of Turbine Blades

Principal Investigator: Dr. Alok Sinha

Sponsors: GUlde 4 Consortium
Air Force Research Lab
MMDA (Modified Modal Domain Analysis): High Fidelity Reduced Order Model
- Geometric Mistuning: variations in blade geometry due to manufacturing tolerances
- Proper Orthogonal Decomposition (POD) of Coordinate Measurement Machine (CMM) data on blades’ geometries
- Validated on an industrial rotor

Flow-excited ribbed panel optimization

Principal Investigator: Micah Shepherd, PhD student, Acoustics
Dr. S.A. Hambric, Advisor

Sponsor: NASA, ONR

1Shepherd, M., Structural-acoustic optimization of structures excited by turbulent boundary layer flow, PhD dissertation in Acoustics, May 2014.
Automated design optimization

Use stochastic, global optimization procedure

Update Design Variables
- Shape
- Properties

Model

Analysis
- Finite Element
- Boundary Element

Objective Function
Flow-excited structure

- Simply-supported edges
- Aluminum
- Pressurization (55 kPa)
- Radiation from plate only
- TBL flow at 216 m/s
Unconstrained Optimization

\[ x/L_x = 0.10 \quad x/L_x = 0.89 \]

\[ \text{Radiated Sound Power (dB re } 1pW/Hz) \]

- Initial (92.9 dB)
- Optimized (83.1 dB)

Frequency (Hz)

April 2014
Constrained Optimization
(Limited center displacement)

\[ \frac{x}{L_x} = 0.48 \quad \frac{x}{L_x} = 0.89 \]

Radiated Sound Power (dB re 1pW/N^2/Hz)

- Optimized w/o constraint (83.1 dB)
- Optimized w/ constraint (84.8 dB)
Acoustic Black Holes for Noise and Vibration Control

29 April 2014

Center for Acoustics & Vibration
Annual Spring Workshop

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Recent Publications:


Motivation

• Increasing needs for lightweight high loss structures
  • Vibration attenuation
  • Transmission loss

• Vehicle applications
  • Interior acoustics
  • Radiated sound power

• New design opportunities
  • Embedded “designed-in” high loss treatments – new systems
  • Advanced composites
  • Multifunctional structures
  • “Add-on” treatments - problematic existing systems
Motivation: Vehicle Applications

Efficient Passive Noise & Vibration Control
Background: Black Hole History

Concept of an object from which light could not escape - *black hole*

- Originally proposed by Pierre Simon Laplace in 1795
- Using Newton's Theory of Gravity, Laplace calculated if an object were compressed into a small enough radius - the escape velocity of that object would be faster than the speed of light
- **Singularity** point has zero volume and infinite density
Black Holes Background: Acoustic Wave Propagation

**Pekeris, JASA 18 (1946),** absence of reflected waves - layered inhomogeneous media w/ sound velocity profile that decays linearly to zero w/ increasing depth

**Mironov, Sov. Phys. Acoust. 34 (1988),** no reflection of waves in plate w/ thickness decreases smoothly to zero over finite interval

“Acoustic Black Holes”

**Krylov, (multiple publications 2001-2013),** power law one-dimensional wedges, two-dimensional ABH feature in plates

More recently many other authors examining fundamental 1D and 2D concepts theoretical and preliminary measurements of single features

**Next Step** – need to examine how to apply ABH’s to practical noise control applications
• Bending Wave Speed
  – Aluminum
  – 6.8 mm (0.268 inch)
  – Power taper = 2.2

\[ c_b = \sqrt{\omega \kappa c_L} \]

\[ = \left[ 2\pi f \left( \frac{h}{2\sqrt{3}} \right) \sqrt{\frac{E}{\rho (1 - \nu^2)}} \right]^{1/2} \]

\[ h(x) = h_f, \quad 0 \leq x \leq L/2 \]

\[ = \varepsilon (L - x)^m, \quad L/2 < x \leq L \]

\[ c_{b\text{-taper}} = \left[ \text{constant} \right] f^{1/2} \left( L - x \right)^{m/2} \]

\[ c_b \rightarrow 0 \quad \text{as} \quad x \rightarrow L \]
• Many authors cite no reflections (ABH effect) with power taper
  – Due to the vanishing wave speed as tip is approached
  – Also cite need for added damping layer to dissipate resulting
    reflections due to “practical manufacturing limitations” (tip thickness
    does not reach zero)

• Singularity at tip of power taper structure
  – Wave speed vanishes (infinite time to reach tip, no reflections)
  – Velocity becomes infinite (the second effect of the singularity)

• The velocity result tells us for practical structures we NEVER
  want the thickness profile to reach zero even if we could
  manufacture it!
Tapered wedge simulations
  • Taper power= 2.2
  • Material= Aluminum
  • Thickness= 6.8 mm (0.268 inch)
  • Material Damping: 2%

Damping layer:
  • Thickness= 2.2 mm
  • Damping : 10%
  • Excitation= harmonic at 5kHz
  • NASTRAN Sol 109 – Time Transient

• Finite thickness at wedge termination – significant reflections
• Addition of damping layer eliminates majority of reflections

With added damping layer
Finite Element / Boundary Element Models & Response Simulations

- Transient excitation (FE)
  - wave attenuation

- Harmonic excitation – Structural acoustic coupling (FE/BE)
  - accelerance / vibration reduction
  - radiated sound power reduction
  - structural intensity
Transient Excitation Results: Vibration Reduction

- Aluminum Plate
  - 6.8 mm thick (0.268 inch)
  - 5 x 5 grid of ABH
  - Material damping: 2%
  - ABH
    - Diameter: 10cm
    - Taper exponent: 2.2
    - Damping layer thk: 2mm
    - Damping: 10%
  - Sinusoidal excitation (5 kHz)
  - NASTRAN Sol 109, Δt=1.e-7

- Vibration completely attenuated ~ 4\textsuperscript{th} ABH Row
Results: Vibration & Radiated Sound Reduction

FE / BE Models

- Aluminum plate thickness = 6.8 mm (0.268 in)
- Panel masses, uniform = 9.09 kg, ABH = 7.58 kg
- Panel length and width = 0.7 m (both panels)
- Aluminum damping loss factor = 0.001
- Damping layer thickness = 2.0 mm
- Damping layer loss factor = 0.1

- Air loading on one side of panels
- Boundary conditions
  - Mechanical – edges unsupported (free)
  - Acoustic – baffled
- Unit force point drive
- Uniform panel critical frequency ~ 1900 Hz
Results: Vibration & Radiated Sound Reduction

- Total radiated sound power
- Surface averaged accelerance

Vibration & sound power reduction (5 to ~20 dB) above 2.5 kHz

ABH panel = 80% mass of uniform panel
Results: Radiated Sound Power Reduction

\[ \left[ \frac{f \times (\text{Dia}_{ABH})}{c_{B\text{-plate}}} \right] \approx 6500 \text{Hz} \]

Significant ABH panel sound power reduction

ABH broadband absorption
Results: Modal Loss Factors

- With air loading
- Uniform plate

$\text{f}_{\text{crit}} \sim 1900 \text{ Hz}$
Results: Modal Loss Factors

- No air loading
- ABH & uniform plates have same damping layers / locations

~37x
Results: Modal Loss Factors

Graph showing the relationship between Loss Factor and Frequency (Hz). The x-axis represents Frequency (Hz) ranging from 0 to 5000, while the y-axis represents Loss Factor ranging from 0 to 0.05.
Results: Modal Loss Factors

Frequency (Hz)

Loss Factor

(0,1) Mode

(1,1) Modes

(2,1) Modes
Results: Energy Flow 2x2 ABH

Point damper

Point force drive

Structural intensity: Local ABH mode ~ 2700 Hz

Structural intensity: Global mode ~ 1000 Hz
Results: Energy Flow 2x2 ABH

ABH side of plate

Structural intensity: Local ABH mode ~ 2700 Hz

Backside of plate
Results: Energy Flow 2x2 ABH

Structural intensity: Local ABH mode ~ 2700 Hz
• Results demonstrated the potential for arrays of Acoustic Black Hole features embedded in structures for vibration and noise control

• Panel with periodic ABHs showed significant vibration and radiated sound power reduction
  • ABH panel mass = 83% of uniform panel mass

• Low frequency energy dissipation mechanisms were characterized
  • Global vs local ABH modal loss factors

• Future noise control work will focus on design sensitivity studies for ABH features & optimization