A SEQUENTIAL ADJOINT VARIABLE METHOD FOR DSA OF NVH PROBLEMS

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INTRODUCTION

- FEA-BEA NVH model: structural dynamic behavior is solved using a FE frequency response analysis (NASTRAN), while driver position sound pressure is calculated using an acoustic BE analysis (COMET).
- A sequential adjoint variable method is developed in DSA such that the adjoint load is calculated from the acoustic BE re-analysis, and the adjoint response is obtained from the structural FE re-analysis.
- Based on the the design sensitivity result, the first design iteration is carried out to obtain an improved design.

WHY
ADJOINT VARIABLE METHOD?

- DDM $\propto$ Number of Design Variables (*NDV*)
- AVM $\propto$ Number of Constraints (*NCT*)

$$NDV \geq NCT$$

- AVM provides the element sensitivity as a side benefit
- Sequential Adjoint Variable Method

Adjoint load $\Longleftarrow$ acoustic BE re-analysis

Adjoint response $\Longleftarrow$ structural FE re-analysis.

NVH ANALYSIS

- Frequency Response Analysis (FEA)

$$j\omega d_u(\boldsymbol{v},\overline{\boldsymbol{z}}^*)+c_u(\boldsymbol{v},\overline{\boldsymbol{z}}^*)+\frac{1}{j\omega}a_u(\boldsymbol{v},\overline{\boldsymbol{z}}^*)=\ell_u(\overline{\boldsymbol{z}}^*), \qquad \forall\overline{\boldsymbol{z}}\in Z$$

$$[\,j\omega\boldsymbol{M}+\kappa\boldsymbol{K}]\{\boldsymbol{v}(\omega)\}=\{\boldsymbol{F}(\omega)\}$$

- Acoustic Analysis (BEA)

$$\iint_{\Omega^s}[-j\rho\omega G(\boldsymbol{x}_S,\boldsymbol{x}_0)v_n(\boldsymbol{x}_S)-\frac{\partial G}{\partial n}p(\boldsymbol{x}_S)]\,d\Omega^S=Cp(\boldsymbol{x}_0)$$

$$[\boldsymbol{A}]\{\boldsymbol{p}_s\}=[\boldsymbol{B}]\{\boldsymbol{v}\}$$

$$p(\boldsymbol{x}_0)=\{\boldsymbol{b}\}^T\{\boldsymbol{v}\}+\{\boldsymbol{e}\}^T\{\boldsymbol{p}_s\}$$

DESIGN SENSITIVITY ANALYSIS (DDM)

- Structural DSA (FEA)

$$j\omega d_{\boldsymbol{u}}(\boldsymbol{v}', \overline{\boldsymbol{z}}^*) + \kappa a_{\boldsymbol{u}}(\boldsymbol{v}', \overline{\boldsymbol{z}}^*) = \ell'_{\delta \boldsymbol{u}}(\overline{\boldsymbol{z}}^*) - j\omega d'_{\delta \boldsymbol{u}}(\boldsymbol{v}, \overline{\boldsymbol{z}}^*) - \kappa a'_{\delta \boldsymbol{u}}(\boldsymbol{v}, \overline{\boldsymbol{z}}^*), \qquad \forall \overline{\boldsymbol{z}} \in Z$$

$$[\, j\omega \boldsymbol{M} + \kappa \boldsymbol{K}]\{\boldsymbol{v}'(\omega)\} = \{\boldsymbol{F}^{fic}(\omega)\}$$

- Acoustic DSA (BEA)

$$b(\boldsymbol{x}_0; \boldsymbol{v}') + e(\boldsymbol{x}_0; p'_S) = Cp'(\boldsymbol{x}_0)$$

$$[\boldsymbol{A}]\{\boldsymbol{p}'_S\} = [\boldsymbol{B}]\{\boldsymbol{v}'\}$$

$$p'(\boldsymbol{x}_0) = \{\boldsymbol{b}(\boldsymbol{x}_0)\}^T\{\boldsymbol{v}'\} + \{\boldsymbol{e}(\boldsymbol{x}_0)\}^T\{\boldsymbol{p}'_S\}$$

BEA matrices are independent of the structural sizing design

DESIGN SENSITIVITY ANALYSIS (DDM) cont.

- Structural Performance Measure

$$\psi_1 = \iint_{\Omega^S} g(\boldsymbol{v}, \boldsymbol{u})\, d\Omega^S$$

$$\psi_1' = \iint_{\Omega^S} (g_{,\boldsymbol{v}}^T \boldsymbol{v}' + g_{,\boldsymbol{u}}^T \delta \boldsymbol{u})\, d\Omega^S$$

- Acoustic Performance Measure

$$\psi_2(\boldsymbol{x}_0) = h\left(p(\boldsymbol{x}_0), \boldsymbol{u} \right)$$

$$\psi_2' = h_{,p} p' + h_{,\boldsymbol{u}}^T \delta \boldsymbol{u}$$

$\boldsymbol{v}'$ from structural DSA and p' from acoustic DSA are used to evaluate sensitivity of ψ_1 and ψ_2.

DESIGN SENSITIVITY ANALYSIS (AVM)

- Structural Performance Measure (Choi and Lee, 1992)
  - Adjoint Equation

$$j\omega d_{\boldsymbol{u}}(\bar{\boldsymbol{\lambda}}, \boldsymbol{\lambda}^*) + \kappa a_{\boldsymbol{u}}(\bar{\boldsymbol{\lambda}}, \boldsymbol{\lambda}^*) = \iint_{\Omega^S} g_{,\boldsymbol{v}}^T \bar{\boldsymbol{\lambda}} \, d\Omega^S, \qquad \forall \bar{\boldsymbol{\lambda}} \in Z$$

  - Sensitivity Computation

$$\psi_1' = \iint_{\Omega^S} g_{,\boldsymbol{u}}^T \delta\boldsymbol{u} \, d\Omega^S + \ell'_{\delta\boldsymbol{u}}(\boldsymbol{\lambda}^*) - j\omega d'_{\delta\boldsymbol{u}}(\boldsymbol{v}, \boldsymbol{\lambda}^*) - \kappa a'_{\delta\boldsymbol{u}}(\boldsymbol{v}, \boldsymbol{\lambda}^*)$$

– Different performance measures have different adjoint loads.

– Sensitivity computation is same for all types of performance measures.

DESIGN SENSITIVITY ANALYSIS (AVM) cont.

- Acoustic Performance Measure
  - Adjoint Load Computation

    $$[\boldsymbol{A}]^T\{\boldsymbol{\eta}\} = \{\boldsymbol{e}\}$$

  - Adjoint Equation

    $$[\,j\omega\boldsymbol{M} + \kappa\boldsymbol{K}]\{\boldsymbol{\lambda}^*\} = \{\boldsymbol{b}\} + [\boldsymbol{B}]^T\{\boldsymbol{\eta}\}$$

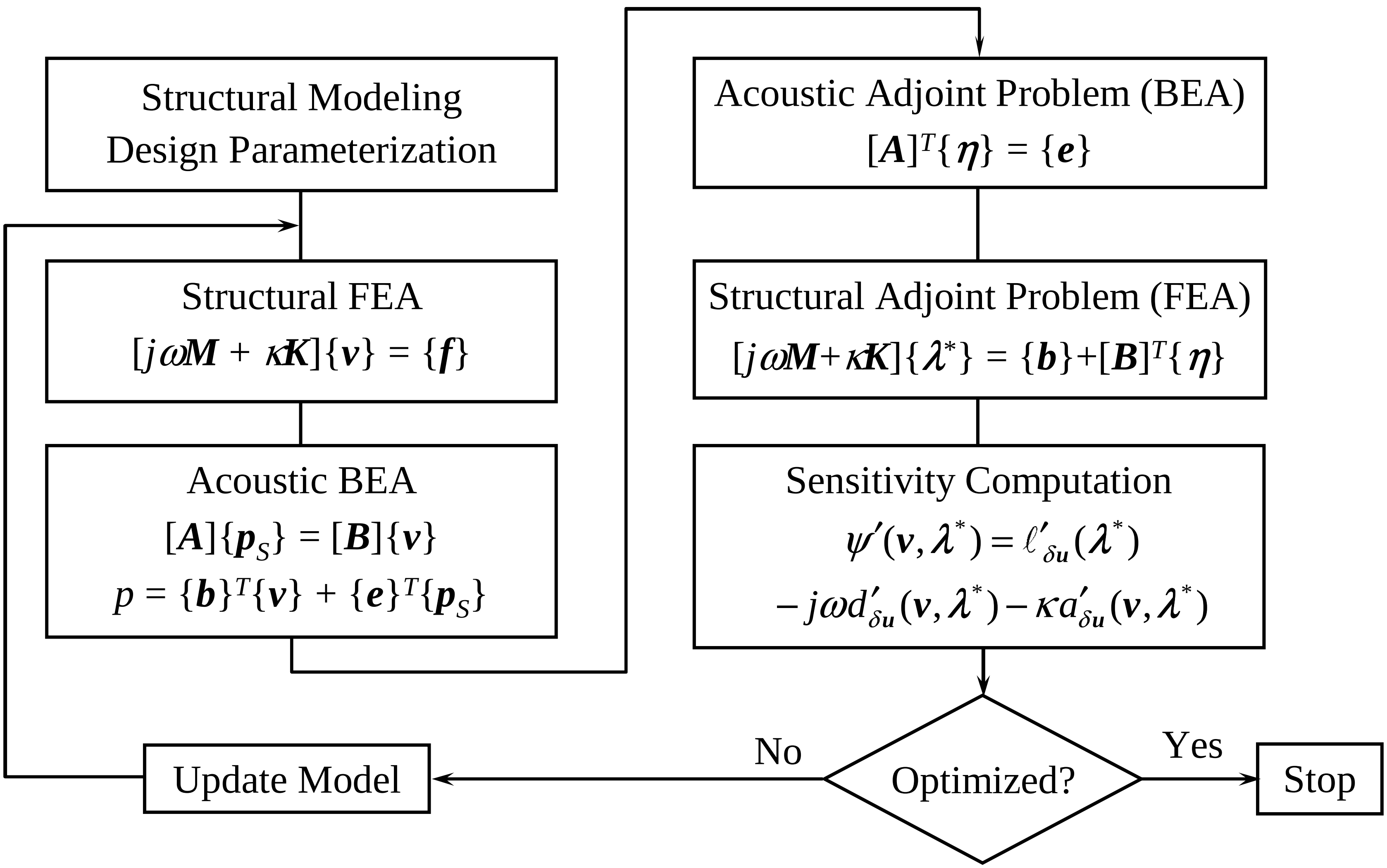
  - Sensitivity Computation

    $$\psi_2' = h_{,\boldsymbol{u}}\delta\boldsymbol{u} + \ell'_{\delta\boldsymbol{u}}(\boldsymbol{\lambda}^*) - j\omega d'_{\delta\boldsymbol{u}}(\boldsymbol{v},\boldsymbol{\lambda}^*) - \kappa a'_{\delta\boldsymbol{u}}(\boldsymbol{v},\boldsymbol{\lambda}^*)$$

– BE re-analysis solves the transposed matrix equation.

– Sensitivity computation process is the same as structural performance measure case.

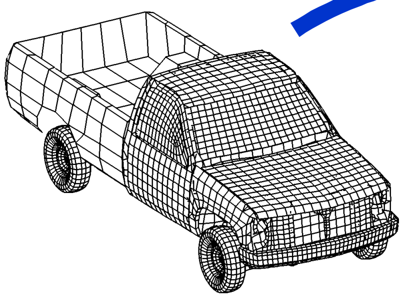
DSA FLOW CHART

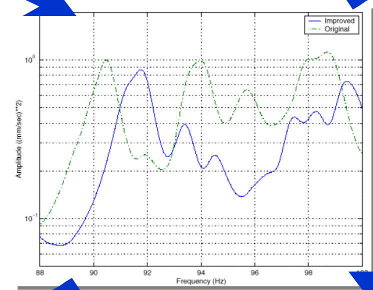


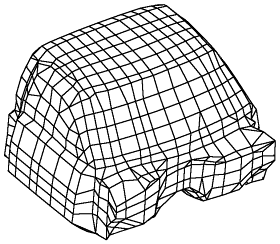
ADJOINT DSA PROCESS

Frequency Response FEA

Acoustic BEA



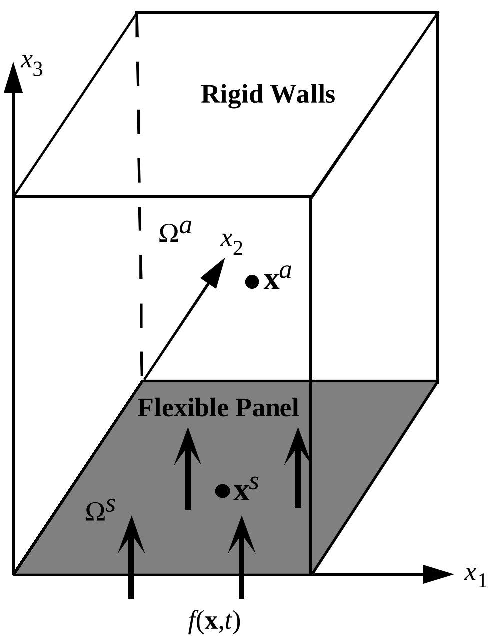


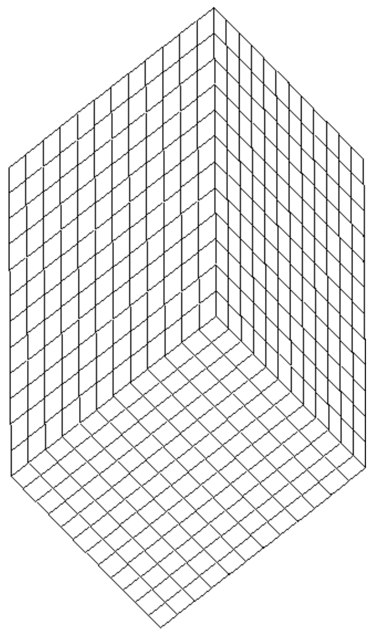


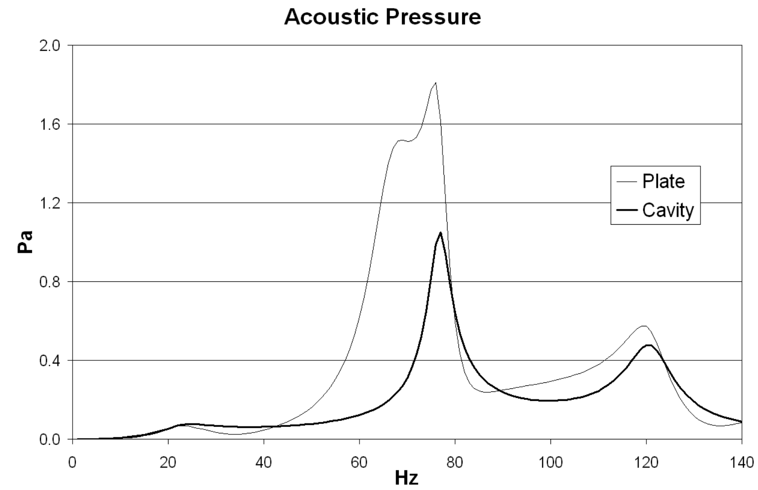
DSA

Adjoint Load

BOX MODEL





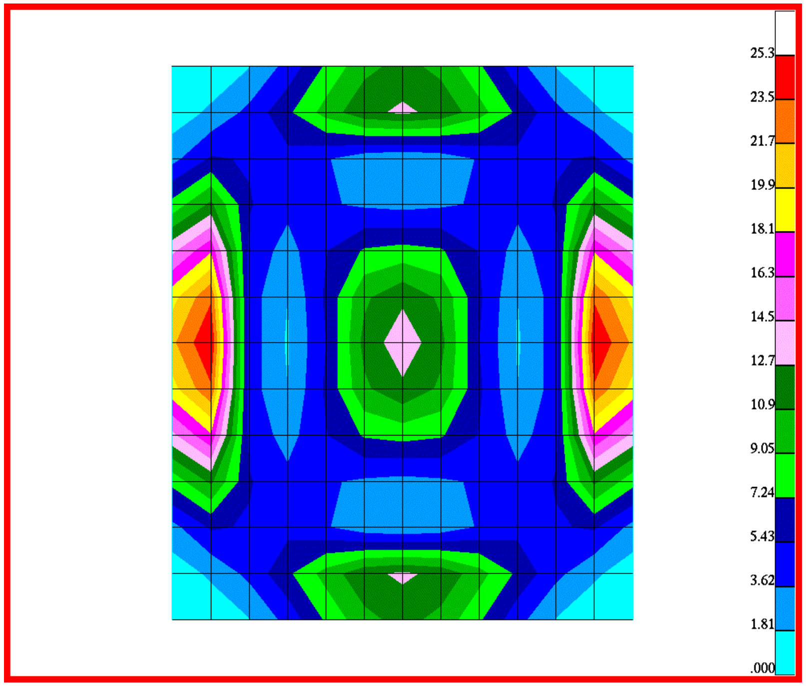


FEA-BEA Model   Pressure Response

Sound Speed = 344 m/sec   Thickness = .01 m

Air Density = 1.205 kg/m^3 Panel Density = 2,700 kg/m^3

BOX MODEL cont.



Element Sensitivity for Cavity Pressure (Negative)

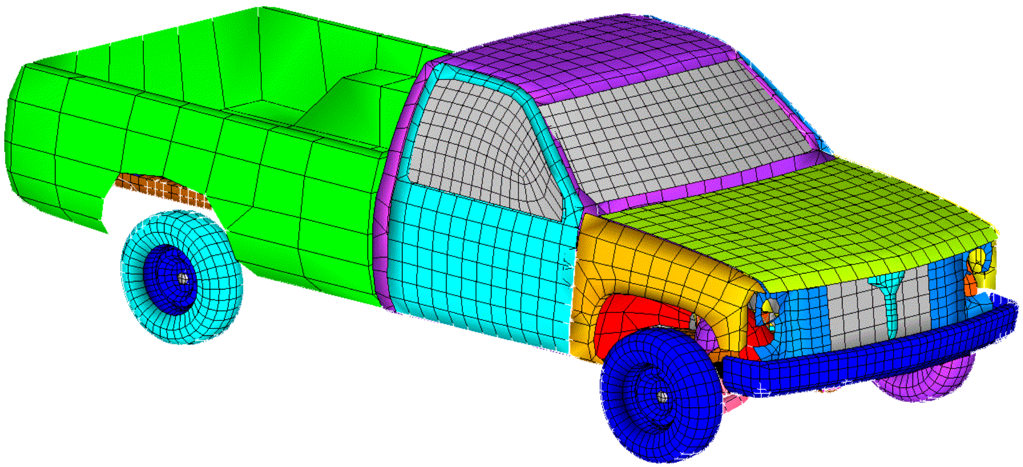
BOX MODEL cont.

Sensitivity Comparison with FDM

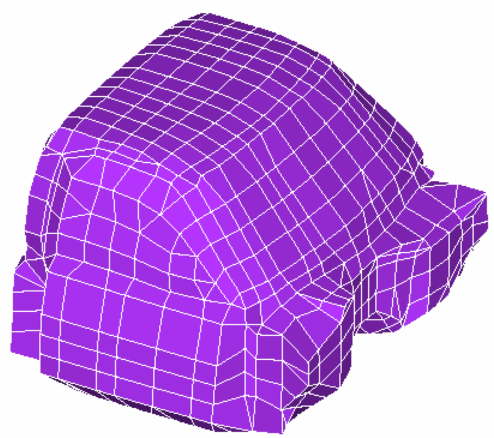
Type	ψ	ψ_τ	$\Delta\psi/\Delta\tau$	ψ'	$\Delta\psi/\Delta\tau/\psi' \times 100$
Displacement at $\boldsymbol{x}^s$	3.2796E-5	3.2756E-5	-0.0404	-0.0402	100.55
Velocity at $\boldsymbol{x}^s$	0.0156608	0.0156416	-19.219	-19.187	100.17
Pressure at $\boldsymbol{x}^s$	1.8118117	1.8096199	-2191.8	-2223.3	98.58
Pressure at $\boldsymbol{x}^a$	0.9901643	0.9889635	-1200.8	-1198.4	100.20

Frequency = 76 Hz

TRUCK MODEL



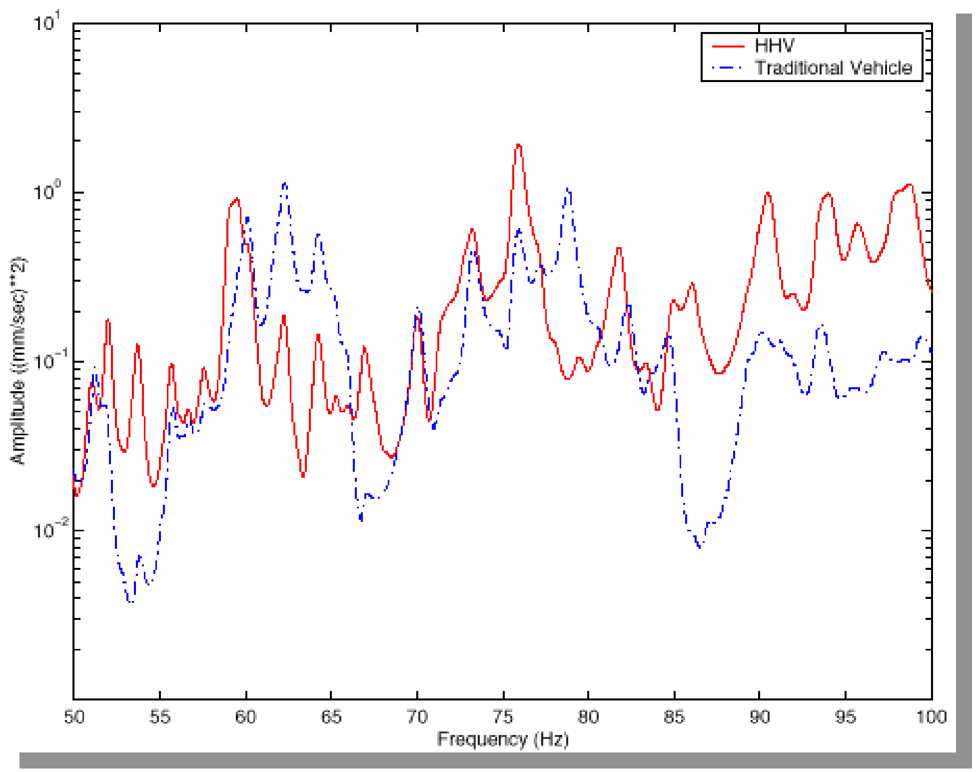
Structural FEA Model



Acoustic BEA Model

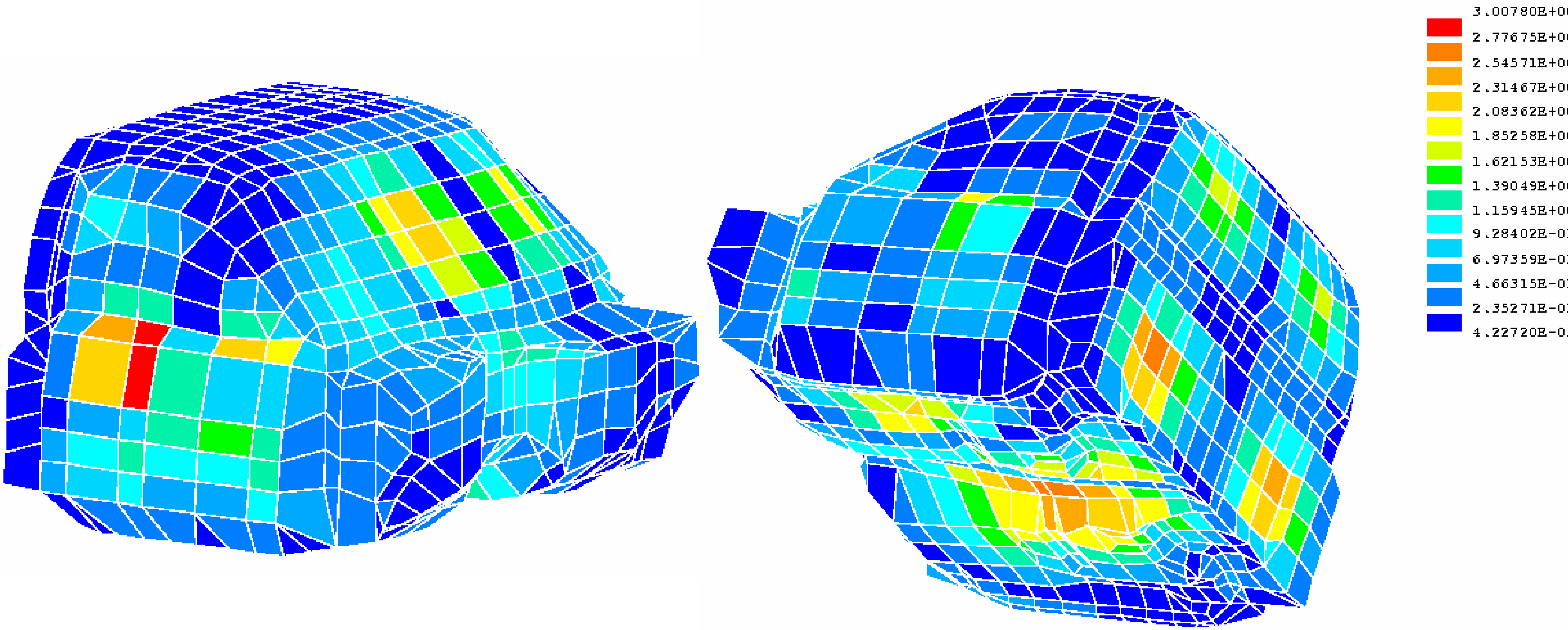
- Design Parameters : 40 Panel Thicknesses
- Performance Measure: Weight, Panel Velocity, Driver Position Pressure

VELOCITY PROFILES



Cabin vibration vs. frequency
(aggregate velocity response
at representative nodes)

CABIN NORMAL VELOCITY



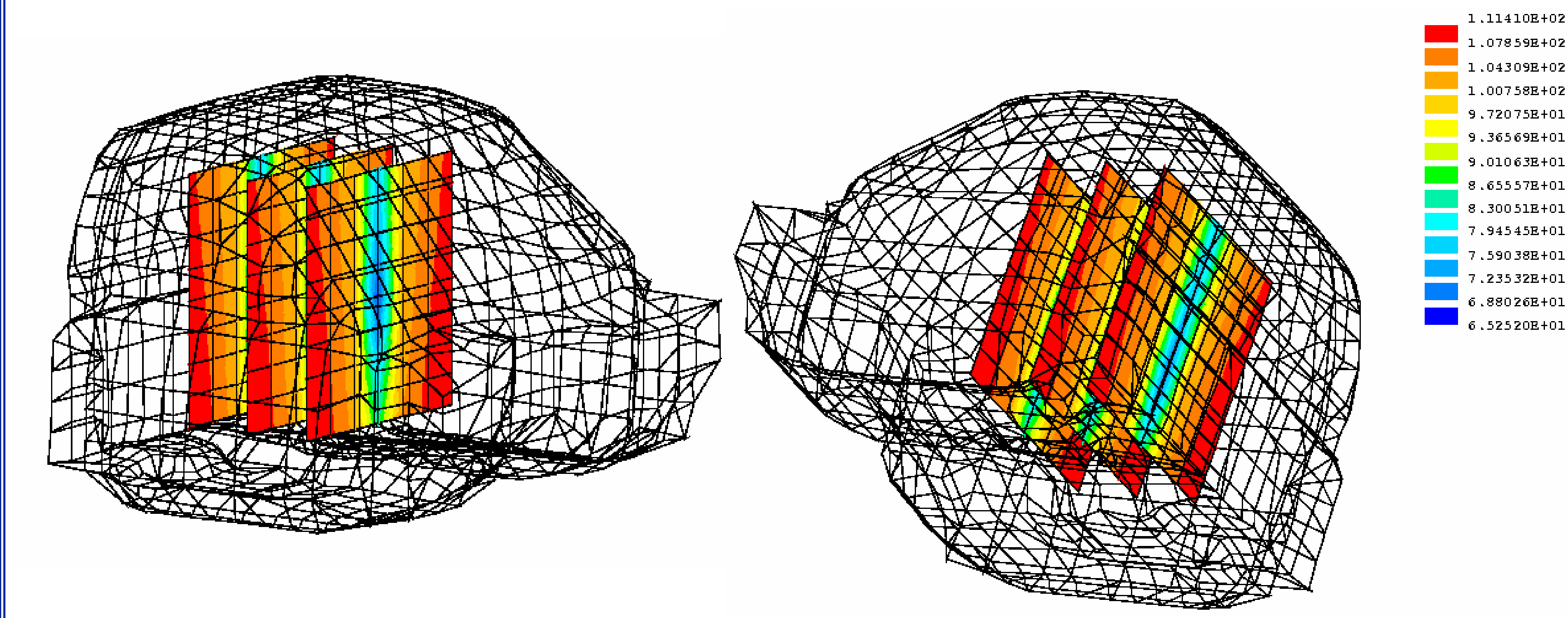
Magnitude of normal velocity, 94Hz

PRESSURE RESULTS

Frequency(Hz)	Pressure ($kg/mm \cdot sec^2$)	Phase Angle (Degree)
47.3	0.64275E-04	66.92
59.5	0.35889E-03	328.99
75.9	0.66052E-04	193.91
81.8	0.41081E-03	264.21
86.0	0.21629E-03	176.18
90.5	0.43862E-03	171.44
94.0	0.75627E-02	178.30
98.7	0.22676E-03	226.07

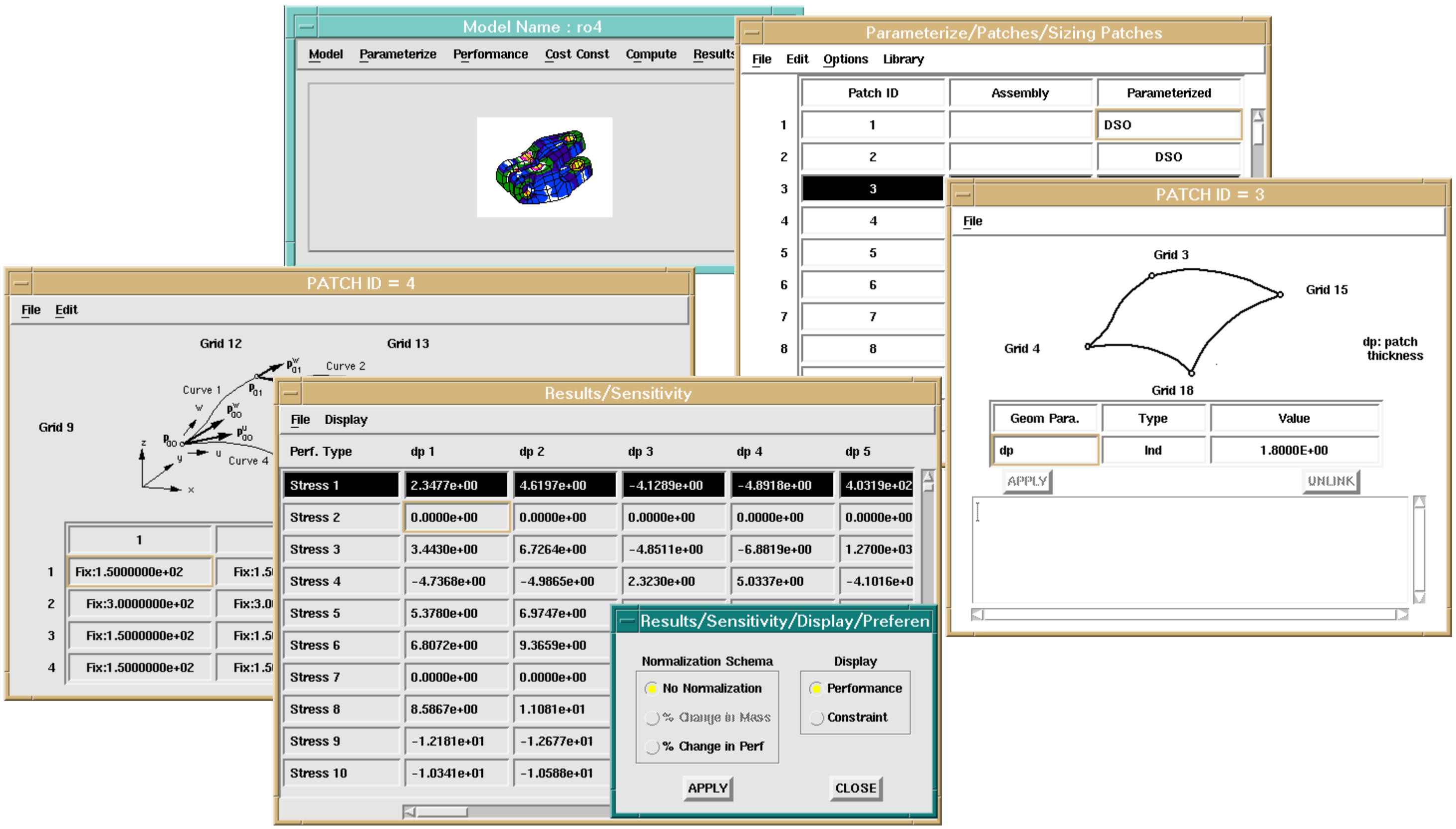
Measure Point: (–800, –440, –370)

CABIN INTERIOR NOISE (dB)



Interior noise levels, 94Hz

DSA AND OPTIMIZATION TOOL (DSO)



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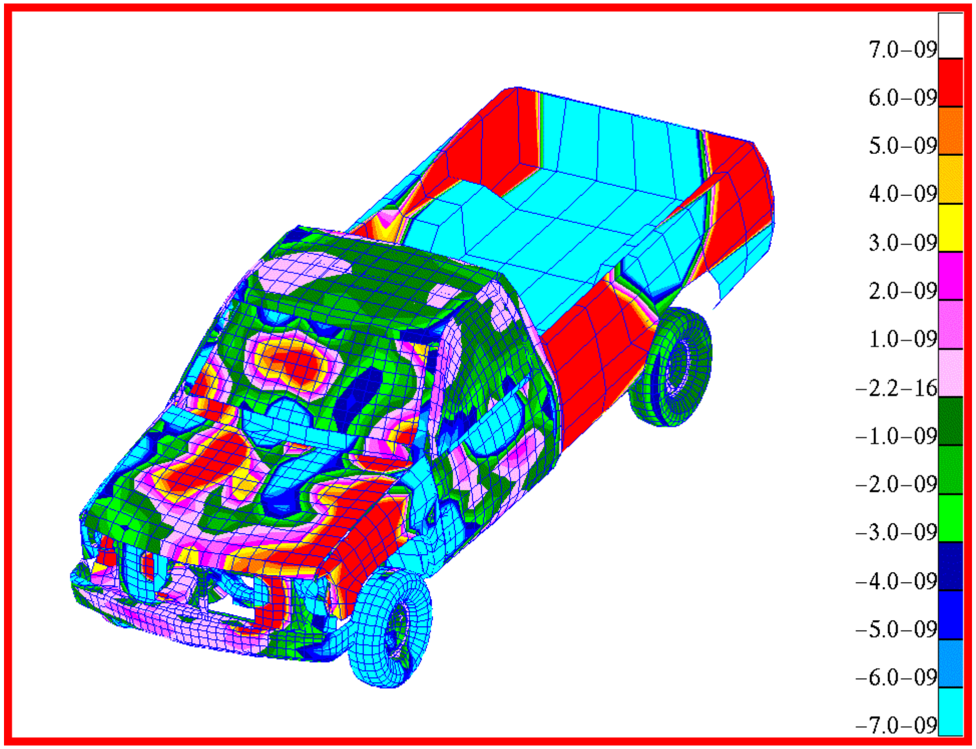
ELEMENT SENSITIVITY PLOT

Frequency = 94 Hz

Element Thickness Design Parameter

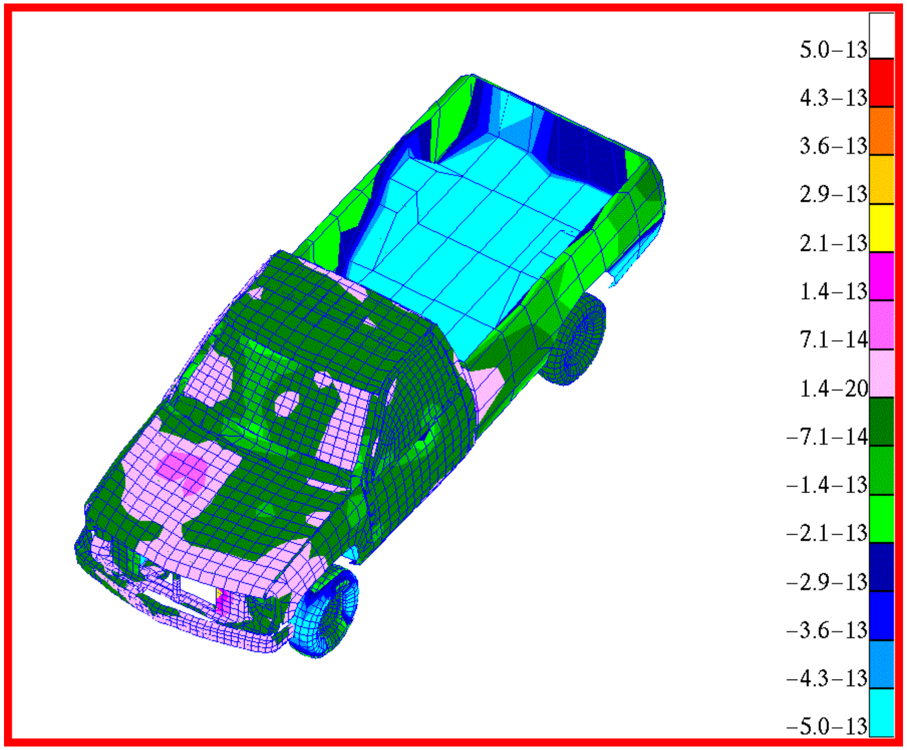


Pressure Sensitivity at (-800,-440,-370)

SPL SENSITIVITY TO MATERIAL PROPERTY

Frequency = 94 Hz

Material Property Design Parameter



Pressure Sensitivity at (-800,-440,-370)

DSA RESUTLS

v_z at the Center of Cabin Roof, Value = .40293
Perturbation = 0.1%, Frequency = 48 Hz

Design	Perturbed	FDM	DSA	Ratio (%)
Bumper	.40292	-3.5739E-3	-3.9091E-3	91.43
Rails	.40196	-3.1287E-1	-3.0824E-1	101.50
Arm LL	.40288	-9.8022E-3	-9.6368E-3	101.72
Arm LR	.40250	-9.0502E-2	-9.6967E-2	93.33
Tire Rim	.39692	-6.1762E-1	-6.4708E-1	95.44
Oil Box	.40293	1.9519E-3	2.0538E-3	95.04
Brake FL	.40289	-6.9373E-3	-6.4794E-3	107.07
Brake FR	.40239	-1.0890E-1	-9.7718E-2	111.45
Rail Conn	.40274	-5.2836E-2	-5.2732E-2	100.20
Arm Conn UL	.40293	-4.1533E-5	-4.1283E-5	100.60
Arm Conn UR	.40293	-1.1367E-5	-1.0735E-5	105.89

DSA RESULTS cont.

v_y at the Center of Right Door, Value = .19688
Perturbation = 0.1%, Frequency = 48 Hz

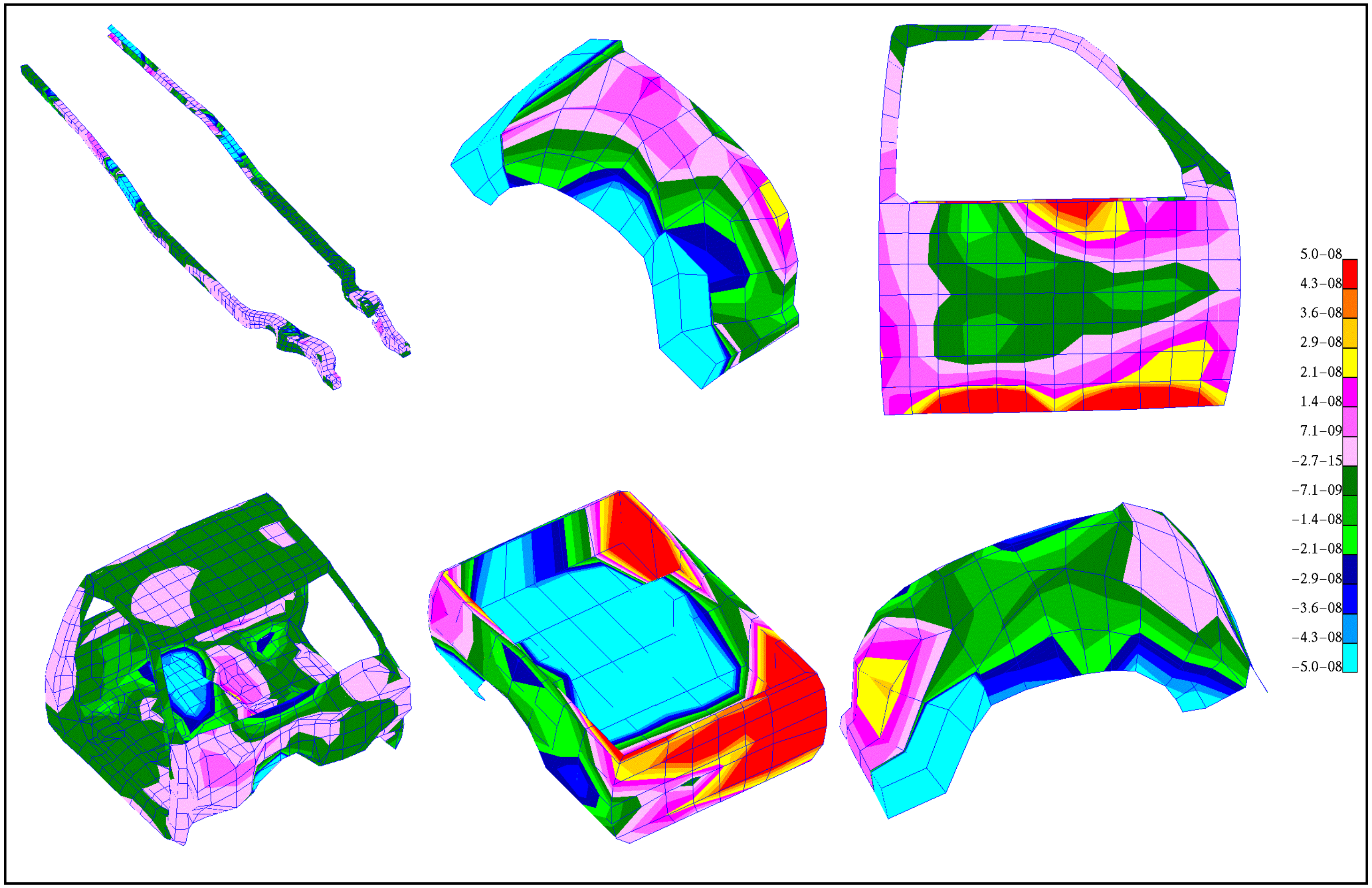
Design	Perturbed	FDM	DSA	Ratio (%)
Bumper	.19686	-7.5637E-3	-7.3715E-3	102.61
Rails	.19648	-1.2939E-1	-1.3230E-1	97.80
Arm LL	.19676	-2.5714E-2	-2.7676E-2	92.91
Arm LR	.19693	1.1412E-2	1.1352E-2	100.53
Tire Rim	.19662	-3.0087E-2	-3.0631E-2	98.22
Oil Box	.19688	-9.5276E-4	-8.8797E-4	107.30
Brake FL	.19676	-3.2308E-2	-2.9284E-2	110.33
Brake FR	.19691	6.8055E-3	6.6268E-3	102.70
Rail Conn	.19701	3.5830E-2	3.6917E-2	97.06
Arm Conn UL	.19686	-3.5325E-4	-3.5081E-4	100.70
Arm Conn UR	.19687	-2.0189E-4	-2.0063E-4	100.63

NORMALIZED PANEL SENSITIVITY

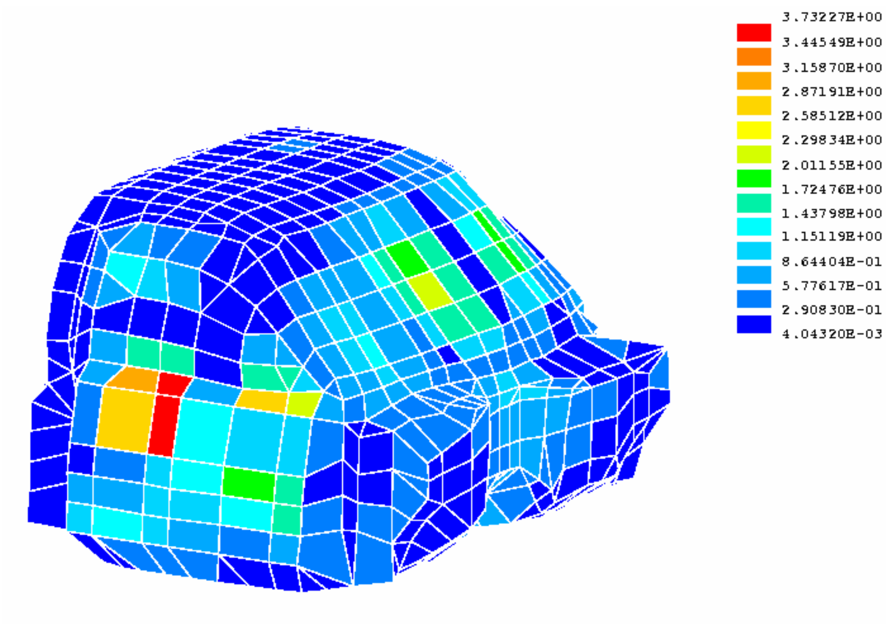
Component	DSA	Component	DSA
Rails	–1.00	Rail MTG	–0.11
Left wheelhouse	–0.82	Rail connectors	–0.10
Right door	0.73	Right fender	–0.07
Cabin	–0.35	Left door	–0.06
Right wheelhouse	–0.25	Bumper	–0.03
Bed	–0.19	Rear glass	0.03

with respect to panel thickness

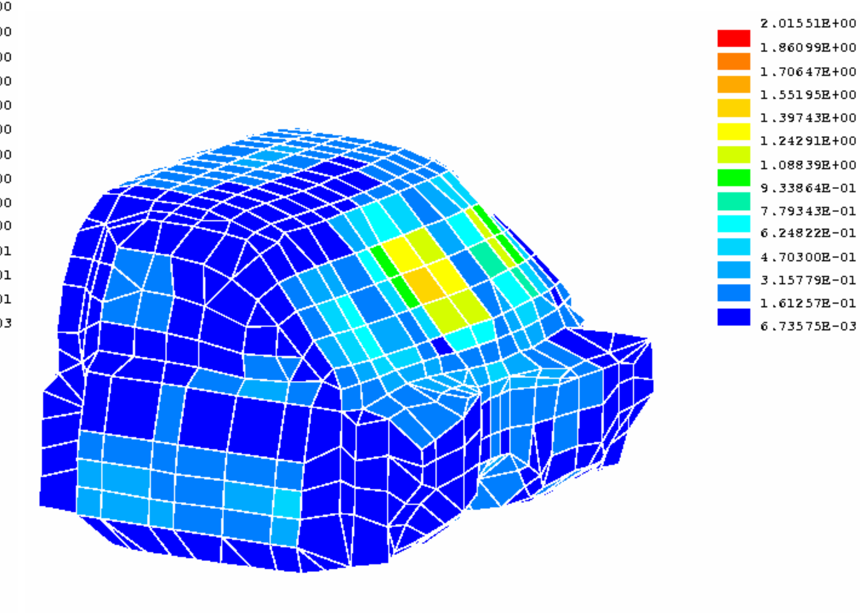
PANEL SENSITIVITY



IMPROVEMENTS IN THE NORMAL VELOCITY



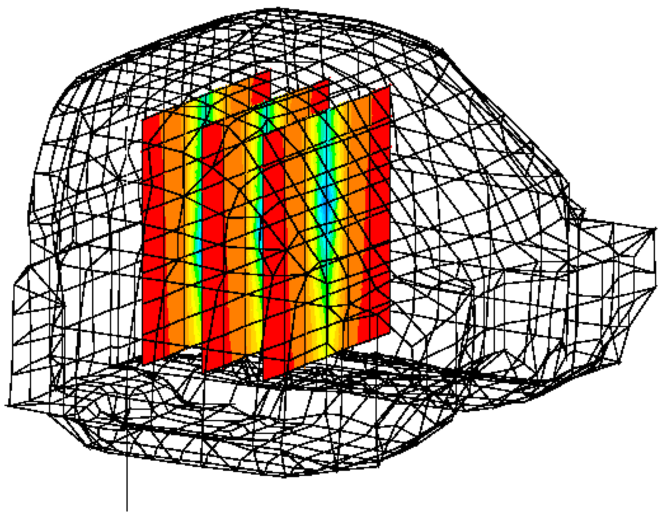
Original design
(Max velocity: 3.7)



Improved design
(Max velocity: 2.0)

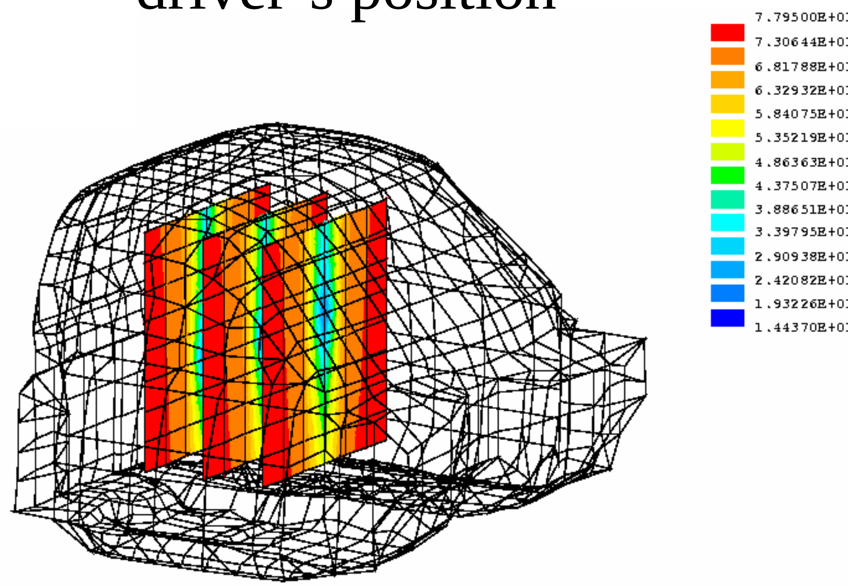
Improved design: Increased the frame thickness by 1.0 mm

COMPARISON IN SOUND PRESSURE LEVEL



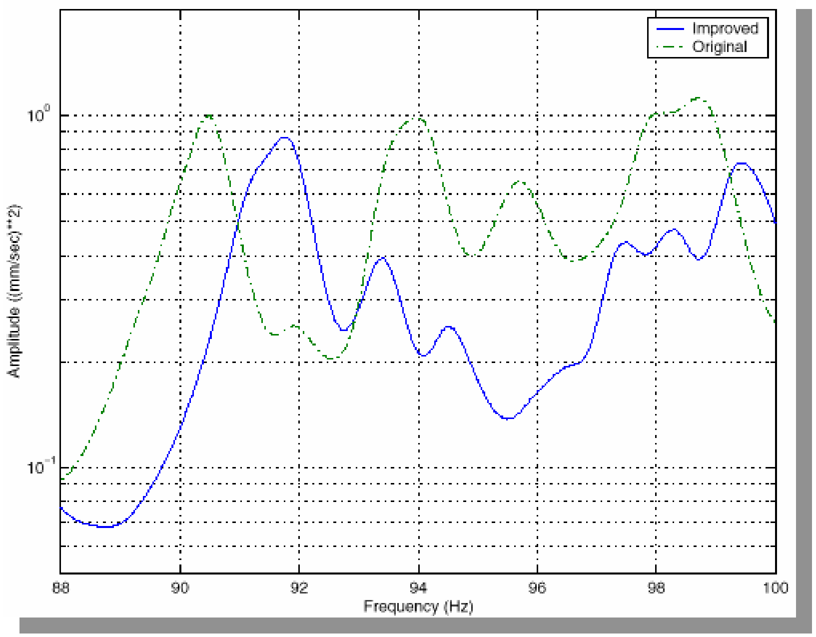
Original design
(Max SPL: 77.8dB)

- SPL at 93.6Hz
- 2.8 dB reduction at the driver's position

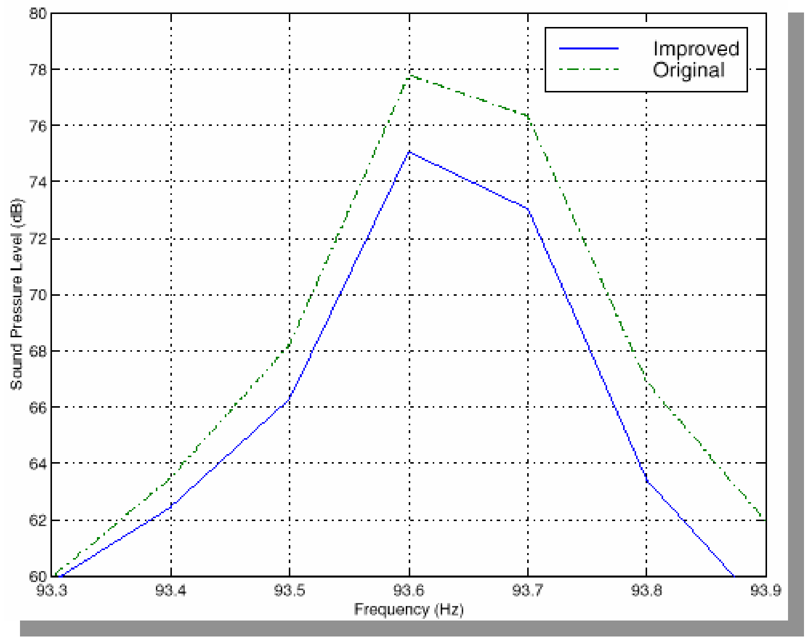


Improved design
(Max SPL: 75.0dB)

IMPROVEMENTS IN PREDICTED VIBRATION AND NOISE



Cabin Vibration



Sound Pressure Level

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CONCLUSIONS AND PLANS

Conclusions

- An efficient sequential design sensitivity analysis is proposed for the FEA-BEA based NVH model.
- The adjoint load is obtained from the BE re-analysis, and the adjoint response is obtained from the structural FE re-analysis.

Future Plans

- Component mode synthesis will be used to reduce the response analysis costs.
- Shape and configuration DSA will be developed to improve NVH performance more effectively.