Structure Borne NVH Basics

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Material Sciences Corporation
NVH Workshop Topic Outline

• Introduction
• Ride Balance in the Ride Range
• NVH Load Conditions
• Low Frequency Basics
• Live Noise Attenuation Demo
• Mid Frequency Basics
• Utilization of Simulation Models
• Closing Remarks
The Fundamental Secret of Structure Borne NVH Performance

Revealed here tonight!
Structure Borne NVH Basics References

Primary References (Workshop Basis: 4 Papers)


Structure Borne NVH Workshop - on Internet

Available online at www.AutoAnalytics.com
Structure Borne NVH Basics References

Supplemental References

   (Also see SAE Video Lectures Series, same topic and author)


NVH Workshop Topic Outline

• Introduction
• Ride Balance in the Ride Range
• NVH Load Conditions
• Low Frequency Basics
Competing Vehicle Design Disciplines

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

Cost, Weight, Investment, Manufacturing
Automotive NVH Frequency Range

Response

Log Frequency

Global Stiffness  "Low"

Local Stiffness + Damping  "Mid"

Absorption + Mass + Sealing  "High"

Structure Borne Noise

Airborne Noise

~ 150 Hz  ~ 1000 Hz  ~ 10,000 Hz
NVH Workshop Topic Outline

- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics
Study of Ride Balance

- Demonstrate the First Order Vehicle Modes
- Demonstrate Transient Response in Time Domain
- Derive Transition into the Frequency Domain
Ride Balance Study

Vehicle Traversing a Bump

Impact at Front Suspension
Followed by Impact at Rear Suspension

Response at Front
Response at Rear

Rear Suspension is in-phase with Front
After one cycle of ride motion, thus minimizing pitch motion.

See Ref. 5, Gillespie
NVH Model of Unibody Passenger Car
Symbolic Outline

Total mass: 2178.2 Kg (4800LBS)
- Sprung mass: 1996.7 Kg
- Unsprung mass: 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
Excitation Bump Profile

Profile Height (mm)

Distance (mm)

Profile

On to 100,380
Pitch at Mid-Car DOF3

Rotation - Radians

Time (sec.)
Pitch Response - Baseline Model

- **Y-axis**: Rotation Radians
- **X-axis**: Frequency (Hz)
- **Graph Title**: Pitch Response - Baseline Model
- **Legend**: Base Model

The graph shows the response of the pitch axis over a range of frequencies from 0.0 to 20.0 Hz. The y-axis represents the rotation in radians, with values ranging from 1.0E-08 to 1.0E-04. The curve indicates a peak at a certain frequency and decreases as the frequency increases.
\[ X(f) = \text{Fourier Transform of} \ X(t) \]
\[ \{X(f)\} = [H(f)] \ast \{F(f)\} \]

Caveat: \( X(t) \) must be Periodic Linear System

Spring Analogy \( (C = 1/K) \)

\[ X = C \ast F \]

\[ F(t) \text{ is Periodic} \]

5 sec
Transform Input Force to $F(f)$

**FFT of the Input Bump**

Amplitude (mm)

- $1.0 \times 10^{-6}$
- $1.0 \times 10^{-5}$
- $1.0 \times 10^{-4}$
- $1.0 \times 10^{-3}$
- $1.0 \times 10^{-2}$
- $1.0 \times 10^{-1}$
- $1.0 \times 10^{0}$
- $4.0 \times 10^{-3}$
- $8.0 \times 10^{-3}$
- $2.0 \times 10^{-2}$
- $2.0 \times 10^{-1}$

Cycles / mm

- $4.0 \times 10^{-3}$
- $8.0 \times 10^{-3}$
- $1.0 \times 10^{-2}$
- $2.0 \times 10^{-2}$
- $2.0 \times 10^{-1}$

20 Hz @ 45 MPH
Transform Input Force to $F(f)$

**FFT of the Input Bump**

Amplitude is Approximately Constant over the Frequency Range

Constant Displacement
Pitch at Mid-Car DOF3

<table>
<thead>
<tr>
<th>RotationRadian</th>
<th>Frequency Hz</th>
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<tbody>
<tr>
<td>1.0E-05</td>
<td>0</td>
</tr>
<tr>
<td>1.0E-06</td>
<td>5</td>
</tr>
<tr>
<td>1.0E-07</td>
<td>10</td>
</tr>
<tr>
<td>1.0E-08</td>
<td>15</td>
</tr>
<tr>
<td>1.0E-09</td>
<td>20</td>
</tr>
</tbody>
</table>

Time Domain FFT
FFT of Input
Ride Balance Study Summary

• Demonstrated the Fundamental Bounce and Pitch Modes in the Ride Range

• Demonstrated Transient Response in Time Domain then Obtained the FRF

• Derived Direct Computation of FRF in the Frequency Domain
NVH Workshop Topic Outline

• Introduction
• Ride Balance in the Ride Range
• NVH Load Conditions
• Low Frequency Basics
Two Main Sources

- Suspension
- Powertrain
Typical NVH Pathways to the Passenger
<table>
<thead>
<tr>
<th>NVH Load Conditions</th>
<th>Inherent</th>
<th>Process Variation</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Suspension Induced</strong></td>
<td>Tar Strip Impacts</td>
<td>Tire/Wheel Unbalance</td>
</tr>
<tr>
<td></td>
<td>Rough Road Surface</td>
<td>Tire Force Variation</td>
</tr>
<tr>
<td><strong>Driveline Induced</strong></td>
<td>Engine Fuel Combustion and Reciprocating Unbalance</td>
<td>Driveshaft and Halfshaft Unbalances</td>
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<tr>
<td></td>
<td>. Idle-in-Gear</td>
<td>Torque Converter Unbalance</td>
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<td></td>
<td>. Constant Speed Cruise</td>
<td></td>
</tr>
<tr>
<td></td>
<td>. WOT Acceleration</td>
<td>Axle Gear Mesh Variation</td>
</tr>
</tbody>
</table>
Suspension Excitation Load Conditions

Wheel Unbalance

Road Roughness
Medium Rough Road Spectrums from Wong, Ref. 7

CONCLUSION: With respect to the Ride Balance example at constant disp., a typical road spectrum disp. decreases and thus exhibits even more response attenuation beyond the ride modes.

*Spectrums were scaled to plot with Velocity
NVH Workshop Topic Outline

• Introduction

• Ride Balance in the Ride Range

• NVH Load Conditions

• Low Frequency Basics
Low Frequency NVH Basics

- Subjective to Objective Relationships
- Single Degree of Freedom Vibration
- Vibration and Noise Attenuation Strategies
Low Frequency NVH Fundamentals
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 )
Single Degree of Freedom Vibration

APPLIED FORCE

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

Transmitted Force

\[ TR = \frac{F_T}{F} = \sqrt{\frac{1 + (2 d_{f_{\text{fn}}})^2}{(1 - f_{\text{fn}}^2)^2 + (2 d_{f_{\text{fn}}})^2}} \]

- \( d = \) fraction of critical damping
- \( f_{\text{fn}} = \) natural frequency \( \sqrt{k/m} \)
- \( f = \) operating frequency
Vibration Isolation Principle

\[ F = F_0 \sin(2\pi ft) \]

Transmissibility Ratio

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

Isolation Region

\[ k \]
\[ c \]
\[ m \]

Applied Force

Transmitted Force
Isolation from an Applied Force

Excitation Force Coming from Engine $F_0$

Transmissibility Force Ratio is $F_T/F_0$

Example:
A 4 Cyl. Excitation for Firing Pulse at 700 RPM has a second order gas pressure torque at 23.3 Hz. Thus, to obtain isolation, the engine roll mode must be below 16.6 Hz.

Support Forces Transmitted to Body
Body on Suspension Single DOF Model

Isolation from Base Excitation

Transmissibility Ratio is $\frac{X_{out}}{X_{in}}$

Example: Vertical Ride Mode at 1.3 Hz provides isolation starting at 1.8 Hz. This provides isolation for the first order Hop and Tramp modes.
Simplified Models from 1 to 8 DOF’s
Enforced Base Motion
8 Degree of Freedom Vehicle NVH Model

- Engine Mass
- Engine Isolator
- Flexible Beam for Body
- Suspension Springs
- Wheels
- Tires
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

• Reduce the Input Forces from the Source
• Provide Isolation
• Mode Management
• Nodal Point Mounting
• Dynamic Absorbers
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

- Reduce the Input Forces from the Source
- Provide Isolation
- Mode Management
- Nodal Point Mounting
- Dynamic Absorbers
Reduction of Input Forces from the Source

**Road Load Excitation**
- Use Bigger / Softer Tires
- Reduce Tire Force Variation
- Drive on Smoother Roads

**Powertrain Excitation**
- Reduce Driveshaft Unbalance Tolerance
- Use a Smaller Output Engine
- Move Idle Speed to Avoid Excitation Alignment
- Modify Reciprocating Imbalance to alter Amplitude or Plane of Action of the Force.
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

• Reduce the Input Forces from the Source
• Provide Improved Isolation
• Mode Management
• Nodal Point Mounting
• Dynamic Absorbers
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly

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

Response at Mid Car

Constant Force Load; $F \sim A$

- 15.9 Hz
- 8.5 Hz
- 7.0 Hz

700 Min. RPM First Order Unbalance
Operation Range of Interest

Velocity (mm/sec)
Frequency Hz
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) vs Frequency (Hz)

- Base Model
- Without Double_ISO

1.414*f_n
Vibration and Noise Attenuation Methods

**Main Attenuation Strategies**

- Reduce the Input Forces from the Source
- Provide 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
- Suspension Hop and Tramp Modes
- Powertrain Modes
- Suspension Longitudinal Modes
- Exhaust Modes

**BODY/ACOUSTIC MODES**

- Body First Torsion
- Body First Bending
- Steering Column First Vertical Bending
- First Acoustic Mode

(See Ref. 1)
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

Velocity (mm/sec)

Frequency (Hz)

5 10 15 20

Vibration and Noise Attenuation Methods

Main Attenuation Strategies

• Reduce the Input Forces from the Source
• Provide 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 mode shape to prevent excitation coming from suspension input motion.
Mount at Nodal Point
First Torsion: Nodal Point Mounting Examples

Passenger sits at node point for First Torsion.

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

- Node Shifted
- Base Model

Velocity (mm/sec)

Frequency (Hz)

Node Shifted Base Model
< is more important than >

Frequency = $1 / T$
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

• Reduce the Input Forces from the Source
• Provide Isolation
• Mode Management
• Nodal Point Mounting
• Dynamic Absorbers
Dynamic Absorber Concept

Auxiliary Spring-Mass-Damper

\[ m = \frac{M}{10} \]
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 = $\sqrt{\frac{k}{m}}$
Baseline Sound Level
63 Hz Dynamic Absorber
63 + 110 Hz Absorbers

[Figure Courtesy of DaimlerChrysler Corporation]
Vibration and Noise Attenuation Methods

**Main Attenuation Strategies**

- Reduce the Input Forces from the Source
- Provide Isolation
- Mode Management
- Nodal Point Mounting
- Dynamic Absorbers
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Sachin Gogate
This looks familiar!
Frequency Range of Interest has changed to
150 Hz to 500 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

- High modal density and coupling in source, path and receiver
- Local Stiffness + Damping
- Global Stiffness
- Absorption + Mass + Sealing
- "Low" ~ 150 Hz
- "Mid" ~ 1000 Hz
- "High" ~ 10,000 Hz

- Mode separation is less practical in mid-frequency
- Effective isolation of energy between source and receiver at key noise paths is the basis of mid-frequency analysis
Mid-Frequency Analysis Character

- Important characteristics of mid frequency analysis

Effective Isolation &

Identifying Key Noise Paths
Mid-Frequency Analysis Character

- Important characteristics of mid frequency analysis

**Effective Isolation**

&

Identifying Key Noise Paths
Isolation Effectiveness

Effectiveness deviates from the classical development as resonances occur in the receiver structure and in the foundation of the source.
• **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, also referred as **Admittance**, characterizes Dynamic Stiffness of the structure at load application point

\[
\text{Mobility} = \frac{\text{Frequency} \times \text{Displacement}}{\text{Force}} = \frac{\text{Frequency}}{\text{Dynamic Stiffness}}
\]
The isolation effectiveness can be quantified by a theoretical model based on a theoretical model based on analysis of mobilities of receiver, isolator and source.

Transmissibility ratio is used to objectively define measure of isolation:

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

Isolation

\[ F_r = Y_i + Y_r + Y_s \]

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

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

2003, 2005 ERRATA

Source

Receiver

Isolator

Source
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.

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

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

\[
F_s = \frac{V}{Y_r + Y_s}
\]
For Effective Isolation (Low TR) the Isolator Mobility must exceed the sum of the Source and Receiver Mobilities.
Mid-Frequency Analysis Character

- Important characteristics of mid frequency analysis

Effective Isolation &

Identifying Key Noise Paths
Identifying Key Noise Paths

- Key noise paths identified by Transfer Path Analysis (TPA)

TPA is a technique to perform phased summation of partial responses through all noise paths to give total tactile or acoustic response under operating loads at a given frequency.

TPA is applicable in both testing and simulation scenarios to identify key noise paths.
Transfer Path Analysis

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

- Total operating Response (Tactile/Acoustic) is summation of partial responses through all noise paths

$$R_t = \sum_{\text{paths}} [R_i] = \sum_{\text{paths}} [F_i \times (R/F)_i]$$

$R_i$ : Partial contribution of path i due to operating force

$(R/F)_i$ : Tactile or Acoustic Transfer Function
Transfer Path Analysis

- TPA allows path rankings based on contribution to total response of noise paths at a given frequency
- TPA thus helps identify key noise paths
- TPA is mainly used for acoustic response in mid frequency range

• TPA allows path rankings based on contribution to total response of noise paths at a given frequency
• TPA thus helps identify key noise paths
• TPA is mainly used for acoustic response in mid frequency range
Designing for Mid Frequency

- Important characteristics of mid frequency analysis

Effective Isolation

&

Identifying Key Noise Paths
Designing for Mid Frequency

While designing a new vehicle, generic targets are set for key parameters along all noise paths in order to achieve effective isolation.

What are these generic targets and key parameters?
Generic Noise Path Targets

Transmissibility along a given noise path \((TR_i)\)

\[
TR = \left| \frac{Y_r + Y_s}{Y_i + Y_r + Y_s} \right|
\]

\[
TR = \left| \frac{1/K_{body} + 1/K_{source}}{1/K_{body} + 1/K_{iso} + 1/K_{source}} \right|
\]
**Generic Noise Path Targets**

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

<table>
<thead>
<tr>
<th>( \frac{K_{body}}{K_{iso}} )</th>
<th>1.0</th>
<th>5.0</th>
<th>Infinite</th>
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<tr>
<td>5.0</td>
<td>0.54</td>
<td>0.28</td>
<td>0.17</td>
</tr>
<tr>
<td>Infinite</td>
<td>0.50</td>
<td>0.17</td>
<td>0.00</td>
</tr>
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</table>

As a generic target, body to bushing stiffness ratio of at least 5.0 and very high source to bushing stiffness ratio (~ infinite) is desired to achieve “good” TR of 0.17
Relationship of Body-to-Bushing Stiffness ratio to Transmissibility

For infinitely stiff (immobile) source

$$TR = \left| \frac{K_{bsg}}{K_{bsg} + K_{body}} \right|$$

Target Min. = 5
Generic Noise Path Targets

Operating loads create forces \( F_i \) into body at all noise paths.

Response \( R_t \) could be either Tactile or Acoustic Response.

\[
R_t = \sum_{\text{paths}} [R_i] = \sum_{\text{paths}} [F_i \times (R/F)_i] = \sum_{\text{paths}} [F_i \times (R/V)_i \times (V/F)_i]
\]
Generic Noise Path Targets

For example, for Acoustic Response $P_t$

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

- For a given force generated at suspension attachment to body, lowering sensitivities $(P/F)$ or $(V/F)$ along a path would reduce total response.

- As a generic target,
  - Acoustic sensitivity $(P/F)$ in 55-60 dBL/N range
  - Structural Mobility $(V/F)$ less than 0.312 mm/sec/N
Generic Noise Path Targets

How does one achieve these generic targets:

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

\[ \frac{K_{\text{source}}}{K_{\text{iso}}} \sim \text{infinite} \]

\[ 55 \text{ dBL/N} < \text{Acoustic Mobility} < 60 \text{ dBL/N} \]

\[ \text{Structural Mobility} < 0.312 \text{ mm/sec/N} \]
Generic Noise Path Targets

How does one achieve

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

- Increase local body attachment stiffness \( (K_{\text{body}}) \) through structural modifications

- Reduce attachment isolator stiffness \( (K_{\text{iso}}) \) while balancing the conflicting requirement of other functionalities such as Ride & Handling

Structural Mobility < 0.312
Generic Noise Path Targets

How does one achieve

\[
\frac{K_{\text{source}}}{K_{\text{iso}}} \sim \text{infinite}
\]

• Increase source side attachment stiffness \((K_{\text{source}})\)

• Reduce attachment isolator stiffness \((K_{\text{iso}})\)

*In automotive structures, it is realistic to expect that source to isolator stiffness ratio is almost infinite since source usually corresponds to stiff structure (such as powertrain or axle)*
Source Excitation Characteristics

Other means of reducing source input to paths

- Add lumped mass on component
  - Applicable when Rigid Body Resonance of Source Side Component are present

- Use tuned absorber
  - Applicable when flexible modes of source side component are present

- Bring nodal points toward path
How does one achieve

- At a given frequency, Acoustic Mobility \((P/F)\) is

\[
P/F = (P/V) \times \frac{\text{Frequency}}{\text{Body stiffness}}
\]

- Based on the above equation, increasing body stiffness usually reduces acoustic mobility

55 < Acoustic Mobility < 60
How does one achieve

- There are situations when increasing body stiffness does not reduce acoustic mobility

- In such cases, means of reducing acoustic mobility are to reduce overall body panel velocity through application of damping and acoustic treatment
Application of Damping Treatment

Effect on sound response of damping treatment applied on key identified contributing panels

![Graph showing sound pressure level (SPL) in decibels (dBA) across different frequencies (Hz) for a floor with and without damping treatment. The graph indicates a 12.5 dBA reduction in SPL for the floor with damping treatment compared to the floor without damping treatment.]

[Figure Courtesy of DaimlerChrysler Corporation]
Designing for Mid Frequency

While designing a new vehicle, generic targets are set for key parameters along *all noise paths* in order to achieve effective isolation.

*is it really necessary to achieve generic targets for all noise paths?*

*Probably Not!!*
Designing for Mid Frequency

Driver’s Ear Noise

Vehicle Level Response

Transfer Path Analysis

Some noise paths are more dominant than others

Impose more strict requirements for these dominant paths and relax requirements for other paths to achieve more “rebalanced” noise

Vehicle Response to Meet NVH Targets

Original Noise Path Contributions

Reduced Noise
Designing for Mid Frequency

Principles to follow

• At the beginning of program, work towards generic targets for key parameters for all noise paths in order to achieve effective isolation.

• As the design is firmed out, evaluate key noise paths using Transfer Path Analysis in order to meet target for all NVH operating load conditions.

• Perform path “rebalancing” to arrive at revised path targets (more strict for dominant paths) for all NVH operating load conditions.
Mid Frequency NVH Goal Achievement Process

1. Initial Sub-System Targets
   - Rebalance Trade-Off
   - Evaluate Sub-System Performance
     - Meet Sub-System Goals?
       - Yes or Time Out
       - Evaluate Vehicle Goals
         - Meet Vehicle Goals?
           - Yes or Time Out
           - No
             - Re-Design Sub-System
               - Yes or Time Out
             - No
               - Rebalance Trade-Off
Mid Frequency NVH Improvement (Sports Utility Vehicle Example)

Full Vehicle Model

[Figure Courtesy of DaimlerChrysler Corporation]

Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)
Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)

Acoustic Cavity Model

[Figure Courtesy of DaimlerChrysler Corporation]
Mid-Frequency NVH improvement

Axle Whine Example: 300-500 Hz

- Front and Rear Axle
- Gear-Pinion Mesh
- Transmission Error

Trimmed Body

Chassis

Interior Acoustic Cavity

SOURCE ------------ PATH ---------------- RECEIVER

[Figure Courtesy of DaimlerChrysler Corporation]
Axle Whine Example

- Design work was focused in the beginning towards achieving generic targets for all noise paths.
- As the design was firmed out, full vehicle analysis revealed under target performance for Driver’s ear SPL response which was dominated by rear excitation.

![Sound Response with Varying Excitation](Figure Courtesy of DaimlerChrysler Corporation)
Axle Whine Example

• Before embarking on identifying the root cause for under-target performance at dominant noise paths, it is a good practice to perform reasonableness check on the response

• Steps for Reasonableness Determination
  
  ➢ Judging the response based on system knowledge
    • Total response content is dominated by rear excitation. This is reasonable since vehicle has IFS and solid axle rear suspension which is harder to isolate for noise

  ➢ Forced mode Animation
    • Operating deformed shape motion is rear axle pitching about ring gear axis. This was expected since input excitation is MTE imposed as enforced angular rotation between ring and pinion gear

  ➢ Disconnect Studies
    • Disconnecting rear suspension noise paths (shock in particular) in pair had the most significant effect on Driver’s SPL response
Axle Whine Example

- **Transfer Path Analysis**

- **Dominant Paths**
  - Rear left shock vertical
  - Rear LCA vertical: Left is positive whereas right is negative contributor
  - Rear Right shock vertical

- The conclusion matches with reasonableness checks

[Figure Courtesy of DaimlerChrysler Corporation]
Axle Whine Example

- Is it high forces or high acoustic sensitivity at shock to body attachment?

\[ R_t = \sum_{\text{paths}} [R_i] = \sum_{\text{paths}} [F_i \cdot (R/F)_i] \]

- The issue is with high forces into the body through shock attachment due to stiff shock bushings.
- Stiff shock bushings gave low body-to-bushing stiffness ratio.

Acoustic sensitivity is better than generic target.

[Figure Courtesy of DaimlerChrysler Corporation]
Axle Whine Example

Solution

- Soften shock vertical bushings by 65%

- To balance this against handling requirement of stiff bushing, local attachment stiffness between shock and body was improved through a new bracket design

- This addition of bracket improved right shock mobility 3 times whereas left shock mobility by 1.5 times thereby improving isolation effectiveness of shock bushing

[Figure Courtesy of DaimlerChrysler Corporation]
Axle Whine Example

Response Improvement due to proposed solution

[Figure Courtesy of DaimlerChrysler Corporation]
Axle Whine Example

How Robust is the proposed solution?

- Parameter variations such as weld deletion in “new bracket” and gage changes were considered to study robustness of solution.

- Response scatter of model with proposal does not overlap baseline model response scatter indicating a robust solution.

- The problem peak has now shifted to a new vehicle speed of 50.7 mph which requires a new contribution analysis.

*Figure Courtesy of DaimlerChrysler Corporation*
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

• Other means of dealing high levels of source input (Tuned dampers, damping treatments, isolator placement at nodal locations) are also effective

• It is important to balance NVH requirement against other functionalities (Ride and Handling, Impact)

• It is important to understand the robustness of design recommendations
NVH Workshop Topic Outline

• Introduction
• Ride Balance in the Ride Range
• NVH Load Conditions
• Low Frequency Basics
• \textit{Live Noise Attenuation Demo}
• Mid Frequency Basics
• \textbf{Utilization of Simulation Models}
• Closing Remarks

\textbf{Alan Duncan}
NVH Workshop Workshop Topic Outline

• Introduction
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Considerations

• Some Agreement: Math Models can be used as Trend Predictors. (but not for absolute levels, yet.)

• Q. How do I know my model is good?
  • ANS. We require correlation work to know the simulation compares to test values to some degree.

• Q. How do I make design decisions before hardware is available?
  • ANS. Correlation must be performed on existing hardware to establish modeling methods to be applied to the future design. *(The Reference Baseline Ref. 3)*

A model of the new design is built with the same Methodology as the Reference Baseline to predict the change in performance as the design process progresses but before prototypes are available.
Considerations

• Q. How do I compare my model to test measurement and how close does it have to be to assure it can be used as a trend predictor?
  • ANS. If model predictions were within the band of variability of the test measurement, for a statistically significant number of samples, this would increase confidence in the predictive capability.

• Q. How wide is the band of variability?
  • ANS. Don’t Ask !!!
Discussion of Product Variability

Topics

• Kompella and Bernhard Observations
• Freyman NVH Scatter Results
• Model Confidence Criteria
• Conclusions
Magnitude of 99 Structure – borne FRF’s for the Rodeo’s for the driver microphone (Ref. 8) ©1993 Society of Automotive Engineers, Inc.
Acoustic scatter numerically determined in the vibro-acoustic behavior of a vehicle due to possible tolerances in the component area and in the production process.
Reference Baseline Confidence Criterion

For *Operating* Response Simulations

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

Simulation Prediction

**Test Upper Bound**
**Test Band Average**
**Test Lower Bound**

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

- Design work was focused in the beginning towards achieving generic targets for all noise paths.

- As the design was firmed out, full vehicle analysis revealed under target performance for Driver’s ear SPL response which was dominated by rear excitation.

![Sound Response with Varying Excitation Graph](Figure Courtesy of DaimlerChrysler Corporation)

- **FR + RR Excitation**
- **RR Excitation Only** (Dominates Total Content)
- **FR Excitation Only**

**Target Level**

Must define Targets for the Simulation to know when goal is reached!!!
Conclusions:

Significant Product Variation exists even in best-in-class vehicles.

Correlation should be considered as being within the band of variability whether test or simulation.

The Confidence Criteria, for operating responses, is a relatively challenging condition to meet when considering the following:

- It uses the same bandwidth as Kompella (Ref. 8), determined from simple FRF’s, while the criteria is for operating responses which are subject to additional variation in the operating loads.
- It assumes that one test will generate the mean response level in the band subject to the condition that a “qualified” median performer will be tested. This requires a test engineer extremely experienced with the vehicle line in order to “qualify” the vehicle.

Best hope for reduced product development times is a coordinated effort of Virtual Vehicle Simulation and Reference Baseline and Physical Prototype Testing to grasp the complexities of NVH responses and the robustness of their sensitivity to variation.
Competing Vehicle Design Disciplines

- Ride and Handling
- Impact CrashWorthiness
- Cost, Weight, Investment, Manufacturing
- NVH
- Durability
The Fundamental Secret of Structure Borne NVH Performance

Revealed here tonight!
Fundamental Secret to Making Money in the Stock Market

Buy Low and Sell High!
The Fundamental Secret of Structure Borne NVH Performance

To Minimize Structure Borne NVH response, always connect Sub-systems at locations where motion is at a Minimum.

Meets Conditions of the Attenuations Strategies

• Minimize the Source Load
• Manage Mode Placement
• Provide Isolation
• Mount at Nodal Points
• Provide Dynamic Absorber
• Reduce Source - Receiver Mobility
The Fundamental Secret of Structure Borne NVH Performance

To Minimize Structure Borne NVH response, always connect Sub-systems at locations where motion is at a Minimum.

That’s All Folks!

Thank You for Attending the SAE Structure Borne NVH Workshop

Your Presenters tonight were:

Alan Duncan, Automotive Analytics, Inc.
Greg Goetchius, Material Sciences Corp.
Sachin Gogate, Altair Engineering, Inc.

Visit: www.AutoAnalytics.com/papers.html to download the Structure Borne NVH Workshop (May 12)