Structure Borne NVH Workshop

Workshop Objectives -
2. Propose Generic Targets.

Intended Audience –
• New NVH Engineers.
• “Acoustics” Engineers seeking new perspective.
• “Seasoned Veterans” seeking to brush up skills.
Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- New Technology
  - Uncertainty and NVH Scatter
- Closing Remarks
Structure Borne NVH Workshop

• Introduction
• Low Frequency Basics
• Mid Frequency Basics
• Live Noise Attenuation Demo
• New Technology
  Uncertainty and NVH Scatter
• Closing Remarks
Competing Vehicle Design Disciplines

- Ride and Handling
- Impact Crash Worthiness
- NVH
- Durability

Cost, Weight, Investment, Manufacturing
Automotive Engineering Objectives are Timeless
Structure Borne NVH Workshop

• Introduction
• **Low Frequency Basics**
• **Mid Frequency Basics**
• *Live Noise Attenuation Demo*
• New Technology
  - Uncertainty and NVH Scatter
• Closing Remarks
Structure Borne Noise and Vibration

Frequency Range: up to 1000 Hz

System Characterization

- Source of Excitation
- Transmission through Structural Paths
- “Felt” as Vibration
- “Heard” as Noise
Automotive NVH Frequency Range

- Structure Borne Noise
- Airborne Noise

Log Frequency

- Global Stiffness + Damping: "Low"
- Local Stiffness + Damping: "Mid"
- Absorption + Mass + Sealing + Damping: "High"

Frequency:
- ~ 150 Hz
- ~ 1000 Hz
- ~ 10,000 Hz
Low Frequency Basics

- **Source-Path-Receiver Concept**
- **Single DOF System Vibration**
- **NVH Source Considerations**
- **Receiver Considerations**
- **Vibration Attenuation Strategies**
  - Provide Improved Isolation
  - Mode Management
  - Nodal Point Mounting
  - Dynamic Absorbers
Low Frequency Basics

• **Source-Path-Receiver Concept**

• Single DOF System Vibration

• NVH Source Considerations

• Receiver Considerations

• Vibration Attenuation Strategies
  
  *Provide Improved Isolation*
  
  *Mode Management*
  
  *Nodal Point Mounting*
  
  *Dynamic Absorbers*
Structure Borne NVH Basics

- **Source**
- **Path**
- **Receiver**
Low Frequency Basics

- **Source-Path-Receiver Concept**
- **Single DOF System Vibration**
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Single Degree of Freedom Vibration

APPLIED FORCE

\[ F = F_0 \sin 2 \pi f t \]

\[ TR = \frac{F_T}{F} = \frac{1 + (2\zeta \frac{f}{f_n})^2}{\sqrt{(1 - \frac{f^2}{f_n^2})^2 + (2\zeta \frac{f}{f_n})^2}} \]

- \( \zeta \) = fraction of critical damping
- \( f_n \) = natural frequency \( \sqrt{k/m} \)
- \( f \) = operating frequency
Vibration Isolation Principle

Transmissibility Ratio

APPLIED FORCE

\[ F = F_0 \sin 2\pi f t \]

\[ TR = \frac{F_T}{F} \]

Isolation Region

F_T Transmitted Force

\[
\begin{align*}
1.0 & , 0.75 & , 0.5 & , 0.375 & , 0.25 & , 0.15 & , 0.1 \\
\end{align*}
\]

\[
\begin{align*}
0 & , 1 & , 2 & , 3 & , 4 & , 5 \\
\end{align*}
\]

Transmitted Force

Frequency Ratio \((f / f_n)\)

0 1 2 3 4 5

Transmissibility Ratio

0 1 2 3 4

Vibration Isolation Principle
Low Frequency Basics

- Source-Path-Receiver Concept
- Single DOF System Vibration
- **NVH Source Considerations**
- Receiver Considerations
- Vibration Attenuation Strategies
  - Provide Improved Isolation
  - Mode Management
  - Nodal Point Mounting
  - Dynamic Absorbers
NVH Source Considerations

Two Main Sources

Suspension

Powertrain
Typical NVH Pathways to the Passenger
## Structure Borne NVH Sources

<table>
<thead>
<tr>
<th></th>
<th>Inherent</th>
<th>Process Variation</th>
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<tbody>
<tr>
<td><strong>Suspension Induced</strong></td>
<td><strong>Tar Strip Impacts</strong></td>
<td><strong>Tire/Wheel Unbalance</strong></td>
</tr>
<tr>
<td></td>
<td><strong>Rough Road Surface</strong></td>
<td><strong>Tire Force Variation</strong></td>
</tr>
<tr>
<td><strong>Driveline Induced</strong></td>
<td><strong>Engine Fuel Combustion and Reciprocating Unbalance</strong></td>
<td><strong>Driveshaft and Halfshaft Unbalances</strong></td>
</tr>
<tr>
<td></td>
<td>. Idle-in-Gear</td>
<td><strong>Torque Converter Unbalance</strong></td>
</tr>
<tr>
<td></td>
<td>. Constant Speed Cruise</td>
<td><strong>Axle Gear Mesh Variation</strong></td>
</tr>
<tr>
<td></td>
<td>. WOT Acceleration</td>
<td></td>
</tr>
</tbody>
</table>
Primary Consideration:

Reduce the Source first as much as possible because whatever enters the structure is transmitted through multiple paths to the receiver.

Transmission through multiple paths is more subject to variability.
Low Frequency Basics

• Source-Path-Receiver Concept
• Single DOF System Vibration
• NVH Source Considerations
• Receiver Considerations
• Vibration Attenuation Strategies
  Provide Improved Isolation
  Mode Management
  Nodal Point Mounting
  Dynamic Absorbers
Receiver Considerations
Subjective to Objective Conversions

Subjective NVH Ratings are typically based on a 10 Point Scale resulting from Ride Testing

Receiver Sensitivity is a Key Consideration

\[ A_2 \approx \frac{1}{2} A_1 \]

Represents 1.0 Rating Change

TACTILE: 50% reduction in motion

SOUND: 6.dB reduction in sound pressure level
( long standing rule of thumb )
Low Frequency Basics

- Source-Path-Receiver Concept
- Single DOF System Vibration
- NVH Source Considerations
- Receiver Considerations

**Vibration Attenuation Strategies**

Provide Improved Isolation
Mode Management
Nodal Point Mounting
Dynamic Absorbers
Symbolic Model of Unibody Passenger Car
8 Degrees of Freedom

Total 2178.2 Kg (4800LBS)
Sprung 1996.7 Kg
Unsprung 181.5 Kg (8.33% of Total)
Powertrain 181.5 Kg

Tires 350.3 N/mm
KF 43.8 N/mm
KR 63.1 N/mm

Beam mass lumped on grids like a beam
M2,3,4 = 2 * M1,5

From Reference 6
2013 Automotive Analytics LLC A.E.Duncan
8 Degree of Freedom Vehicle NVH Model

- Engine Mass
- Engine Isolator
- Flexible Beam for Body
- Suspension
- Springs
- Wheels
- Tires
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly

$F_{\text{eng}}$

Forces at Powertrain could represent a First Order Rotating Imbalance
Engine Isolation Example

Response at Mid Car

Constant Force Load; $F \sim A$

Frequency (Hz)

Velocity (mm/sec)

15.9 Hz
8.5 Hz
7.0 Hz

700 Min. RPM First Order Unbalance
Operation Range of Interest

2013 SAE NVC Structure Borne Noise Workshop
Engine Isolation Example

Response at Mid Car

Constant Force Load; \( F \sim A \)

Engine Idle Speed
Operating Shapes at 700 RPM

Highest Body Movement

5.0 10.0

Frequency (Hz)

0.0001 0.0010 0.0100 0.1000 1.0000

Velocity (mm/sec)

700 Min. RPM First Order Unbalance

Operation Range of Interest

3 1 8 2 6 4 7 5 3

15.9 Hz 8.5 Hz 7.0 Hz

Engine Idle Speed

Operating Shapes at 700 RPM

Lowest Body Movement
Concepts for Increased Isolation

“Double” isolation is the typical strategy for further improving isolation of a given vehicle design.

Second Level of Isolation is at Subframe to Body Mount

Subframe is Intermediate Structure

Suspension Bushing is first level
8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect

Wheel Mass Removed
Double Isolation Example

Vertical Response at DOF3

Velocity (mm/sec)

Base Model

Without Double_ISO

Frequency Hz

5.0 10.0 15.0 20.0

0.0E+00 1.0E+00 2.0E+00 3.0E+00 4.0E+00 5.0E+00 6.0E+00

1.414*f_n
Low Frequency Basics

- Source-Path-Receiver Concept
- Single DOF System Vibration
- NVH Source Considerations
- Receiver Considerations

- Vibration Attenuation Strategies
  Provide Improved Isolation
  Mode Management
  Nodal Point Mounting
  Dynamic Absorbers
**Mode Management Chart**

**EXCITATION SOURCES**
- Inherent Excitations (General Road Spectrum, Reciprocating Unbalance, Gas Torque, etc.)
- Process Variation Excitations (Engine, Driveline, Accessory, Wheel/Tire Unbalances)

**CHASSIS/POWERTRAIN MODES**
- Ride Modes
- Powertrain Modes
- Suspension Hop and Tramp Modes
- Suspension Longitudinal Modes
- Exhaust Modes

**BODY/ACOUSTIC MODES**
- Body First Bending
- Body First Torsion
- Steering Column First Vertical Bending
- First Acoustic Mode

**EXCITATION SOURCES**
- First Order Wheel/Tire Unbalance
- V8 Idle Hot - Cold

(See Ref. 1)

2013 Automotive Analytics LLC  A.E.Duncan
Beam Stiffness was adjusted to align Bending Frequency with Suspension Modes and then progressively separated back to Baseline.
8 DOF Mode Separation Example

Response at Mid Car

- 18.2 Hz Bending
- 13.0 Hz Bending
- 10.6 Hz Bending

Frequency (Hz) vs. Velocity (mm/sec)
All Operating Shapes at 10.6 Hz

Highest Body Bending
Low Frequency Basics

• **Source-Path-Receiver Concept**
• **Single DOF System Vibration**
• **NVH Source Considerations**
• **Receiver Considerations**

• **Vibration Attenuation Strategies**
  - Provide Improved Isolation
  - Mode Management
  - **Nodal Point Mounting**
  - Dynamic Absorbers
Mount at Nodal Point

*First Bending*: Nodal Point Mounting Example

Locate wheel centers at node points of the first bending modeshape to prevent excitation coming from suspension input motion.
Passenger sits at node point for First Torsion.

Transmission Mount of a 3 Mount N-S P/T is near the Torsion Node.

2013 Automotive Analytics LLC  A.E.Duncan
Mount system is placed to support Powertrain at the Nodal Locations of the First order Bending Mode. Best compromise with Plan View nodes should also be considered.
8 Degree of Freedom Vehicle NVH Model

Bending Node Alignment with Wheel Centers

Redistribute Beam Masses to move Node Points to Align with points 2 and 4
First Bending Nodal Point Alignment

Response at Mid-Car

Velocity (mm/sec)

Node Shifted

Base Model

Frequency Hz

5.0 10.0 15.0 20.0
Response at Mid-Car

Node Shifted Base Model

First Bending Nodal Point Alignment

Node Shifted Model

Operating Shapes at 18.2 Hz

No Residual Body Bending
Diagnosis: Increase at 10.2 Hz

Body Bends up at Downward Position of Cycle
Diagnosis: Increase at 10.2 Hz

Concentration - Undeformed Position

Motion Experienced when Bending is Present

Motion Experienced when Bending is Removed

Up Position

Down Position

Conclusion: Response Increases when a Beneficial mode is Removed.
Low Frequency Basics

• Source-Path-Receiver Concept
• Single DOF System Vibration
• NVH Source Considerations
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Dynamic Absorber Concept

Auxiliary Spring-Mass-Damper

m = M / 10

SDOF

2DOF

Displacement mm

Y_0

Frequency Hz

0.0 0.5 1.0 1.5 2.0

0.0 0.5 1.0 1.5 2.0

Y_0

M

M
Powertrain Example of Dynamic Absorber

Anti-Node Identified at end of Powerplant

Absorber attached at anti-node acting in the Vertical and Lateral plane.

Tuning Frequency = \( \sqrt{\frac{k}{m}} \)

[Figure Courtesy of DaimlerChrysler Corporation]
Baseline Sound Level
63 Hz Dynamic Absorber
63 + 110 Hz Absorbers

(Sound dBA Baseline -VS- Dual Absorbers @ 63 & 110 Hz)

10 dB

2nd Order Freq. -Hz-
Low Frequency Basics - Review

- **Source-Path-Receiver Concept**
- **Single DOF System Vibration**
- **NVH Source Considerations**
- **Receiver Considerations**
- **Vibration Attenuation Strategies**

  - Provide Improved Isolation
  - Mode Management
  - Nodal Point Mounting
  - Dynamic Absorbers
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• Low Frequency Basics
• Mid Frequency Basics  Jianmin Guan
• *Live Noise Attenuation Demo*
• New Technology
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Mid Frequency NVH Fundamentals

This looks familiar!

Frequency Range of Interest has changed to 150 Hz to 1000 Hz
Typical NVH Pathways to the Passenger

Noise Paths are the same as Low Frequency Region

PATHS FOR STRUCTURE BORNE NVH
Mid-Frequency Analysis Character

- **Structure Borne Noise**

- **Airborne Noise**
  - Mode separation is less practical in mid-frequency
  - New Strategy is Effective Isolation: Achieved by reducing energy transfer locally between source and receiver at key paths.

- High modal density and coupling in source, path and receiver

- Local Stiffness + Damping

- "Mid"

- \(~ 150 \text{ Hz} \) \( \, \text{~ 1000 \text{ Hz} } \) \( \, \text{~ 10,000 \text{ Hz} } \)

- Log Frequency
Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths
Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths
Isolation Effectiveness

Classical SDOF: Rigid Source and Receiver

“Real Structure” Flexible (Mobile) Source and Receiver

Isolation Region

Effectiveness deviates from the classical development as resonances occur in the receiver structure and in the foundation of the source.
Mobility

- **Mobility** is the ratio of velocity response at the excitation point on structure where point force is applied

\[
\text{Mobility} = \frac{\text{Velocity}}{\text{Force}}
\]

- **Mobility**, related to **Admittance**, characterizes **Dynamic Stiffness** of the structure at load application point

\[
\text{Mobility} = \frac{\text{Frequency} \times \text{Displacement}}{\text{Force}}
\]

\[
= \frac{\text{Frequency}}{\text{Dynamic Stiffness}}
\]
Isolation

• The isolation effectiveness can be quantified by a theoretical model based on analysis of mobilities of receiver, isolator and source

• Transmissibility ratio is used to objectively define measure of isolation

\[
TR = \frac{\text{Force from source with isolator}}{\text{Force from source without isolator}}
\]

\[
F_s = \frac{V}{Y_i + Y_r + Y_s}
\]

\[
F_s = \frac{V}{Y_r + Y_s}
\]
TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}

\begin{align*}
TR = \left| \frac{(Y_r + Y_s)}{(Y_i + Y_r + Y_s)} \right|
\end{align*}

- $Y_r$: Receiver mobility
- $Y_i$: Isolator mobility
- $Y_s$: Source mobility

- *For Effective Isolation (Low TR) the Isolator Mobility must exceed the sum of the Source and Receiver Mobilities.*

Recall that $\gamma \propto \frac{1}{K}$
Designing Noise Paths

\[ TR = \left| \frac{\frac{1}{K_{\text{body}}} + \frac{1}{K_{\text{source}}}}{\frac{1}{K_{\text{body}}} + \frac{1}{K_{\text{iso}}} + \frac{1}{K_{\text{source}}}} \right| \]

<table>
<thead>
<tr>
<th></th>
<th>( K_{\text{body}} )</th>
<th>( K_{\text{source}} )</th>
<th>( K_{\text{iso}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.67</td>
<td>0.55</td>
<td>0.51</td>
</tr>
<tr>
<td>5.0</td>
<td>0.55</td>
<td>0.29</td>
<td>0.20</td>
</tr>
<tr>
<td>20.0</td>
<td>0.51</td>
<td>0.20</td>
<td>0.09</td>
</tr>
</tbody>
</table>

Generic targets:
- body to bushing stiffness ratio of at least 5.0
- source to bushing stiffness ratio of at least 20.0
Body-to-Bushing Stiffness Ratio Relationship to Transmissibility

For a source ratio of 20

Target Min. = 5 gives TR = .20

Transmissibility Ratio TR

0 0.1 0.2 0.3 0.4 0.5 0.6
1 2 3 4 5 6 7 8 9 10

Stiffness Ratio; $K_{body}/K_{iso}$
Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths
Identifying Key NVH Paths

Key NVH paths are identified by Transfer Path Analysis (TPA)

Operating loads

Total Acoustic Response is summation of partial responses over all noise paths

\[ P_t = \sum_{\text{paths}} [P_i] = \sum_{\text{paths}} [(P/F)_i \times F_i] \]
Identifying Key NVH Paths

TPA Example: Contribution at One Transfer Path

Partial response from a particular path: \( P_i = (P/F)_i \times F_i \)
Identifying Key NVH Paths

TPA Example: Sum of Key Transfer Paths at One Peak

Total Response: $P_t = \sum \text{paths} [P_i] = \sum \text{paths} [(P/F)_i * F_i]$
Attenuating Key NVH Paths

TPA Example: Identifying Root Cause of Dominant Paths
Attenuating Key NVH Paths

TPA Example: Dominant Paths over Frequencies
Designing Noise Paths

TPA Example: Cascading Vehicle Targets to Subsystems

Once the dominant noise paths and root cause have been identified, the task is reduced to solving problems of:
1. High Force
2. High Transfer Function
3. High Point Mobility

Limit TF to a 55 dB target
See changes in response

TF<sub>i</sub>

P/T Load
Crank torque
Designing Noise Paths

Operating loads create Forces ($F_i$) into body at All noise paths

$$P_t = \sum_{\text{paths}} [P_i] = \sum_{\text{paths}} [F_i \times (P/F)_i] = \sum_{\text{paths}} [F_i \times (P/V)_i \times (V/F)_i]$$

Measurement Parameters

<table>
<thead>
<tr>
<th>P/F</th>
<th>Acoustic Sensitivity</th>
<th>Generic Targets</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>50 - 60 dBL/N</td>
<td></td>
</tr>
<tr>
<td>V/F</td>
<td>Structural Point Mobility (Receiver Side)</td>
<td>0.2 to 0.3 mm/sec/N</td>
</tr>
</tbody>
</table>
“Downstream” Effects: Body Panels

Recall for Acoustic Response $P_t$

$$P_t = \sum_{paths} [P_i] = \sum_{paths} [F_i \times (P/V)_i \times (V/F)_i]$$

$(P/V)_i \Rightarrow “Downstream” \ (Body \ Panel) \ System \ Dynamics: \ Three \ Main \ Effects:$

1. Panel Damping

2. Panel Stiffness

3. Panel Acoustic Contribution
Generic Noise Path Targets

Primary: Minimize the Source Force

\[ \frac{K_{\text{body}}}{K_{\text{iso}}} > 5.0 \]

\[ \frac{K_{\text{source}}}{K_{\text{iso}}} > 20.0 \]

Structural Mobility

\[ < 0.2 \text{ to } 0.3 \text{ mm/sec/N} \]

Acoustic Sensitivity

\[ < 50 - 60 \text{ dBL/N} \]

Panel Damping Loss Factor

\[ > .10 \]
Final Remarks on Mid Frequency Analysis

- Effective isolation at dominant noise paths is critical
- Reduced mobilities at body & source and softened bushing are key for effective isolation
- Mode Separation remains a valid strategy as modes in the source structure start to participate
- Other means of dealing with high levels of response (Tuned dampers, damping treatments, isolator placement at nodal locations) are also effective
Structure Borne NVH: Concepts Summary

- **Source-Path-Receiver as a system**
  1. Reduce Source
  2. Rank and Manage Paths
  3. Consider Subjective Response

- Effective Isolation
- Mode Management
- Nodal Point Placement
- Attachment Stiffness
- “Downstream” (Body Panel) Considerations
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Greg Goetchius
Toolbox Demo Noise Test Results

1) Baseline: Imbalance, No Isolation
2) Imbalance + Isolation
3) No Imbalance, No Isolation
4) No Imbalance, No Isolation + Damping
5) No Imbalance + Isolation + Damping
6) #5 + Absorption
7) #6 + Insulator Mat

SPL (dBA):
- Baseline: 85 dBA
- Imbalance + Isolation: 61 dBA
- No Imbalance, No Isolation: 68 dBA
- No Imbalance, No Isolation + Damping: 63 dBA
- No Imbalance + Isolation + Damping: 55 dBA
- #5 + Absorption: 47 dBA
- #6 + Insulator Mat: 39 dBA
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 NVH Scatter and Uncertainty

Closing Remarks

Alan Duncan
NVH Scatter and Uncertainty

- Scatter 20 Years Ago
- Scatter 10 Years Ago
- New Technology to Address Scatter
NVH Scatter and Uncertainty

• Scatter 20 Years Ago

• Scatter 10 Years Ago

• New Technology to Address Scatter
Magnitude of 99 Structure Borne Noise Transfer Functions for Rodeo’s at the Driver Microphone Measurements from Kompella and Bernhard (Ref. 8) ©1993 Society of Automotive Engineers, Inc.
Freymann, BMW NVH Scatter Results

Experimentally detected Scatter in low frequency vibroacoustic behavior of production vehicles.

Acoustic Scatter from Simulation of the vibroacoustic behavior of a vehicle due to possible tolerances in the component area and in the production process.

12 dB Variation

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CONCLUSIONs: From K/B and BMW Studies

It is not highly probable:
1. that a Single Test will represent the Mean response
2. that a CAE Simulation will match a Single Test

Variability observed from multiple tests of “identical” vehicles is important in understanding the degree to which the Test (or Simulation) of a Design is representative of the Mean response. How many Tests (or Simulations) of a Design would be required for the result to be considered statistically significant?

Scatter is the “Physics”
NVH Scatter and Uncertainty

• Scatter 20 Years Ago

• Scatter 10 Years Ago

• New Technology to Address Scatter
Reference Baseline Confidence Criterion
For Operating Response Simulations

Test Variation Band
10. dB; 50-150 Hz
20. dB; 150-500 Hz

Sound FRF

Frequency Hz

Simulation Prediction

Test Upper Bound
Test Band Average
Test Lower Bound

Confidence Criterion: Simulation result must fall within the band of test variation.

ABSTRACT

Many papers have been published on variation in noise and vibration as well as transfer function characteristics between individual vehicles with nominally identical design [1], [2] and [3]. However, prediction of Noise Vibration and Harshness (NVH) properties is mostly based on detailed, transfer functions are usual and typical in serial production of road vehicles ([1] and [2]), aircraft, ships, appliances etc.

This variability is often considered to be due to tolerances in assembly or imperfections of supplied components, and QC-programs are introduced by many car manufacturers, with the aim to minimize the number of cars with unacceptable NVH properties by reducing this variability.

(Ref. 15)
Figure 1: a) The complex FRF and its modal components, at one position and frequency.
NVH Scatter and Uncertainty

• Scatter 20 Years Ago

• Scatter 10 Years Ago

• New Technology to Address Scatter
Goal: Define a Modeling Methodology that:
1. Accounts for NVH Scatter
2. Quantifies Uncertainty with Statistical Significance using a Stochastic Model
3. Accounts for the Combined Effect of Modeling and Manufacturing Uncertainty

(Ref. 12)
(See Ref. 11 for Model Details)
Non-Parametric Probabilistic Simulation; Soize, Durand, Gagliardini

Standard Eqn. for Structural-Acoustic Coupling Modal Model


The scatter created is a function of:
Dispersion Parameter: \( \delta \)
Once determined, \( \delta \) is a constant controlling the amplitude level of scatter.

\[ \begin{bmatrix} A^s_n(\omega) & C_{n,m} \\ \omega^2 [C_{n,m}]^T & A^a_m(\omega) \end{bmatrix} \begin{bmatrix} Q^s(\omega) \\ Q^a(\omega) \end{bmatrix} = \begin{bmatrix} F^s(\omega) \\ F^a(\omega) \end{bmatrix}, \quad (12) \]

in which the random complex matrices \([A^s_n(\omega)]\) and \([A^a_m(\omega)]\) are defined by
\[ [A^s_n(\omega)] = -\omega^2[M^s_n] + i\omega[D^s_n] + [K^s_n] \]
and where \([A^a_m(\omega)] = -\omega^2[M^a_m] + i\omega[D^a_m] + [K^a_m] \).

Let \( H \) be the positive-definite symmetric \((\nu \times \nu)\) random matrix representing \([G_{M^s_n}], [G_{D^s_n}], [G_{K^s_n}], [G_{M^a_m}], [G_{D^a_m}], [G_{K^a_m}]\) or \([G_{C_{n,m}}]\). The following algebraic representation of random matrix \([G_H]\) allows independent realizations of \([G_H]\) to be constructed. Random matrix \([G_H]\) is written \([G_H] = \left[L_H\right]^T[L_H]\) in which \([L_H]\) is a random upper triangular \((\nu \times \nu)\) real matrix whose random elements are independent random variables defined as follows:

1. For \( j < j' \), the real-valued random variable \([L_H]_{jj'}\) is written as \([L_H]_{jj'} = \sigma_j U_{jj'}\) in which \(\sigma_j = \delta_H (\nu + 1)^{-1/2}\) and where \(U_{jj'}\) is a real-valued Gaussian random variable with zero mean and variance equal to 1. The parameter \(\delta_H\) controlling the dispersion level of random matrix \([H]\) is such that
\[ \delta_H = \sqrt{\text{E} \left\{ \| [G_H] - [I] \|_F^2 \right\} / \nu} \]
in which \([I]\) is the identity matrix and where the subindex \( F \) corresponds to the Frobenius norm.

2. For \( j = j' \), the positive-valued random variable \([L_H]_{jj'}\) is written as \([L_H]_{jj'} = \sigma_j \sqrt{2 V_j}\) in which \(\sigma_j\) is defined above and where \(V_j\) is a positive-valued gamma random variable whose probability density function \(p_{V_j}\) with respect to \(dv\) is written as
\[
p_{V_j}(v) = 1_R(v) \frac{1}{\Gamma\left(\frac{\nu+1}{2} + \frac{1-i}{2} \right)} v^{\frac{\nu+1}{2}} e^{-v},
\]

7 Dispersion Constants are needed:
3 for Structure: M, D, K
3 for Acoustics: M, D, K
1 for Coupling: \(C_{n,m}\)
CONCLUSIONS: Converged at 200 REALIZATIONS and with Struct to 400 and Fluid to 350 Hz

Increasing No. of Modes

Monte Carlo Randomization

Converges at 200 Realizations of the Random Matrices.

Figure 4 Graph of the function $n_F \rightarrow \text{conv}^F(n_F)$ related to the convergence of the random acoustic generalized coordinates.
CONCLUSION:
- The Model shows 95% Confidence Bands with increasing Scatter at higher frequency similar to K-B Study.
Structural-acoustic modeling of automotive vehicles in presence of uncertainties and experimental identification and validation

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(Received 25 September 2007; revised 5 February 2008; accepted 8 June 2008)

The design of cars is mainly based on the use of computational models to analyze structural vibrations and internal acoustic levels. Considering the very high complexity of such structural-acoustic systems, and in order to improve the robustness of such computational structural-acoustic models, both model uncertainties and data uncertainties must be taken into account. In this context, a probabilistic approach of uncertainties is implemented in an adapted computational structural-acoustic model. The two main problems are the experimental identification of the parameters controlling the uncertainty levels and the experimental validation. Relevant experiments have especially been developed for this research in order to constitute an experimental database devoted to structural vibrations and internal acoustic pressures. This database is used to perform the experimental identification of the probability model parameters and to validate the stochastic computational model. © 2008 Acoustical Society of America. [DOI: 10.1121/1.2953316]

PACS number(s): 43.40.At, 43.40.Hb, 43.40.Qi, 43.40.Sk [JGM]

Pages: 1513–1525

I. INTRODUCTION

In the automotive industry, computational structural-acoustic models are nowadays intensively used to analyze the structural-acoustic behavior of vehicles in terms of structural vibrations and internal acoustic levels mainly for the low-frequency range. The present evolution is to extend such computational models to the medium frequency range. In computational structural-acoustic models with experiments. It should be noted that a very few complete and documented experimental databases are available in the literature. Only some elements concerning two databases can be found in the literature (Wood and Joachim, 1987; Kompella and Bernard, 1996). Nevertheless, these two experimental databases cannot easily be used because there are no available comput
FIG. 10. Finite element mesh of the computational structural acoustic model.

(See Ref. 11 for Model Details)
FIG. 16. Comparisons of the stochastic computational model results with the experiments. Graphs of the root mean square of the acoustic pressures averaged in the cavity in dB scale: experiments for the 30 configurations (gray lines); Mean computational model (dashed line); mean value of the random response (mid thin solid line); 95% confidence region: the upper and lower envelopes are the upper and lower thick solid lines.

NOTE: Acoustic Dispersion Parameters are determined here with Mean Structure held Invariant.
FIG. 19. Comparisons of the stochastic computational model results with the experiments for observation Obs6. Graphs of the moduli of the FRFs in dB scale: experiments for the 20 cars (gray lines); Mean computational model (dashed line); mean value of the random response (mid thin solid line); confidence region: the upper and lower envelopes are the upper and lower thick solid lines.

NOTE: Structure Dispersion Parameters are determined here with Mean Acoustic held Invariant.
FIG. 20. Comparisons of the stochastic computational model results with the experiments for the booming noise. Graphs of the moduli of the FRFs in dB(A) scale: experiments for the 20 cars (thin gray lines); mean value of the experiments (thick gray line); Mean computational model (dashed line); Mean value of the random response (mid thin solid line); 95% confidence region: the upper and lower envelopes are the upper and lower thick solid lines.
FIG. 20. Comparisons of the stochastic computational model results with the experiments for the booming noise. Graphs of the moduli of the FRFs in dB(A) scale: experiments for the 20 cars (thin gray lines); mean value of the experiments (thick gray line); Mean computational model (dashed line); Mean value of the random response (mid thin solid line); 95% confidence region: the upper and lower envelopes are the upper and lower thick solid lines.

CONCLUSIONS:
. The 95% Confidence Bands encapsulate the Measured Scatter of 20 Vehicles.
. Half-Bandwidth Scatter (between Mean and Upr 95%) is similar to K-B Half-Bands.

NOTE: The Method has Quantified the Model and Manufacturing Combined Uncertainty.
AN INDUSTRIAL IMPLEMENTATION OF NON-PARAMETRIC STOCHASTIC MODELLING OF VEHICLE VIBROACOUSTIC RESPONSE

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\textsuperscript{(Ref. 14)}
FIG. 4: Computed booming noise inside a car vs. rpm, using only structure-borne excitation compared with the measured value (brown bold). Thin dotted black: deterministic computation; green: median value; thin dotted blue: lower bound with a 95% probability; dotted red: upper bound with a 95% probability.

All Test Peaks in Band: Results imply a Countermeasure is needed.
A-B Design Decision using Deterministic Model - vs - Mean Stochastic Model

FIG. 6: Booming noise inside a car vs rpm. Deterministic models. Comparison of a baseline configuration (green) with a modification set (blue bold).

FIG. 7: Booming noise inside a car vs rpm. Stochastic modelling. Comparison of the median value of a baseline configuration (green) with a modification set (blue bold).
Conclusions:
Observations about Scatter:
Kompella and Bernhard Test Observations are still relevant after 20 Years.

Scatter-like NVH Variation exists even in Best-in-Class vehicles.

Observations from Soize, Durand, Gagliardini, et. al. Papers
The Non-Parametric stochastic computational model with dispersion parameters derived from a test database accounts for NVH Scatter due to combined Modeling and Manufacturing Uncertainties.

The Upper, Lower, and Mean Confidence Probabilities lead to more precise assessment of the effects of NVH scatter.

A database of dispersion parameters enables a virtual product development process assuring robust NVH performance.

The model configuration lends itself to an automated computational methodology driving a robust virtual product development process.
Structure Borne NVH Workshop

• Introduction
• Low Frequency Basics
• Mid Frequency Basics
• Live Noise Attenuation Demo
• New Technology
  Uncertainty and NVH Scatter
• Closing Remarks: Q & A

Alan-Greg-Jimi
Thank You for Your Time!

Q & A
Structure Borne NVH References

Primary References (Workshop Basis: 4 Papers)


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