SAND2012-8176C

Acoustics Research at Sandia

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Fall-2012 Meeting of the
North Carolina Chapter of the Acoustical Society of America
Raleigh, NC
September 28th 2012

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Outline

- History of Acoustic Testing
- Modeling Direct Field Acoustic Tests (DFAT)
- Sierra/SD
- Conclusions

History of Vibroacoustic Testing at Sandia

- **According to urban legend Sandia had both a reverberant acoustic chamber and progressive wave tubes in the 1960's or 1970's but they were dismantled.**
- **During the 1990s a vibroacoustic testing facility (VATF) was constructed using a reverberant acoustic field (RAF) approach:**
  - Chamber Dimensions: 21.6 ft. × 24.6 ft. × 30.1 ft.
  - Volume: 16,000 ft^3
  - Walls: painted 18 in. thick concrete reinforced
  - Schroeder Frequency: ~ 280 Hz
    - Using the strict 3 resonances in one resonance half-width requirement

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History of Vibroacoustic Testing at Sandia

- **The VATF was designed to perform satellite testing up to levels compatible with the Space Shuttle (145dB OASPL) and it achieved > 158 dB OASPL during initial startup tests.**
  - Acoustic Sources: Wyle WAS-3000 & Ling EPT200 Nitrogen horns
  - Fluid: dry nitrogen generated from LN_2
  - Mechanical: could accommodate 2 Unholtz-Dickie T1000 shakers
    - There were plans to place them in a pit so test articles could be wheeled directly in, but the pit is currently filled in and the shakers were instead housed in a vented shroud to protect them from the acoustic noise while still allowing for air cooling.

History of Vibroacoustic Testing at Sandia

- **A lack of demand coupled with the safety and reliability issues (operating the L-N_2 system, oxygen deprivation, etc.) caused the facility to be mothballed in the late 1990s**
- **In the 2000's Paul Larkin (now at MSI) resurrected the facility using electrodynamic speakers to generate the acoustic field**

Nitrogen Horns

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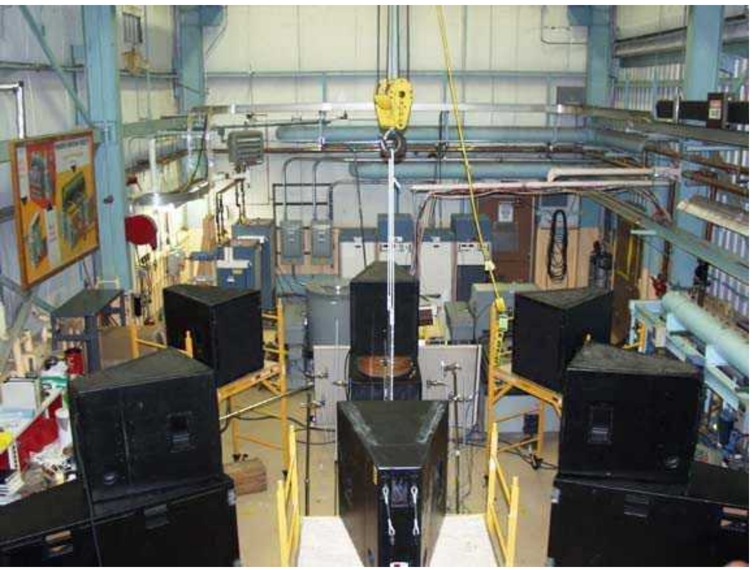
Current Electrodynamic Capability

- **Loudspeakers:**
  - Six Full-Freq-Range VA4s
  - Two Mid-Freq-Range M4 Horns
  - Four Low-Freq-Range Quake Subs

- **Amplifiers:**
  - Five Crown MA-5002 VZ
    - Stereo (2 channel each)
- **Loudspeakers can be placed anywhere within the chamber**
  - Typically placed in corners to excited many modes
- **Spectral Dynamics JAGUAR closed-loop control system**

Hot Topic: Direct Field Acoustic Testing

Sandia R&D/Explore:



Commercial: Maryland Sound International

- **Goal: bring the high-amplitude reverb field to the test article, rather than vice-versa**

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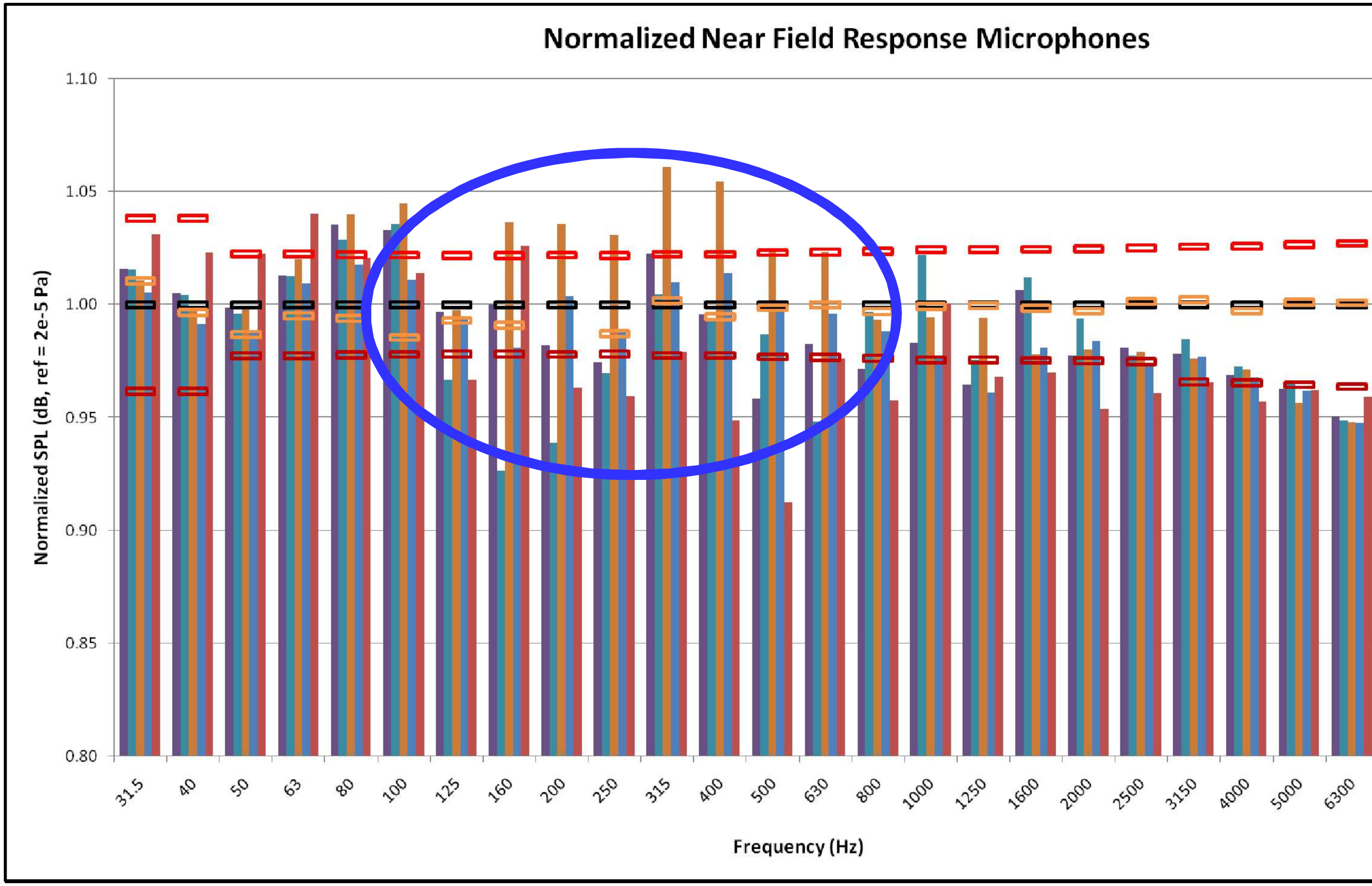
SIMO DFAT Experimental Microphone Levels



Channel	Location
15	SE
16	SW
17	Center
18	NW
19	Inside

- **Large spatial variation of acoustic level during a Direct Field Acoustic Test at Sandia. WHY?**

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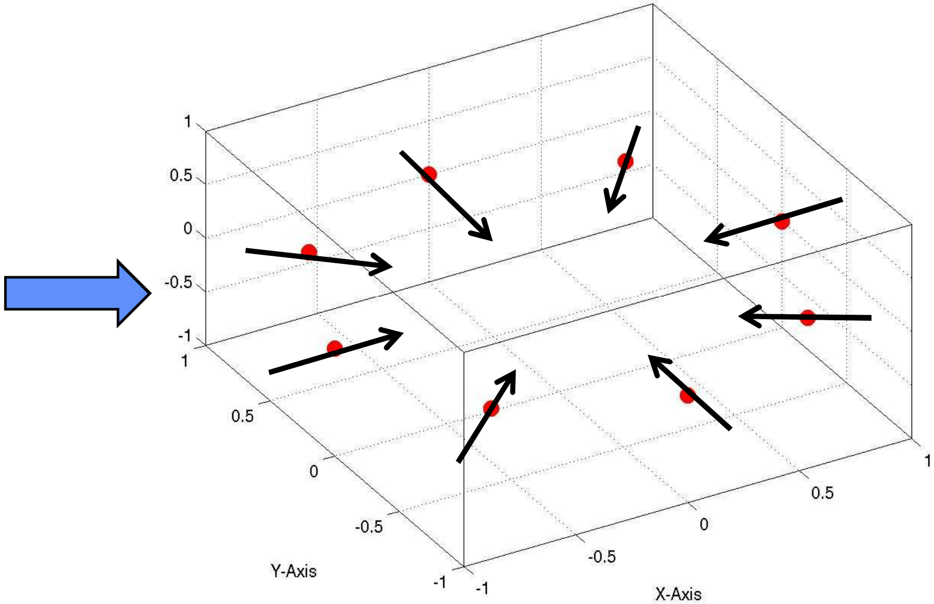
Modeling *SIMO* Direct Field Acoustic Tests



- **Loudspeakers generate spherically spreading waves**
- **Model using propagating plane waves (simple)**
  - **Neglects directivity of loudspeaker, diffraction, etc., yet not a poor approximation (actually DFAT ideal goal)**
- **Plane waves originate at infinity**

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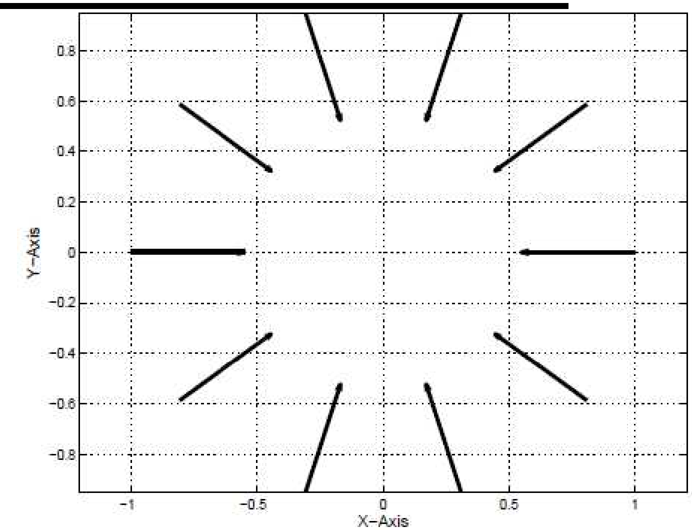
Modeling Direct Field in 2D

- **Model a two-dimensional field for simplicity:**

$$p_{total}(\vec{x},\omega,t) = \mathrm{Re}\left\{ \sum_{q=1}^{N} \left|p_q(\omega)\right| e^{i\left(\omega t - k\hat{n}_q\cdot\vec{x}+\varphi_q\right)} \right\}$$

- **Uniformly distribute:**

$$\hat{n}_q = -\cos\left(\frac{2\pi q}{N}\right)\hat{e}_x - \sin\left(\frac{2\pi q}{N}\right)\hat{e}_y$$



(a) $N = 10$ plane waves

- **Let all plane waves have same amplitude and time average:**

$$\frac{P_o(\omega)}{\sqrt{N}} = \left|p_q(\omega)\right|$$

$$\overline{p^2_{total}}(\vec{x},\omega) = \frac{P_0(\omega)^2}{2}\left(1+\frac{2}{N}\sum_{m=1}^{N}\sum_{n=m+1}^{N}\mathrm{Re}\left\{ e^{-ik(\hat{n}_m-\hat{n}_n)\cdot\vec{x}+i(\varphi_m-\varphi_n)}\right\}\right)$$

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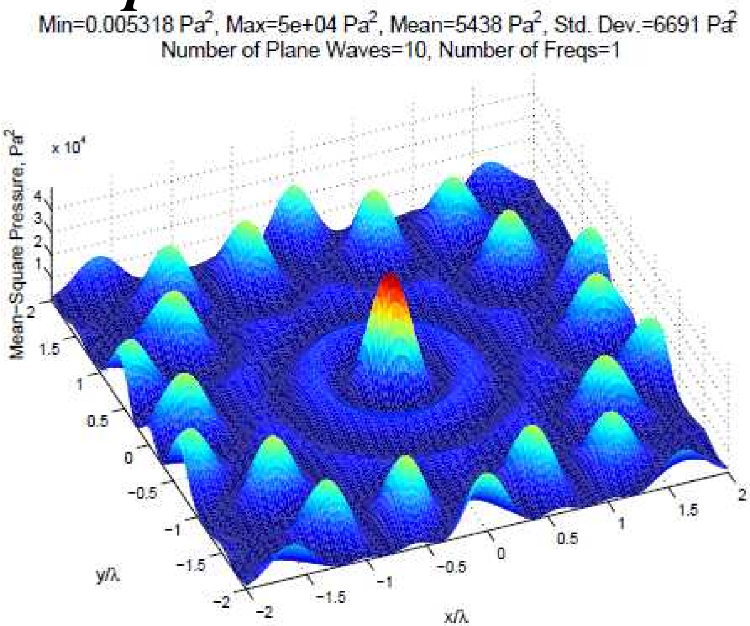
Modeling *Pure-Tone* Direct Field

- **For Direct Field Test same phase throughout:**

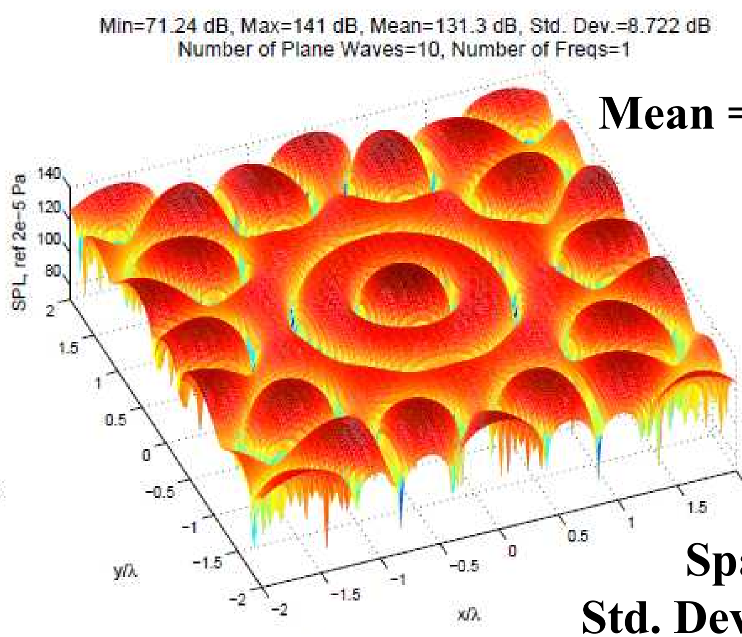
$$\varphi_q = \varphi$$

Ten Sources, Single Frequency:

Mean-Square Total Pressure: → ***Sound Pressure Level:***



(a) Mean-Square Pressure, Pa^2



(b) Sound Pressure Level, dB

Mean = 131.3 dB

Spatial Std. Dev. = 8.7 dB

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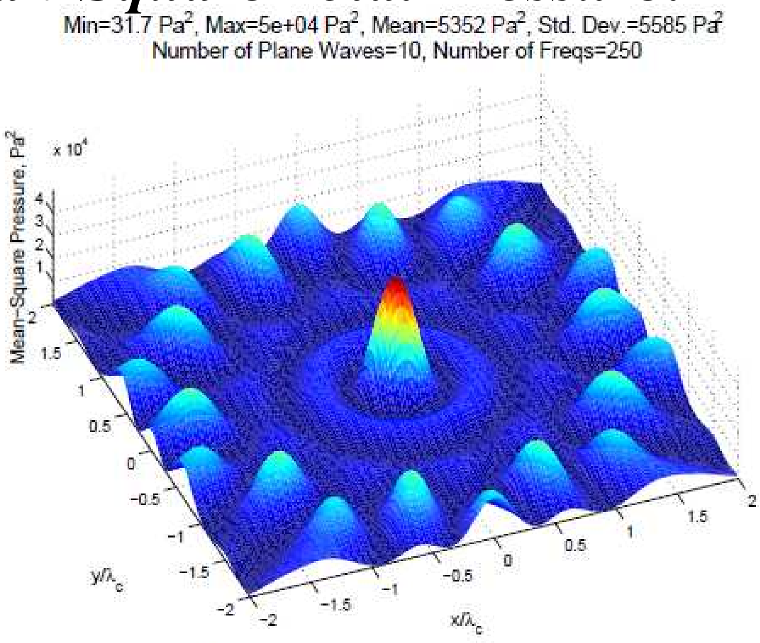
Modeling *Broadband* Direct Field

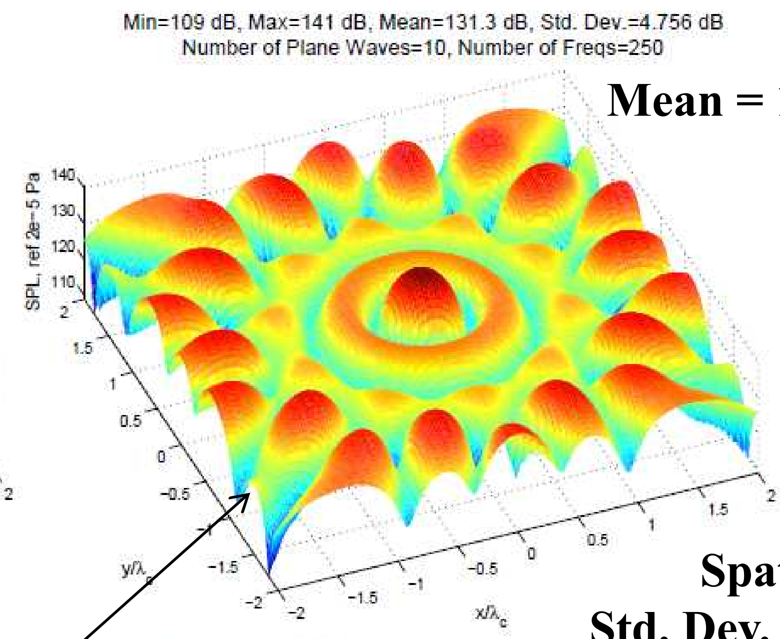
Ten Sources, 1/3 Octave, <u>250 Frequencies in Band</u>:

(P_o same as before)

Mean-Square Total Pressure: ⟶ ***Sound Pressure Level:***



(a) Mean-Square Pressure, Pa^2



(b) Sound Pressure Level, dB

Mean = 131.3 dB

Spatial Std. Dev. = 4.7 dB

Note: ~ same mean, lower std. dev.

λ_c is wavelength at center frequency of band

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Definition of Diffuse Field (Reverb Chamber)

1. **Equal probability of energy flow in all directions**

   Statistical parameters spatially homogeneous and isotropic

2. **Comprises an infinite number of propagating plane waves with *random phase relations*, arriving uniformly from all directions**

   At any point for pure-tone field, phase relations comprised of fixed set of random variables

From: Jacobsen, F., "The Diffuse Sound Field - statistical considerations concerning the reverberant field in the steady state," Technical Report 27, Technical University of Denmark, 1979

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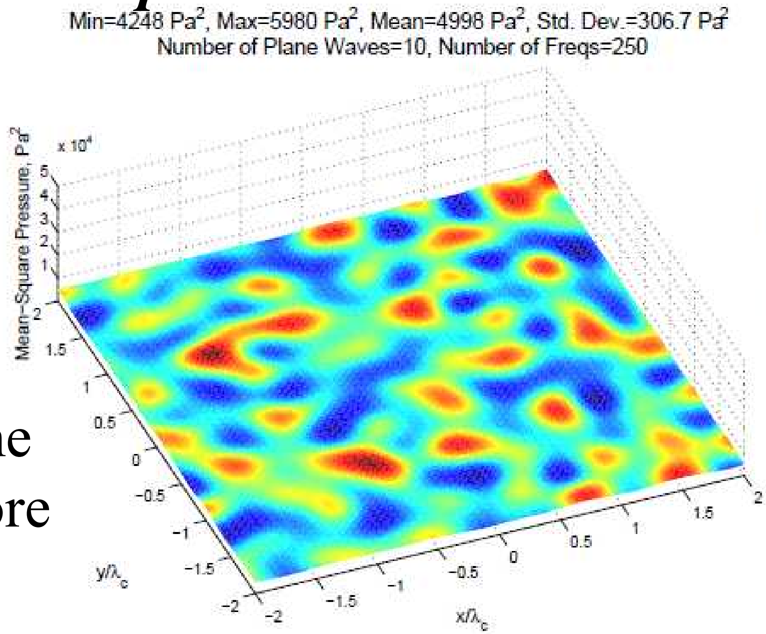
Modeling *Broadband* Diffuse Field

- **For Diffuse Field uniformly distributed random phase:**

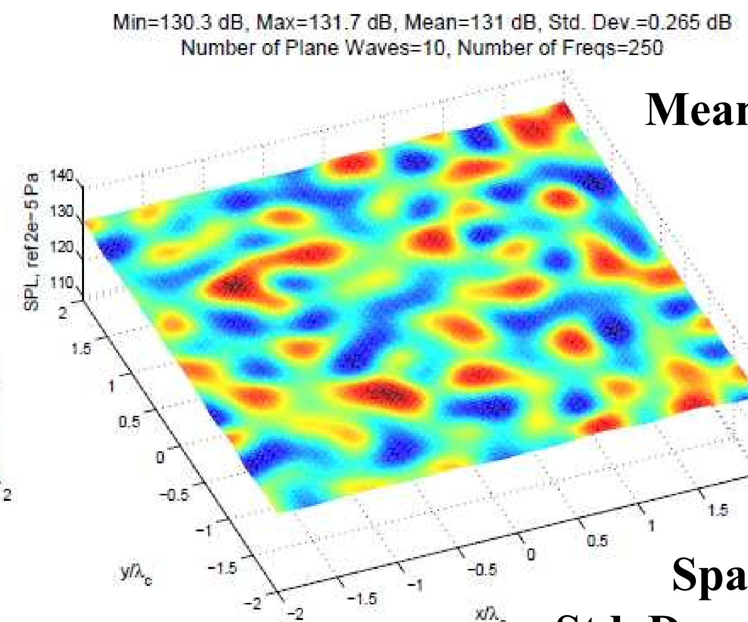
$$\varphi_q \in [0, 2\pi)$$

10 Sources, 1/3 Octave, 250 Frequencies in Band:

Mean-Square Total Pressure: $\longrightarrow$ ***Sound Pressure Level:***



(a) Mean-Square Pressure, Pa^2



(b) Sound Pressure Level, dB

P_o same as before

Mean = 131 dB

Spatial Std. Dev. = 0.26 dB

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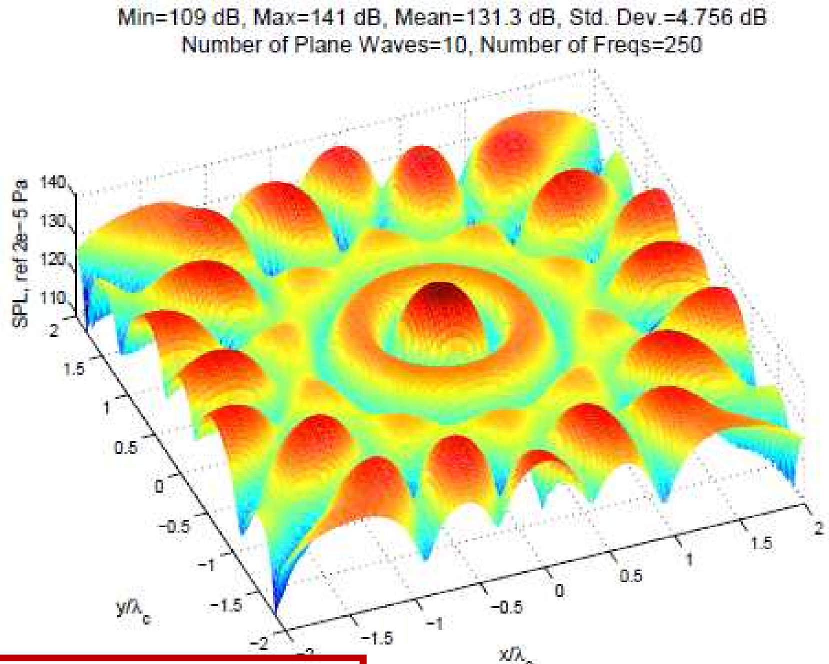
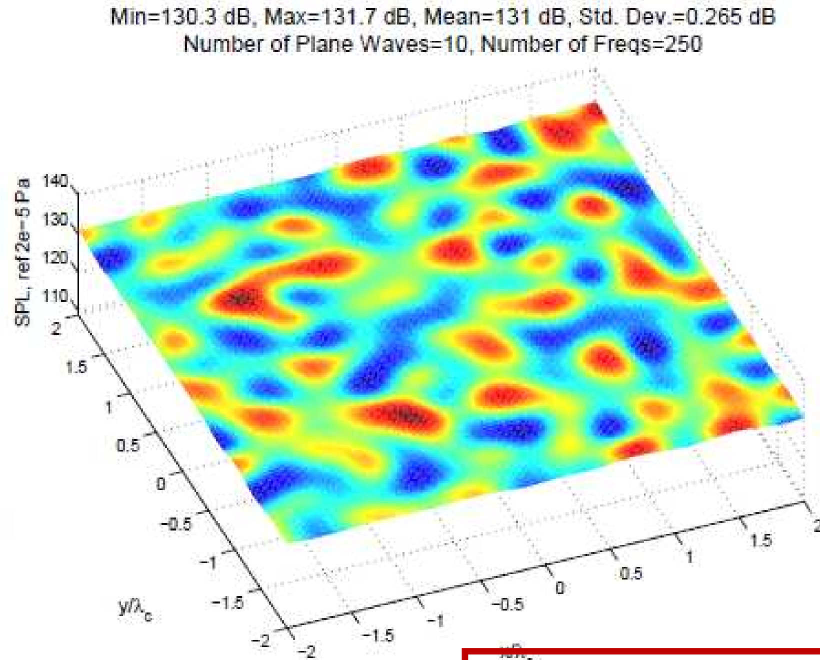
Cause of Relative Difference of Fields

- **Why is broadband diffuse field relatively more isotropic?**

DFT Sound Pressure Level:



Diffuse Sound Pressure Level:



Spatial Std. Dev. = 4.75 dB

Isophase Versus Random Phase Among Plane Waves

Spatial Std. Dev. = 0.265 dB

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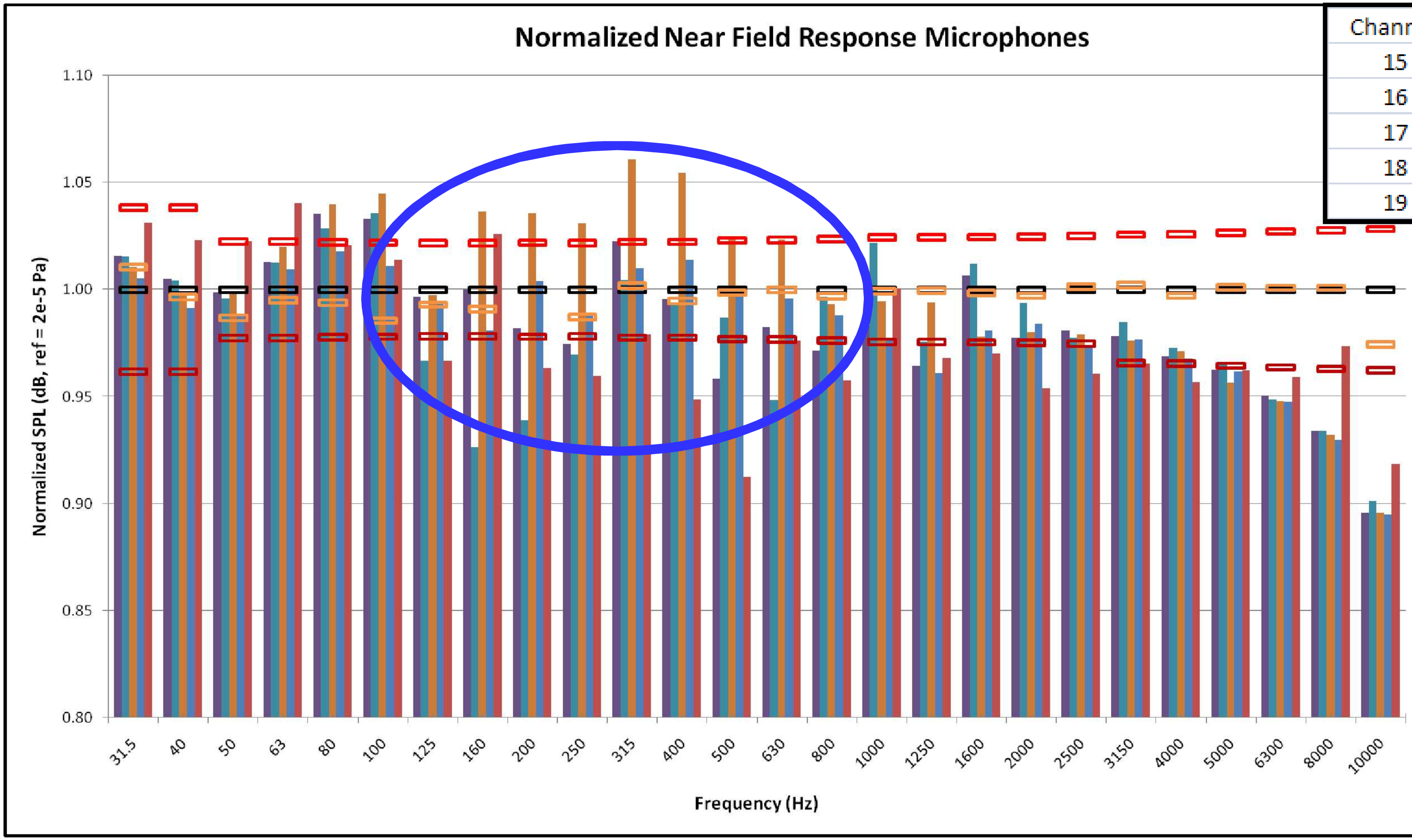
Return to DFAT Experimental Data



Channel	Location
15	SE
16	SW
17	Center
18	NW
19	Inside

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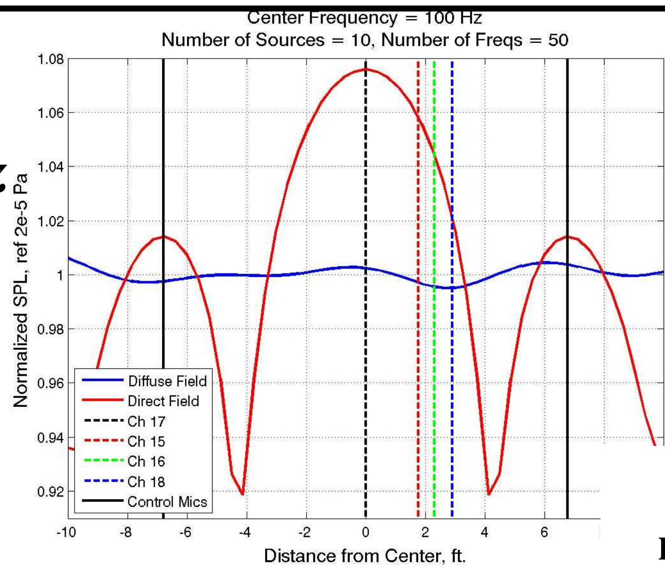
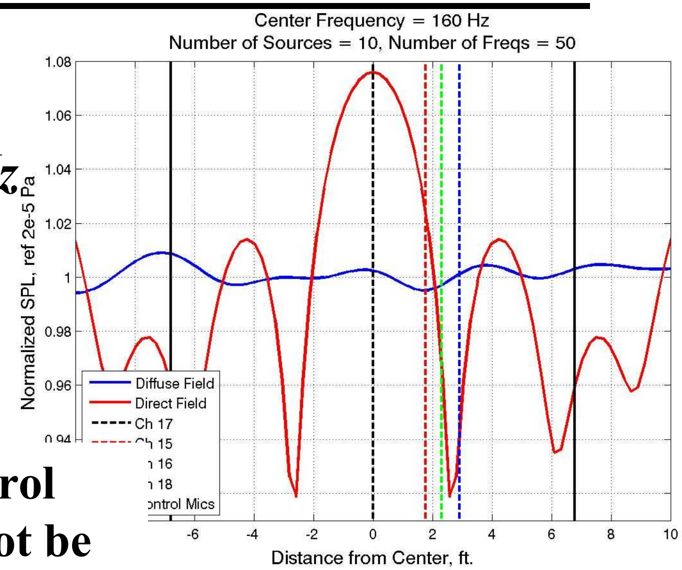
Microphone Location vs. Frequency

f_c=100 Hz

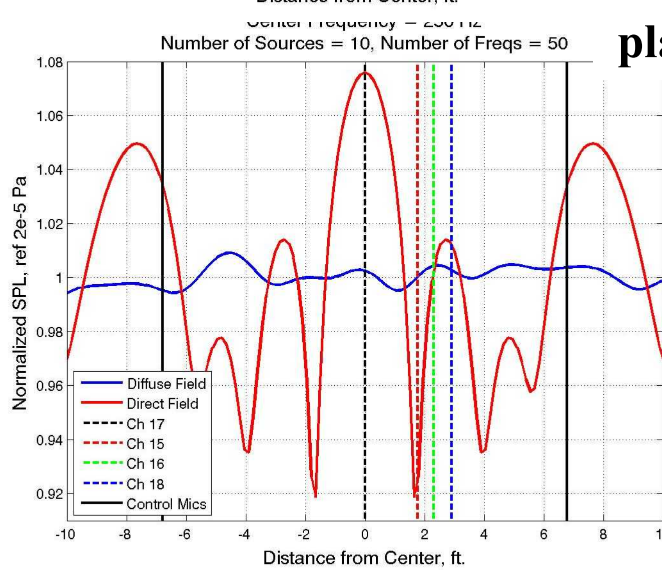


f_c=160 Hz

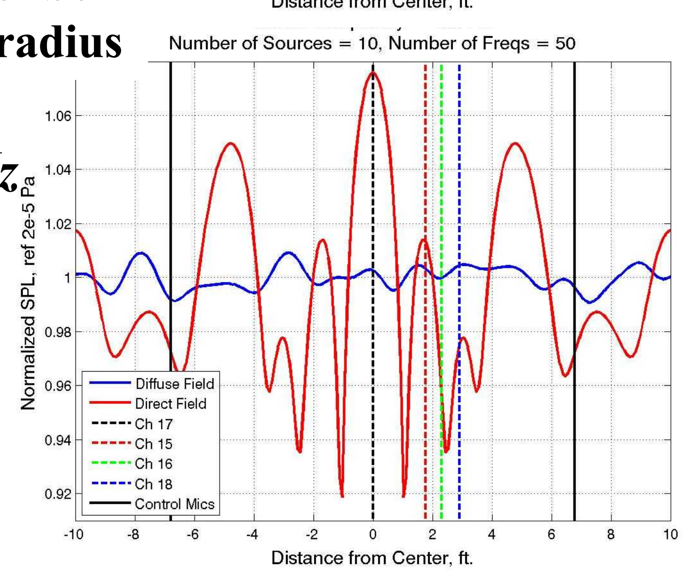


Implies control mics should not be placed at same radius

f_c=250 Hz

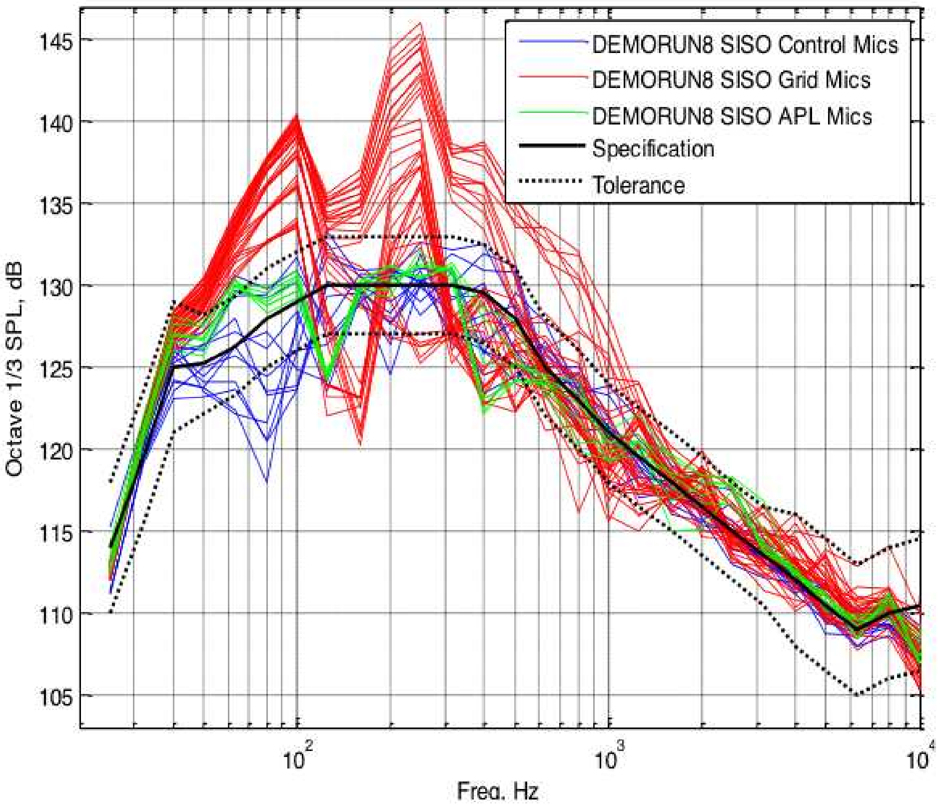


f_c=400 Hz

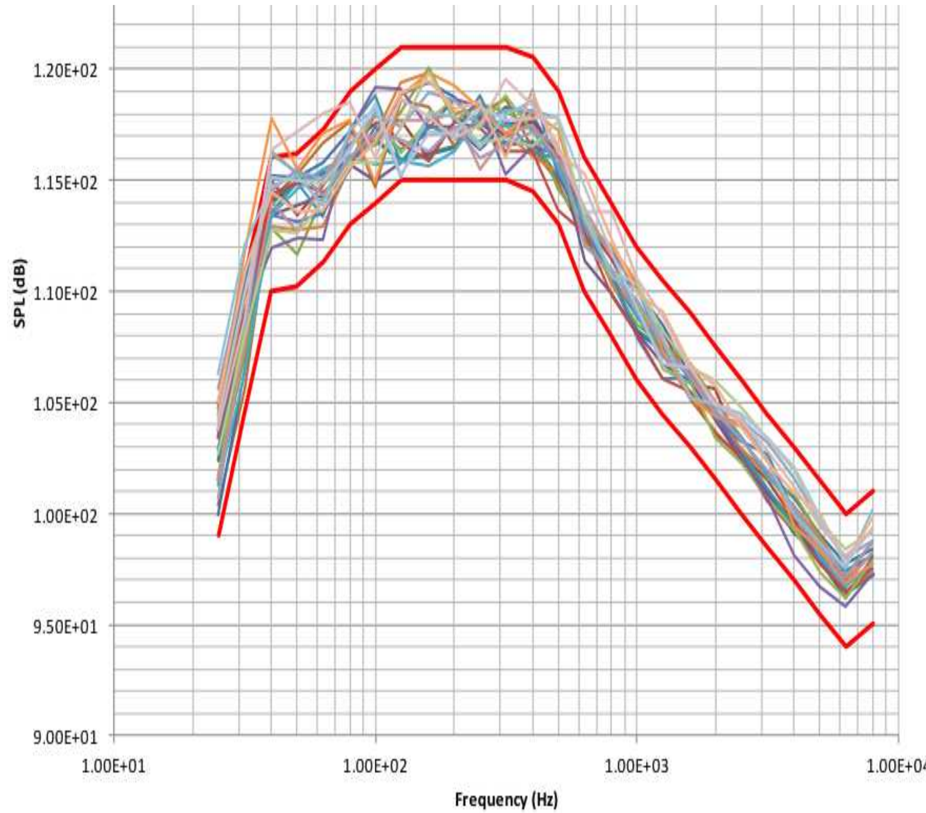


Industry Response to SIMO DFAT Research

- **Development of MIMO DFAT**



SIMO



MIMO

From: P. Larkin, *Direct Field Acoustic Testing*, Spacecraft and Launch Vehicles Workshop, June 19-21, 2012

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Topic Change

Sierra/SD
Massively Parallel Finite Elements

History and Intent

- **Sierra/SD was created in 1990's as part of the Accelerated Strategic Computing Initiative (ASCI) of the US Dept. of Energy**
- **Intended for *extremely* complex finite element analysis**
  - Models with 10s or 100s of millions of DOF
- **Scalability**
  - Ability to solve *n*-times larger problem using *n*-times more compute processors in nearly constant CPU time
- **Code portability**

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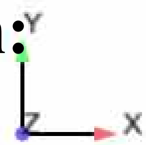
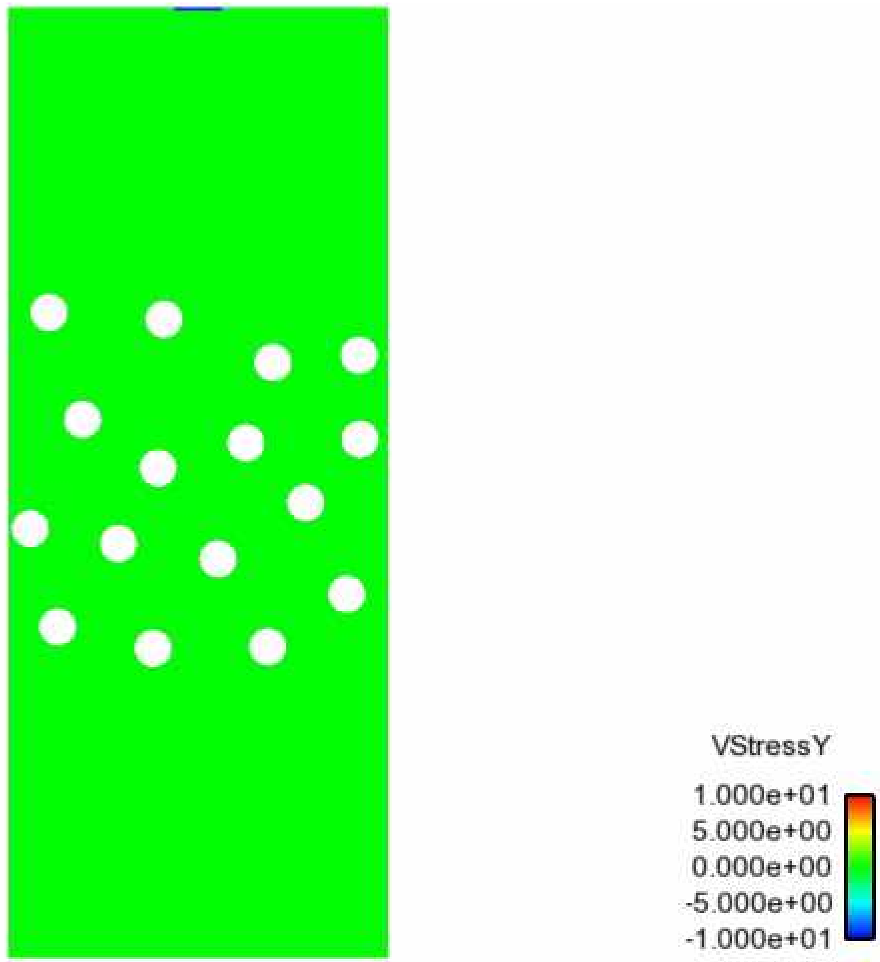
An Illustration of Intent: 1μs Pulse

- **Ultrasonic wave propagation in elastic plate**
  - 4x10x1 in. Aluminum
  - 1 MHz FRF shown (λ=0.25 in.)
- **Examine hole size/shape effects on scattering**
  - Visualize diffuse field development in elastic solids
- **For results shown:**
  - 32 elements/λ
  - 57,255,317 nodes
  - 343,531,902 degrees of freedom





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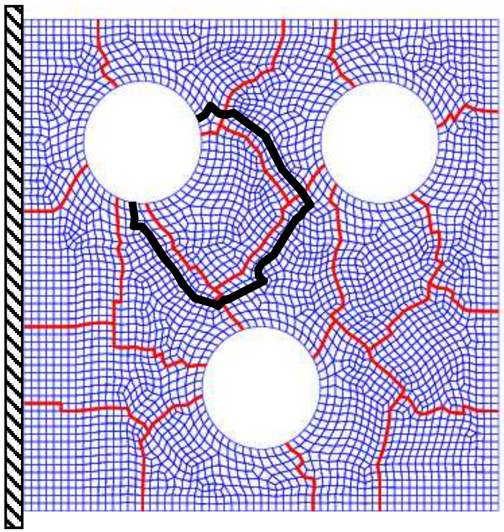
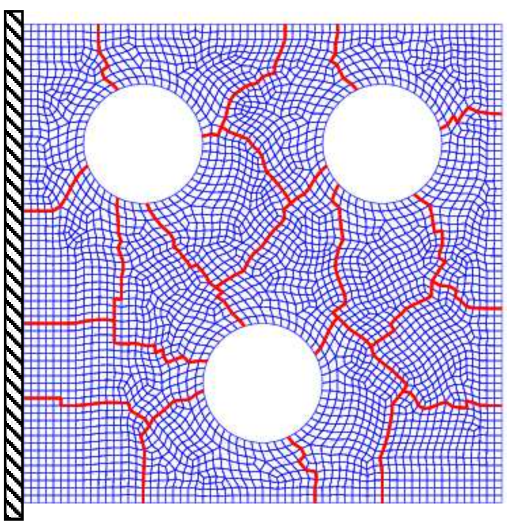
To Meet ASCI Requirements

- **Massively Parallel**
  - Distribution of processors (nodes), each with own memory, linked together by a specialized network communication system
- **Employ Domain Decomposition Methods**
  - First performed by Schwarz in the 1870s
- **Began First Using FETI-DP solver**
  - "Finite Element Tearing and Interconnecting" *(C. Farhat, et al., 2000)*
  - Versatile iterative solver
- **Current Solvers:**
  - FETI-DP and FETI-DPH
  - GDSW *(C. Dohrmann, et al., 2007)*
  - Others

Domain Decomposition



Schwarz Methods (Overlapping)



Schur Complement Methods (Iterative Substructuring)

- **Decompose model into smaller subdomains**
- **Each subdomain is often assigned to one processor**
- **Two-level methods have "local" subdomain solves and "global" coarse solve**
- **Solve using preconditioned conjugate gradients or GMRES**

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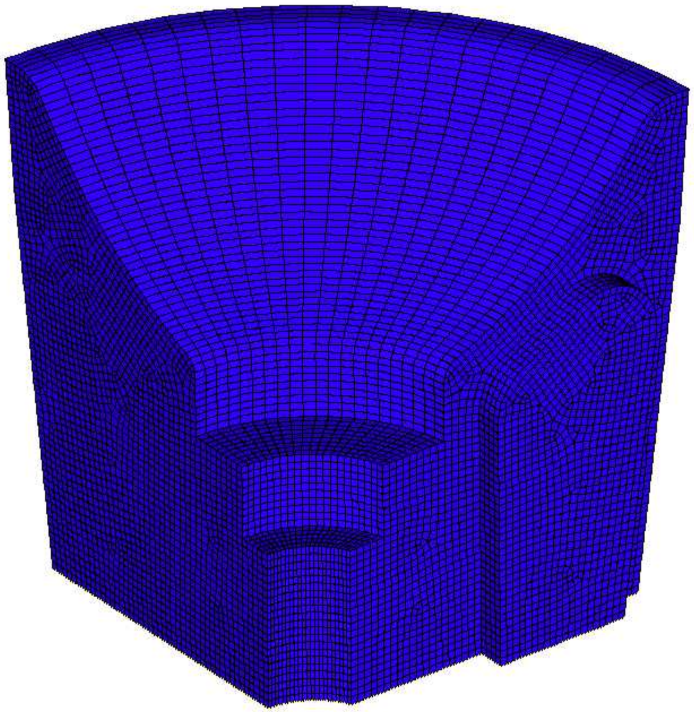
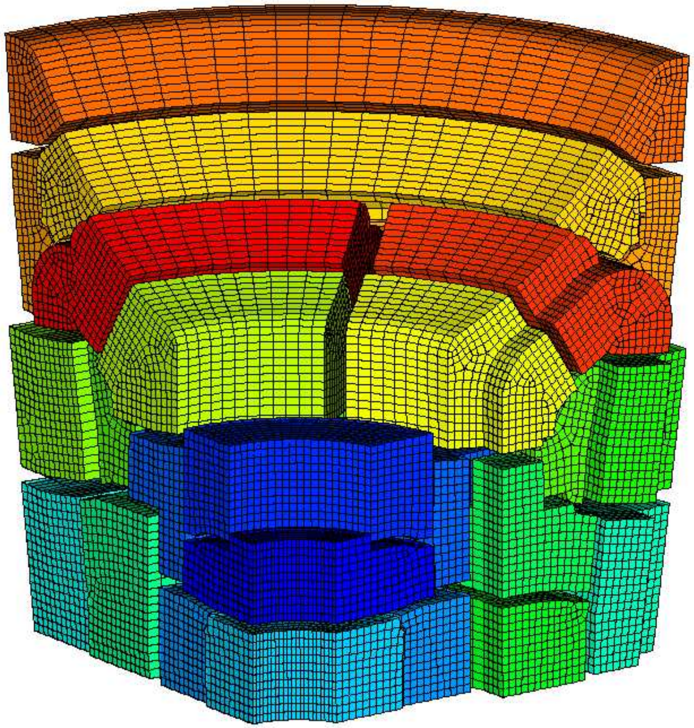
Domain Decomposition Example

Single Mesh Decomposed Into 20 Meshed Subdomains





Current State of High Performance Computing

- **1.37 petaFLOPS capability system, built by Cray, Inc**
- **Installed 2010-2011 at Los Alamos National Laboratory**
- **Compute nodes: 8,944**
  - Each compute node: 2 AMD G34 Opteron Magny-Cours 2.4 GHz 8 core processors for a total of 143,104 cores

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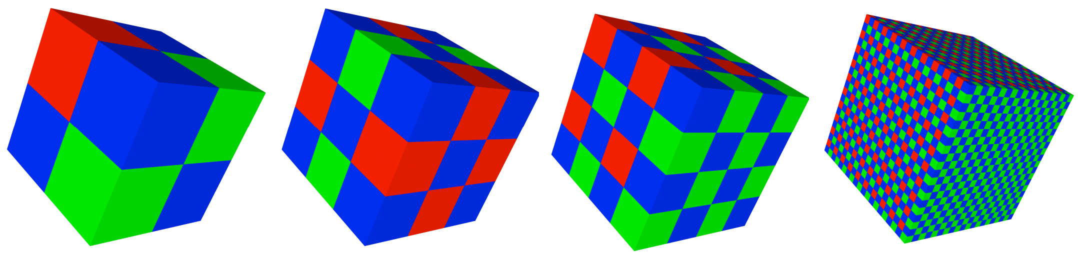
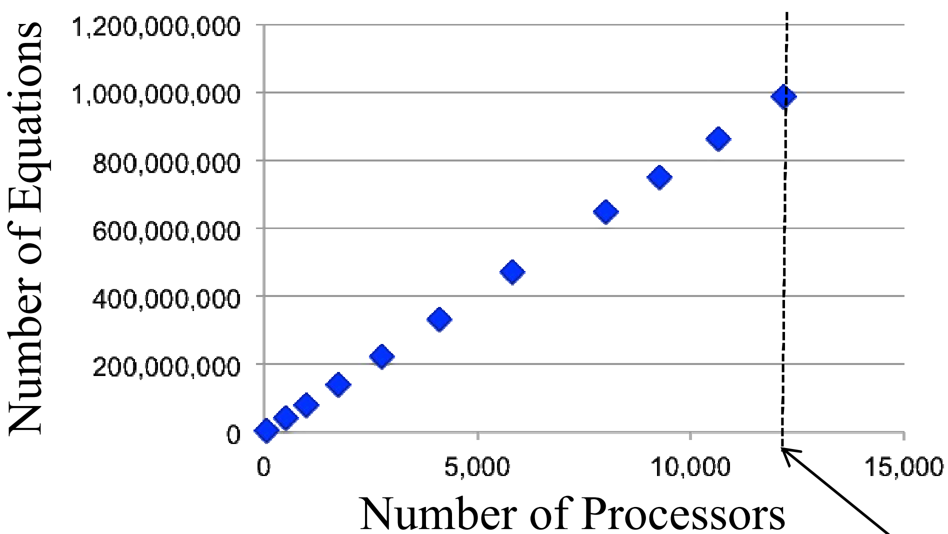
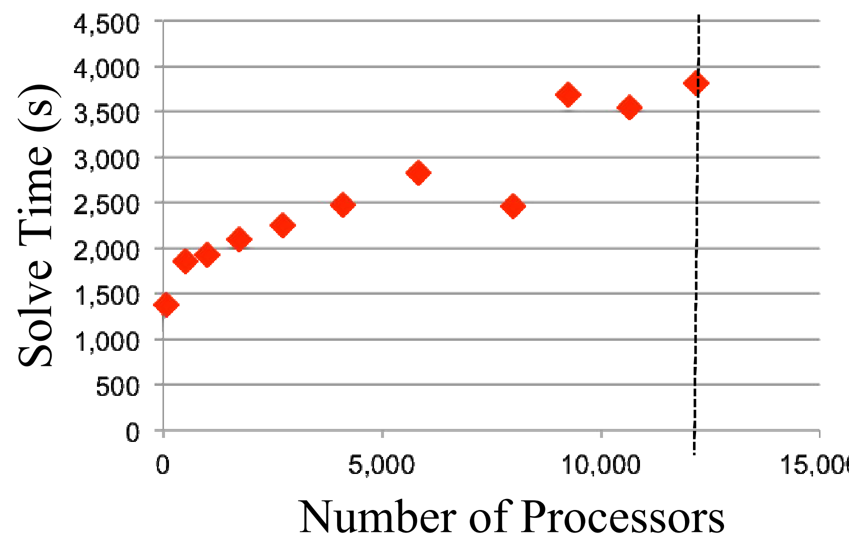
Eigenvalue Scaling Studies

Scaling studies were performed to characterize solver performance to 1 billion equations, well beyond previous work







Hit 32-bit integer limitation in Sierra

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Sierra/SD Solution Methods

- **Linear and Nonlinear Statics and Transient Dynamics**
- **Eigenanalysis**
  - Real and complex (quadratic)
- **Direct Frequency Response**
- **Random Vibration Analysis**
- **Modal Based Solutions for Transient Dynamics, SRS, Frequency Response**
- **Coupled Nonlinear-Linear Analysis**
  - With Adagio/Presto *(Sandia in-house codes)*

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Structural Acoustics

- **Formulations for Structural Acoustics:**

Scalar Based

  - Velocity potential formulation *(Everstine, 1981, 1997)*
  - Mixed pressure-potential symmetric formulation *(Felippa & Ohayon, 1990; Pinsky, 1991; Ohayon 1996)*

Vector Based

  - Displacement-based formulation *(Hamdi & Ousset 1978; Belytschko, 1980; Wilson, 1983; Chen 1990; Bermudez 1994)*
  - Space-time formulation *(Harari et al., 1996; Thompson and Pinsky, 1996)*

  - Others …

- **All fully-coupled formulations (monolithic)**

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Structural Acoustics Formulation

- **Applied two-field formulation of Everstine[1]**
  - Structural displacement
  - Fluid velocity potential
- **Exterior problems straightforward**
  - Compared to other formulations
- **Symmetric, indefinite matrices**
  - Best suited for domain decomposition-based solvers
- **Results in 2^{nd} order equations**
  - Compatible with Newmark beta and alpha time integration
- **Added by Tim Walsh beginning in 2003**

[1] G. C. Everstine, "Finite Element Formulations For Structural Acoustics Problems," *Computers & Structures* **65**: 307-321, (1997).

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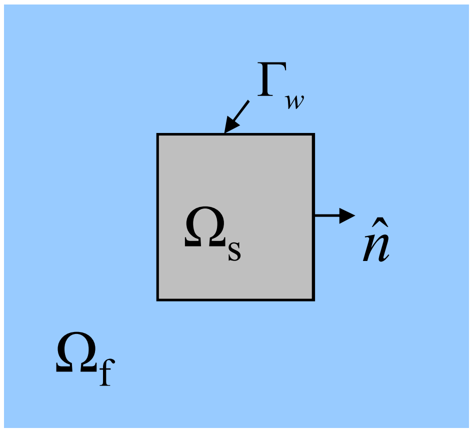
Structural Acoustics Formulation



Structure: $\rho_s \dfrac{\partial^2 \vec{u}}{\partial t^2} - \vec{\nabla} \cdot \tau = \vec{f}(\vec{x}, t) \quad \Omega_s \times [0, T]$

Fluid: $\nabla^2 \varphi - \dfrac{1}{c^2} \dfrac{\partial^2 \varphi}{\partial t^2} = 0 \quad \Omega_f \times [0, T]$

Fluid-Structure B.C.'s:

$$\tau \cdot \hat{n} = -\frac{\partial \varphi}{\partial t}$$

$$\rho_f \frac{\partial \vec{u}}{\partial t} \cdot \hat{n} = -\vec{\nabla} \varphi \cdot \hat{n}$$

- **Resulting time domain finite element form:**

$$\begin{bmatrix} M_s & 0 \\ 0 & \tilde{M}_f \end{bmatrix} \begin{Bmatrix} \ddot{u} \\ \ddot{\varphi} \end{Bmatrix} + \begin{bmatrix} C_s & L \\ L^T & \tilde{C}_f \end{bmatrix} \begin{Bmatrix} \dot{u} \\ \dot{\varphi} \end{Bmatrix} + \begin{bmatrix} K_s & 0 \\ 0 & \tilde{K}_f \end{bmatrix} \begin{Bmatrix} u \\ \varphi \end{Bmatrix} = \begin{Bmatrix} f_s \\ \tilde{f}_f \end{Bmatrix}$$

Coupling occurs
in damping matrix

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Structural Acoustics Solvers/Capabilities

- **Full massively parallel functionality**
- **Hex, wedge, and tetra acoustic elements**
- **Acoustic coupling with 3D and shell (2D) structural elements**
- **Allows for mismatched acoustic/solid meshes**
  - Inconsistent Tying
  - Standard Mortars
- **Solution Procedures:**
  - Frequency Response (frequency-domain)
  - Transient (time-domain)
  - Eigenvalue Analysis (real and quadratic)
  - Material, shape and force inversion (joint work with Wilkins Aquino at Duke)
- **Nonlinear Acoustics – Kuznetsov Equation**
  - Recently coupled to linear structures

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Quadratic Eigenvalue Problem

- **Eigenanalysis formulation:**

$$\lambda^2 \begin{bmatrix} M_s & 0 \\ 0 & \tilde{M}_f \end{bmatrix} \begin{Bmatrix} u \\ \varphi \end{Bmatrix} + \lambda \begin{bmatrix} C_s & L \\ L^T & \tilde{C}_f \end{bmatrix} \begin{Bmatrix} u \\ \varphi \end{Bmatrix} + \begin{bmatrix} K_s & 0 \\ 0 & \tilde{K}_f \end{bmatrix} \begin{Bmatrix} u \\ \varphi \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

  - Coupling within damping matrix brings about complex eigenvalues for structural acoustics (non-diagonalizable)

- **Solve by converting to state-space form:**

$$\begin{bmatrix} M & 0 \\ 0 & K \end{bmatrix} \{w\} = \begin{bmatrix} 0 & M \\ -M & -C \end{bmatrix} \{\dot{w}\} \quad \text{where } w = \begin{Bmatrix} \dot{r} \\ r \end{Bmatrix}$$

- **Depending on BC's, must solve both right *and* left eigenvalue problem**

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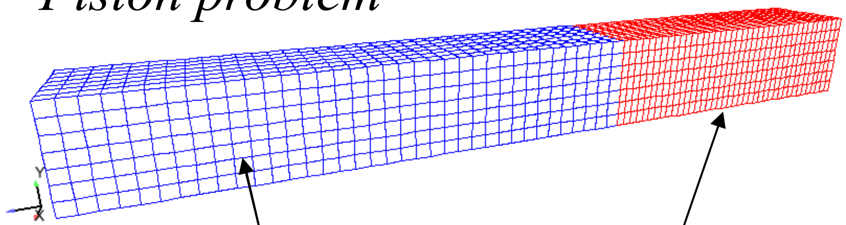
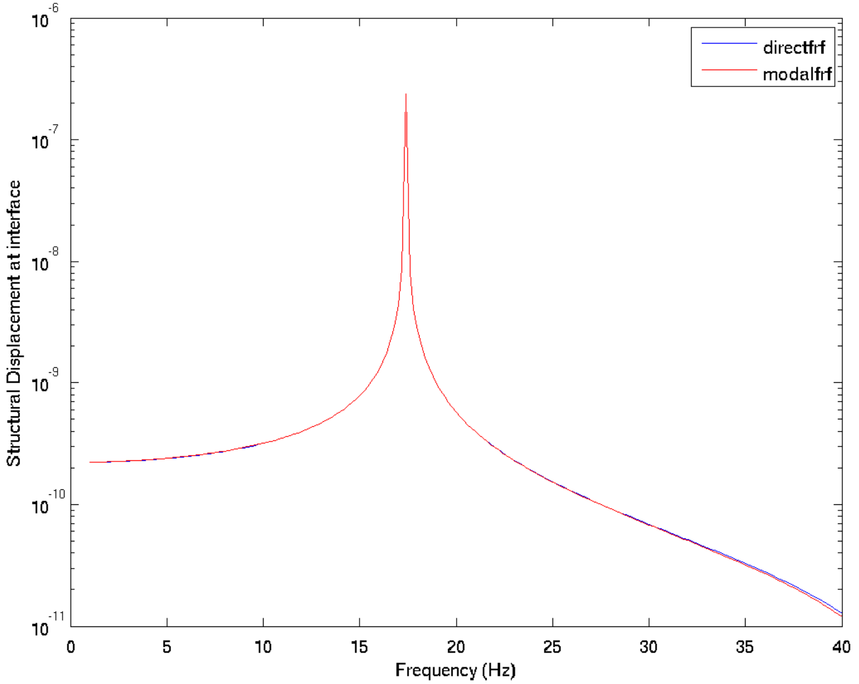
Complex Eigenvalue Modal Analysis

Piston problem



A comparison of structural displacement from directFRF vs CmodalFRF



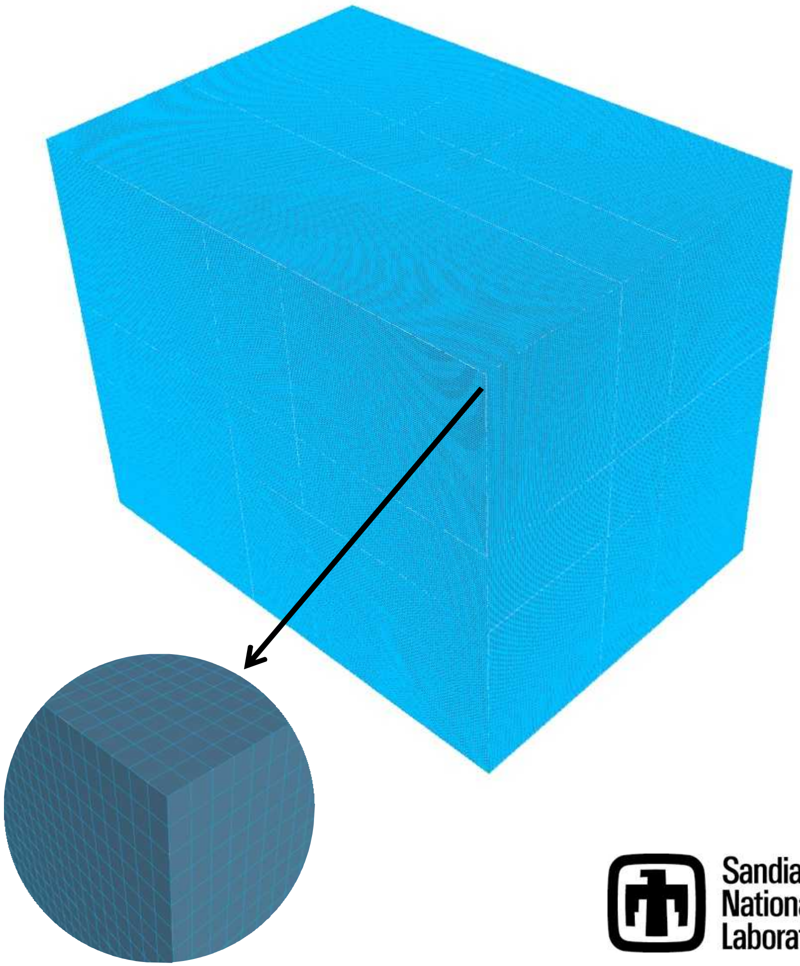
- **DirectFRF:**

$$u(\omega) = \frac{F(\omega)}{-\omega^2 [M] + i\omega [C] + [K]}$$

- **ComplexModalFRF:**
  - Use complex modes from quadratic eigenvalue solution

Transient Excitation of Reverb Chamber

- **16,000 ft^3 reverb chamber**
  - Wall BCs consistent with real chamber
- **Meshed 10 ele / λ at 1 kHz**
  - ~ 11.33 million nodes
- **Excited with 1 kHz sine**
  - 1000 time steps at dt = 0.0001 s
- **Used 800 processors**
  - Took 15 minutes to complete



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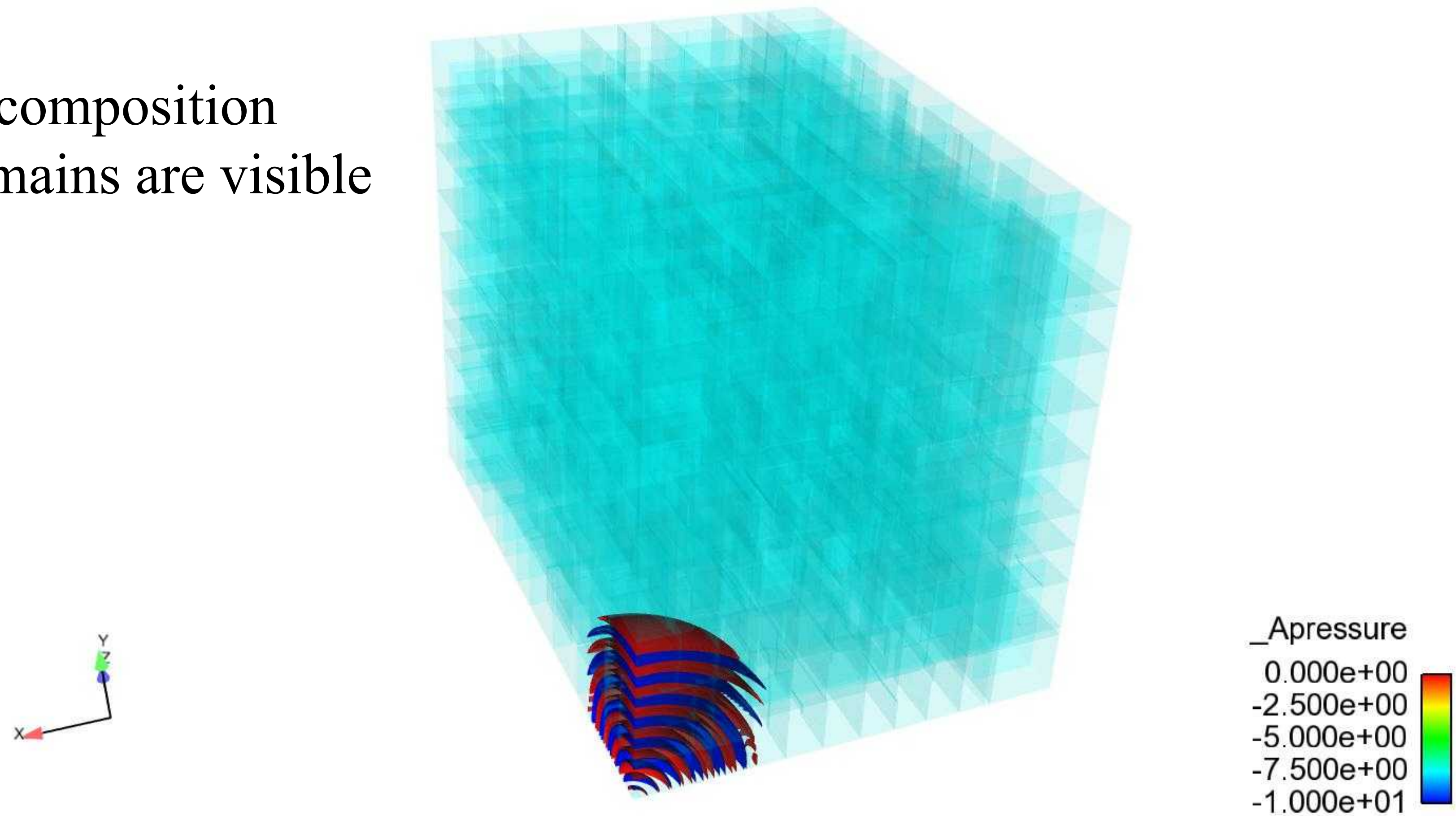
Transient Excitation of Reverb Chamber

Decomposition domains are visible



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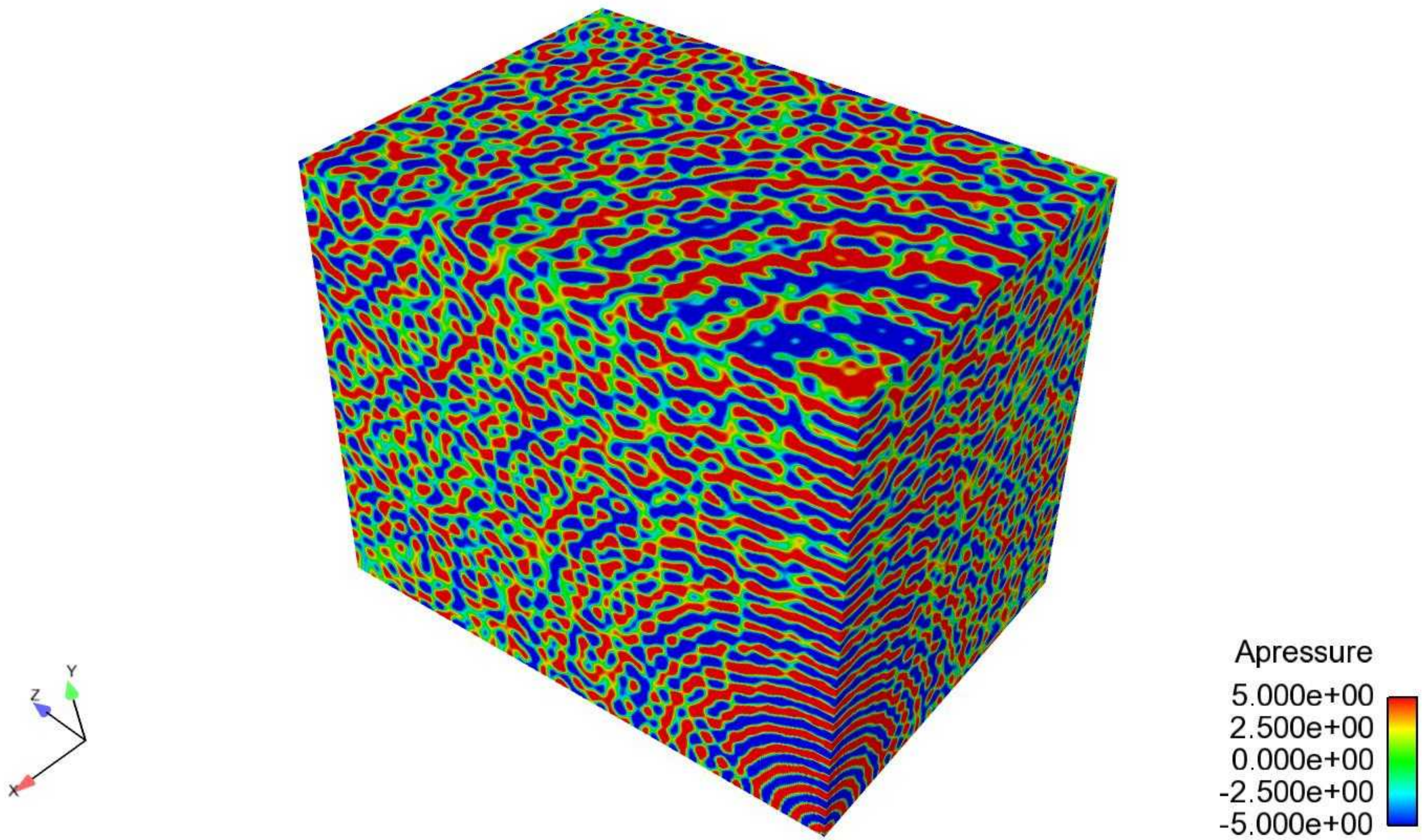
Transient Excitation of Reverb Chamber



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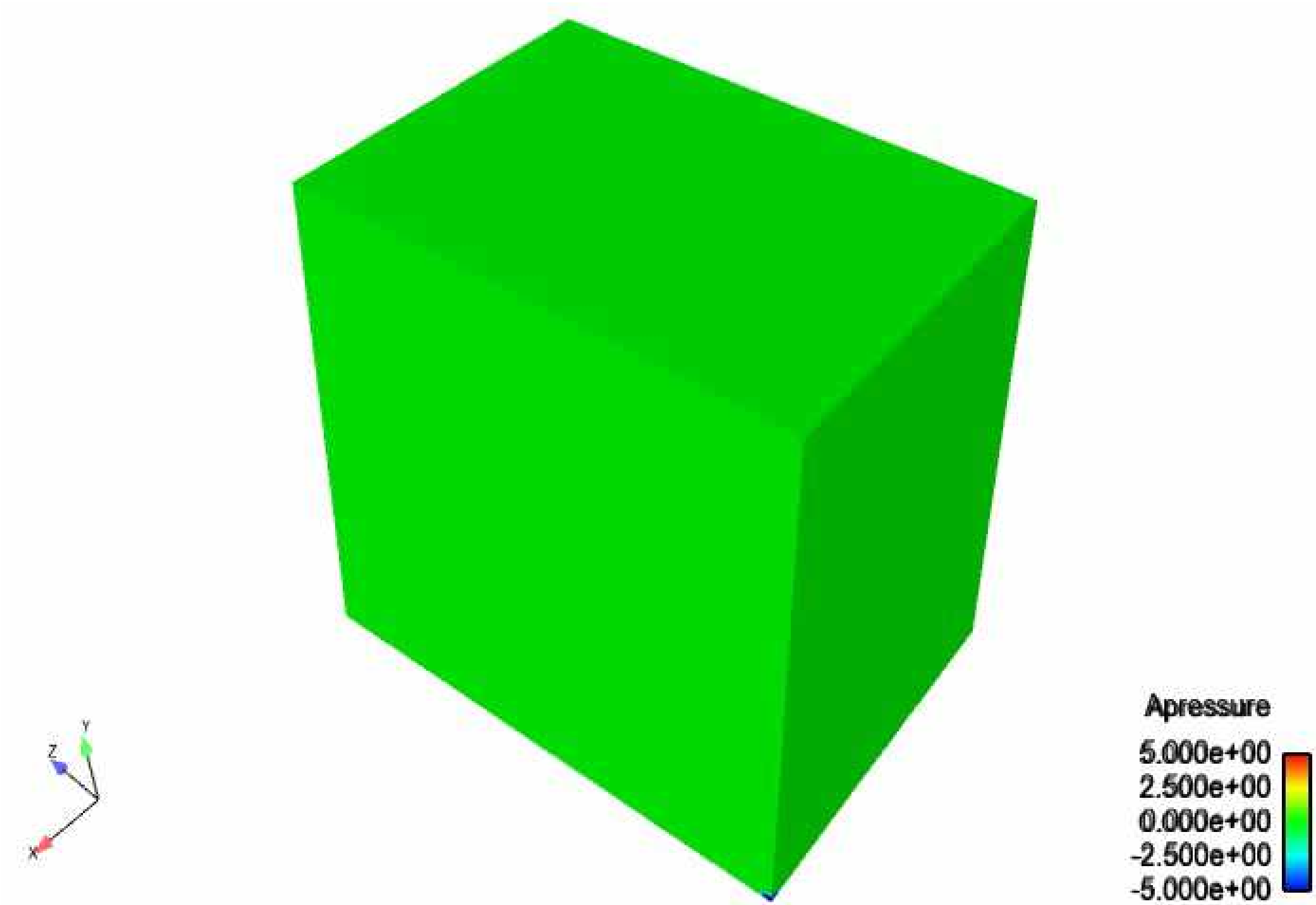
Transient Excitation of Reverb Chamber



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Future Capabilities of Sierra/SD

- Develop parallel solver for structural acoustic Helmholtz equation
- Extend inverse methods to structural acoustics for both time and frequency domain
- Explore special elements for high frequency acoustics
- GDSW three-level parallel solver for problems requiring over 100,000 processors (available now)

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Overall Conclusions

- Sandia has history of vibroacoustic testing
- Current 16,000 ft^3 acoustic reverb chamber
- Research into SIMO Direct Field Acoustic Testing
- Results drove improved test methodology (MIMO)
- Massively Parallel Structural Dynamics and Acoustics Finite Element Software
- Currently pushing the limits of FEA

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Backup Slides

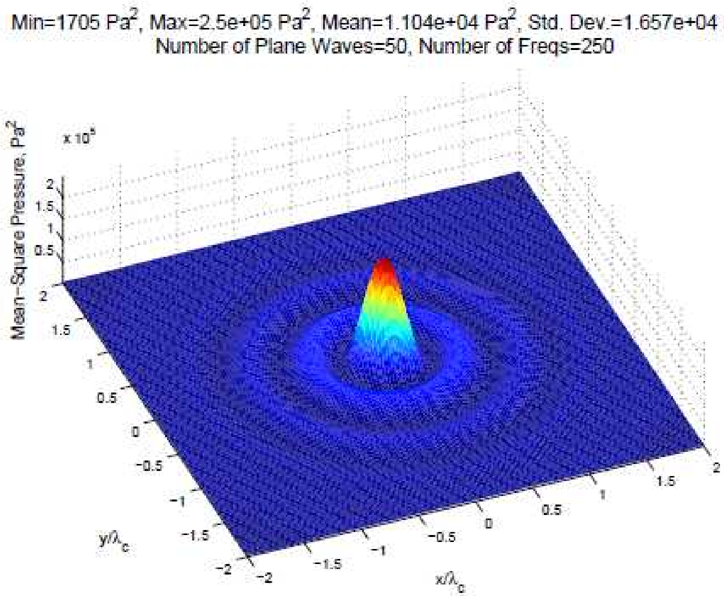
- **Backup Slides**

Converged Broadband Direct Field

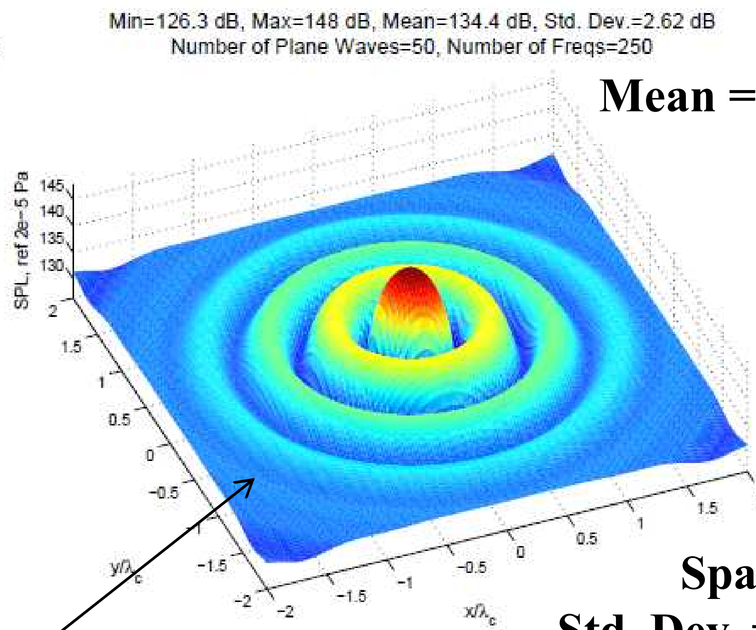
50 Sources, 1/3 Octave, 250 Frequencies in Band:

(P_o same as before)

Mean-Square Total Pressure: ⟶ ***Sound Pressure Level:***



(a) Mean-Square Pressure, Pa^2



(b) Sound Pressure Level, dB

Mean = 134.4 dB

Spatial Std. Dev. = 2.62 dB

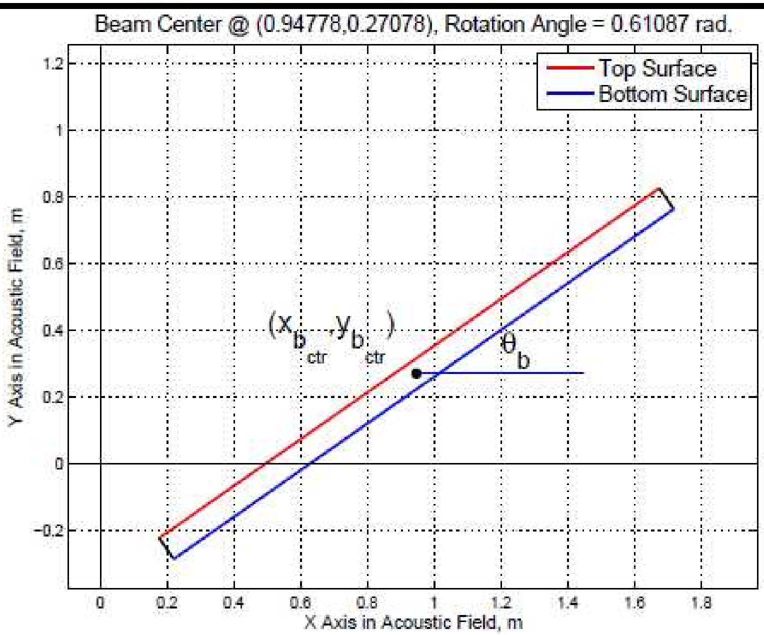
Note: higher mean, std. dev. lower, however, higher center peak

λ_c is wavelength at center frequency of band

Transverse Response of Bernoulli-Euler Beam

$$EI\frac{\partial^4 w}{\partial x_b^4} + \rho A\frac{\partial^2 w}{\partial t^2} = \tilde{F}(x_b, t)$$

$$w(x_b, \omega_0, t) = \sum_{r=1}^{\infty} \tilde{C}_r \sin\left(\frac{r\pi x_b}{L}\right) e^{i\omega_0 t}$$



- **Locate finite simply-supported beam within fields**
  - **Simple structure for analysis**
- **Compare mid-span RMS acceleration response due to direct and diffuse acoustic field loading**

Assumptions for Fluid-Structure Interaction

- 1. Incident plane waves having a non-zero normal component to the beam surface shall double in amplitude at the surface to account for constructive interference with the reflected wave.
- 2. The beam shall be modeled as if baffled; the propagating plane waves are not allowed to pass over the beam (out of the plane).
- 3. Acoustic loading on the ends of the beam (which would induce longitudinal response) is neglected.
- 4. One-way coupling by forcing fluid-loading parameter $\beta = \rho_0 c / \rho h \omega \ll 1$

Modal Analysis Solution

- **Time-harmonic acoustic field**

after much math, beam displacement…

Acoustic field frequency

Sum over beam modes

Sum over acoustic plane waves

$$w(x_b, \omega_0, t) = \frac{-2L^4 \pi P_o}{\rho h \sqrt{N}} \mathrm{Re}\left\{ \sum_{r=1}^{\infty} \left[\frac{r \sin(\frac{r\pi x_b}{L})}{a^2 r^4 \pi^4 - L^4 \omega_0^2} \sum_{n=1}^{N} \left\{ \frac{e^{i\phi_n}(1 - e^{-ikL\hat{n}_n \cdot \hat{t}}(-1)^r)}{\pi^2 r^2 - L^2 k^2 (\hat{n}_n \cdot \hat{t})^2} \cdot \left(g_{top}(\hat{n}_n) e^{-ik\hat{n}_n \cdot \vec{X}_{top}} - g_{bottom}(\hat{n}_n) e^{-ik\hat{n}_n \cdot \vec{X}_{bottom}} \right) \right\} \right] e^{i\omega_0 t} \right\}$$

Location along beam

Invokes Assumption 1 (doubling at surface)

Beam rotation θ_b removed for clarity

- **Time average as before and extend to broadband**

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Beam and Fluid Properties

- **For beam (softened 6061-T6 aluminum):**
  - **Length = 1.8288 m**
  - **Density = 2700 kg/m^3**
  - **Young's modulus = 70E6 Pa**
- **For fluid:**
  - **Sound speed = 343 m/s**
  - **Density = 1.19 kg/m^3**
- **1/3 Octave band centered at 400 Hz**
  - **Beam resonances in band: 374 and 426 Hz**
- **Fluid-loading parameter:**

$$\beta_{max} = \rho_0 c / \rho h 2\pi f_l = 8.8604 \times 10^{-4} \ll 1$$

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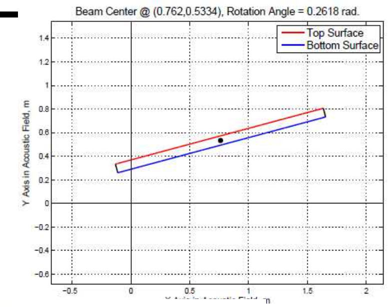
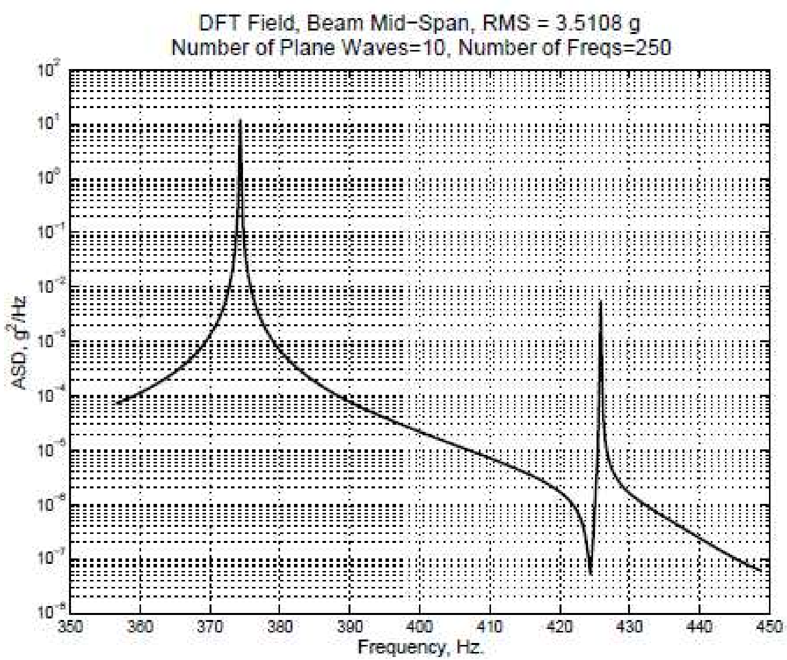
Mid-Span RMS Accel. Comparison

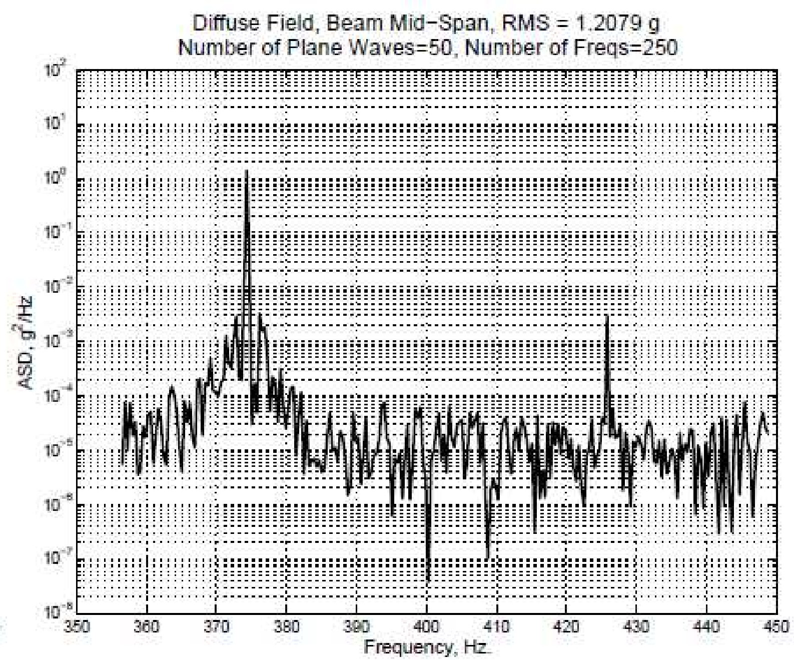
- Beam center (0.76, 0.53), θ_b= 15 deg.
- 10 Sources, 1/3 Octave, 250 Frequencies
- Same plane wave amplitudes for both fields





(a) Direct Acoustic Field Loading

RMS = 3.5 g



(b) Diffuse Acoustic Field Loading

RMS = 1.2 g

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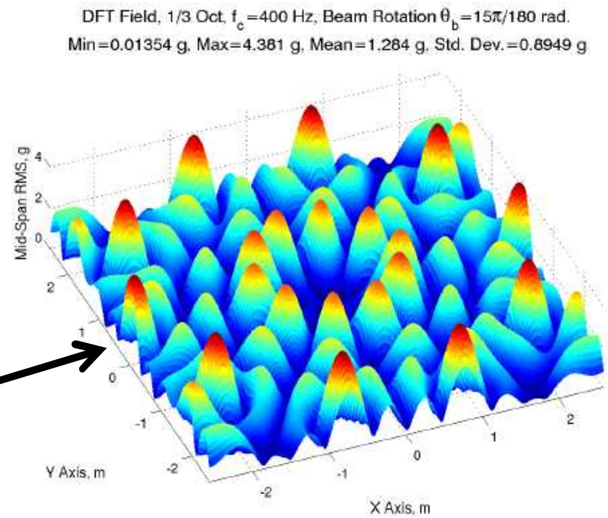
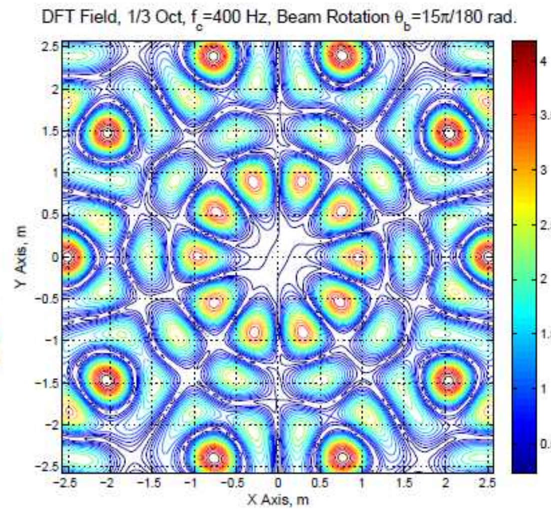
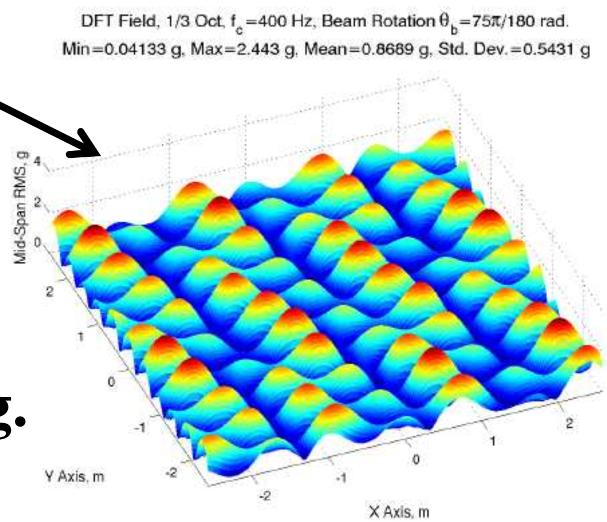
Mid-Span RMS Accel. over DFAT Field

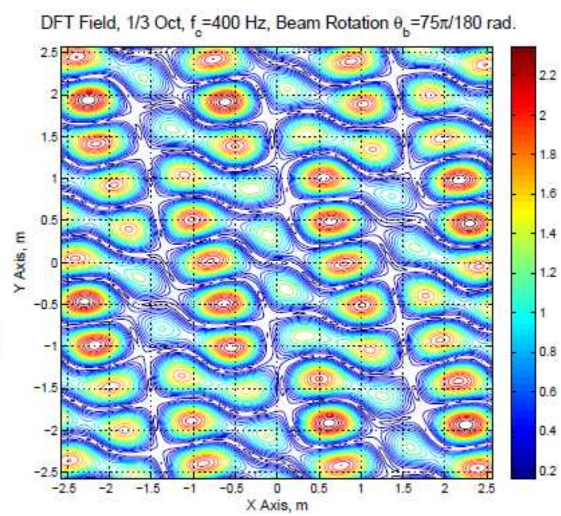
θ_b= 15 deg.





Mid-Span RMS accel. of beam when placed at specific points in DFAT field





θ_b= 75 deg.

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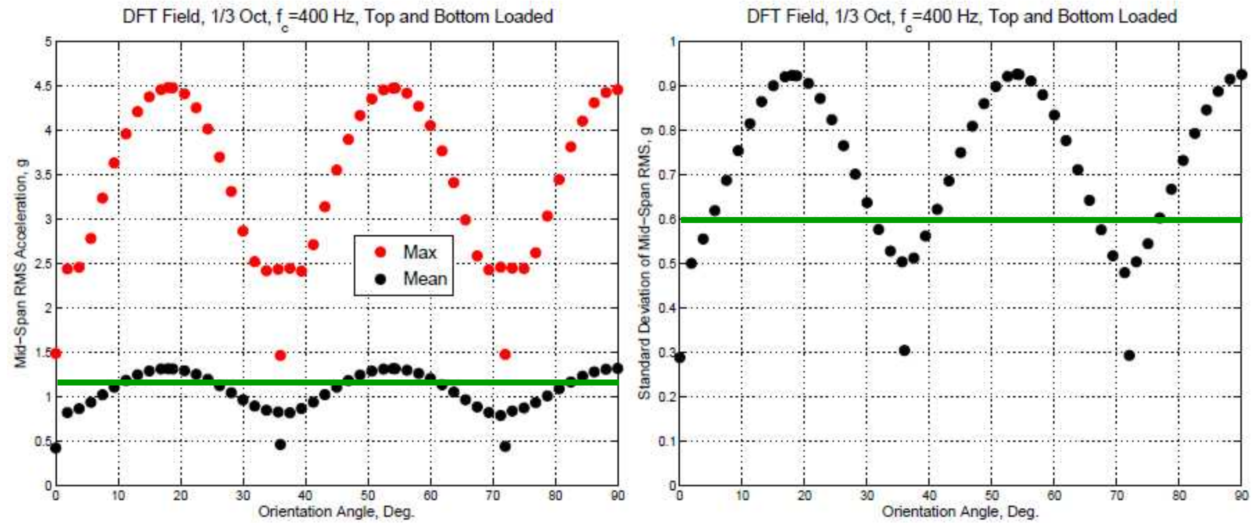
RMS Response Variation with Orientation

Direct acoustic field:

For each orientation angle: mean, max and std. dev of beam response over entire field.

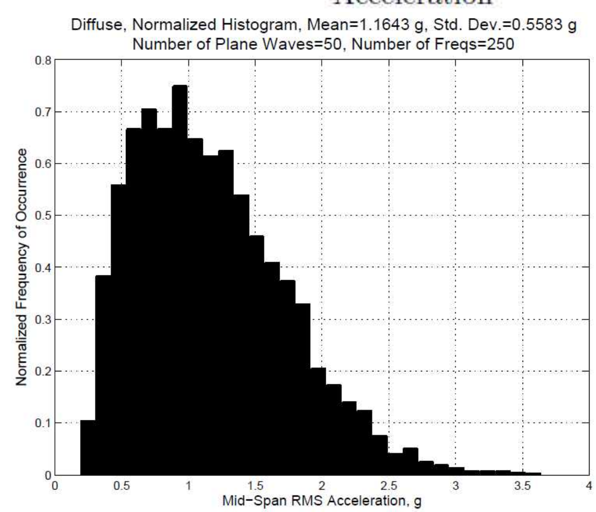


(a) Mean and Maximum Root-Mean-Square Acceleration

(b) Standard Deviation of Root-Mean-Square Acceleration

For diffuse field, marginal probability of response same for all locations and orientations:

(Mean = 1.16 g, Std. Dev = 0.6 g)



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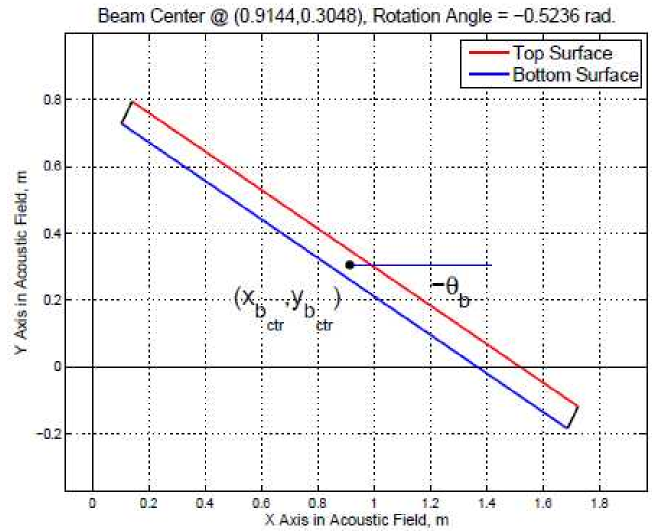
Response Variation with Only Top Loaded



- **Compare mid-span response with only the top surface of beam loaded by acoustic fields**
  - **- Simulates excitation of exterior surface of structure**
- **Orientation angles range from -90 to 0 deg.**
- **Restrict beam location to Quad I of acoustic field**

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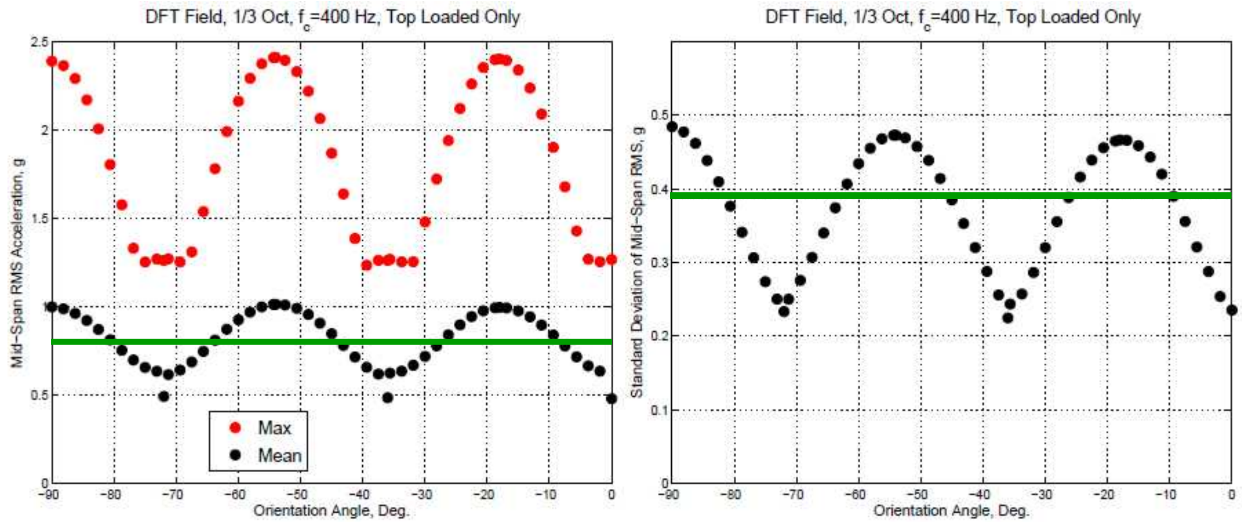
RMS Response Variation with Orientation

Direct acoustic field,

<u>top only loaded</u>:

For each orientation angle: mean, max and std. dev of beam response over entire field.

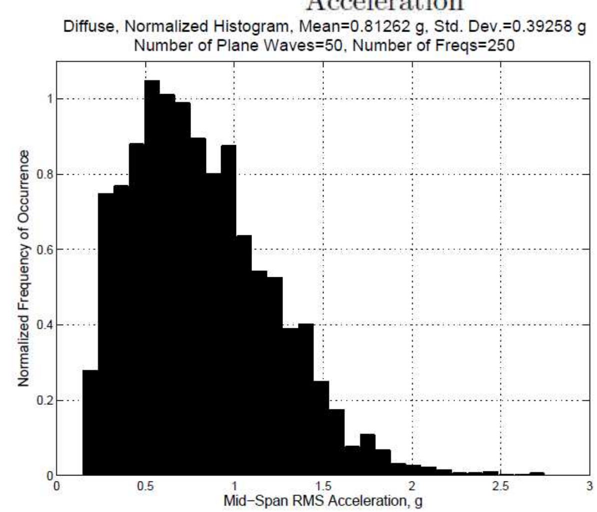


(a) Mean and Maximum Root-Mean-Square Acceleration

(b) Standard Deviation of Root-Mean-Square Acceleration

For diffuse field, again marginal probability of response same for all locations and orientations:

(Mean = 0.81 g, Std. Dev. = 0.4 g)



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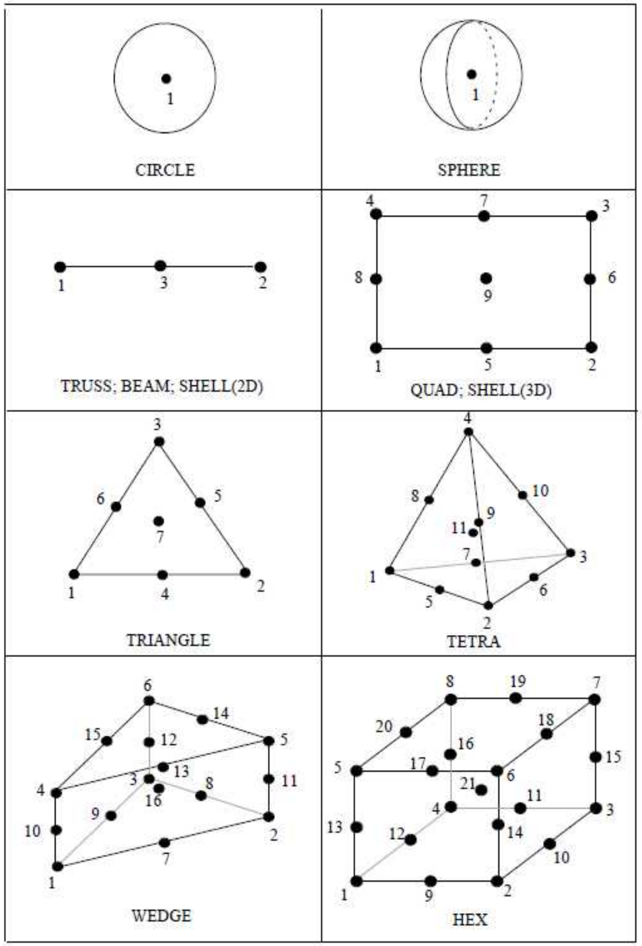
Large Element Library

- **Solid Elements**
  - Hexahedral, Tetrahedral, Wedge
- **Shell Elements**
  - Triangle, Quadrilateral, HexShell (hybrid)
- **Bar/Beam Elements**
  - Beam, Truss, Spring, Dashpot
- **Point Elements**
  - Conmass (concentrated mass)
- **Specialty Elements**
  - Iwan, Hys, Shys, Joint2G, Gap



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