Structural-Acoustic Tutorial

Part II – Structural-Acoustic Finite Element Analysis & Automotive Applications

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Orlando, Florida

Overview

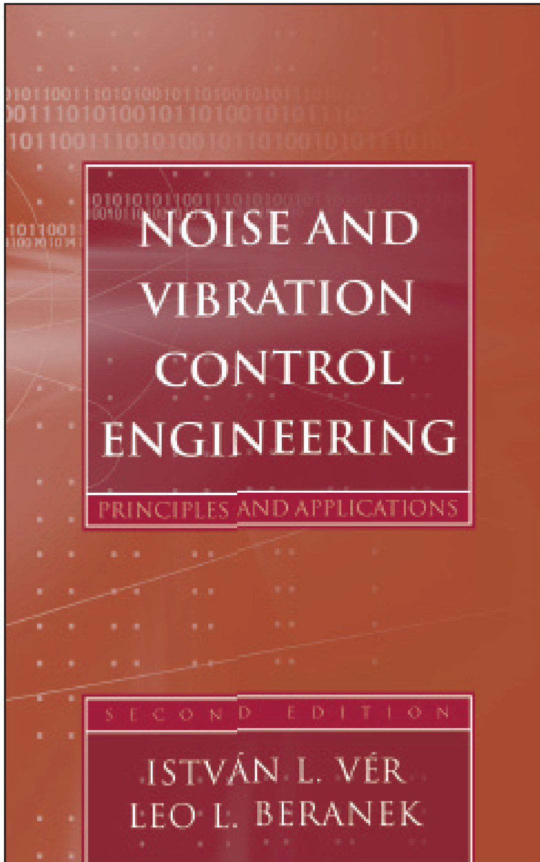
- Instructor
- Acoustic Finite Element Analysis
  - FE Formulation of Equations-of-Motion
  - Cavity Modal Analysis
  - Flexible Wall Excitation (FRFs & Panel Participation)
- Acoustic Absorption & Interior Trim
  - Acoustic Damping Models
  - Example Vehicle Applications
- Structural-Acoustic Finite Element Analysis
  - Coupled Structural-Acoustic FE Formulation
  - Modal Response, Modal and Panel Participation
  - Flexible Wall Coupling Effect
  - Vehicle Validation
  - Example Vehicle Application

Instructor

- Dr. Shung H. (Sue) Sung
  - B.S. - Civil Engineering, National Taiwan University
  - M.S. & Ph.D. - Aero & Astro Engr., Purdue University
  - General Motors R&D Center – Technical Staff 32 years
  - ASME Fellow, AIAA, SAE, ASA, INCE Member

- Major Research
  - Structural-acoustic finite element analysis for vehicle CAE N&V design
  - Vehicle and engine FE modeling for radiated noise prediction
  - Power flow FE method for high frequency N&V Analysis
  - Statistical regression energy method for early vehicle N&V design

Major Source



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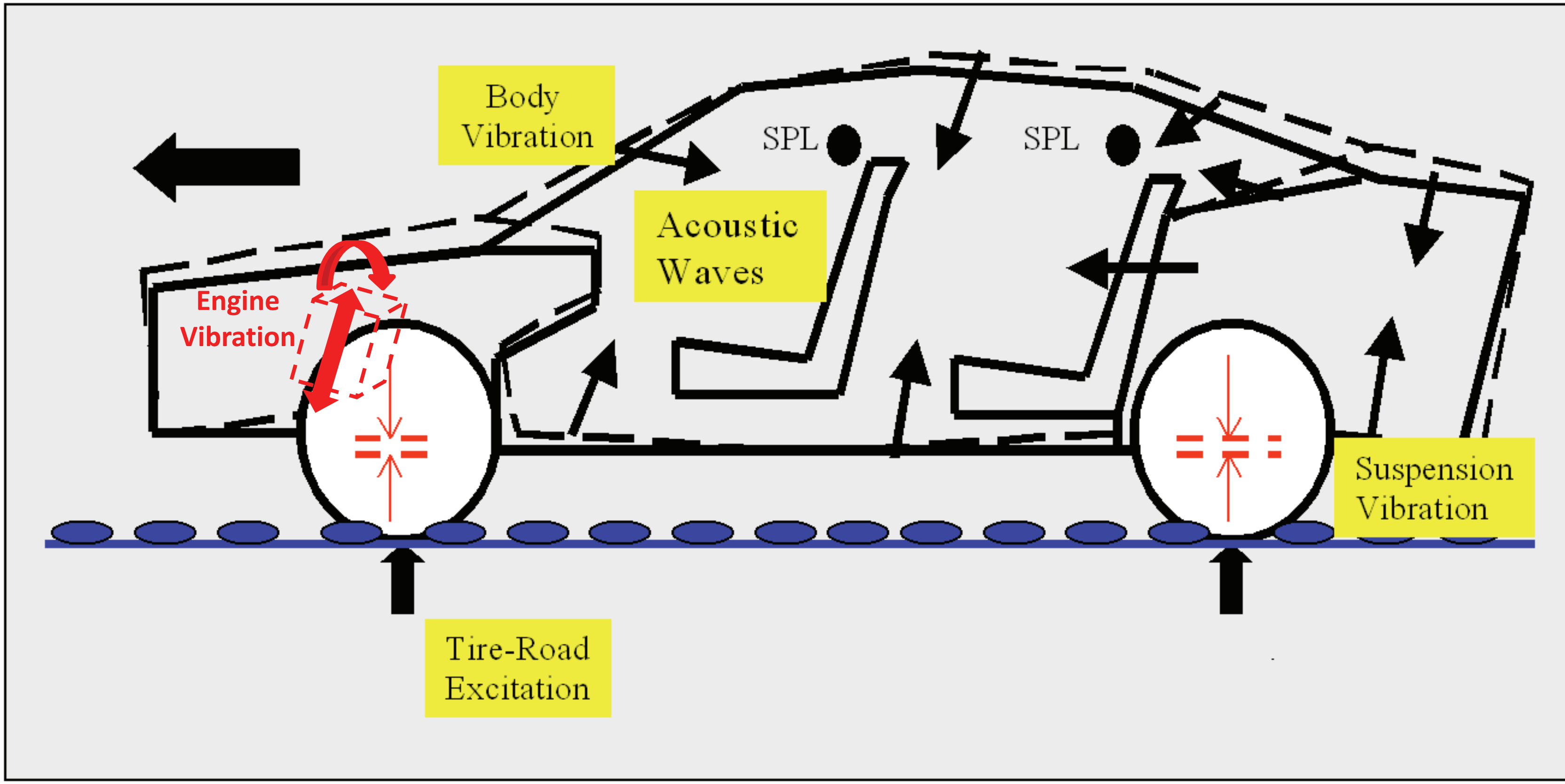
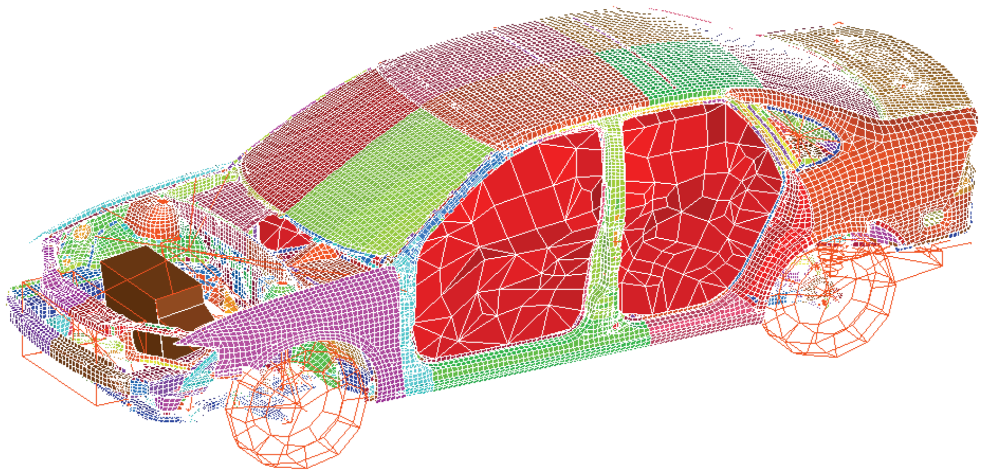
Contents:

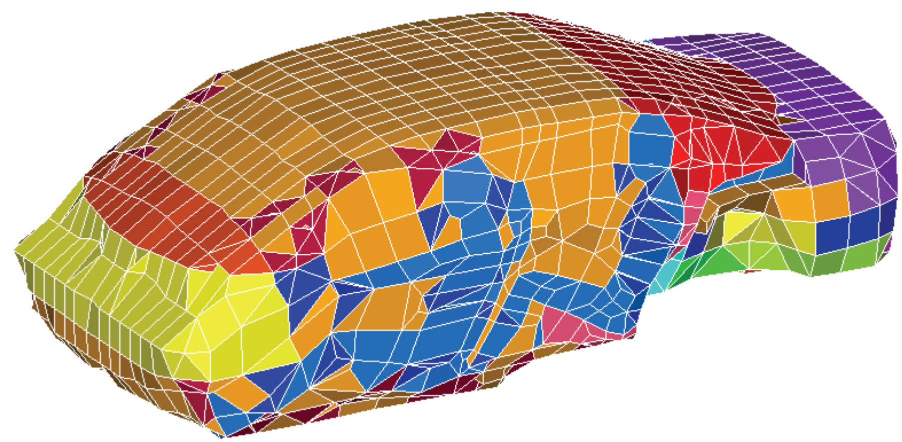
Vehicle Interior Noise Paths



Vehicle Structural & Acoustic Models

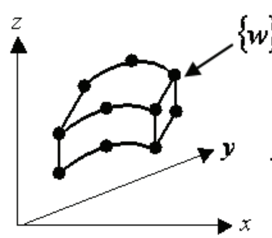
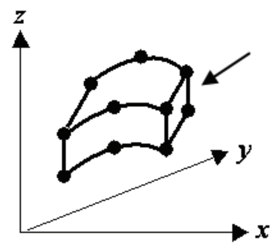


Vehicle Structural Model

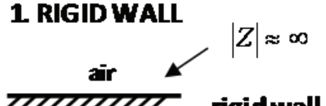
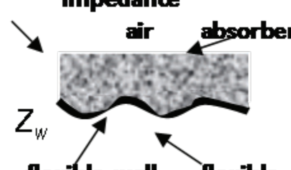


Compartment Acoustic Model

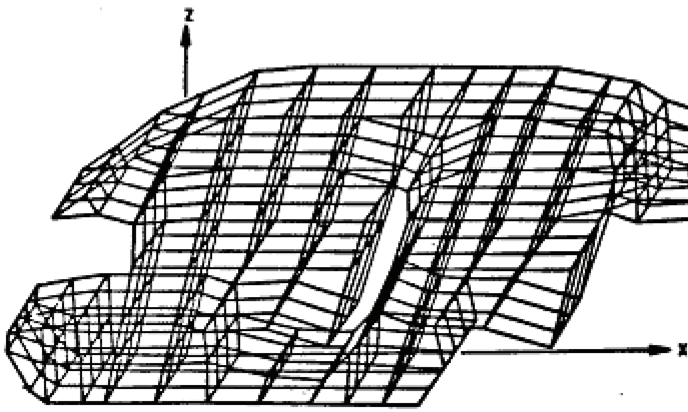
Structural-Acoustic FE Analysis

DESCRIPTION	STRUCTURE	ACOUSTIC CAVITY
FINITE ELEMENT	 $\vec{w}(\vec{r},t)=\{N\}^T\{w\}$	 $p(\vec{r},t)=\{\theta\}^T\{p\}$
CONSTITUTIVE EQUATIONS	$\begin{pmatrix}\sigma_{xx}\\ \tau_{xy}\\ \tau_{xz}\end{pmatrix}=\begin{pmatrix}G_{11} & G_{14} & G_{16}\\ G_{14} & G_{44} & G_{46}\\ G_{16} & G_{46} & G_{66}\end{pmatrix}\begin{pmatrix}\varepsilon_{xx}\\ \gamma_{xy}\\ \gamma_{xz}\end{pmatrix}$	$\begin{pmatrix}u_x\\ u_y\\ u_z\end{pmatrix}=\begin{pmatrix}-1/\rho & 0 & 0\\ 0 & -1/\rho & 0\\ 0 & 0 & -1/\rho\end{pmatrix}\begin{pmatrix}\partial p/\partial x\\ \partial p/\partial y\\ \partial p/\partial z\end{pmatrix}$
EQUILIBRIUM EQUATIONS	$div\,\sigma=-\rho_s\ddot{w}+f$	$div(\frac{1}{\rho_0}\nabla p)=\frac{1}{B_0}\ddot{p}+s$
BOUNDARY CONDITIONS	$\sigma_n=p$	$\frac{1}{\rho_0}\partial p/\partial n=-\ddot{u}_n=-\ddot{w}_n$
FINITE ELEMENT EQUATIONS	$([K]-\omega^2[M])\{w\}+[A]\{p\}=\{F\}$	$([H]-\omega^2[Q])\{p\}+\omega^2[A]^T\{w\}=\{q\}$
FINITE ELEMENT MATRICES	$[K]=\int_V\{\nabla N\}^T[G]\{\nabla N\}\,dV$ $[M]=\int_V\rho_s\{N\}\{N\}^T\,dV$ $[A]=\int_S\{N\}\{\theta\}^T\,dS$ $\quad\{F\}=\int_A\{f\}\{N\}^T\,dA$	$[H]=\int_\Omega\frac{1}{\rho_0}\{\nabla N\}^T\{\nabla N\}\,d\Omega$ $[Q]=\int_\Omega\frac{1}{B_0}\{N\}\{N\}^T\,d\Omega$ $[A]^T=\int_S\{\theta\}\{N\}^T\,dS$ $\quad\{q\}=\int_{A_0}\{s\}\{\theta\}^T\,dA_0$

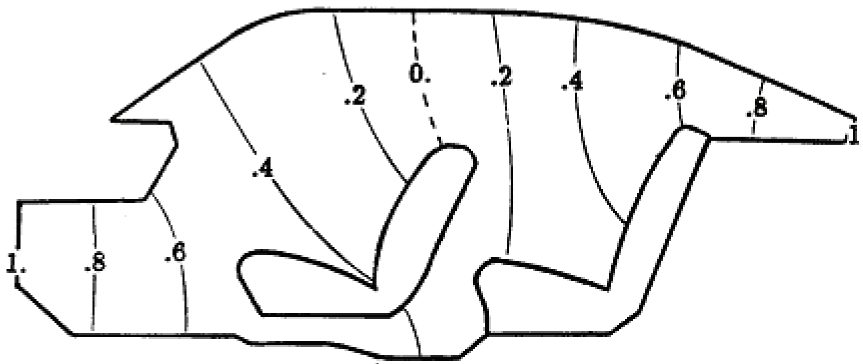
Acoustic Boundary Conditions

TYPE	BOUNDARY CONDITION	AIR PARTICLE VELOCITY
1. RIGID WALL $\lvert Z\rvert \approx \infty$ air, rigid wall 	$\frac{\partial p}{\partial n} = 0$	$u = 0$
2. FLEXIBLE WALL $\dot{w}$ – wall surface-normal velocity; air, flexible wall 	$\frac{1}{\rho}\frac{\partial p}{\partial n} = -\ddot{u}$	$u = \dot{w}$
3. ABSORBER ON RIGID WALL Z_a – absorber impedance; air, absorber, rigid wall 	$\frac{1}{\rho}\frac{\partial p}{\partial n} = -\frac{1}{Z_a}\frac{\partial p}{\partial t}$	$u = \frac{p}{Z_a}$
4. ABSORBER ON FLEXIBLE WALL Z_a – absorber impedance; Z_w – flexible wall impedance; air, absorber, flexible wall 	$\frac{1}{\rho}\frac{\partial p}{\partial n} = -\frac{1}{Z_a}\frac{\partial p}{\partial t} - \ddot{w} = -\left(\frac{1}{Z_a} + \frac{1}{Z_w}\right)\frac{\partial p}{\partial t}$	$u = \frac{p}{Z_a} + \dot{w} = \frac{p}{Z_a} + \frac{p}{Z_w}$
5. OPEN – PRESSURE RELEASE air 	$p = 0$	$u = 0$
6. OPEN – PLANE WAVE $Z_{air} = \rho c$; air 	$\frac{1}{\rho}\frac{\partial p}{\partial n} = -\frac{1}{Z_{air}}\frac{\partial p}{\partial t}$	$u = \frac{p}{Z_{air}}$

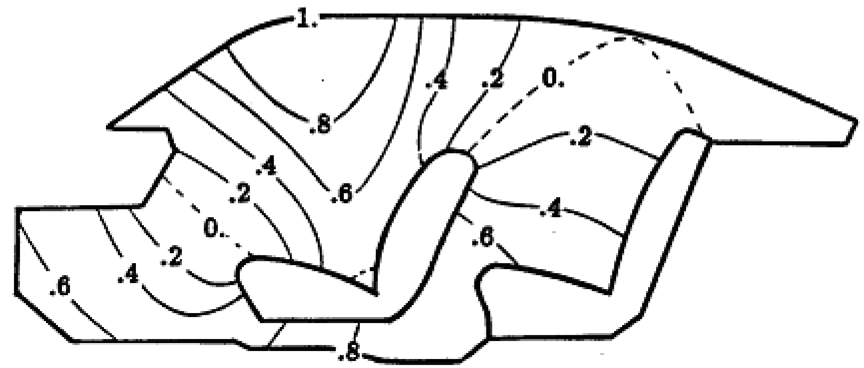
Compartment Cavity Modes



(a) Acoustic Finite Element Model



(b) First Resonant Mode at 73 Hz



(c) Second Resonant Mode at 130 Hz

RIGID WALL MODES

$$\left([H]-\omega^2[Q]\right)\{p\}=0$$

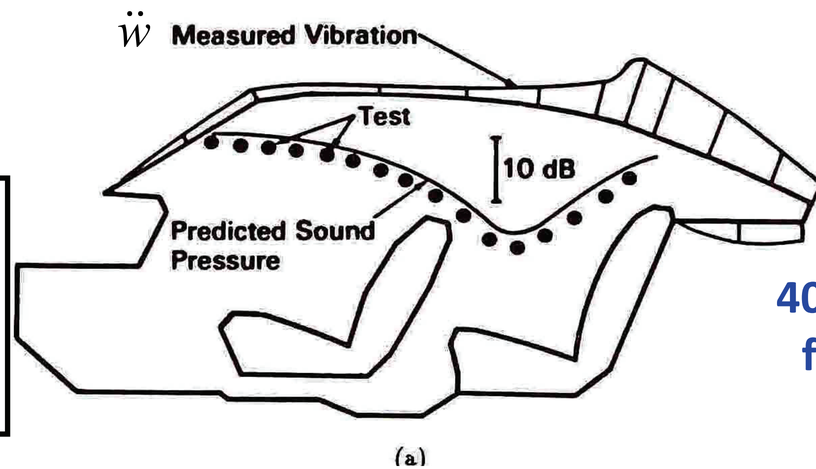
$$\frac{\partial p}{\partial n}=0$$

SPL & Panel Contributions Prescribed Structural Excitation

FORCED RESPONSE

$$([H]-\omega^2[Q])\{p\}=\{q\}$$

$$\{q\}=\int_{A_O}\{\ddot{w}\}\{\theta\}^T\,dA$$



(a) Interior SPL Spatial Variation

40 Hz Structural Vibration for Shaker Excitation of Vehicle

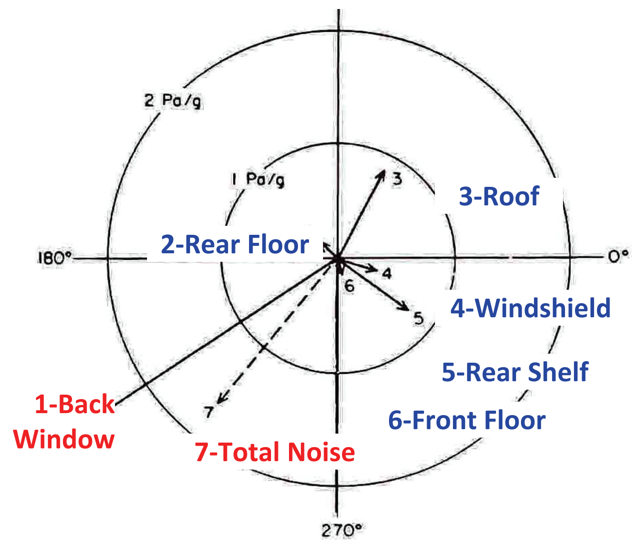
SUMMED PANEL FORCED RESPONSE

$$p=\sum_{i=1}^{L}p_i,$$

$$([H]-\omega^2[Q])\{p_i\}=\{q_i\},$$

$$\{q_i\}=\int_{A_i}\{\ddot{w}\}\{\theta\}^T\,dA_i,$$

$$A_0=\sum_{i=1}^{L}A_i$$

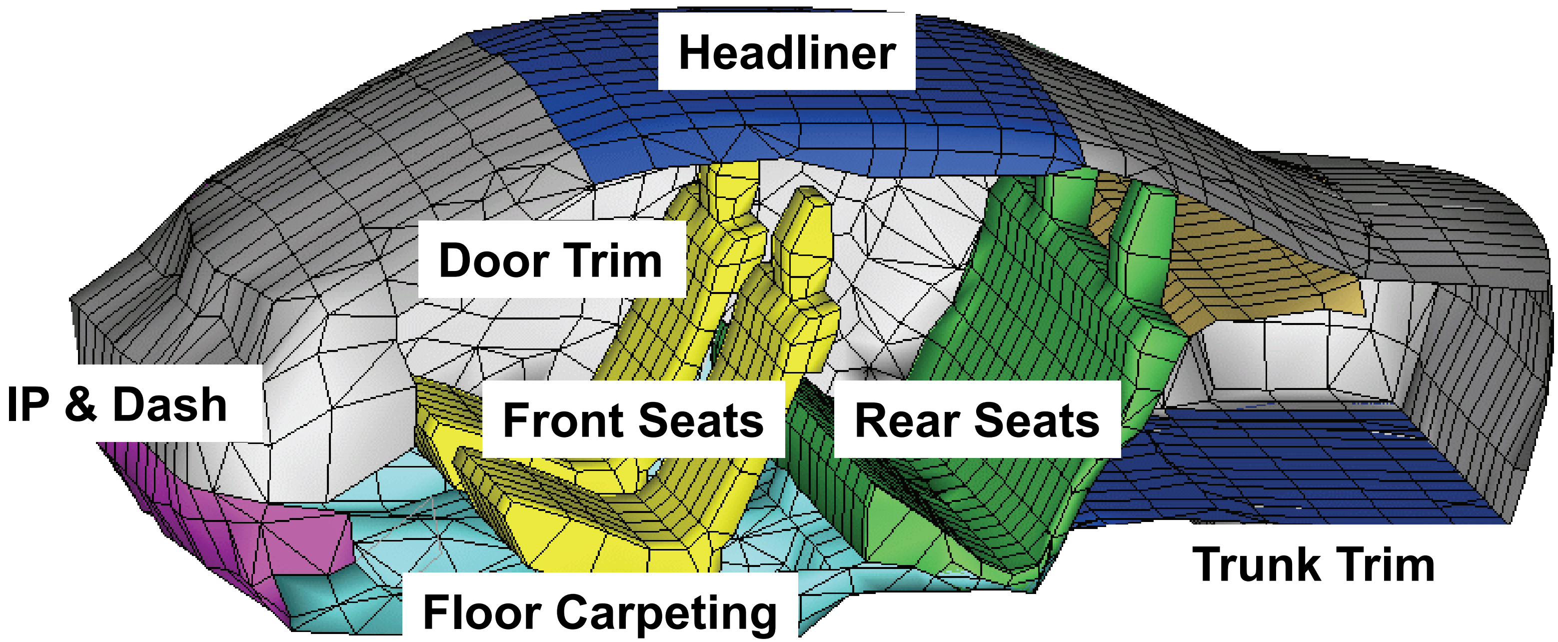


(b) Panel Contribution Polar Diagram

Sound Pressure Level (SPL) at the Driver's Ear

Vehicle Interior Acoustic Trim

Acoustic Finite Element Model

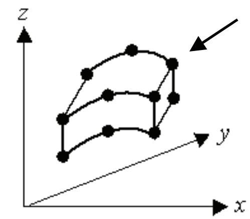


"Correlation of an Acoustic Finite Element Model of the Automobile Passenger Compartment Using Loudspeaker Excitation", S. H. Sung, D. J. Nefske, and D. A. Feldmaier, ASME Paper IMECE2007-42735.

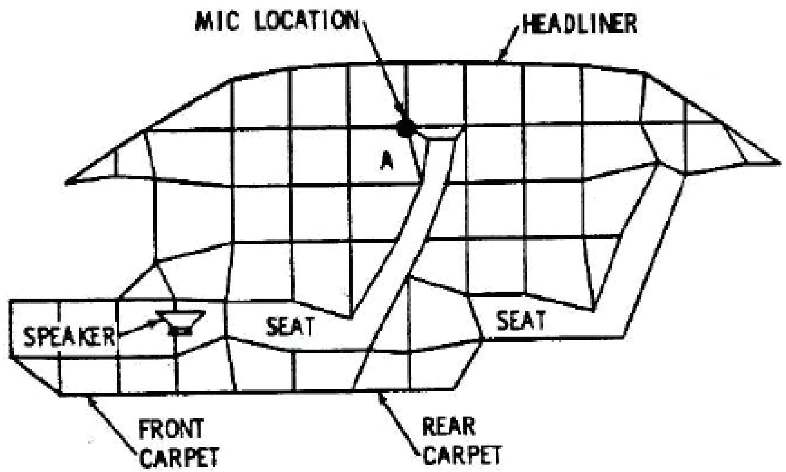
Acoustic Damping Modeling Methods

Item	FEM Modeling Method	Implementation	Modeling & Computation
Overall Acoustic Damping	Modal Damping (Interior Loss Factor/ Reverberation Time)	Estimated or Measured Modal Damping Frequency Dependent	Very Fast
Vehicle Seats	"Heavy Air" Equivalent-Acoustic Model 3D Acoustic Elements	Matched to Test Data Frequency Independent	Very Fast
Interior Trim (including Seats)	Acoustic Impedance Model CAABSF Elements	Measured Impedance Frequency Dependent	Medium Fast
Interior Trim (including Seats)	Equivalent-Acoustic Modeling 3D Acoustic Elements	Measured Impedance or Poroelastic Material Data Frequency Dependent	Medium Fast
Interiror Trim (including Seats)	Poroelastic Material Modeling Special 3D Element/Code	Measured Poroelastic Material Properties (RAYON, NOVA, ACTRAN, EXEL, etc)	Can be Time Consuming

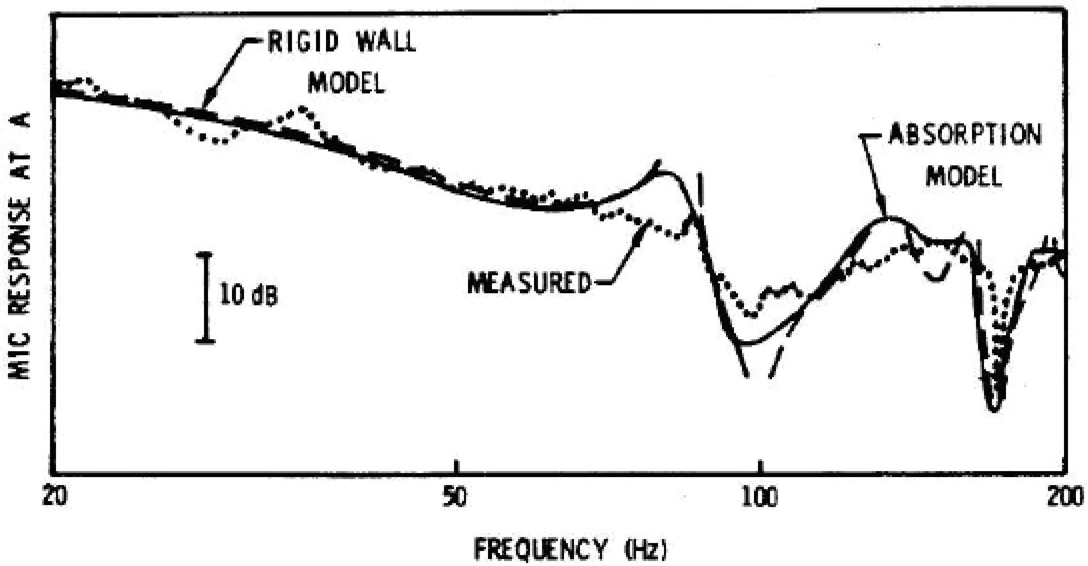
Acoustic Impedance FE Formulation

FINITE ELEMENT	 $\{u\},\{p\}$ $p(\vec{r},t)=\{\theta\}^T\{p\}$	
MOMENTUM, CONTINUITY EQUATIONS	$-\dot{\bar{u}}=\frac{1}{\rho_0}\nabla p$ $\quad -div\;\bar{u}=\frac{1}{B_0}\partial p/\partial t$	
EQUILIBRIUM EQUATION	$div(\frac{1}{\rho_0}\nabla p)=\frac{1}{B_0}\ddot{p}$	
BOUNDARY TYPE	**2. FLEXIBLE WALL** **w – wall surface-normal displacement** 	**4. ABSORBER ON FLEXIBLE WALL** **Z_a – absorber impedance** 
BOUNDARY CONDITION	$\frac{1}{\rho_0}\partial p/\partial n=-\ddot{w}$	$\frac{1}{\rho_0}\partial p/\partial n=-\frac{1}{Z_a}\partial p/\partial t-\ddot{w}$
FINITE ELEMENT EQUATIONS	$([H]-\omega^2[Q])\{p\}=\{q\}-[A]^T\{\ddot{w}\}$	$([H]-\omega^2[Q])\{p\}+i\omega[D]\{p\}=\{q\}-[A]^T\{\ddot{w}\}$
FINITE ELEMENT MATRICES	$[H]=\int_\Omega\frac{1}{\rho_0}\{\nabla N\}^T\{\nabla N\}\,d\Omega$ $\quad [Q]=\int_\Omega\frac{1}{B_0}\{N\}\{N\}^T\,d\Omega$ $\quad [D]=\int_A\frac{1}{Z_a}\{N\}\{N\}^T\,dA$	

Acoustic Impedance Model Application

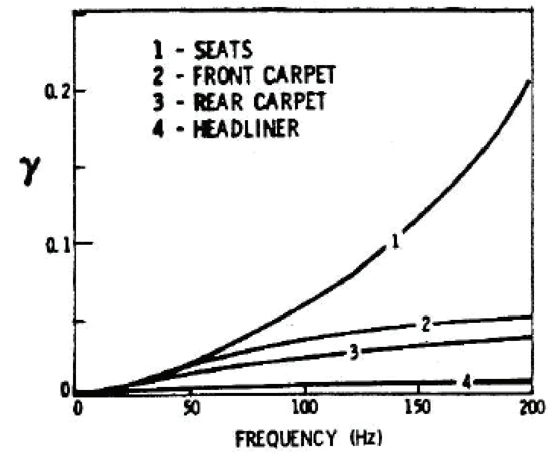


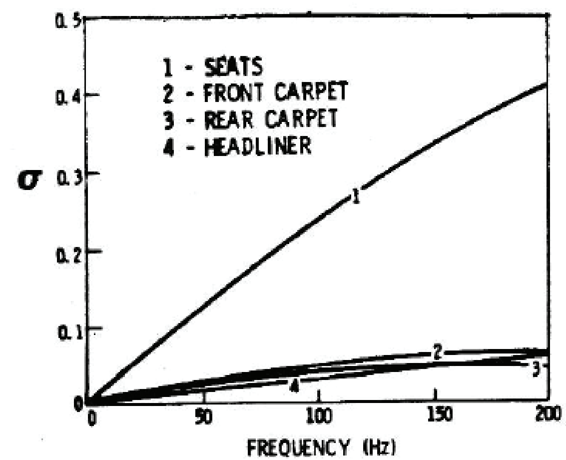
$$([H]-\omega^2[Q])\{p\}+i\omega[D]\{p\}=\{q\} \qquad [D]=\int_A \frac{1}{Z_a}\{N\}\{N\}^T\, dA$$



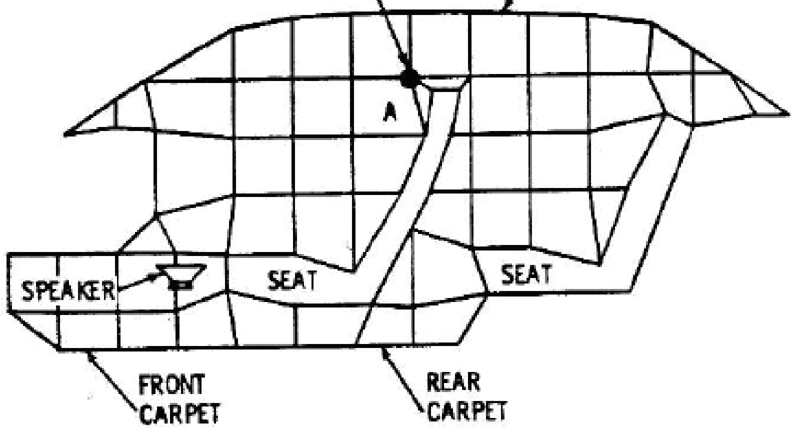
Measured Impedance/Admittance

$$\rho c/Z_a = \gamma + i\sigma$$

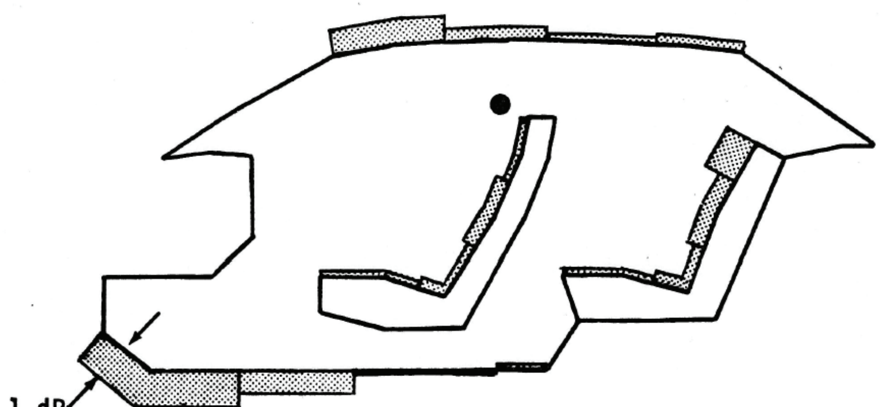




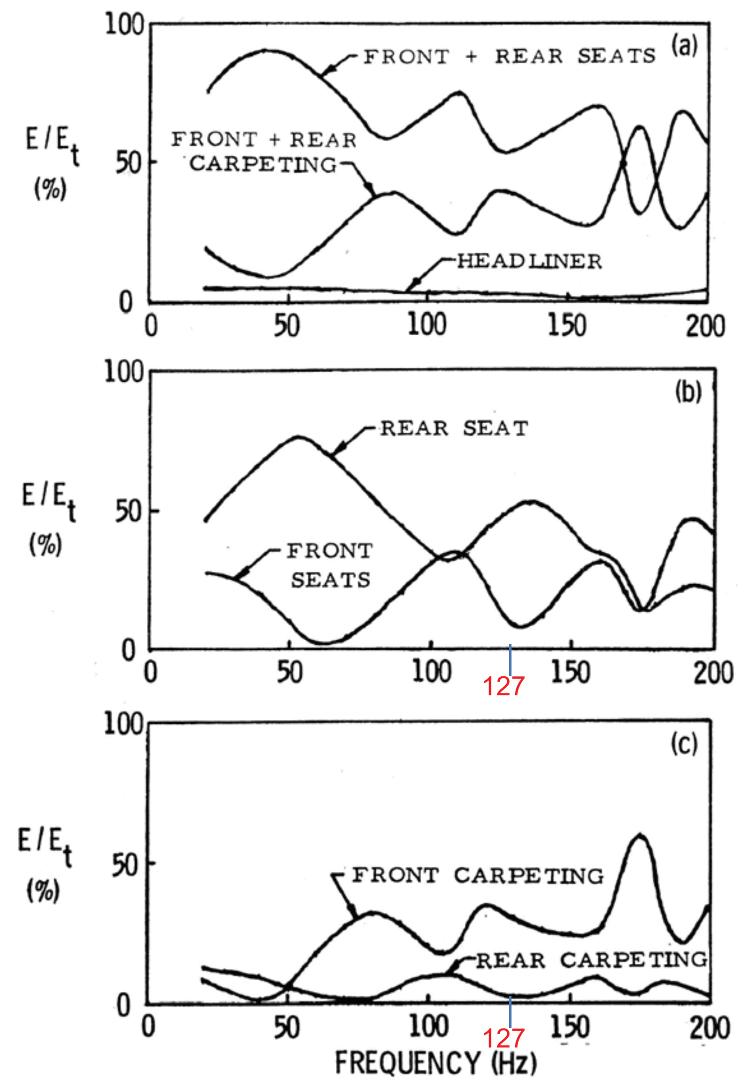
Acoustic Impedance Model Application



ABSORPTION INFLUENCE DIAGRAM FOR INTERIOR NOISE REDUCTION **AT 127 Hz**



PERCENT OF TOTAL ACOUSTIC ENERGY ABSORPTION BY SEATS, CARPETING, AND HEADLINER



Equivalent-Acoustic Trim Modeling

Equivalent-Acoustic Equation-of-Motion

$$\frac{1}{\rho_e(\omega)}\nabla^2 p_a + \frac{\omega^2}{B_e(\omega)} p_a = 0 \qquad \frac{\partial p_a}{\partial n} = -i\omega\rho_e(\omega)u_a$$

Equivalent-Acoustic Finite Element Equation

$$\left[\frac{1}{\rho_e(\omega)}[H_a] - \frac{\omega^2}{B_e(\omega)}[Q_a]\right]\{p_o\} = i\omega\{q_a\}$$

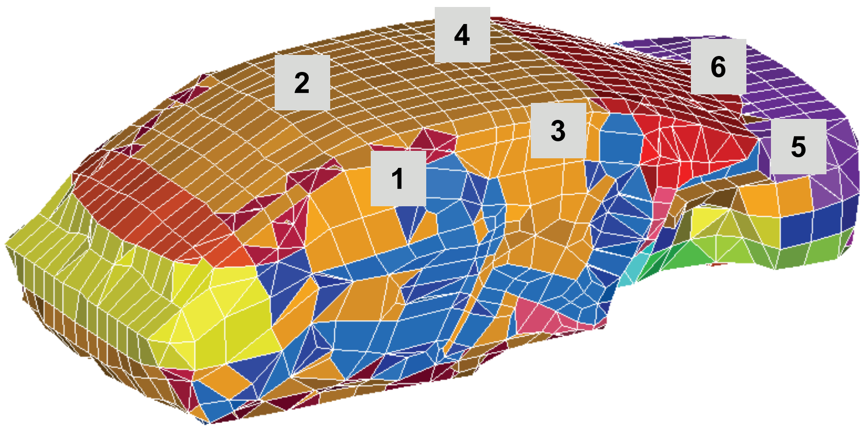
Effective Density & Bulk Modulus from Impedance

$$Z_c(\omega) = \rho_e(\omega)c_e(\omega) = \sqrt{\rho_e(\omega)B_e(\omega)}$$

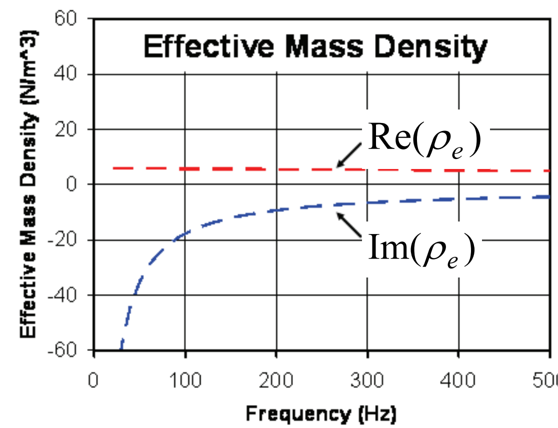
$$\rho_e(\omega) = -Z^2(\omega)\tan^2 k(\omega)l / B_e(\omega)$$

Equivalent-Acoustic Model Application

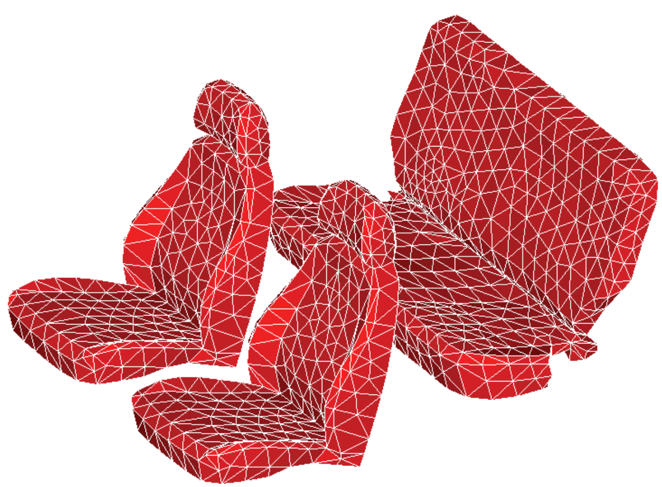
Acoustic Cavity FE Model

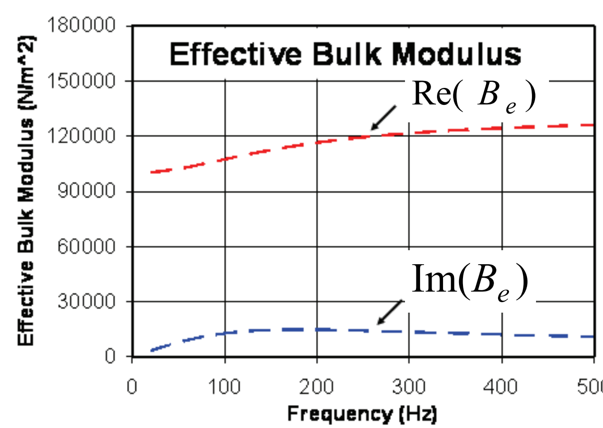


Effective Mass Density & Bulk Modulus from Measured Impedance



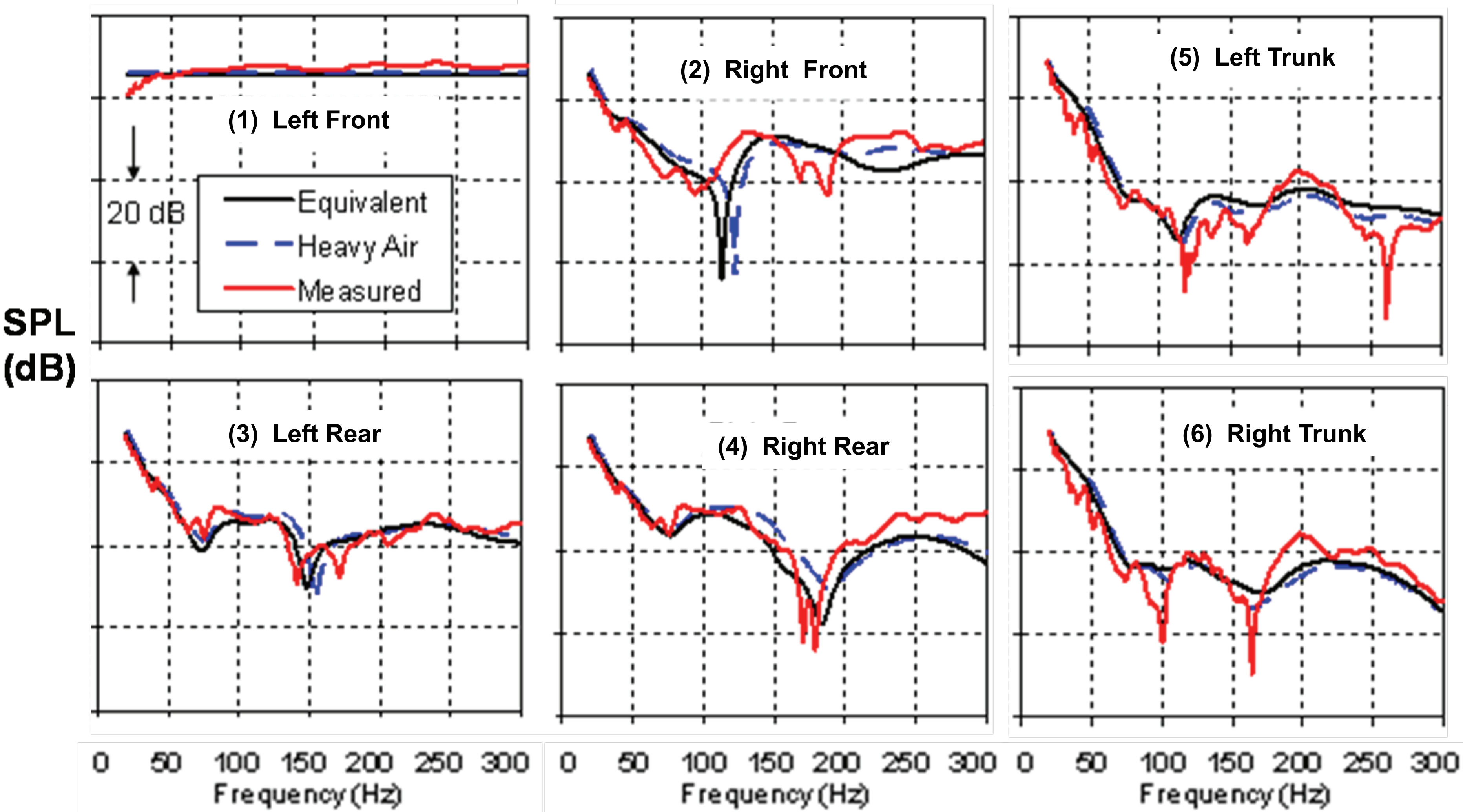
Equivalent-Acoustic Seat Models





Predicted SPL in Passenger Compartment

Driver's Ear Loudspeaker



Coupled Structural-Acoustic Analysis

Un-Symmetric Form

$$\begin{bmatrix} M & 0 \\ -A^T & Q \end{bmatrix} \begin{Bmatrix} \ddot{w} \\ \ddot{p} \end{Bmatrix} + \begin{bmatrix} C & 0 \\ 0 & D \end{bmatrix} \begin{Bmatrix} \dot{w} \\ \dot{p} \end{Bmatrix} + \begin{bmatrix} K & A \\ 0 & H \end{bmatrix} \begin{Bmatrix} w \\ p \end{Bmatrix} = \begin{Bmatrix} F(t) \\ G(t) \end{Bmatrix} \qquad (1)$$

Structural-Acoustic Coupling

$$[A]^T = \int_S \{\theta\}\{N\}^T \, dS$$

Symmetric Form

Let $p = \dot{q}$ and integrate fluid equation with minus sign with $\tilde{G} = -\int_0^t G(\tau) d\tau$

$$\begin{bmatrix} M & 0 \\ 0 & -Q \end{bmatrix} \begin{Bmatrix} \ddot{w} \\ \ddot{q} \end{Bmatrix} + \begin{bmatrix} C & A \\ A^T & -D \end{bmatrix} \begin{Bmatrix} \dot{w} \\ \dot{q} \end{Bmatrix} + \begin{bmatrix} K & 0 \\ 0 & -H \end{bmatrix} \begin{Bmatrix} w \\ q \end{Bmatrix} = \begin{Bmatrix} F(t) \\ \tilde{G}(t) \end{Bmatrix} \qquad (1)$$

Structural-Acoustic Coupling & Symmetric Form Automated in MSC/NASTRAN, Altair RADIOSS, other software

Modal Response Solution

Uncoupled Structural & Acoustic Mode Transformation

$$\{u\}=[\phi]\{\xi\},\quad \{p\}=[\psi]\{\zeta\} \qquad \begin{bmatrix}\phi & 0\\ 0 & \psi\end{bmatrix}^T \times \quad Eq.(1)$$

Modal Frequency Response Equations

$$-\omega^2\begin{bmatrix} m & 0\\ -a^T & q\end{bmatrix}\begin{Bmatrix}\xi\\ \varsigma\end{Bmatrix}+i\omega\begin{bmatrix} c & 0\\ 0 & d\end{bmatrix}\begin{Bmatrix}\xi\\ \zeta\end{Bmatrix}+\begin{bmatrix} k & a\\ 0 & h\end{bmatrix}\begin{Bmatrix}\xi\\ \zeta\end{Bmatrix}=\begin{Bmatrix}F_0\\ G_0\end{Bmatrix} \qquad (2)$$

$$a = [\phi]^T[A][\varphi]$$

Mode & Panel Participation

Acoustic Modes (L *Acoustic Modes)*

$$\{p\}=[\psi]\{\zeta\}=\sum_{i=1}^{L}\{p_k\}$$

Structural Modes (M Structural Modes)

$$\{p\}=[\varphi][Z_a][a]\{\zeta\}=\sum_{j=1}^{M}\{p_j\}$$

Boundary Panel (N Panel Grids)

$$\{p\}=[\psi][Z_a][\psi]^T\sum_{k=1}^{N}\lfloor A\rfloor[\phi]\{\zeta\}=\sum_{k=1}^{N}\{p_k\}$$

$$[Z_a]=(-\omega^2[q]+i\omega[[d]+[h])^{-1}$$

Structural-Acoustic Coupling Effect on Station Wagon Tailgate Modes

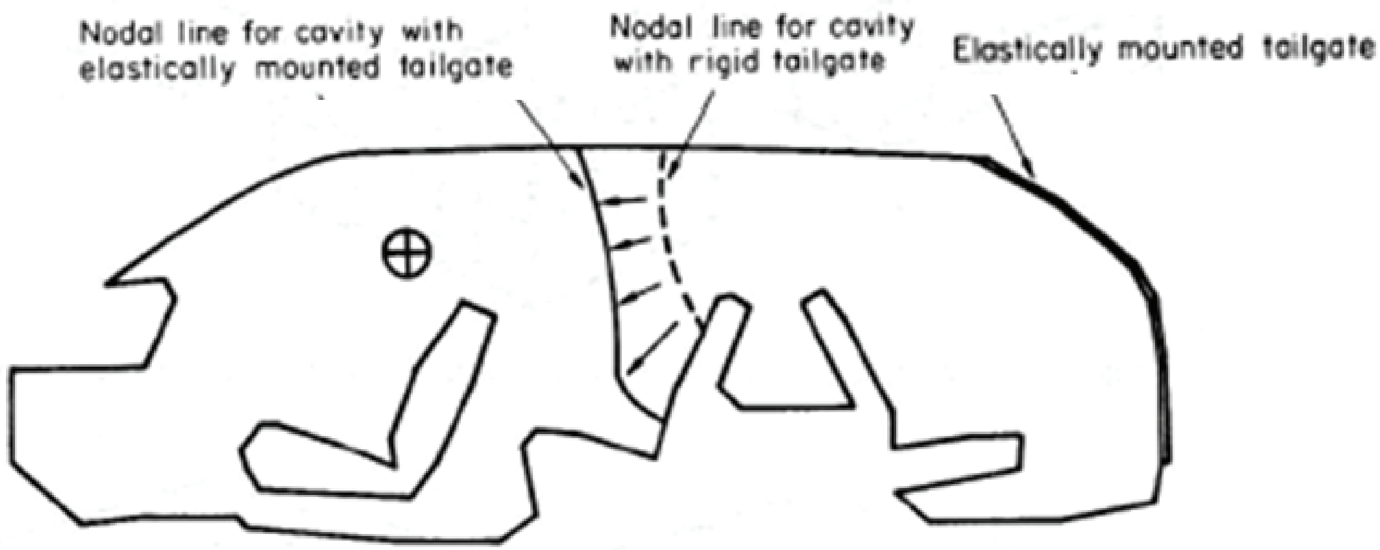
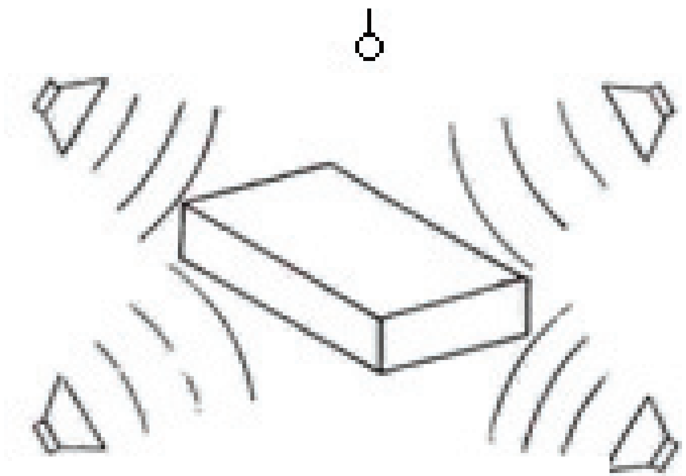


TABLE 1

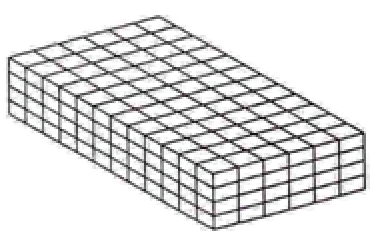
Elastic boundary effects on cavity frequencies of station wagon

Boundary which is non-rigid	Frequency of tailgate mode (*in vacuo*/coupled)	Cavity frequency (rigid wall/coupled)
Tailgate 22·7 kg	30/28·7	41·7/44·8
	40/35·6	41·7/47·4
	50/53·4	41·7/39·2
	60/61·7	41·7/40·5

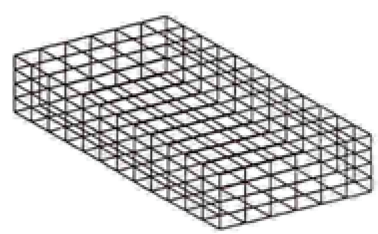
Structural-Acoustic Coupling Effect on Noise Response in a Box



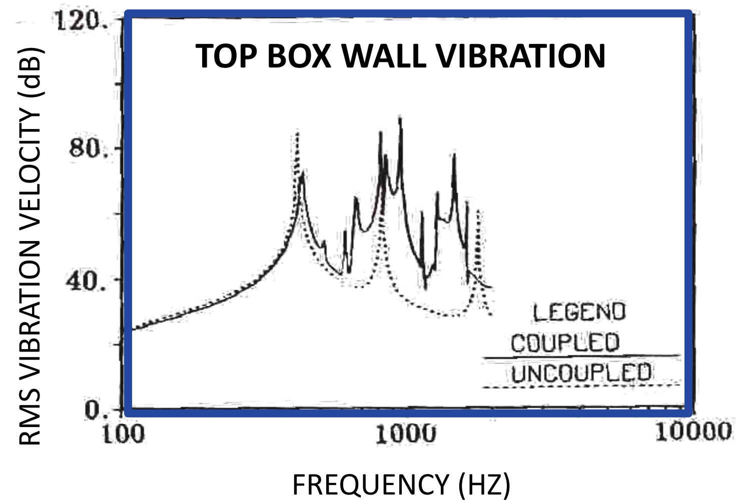
(a) Uniform Noise Environment

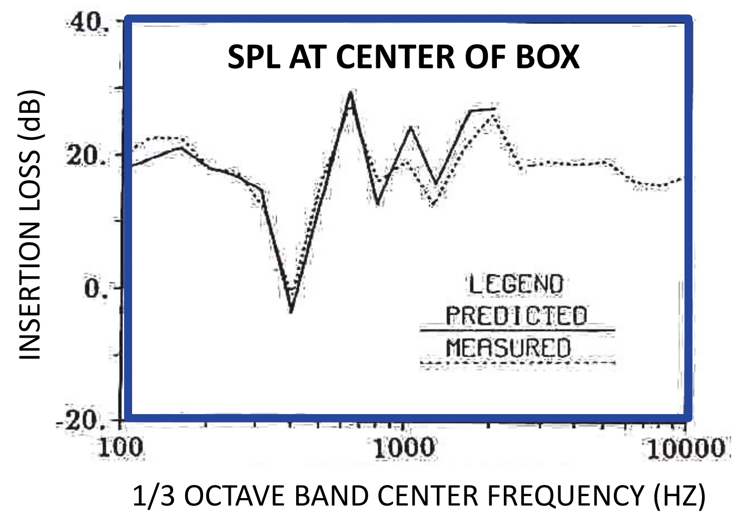


Structural FE Model of Aluminum Box (1.6 mm wall thickness)



Interior Cavity FE Model of Aluminum Box (300 x 150 x 5 mm)

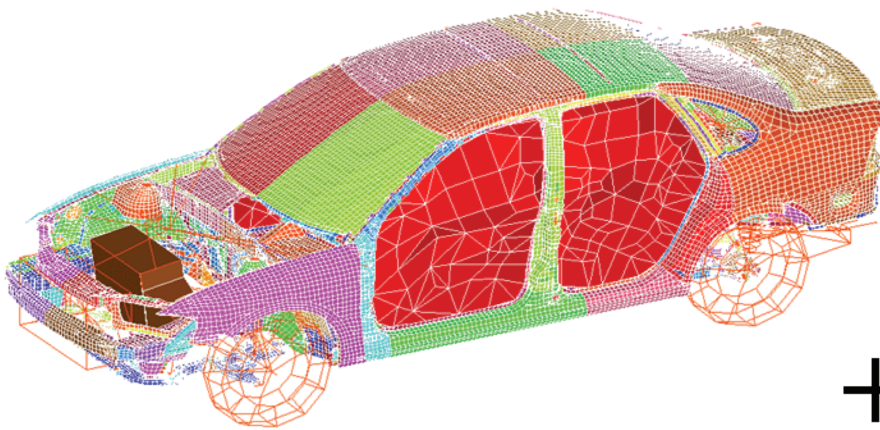




Validation of a Vehicle Structural-Acoustic System Finite Element Model

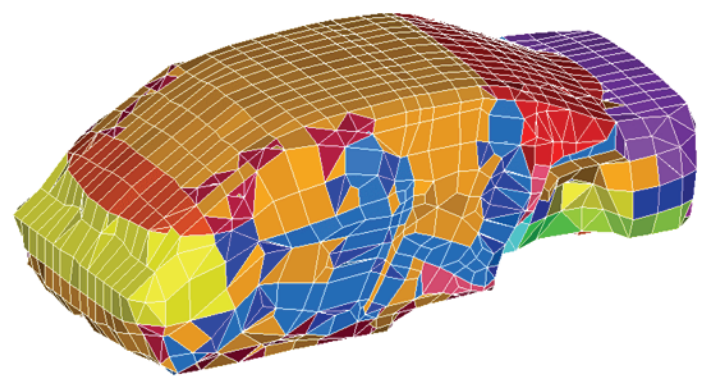
"A Structural-Acoustic Finite Element Method for Predicting Automotive Vehicle Interior Road Noise," S. H. Sung, D. J. Nefske, D. A. Feldmaier, IMECE2009-11065, 2009.

Vehicle Structural & Acoustic Models



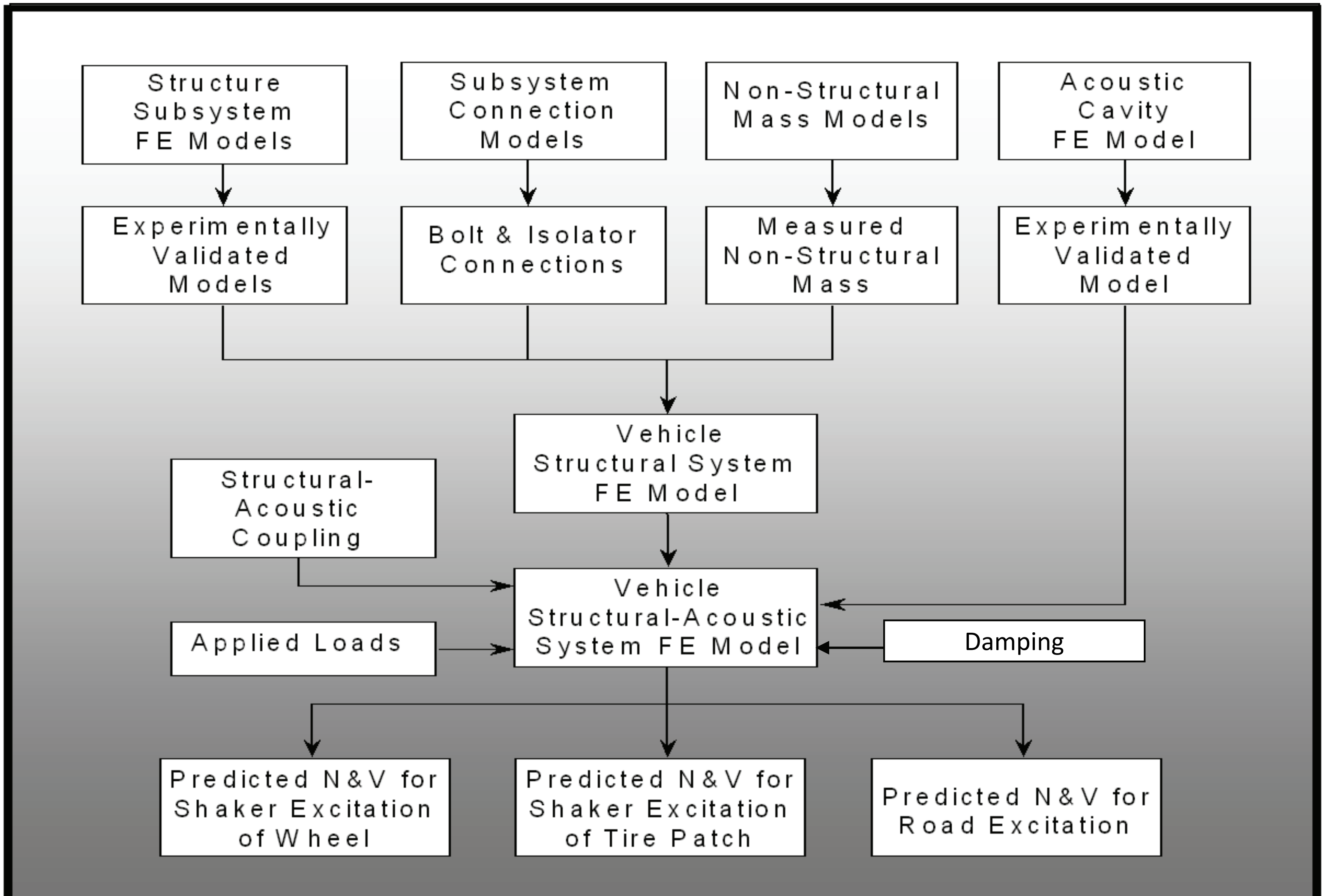
Vehicle Structural Model
(107,400 Grids)

+



Compartment Acoustic Model
(5,300 Grids)

Structural-Acoustic Vehicle System Model Development & Solution Procedures



Finite Element Equations-of-Motion

Vehicle Structural System Model

Coupled Trimmed Body (1) and Chassis (2) Subsystem FE Models

$$\begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix}\begin{Bmatrix} \ddot{w}_1 \\ \ddot{w}_2 \end{Bmatrix}+\begin{bmatrix} C_1 & 0 \\ 0 & C_2 \end{bmatrix}\begin{Bmatrix} \dot{w}_1 \\ \dot{w}_2 \end{Bmatrix}+\begin{bmatrix} K_1+K_{12} & -K_{12} \\ -K_{12} & K_2+K_{12} \end{bmatrix}\begin{Bmatrix} w_1 \\ w_2 \end{Bmatrix}=\begin{Bmatrix} F_1 \\ F_2 \end{Bmatrix}$$

$+$

Compartment Acoustic Cavity Model

Compartment Cavity (o) and Interior Trim (a) Equivalent-Acoustic FE Models

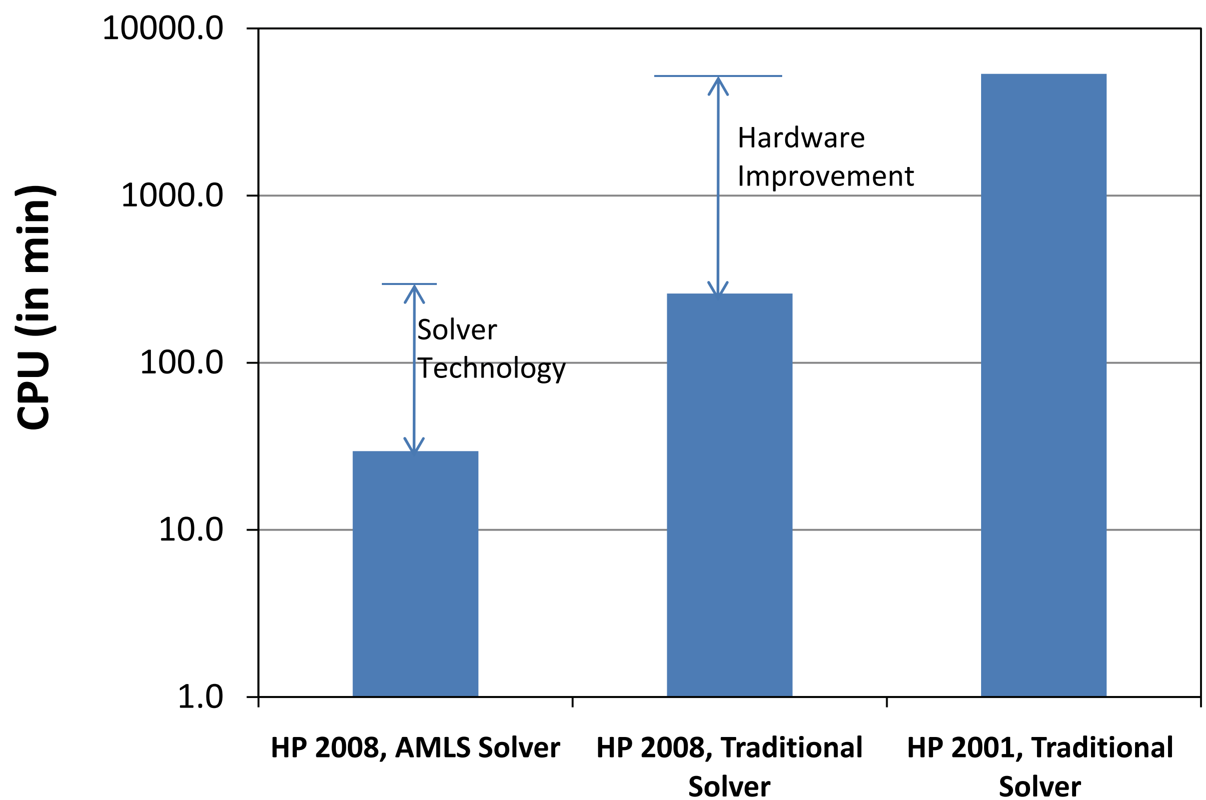
$$\begin{bmatrix} Q_o & \tilde{Q}_{oa} \\ \tilde{Q}_{ao} & Q_a \end{bmatrix}\begin{Bmatrix} \ddot{p}_o \\ \ddot{p}_a \end{Bmatrix}+\begin{bmatrix} D_o & 0 \\ 0 & 0 \end{bmatrix}\begin{Bmatrix} \dot{p}_o \\ \dot{p}_a \end{Bmatrix}+\begin{bmatrix} H_o & \tilde{H}_{oa} \\ \tilde{H}_{ao} & H_a \end{bmatrix}\begin{Bmatrix} p_o \\ p_a \end{Bmatrix}=i\omega\begin{Bmatrix} q_o \\ 0 \end{Bmatrix}$$

Vehicle Structural-Acoustic System FE Model

$$\begin{bmatrix} M & 0 \\ -A^T & Q \end{bmatrix}\begin{Bmatrix} \ddot{w} \\ \ddot{p} \end{Bmatrix}+\begin{bmatrix} C & 0 \\ 0 & D \end{bmatrix}\begin{Bmatrix} \dot{w} \\ \dot{p} \end{Bmatrix}+\begin{bmatrix} K & A \\ 0 & H \end{bmatrix}\begin{Bmatrix} w \\ p \end{Bmatrix}=\begin{Bmatrix} F(t) \\ G(t) \end{Bmatrix}$$

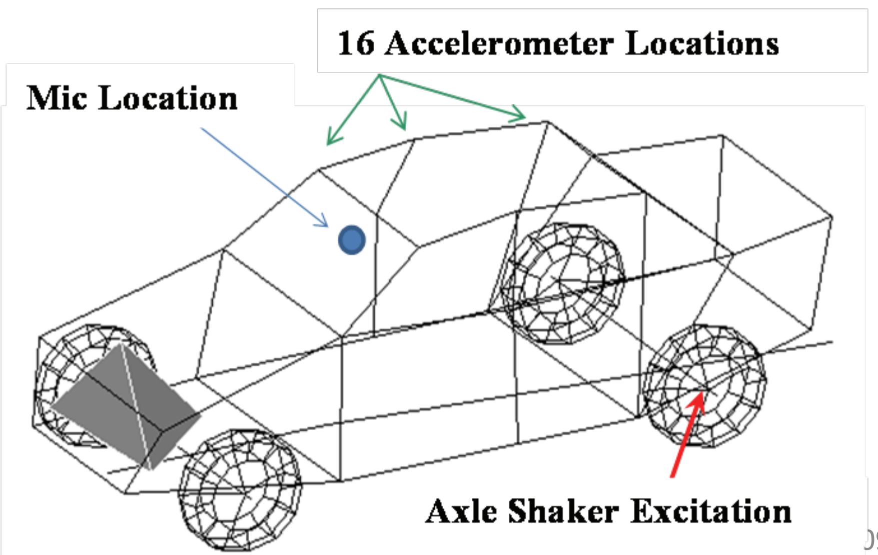
CPU Improvement with New Hardware and AMLS Solver Technology

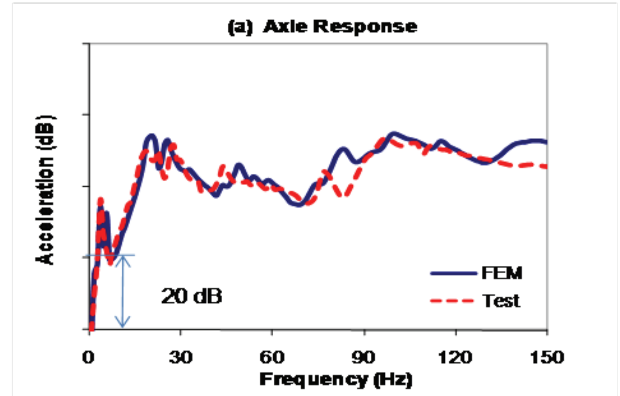
CPU of Structural-Acoustic Model with 650K DOF

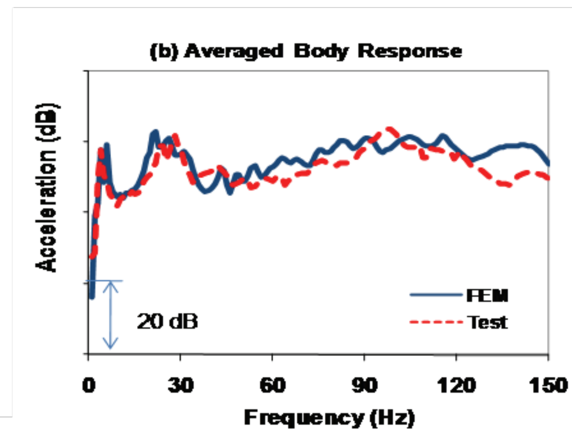


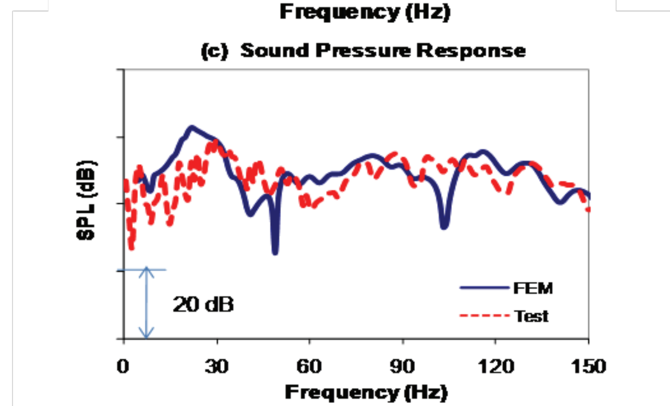
AMLS – Automated Multi-Level Substructuring

Axle Shaker Excitation

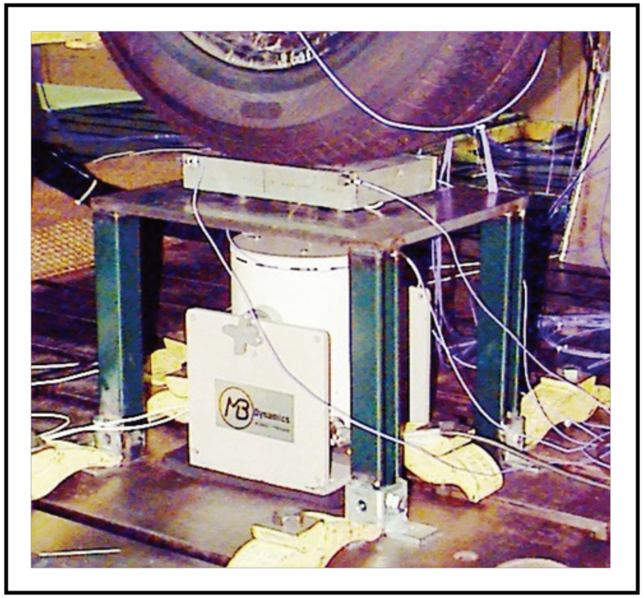


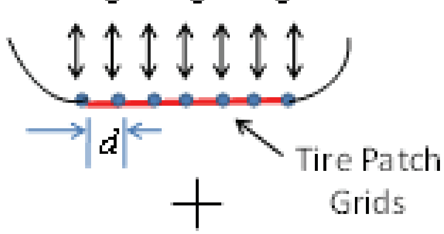






Tire Patch Excitation



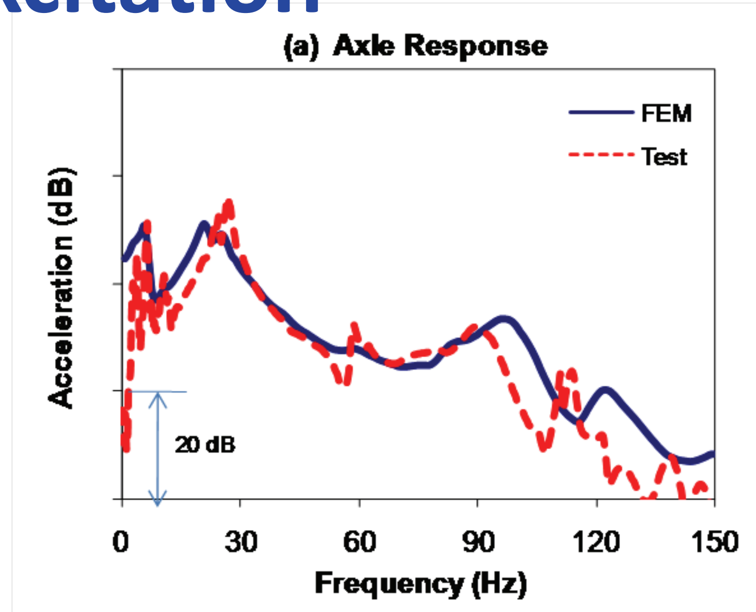


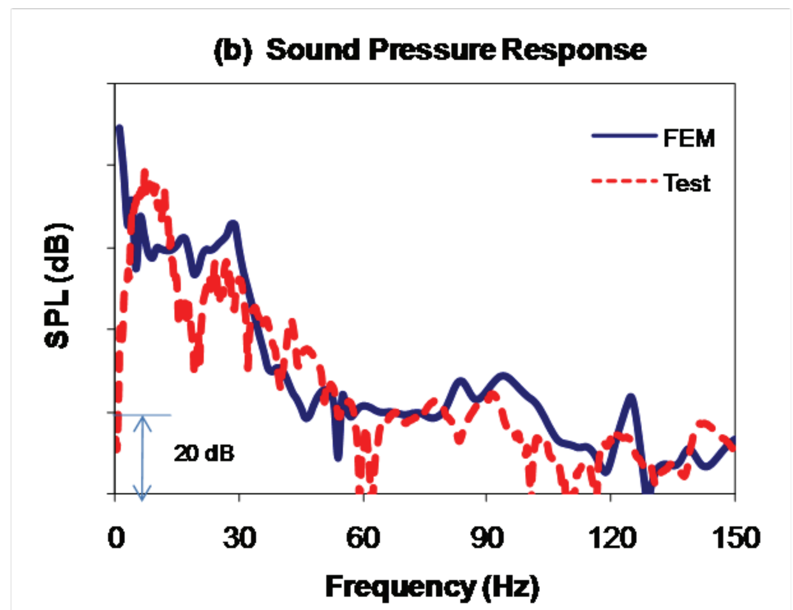
$a_o = F_o / M_o$

d - Grid Spacing

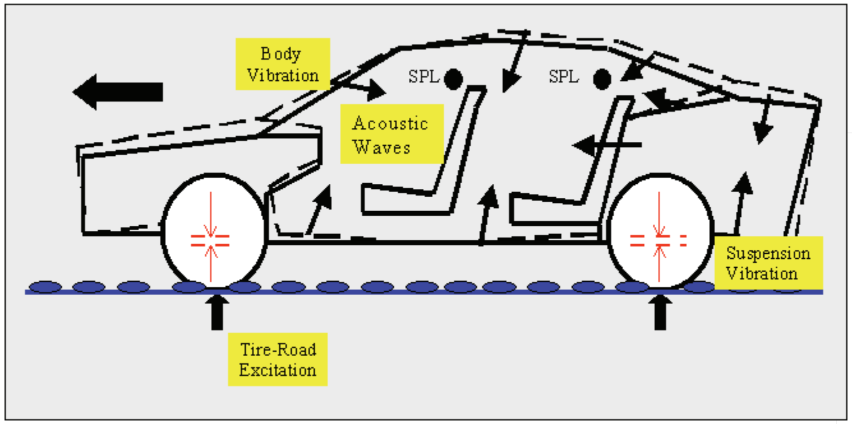
M_o - Large Mass at Each Grid

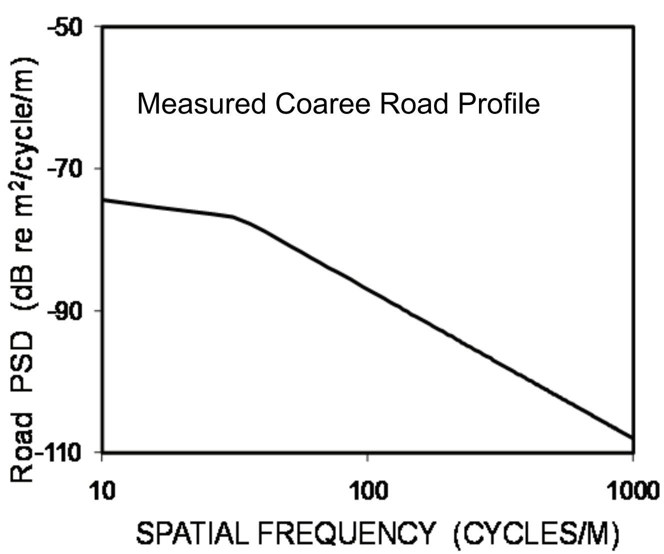
F_o - Applied Load at Each Grid



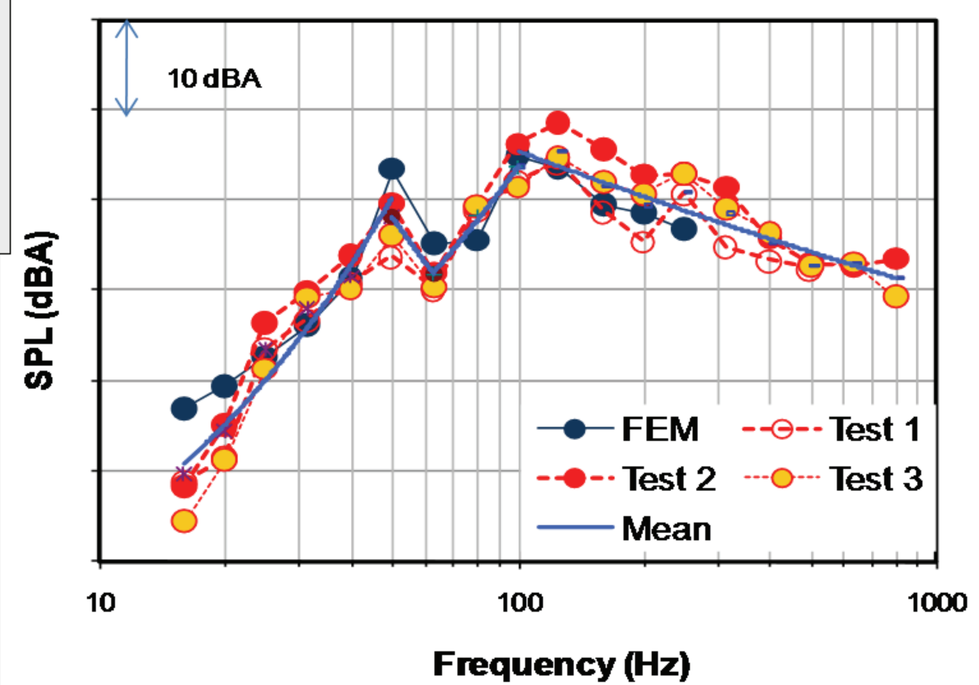


On-Road Vehicle Interior Noise Response (60 km/h on Coarse Road)



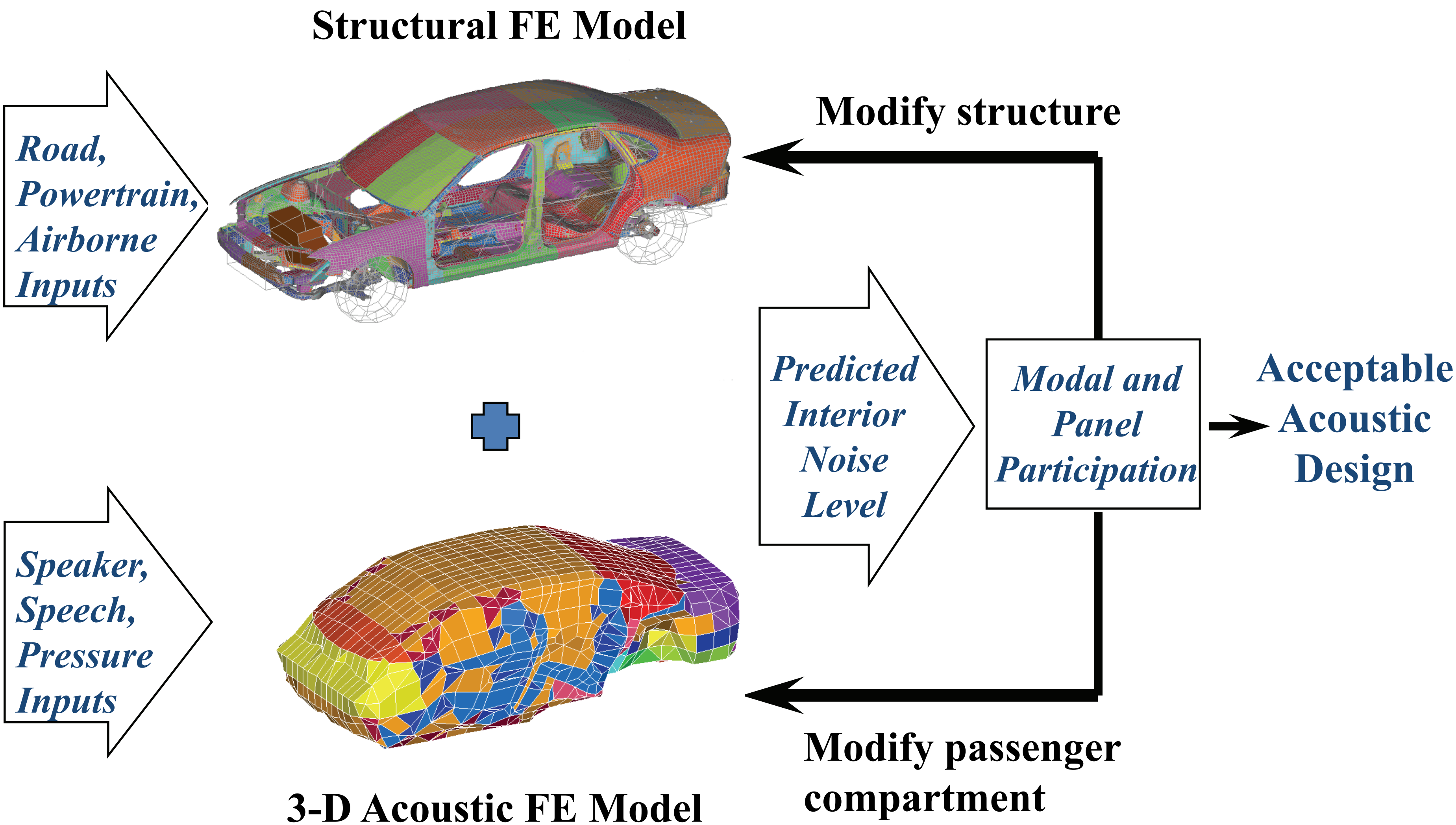


Interior Sound Pressure Level at Front Passenger Ear Location

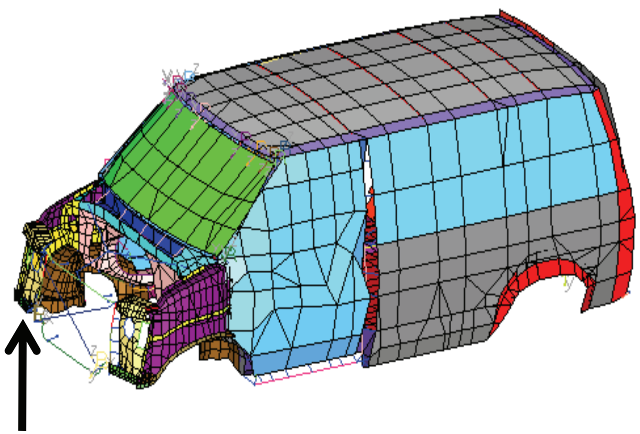


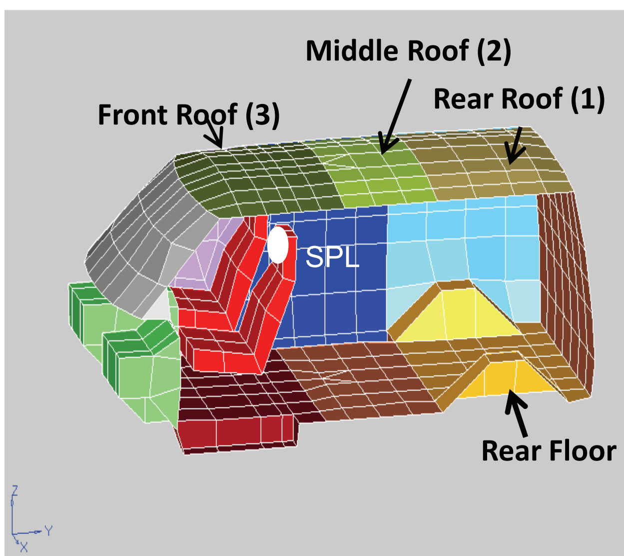
Application of Structural-Acoustic Vehicle System Finite Element Model in Automotive Vehicle Design

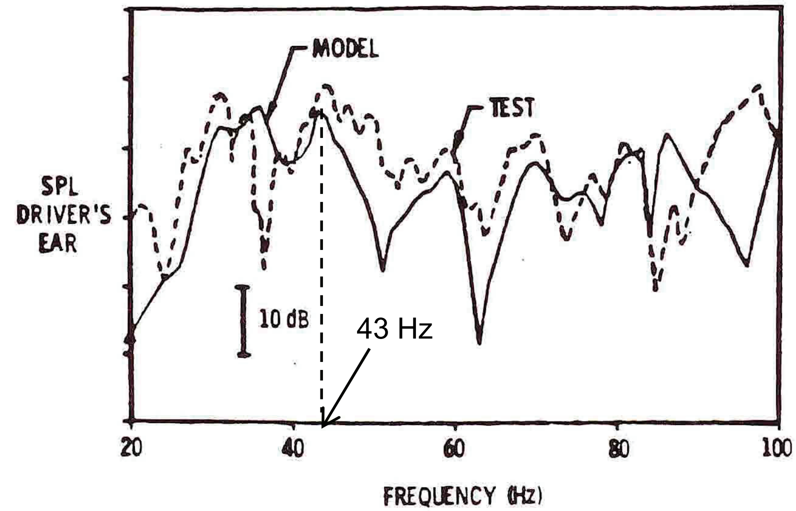
Structural-Acoustic Procedure

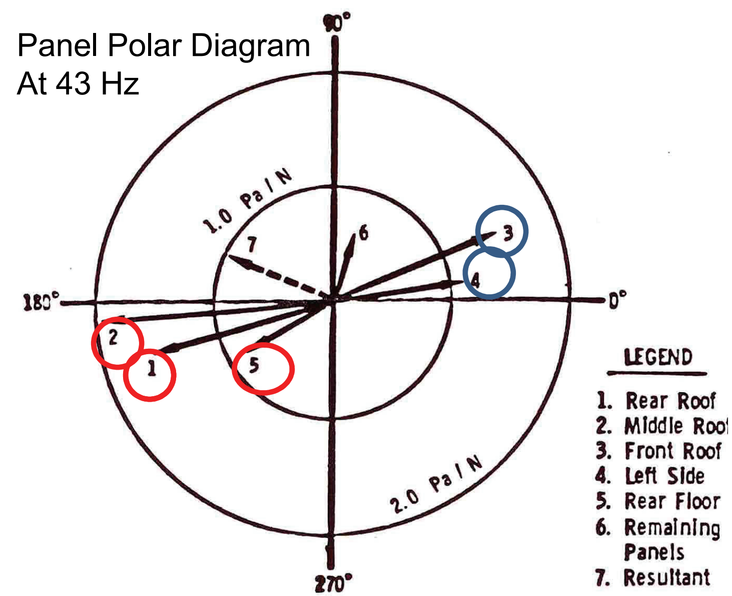


FRF and Panel Polar Diagram

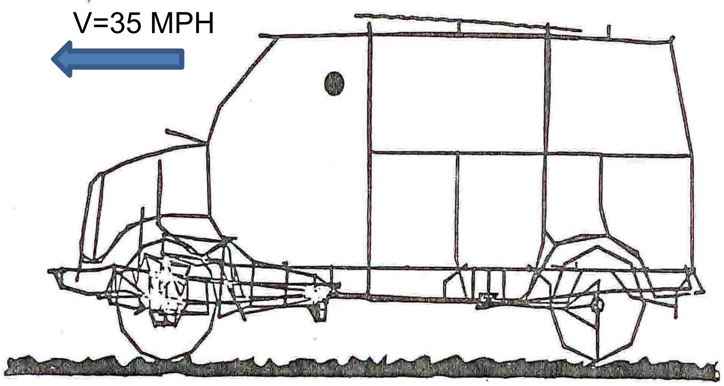


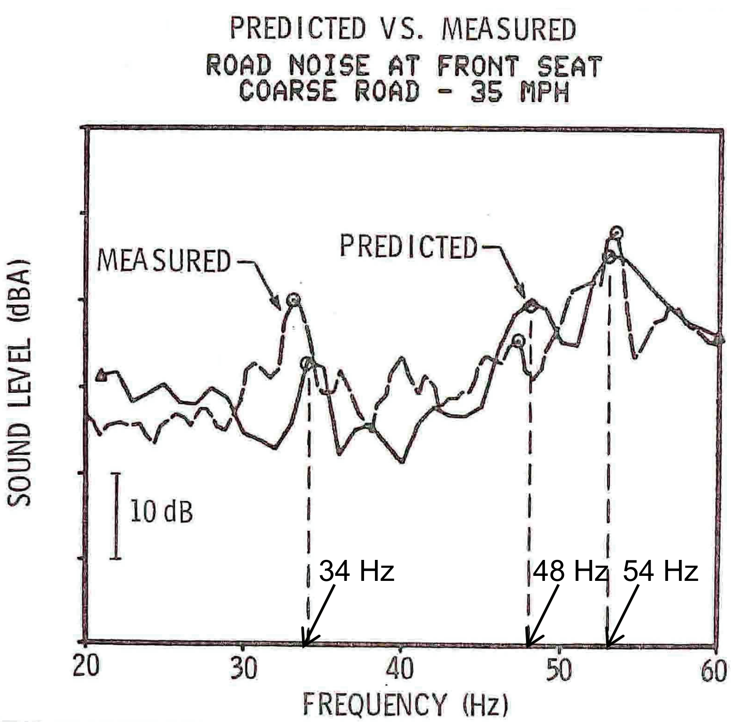




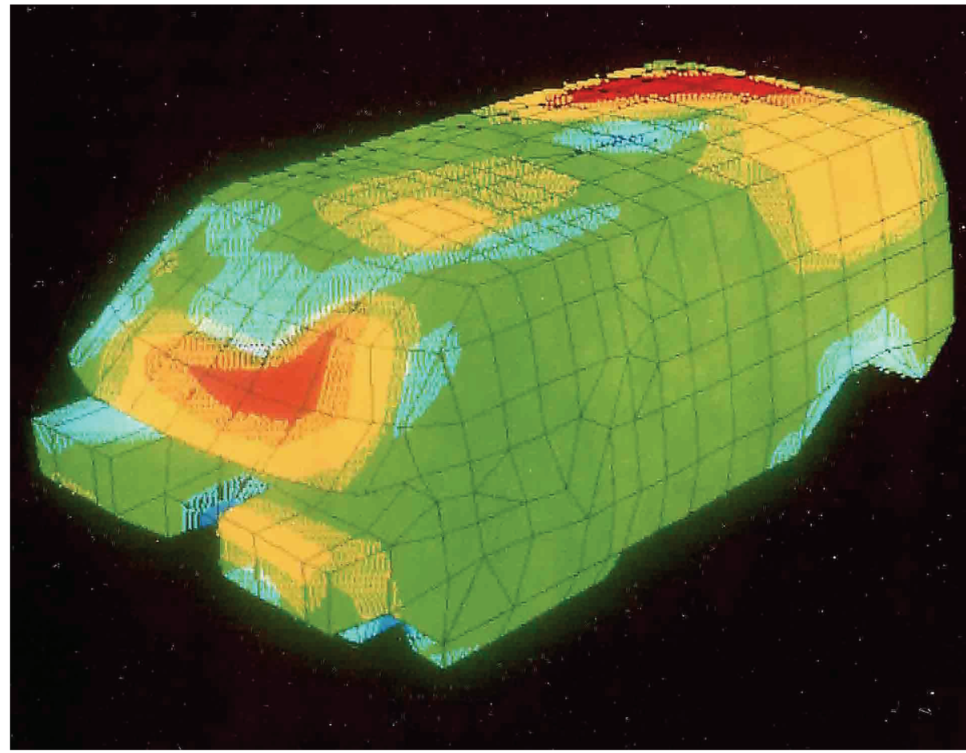


Modal and Panel Participation



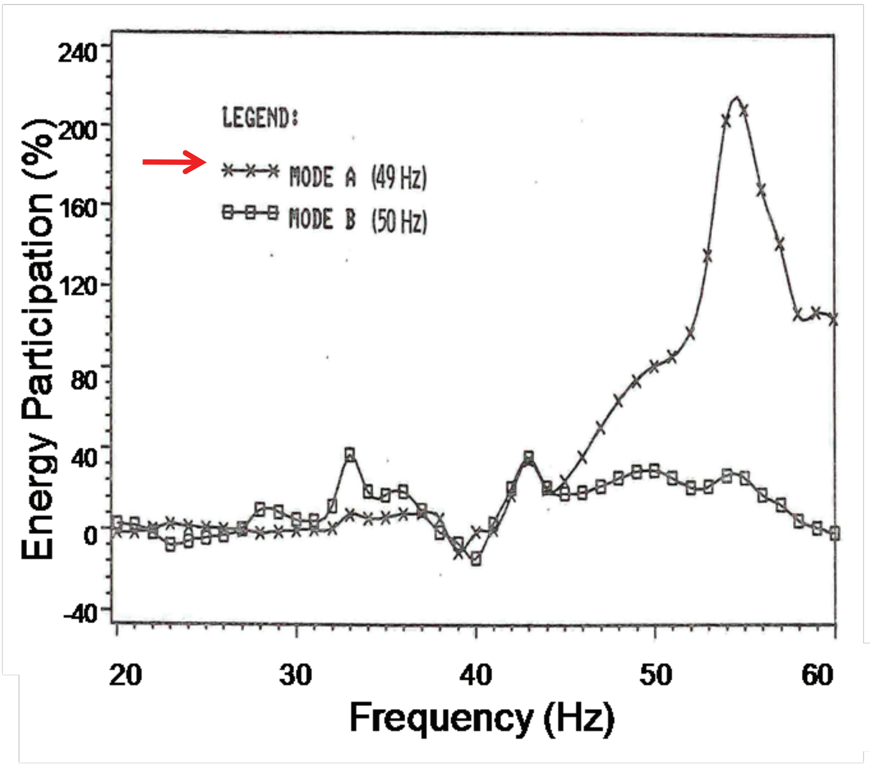


Vehicle Body Panel Participation
to interior road noise response at 54 Hz



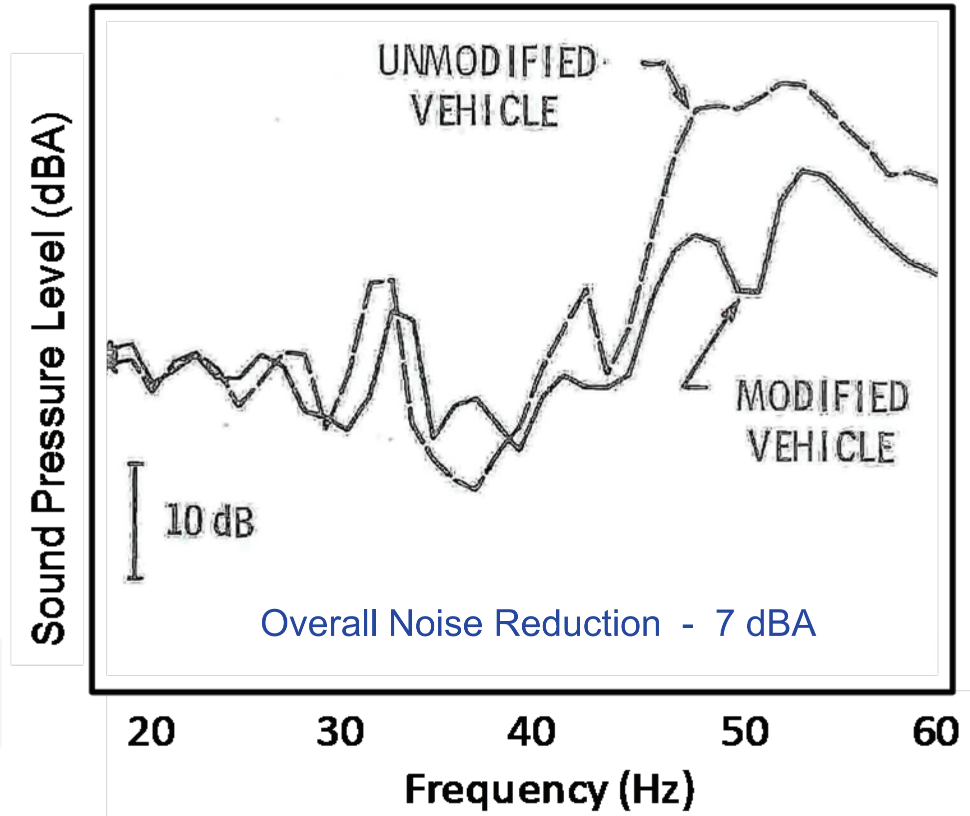
Application in Vehicle Design

Dominant Body Mode Participation



Predicted

35 MPH Coarse Road Noise at Front Seat



Summary

- The structural-acoustic finite element method has been established and validated for modeling complex automotive vehicle systems.

- Structural-acoustic modeling, automated structural-acoustic coupling, modal/panel participation, and AMLS are available in commercially available software (e.g., MSC/Nastran, RADIOSS)

- Large structural-acoustic system models are easily solved using advanced solver technology and computer hardware

- More research work is needed to represent interior trim and other passenger compartment characteristics in the structural-acoustic vehicle system model.

- Robust and standardized design procedures are required to facilitate noise and vibration design in the early vehicle design and development stages.

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