SAE 2013 NVH Conference
Structure Borne NVH Workshop

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NVH Specialist
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NVH Specialist

Jianmin Guan — Altair Engineering
NVH Manager

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Structure Borne NVH Workshop

Workshop Objectives -

1. Review Basic Concepts of Automotive Structure Borne Noise.
2. Propose Generic Targets.
3. Present New Technology Example.

Intended Audience –

- New NVH Engineers.
- “Acoustics” Engineers seeking new perspective.
- “Seasoned Veterans” seeking to brush up skills.

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Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- New Technology

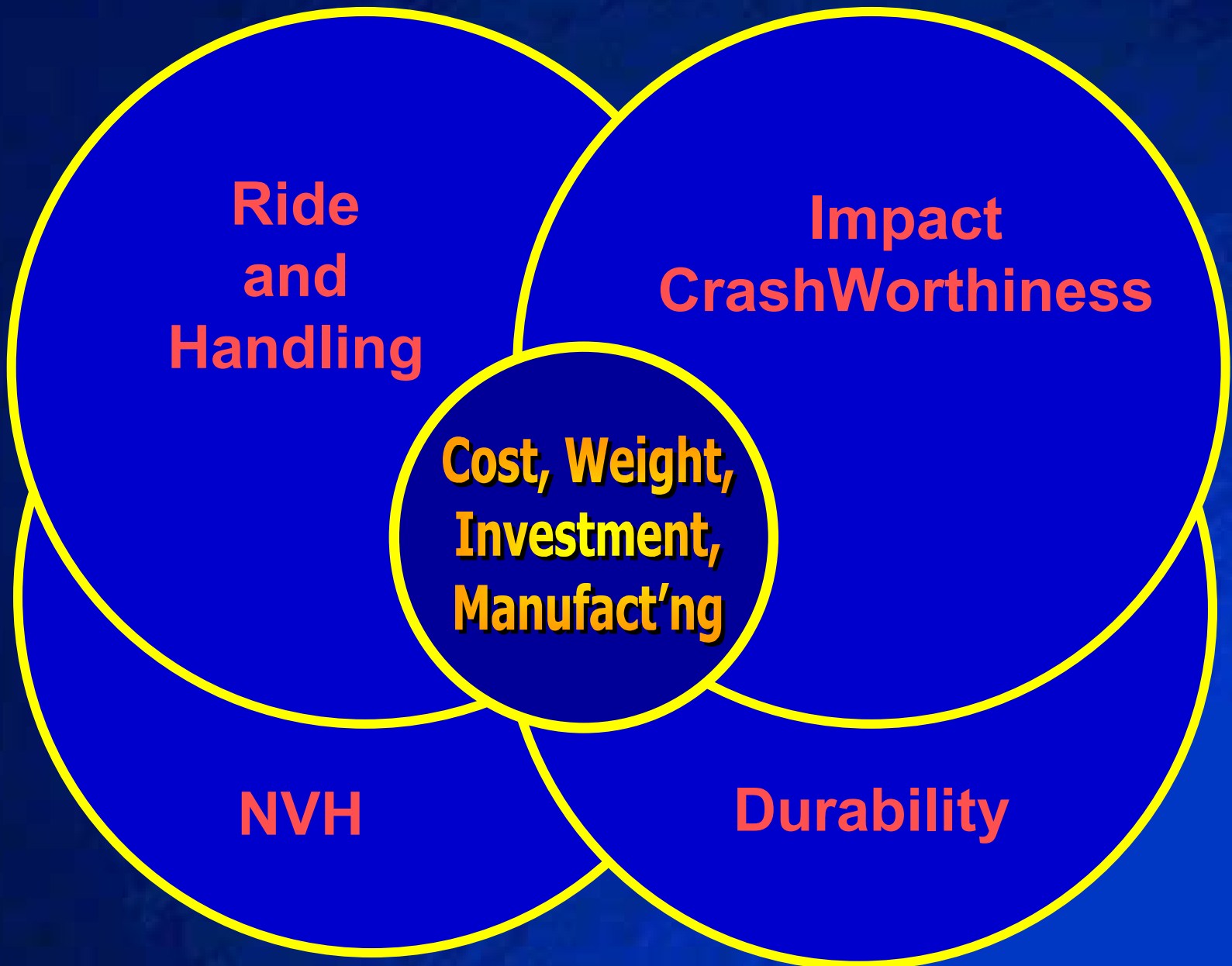
  Uncertainty and NVH Scatter
- Closing Remarks

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



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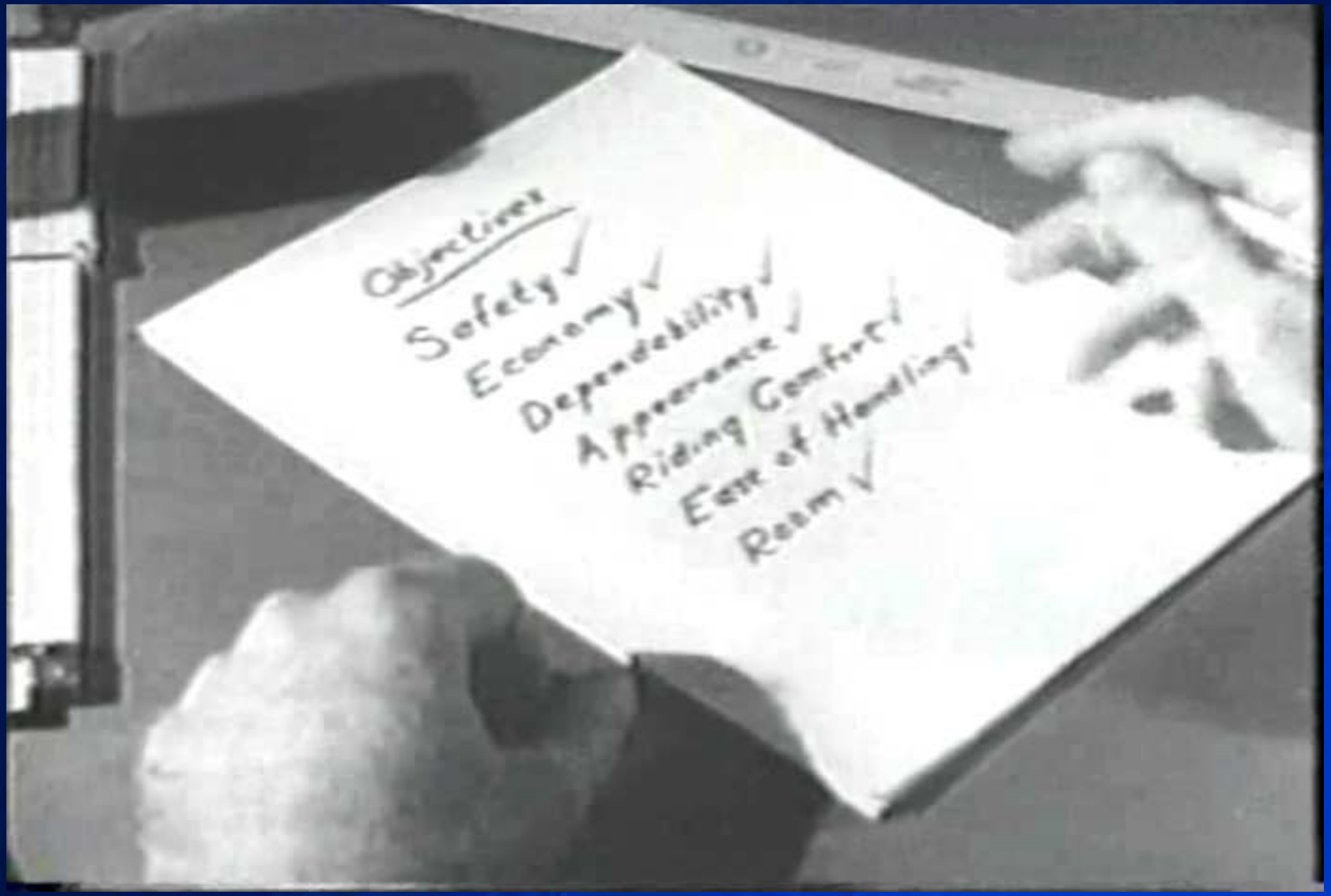
Automotive Engineering Objectives are Timeless



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Structure Borne NVH Workshop

- Introduction
- **Low Frequency Basics** Alan Duncan
- Mid Frequency Basics
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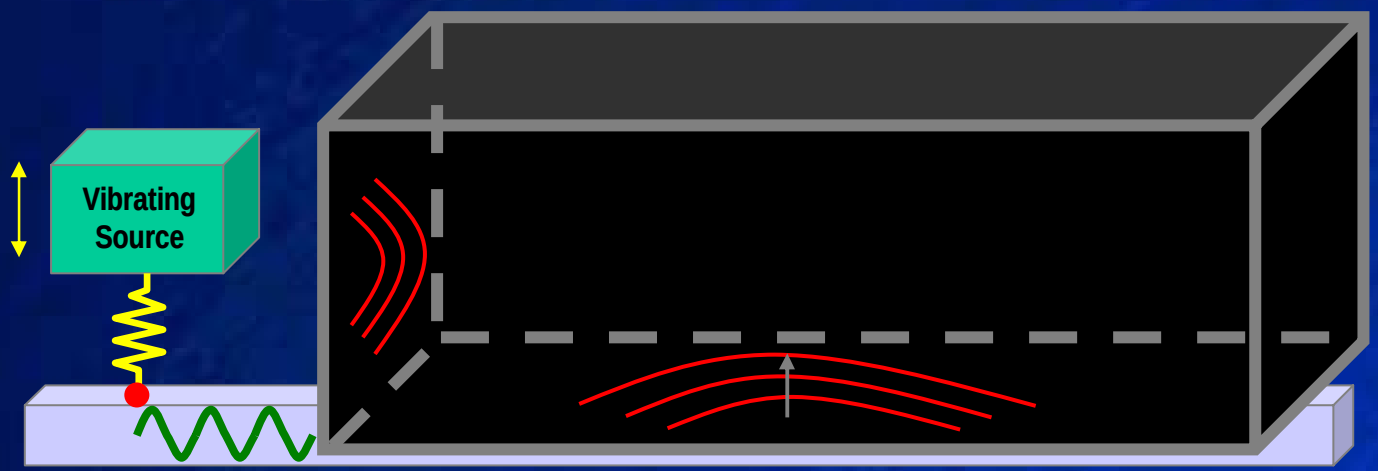
Structure Borne Noise and Vibration



Frequency Range: up to 1000 Hz

System Characterization

- Source of Excitation
- Transmission through Structural Paths
- "Felt" as Vibration
- "Heard" as Noise

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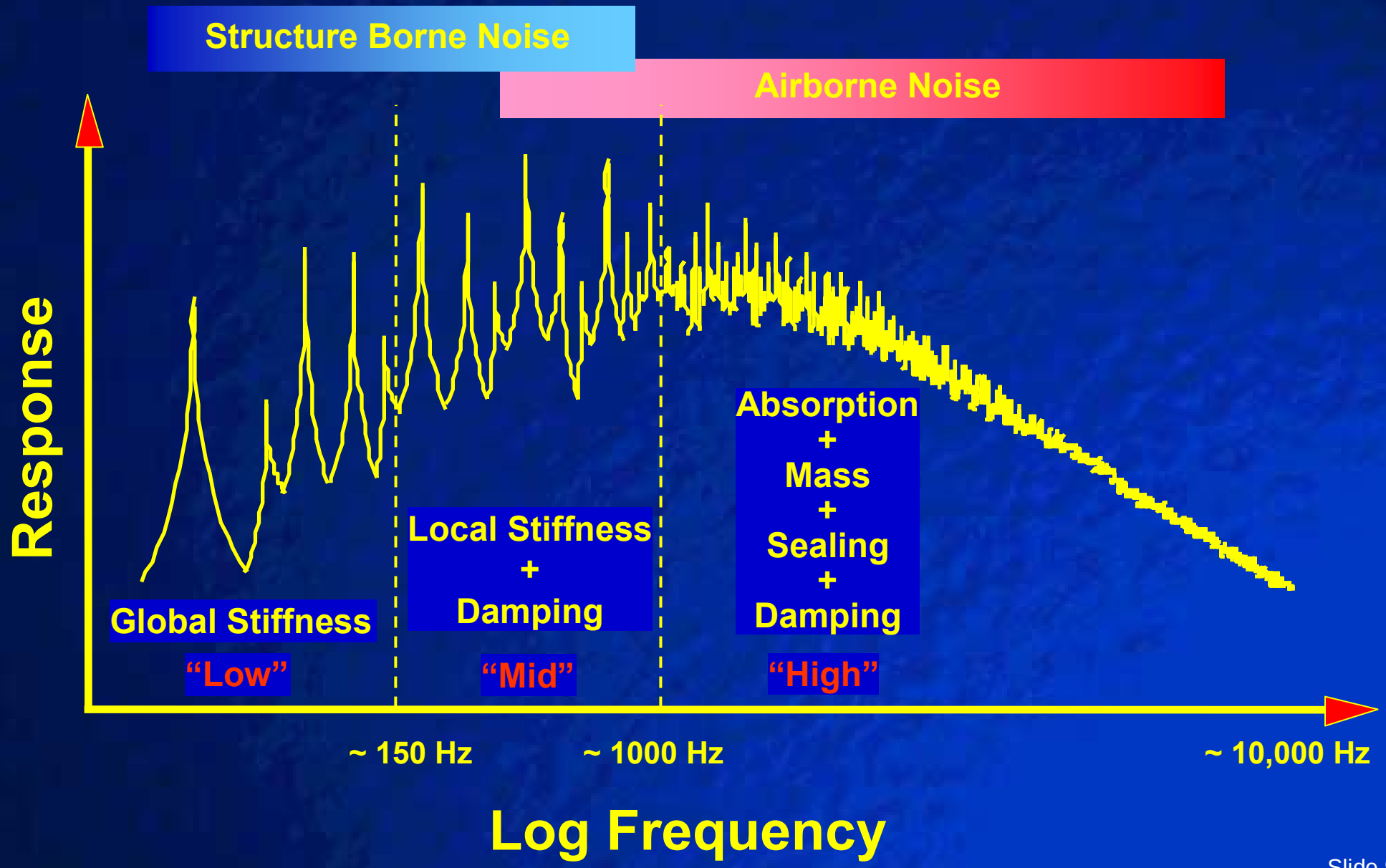
Automotive NVH Frequency Range



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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

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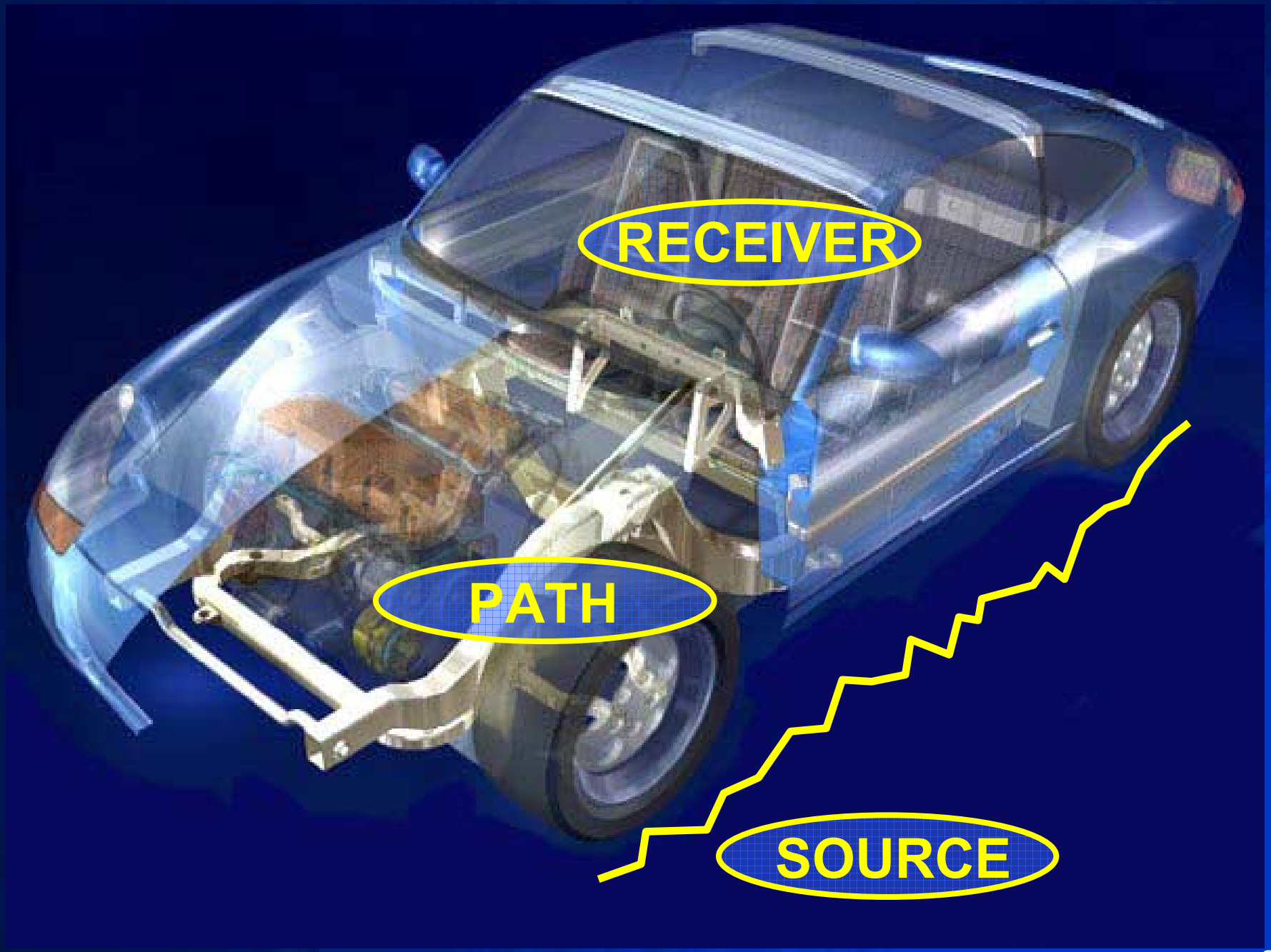
Structure Borne NVH Basics



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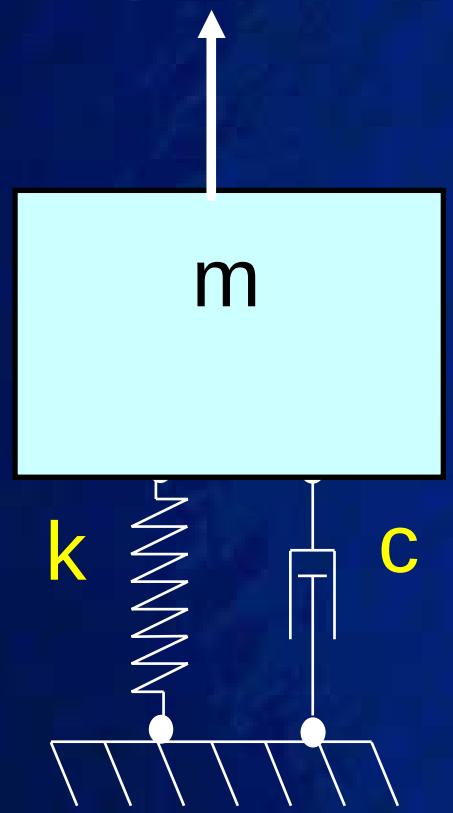
Single Degree of Freedom Vibration

APPLIED FORCE

$$F = F_o \sin 2\pi f t$$



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

Transmitted Force

F_T

ζ = fraction of critical damping

f_n = natural frequency $\sqrt{k/m}$

f = operating frequency

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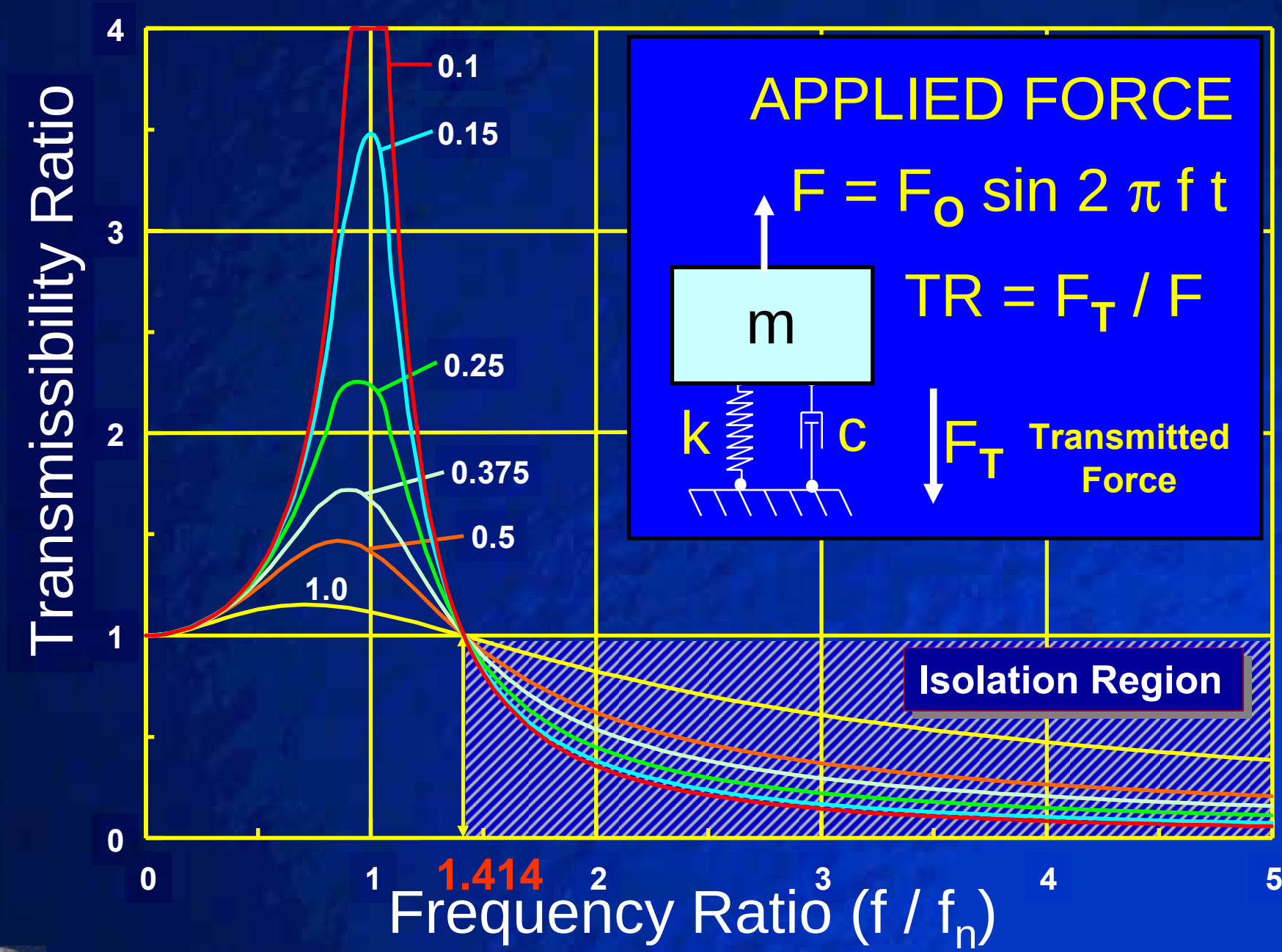
Vibration Isolation Principle



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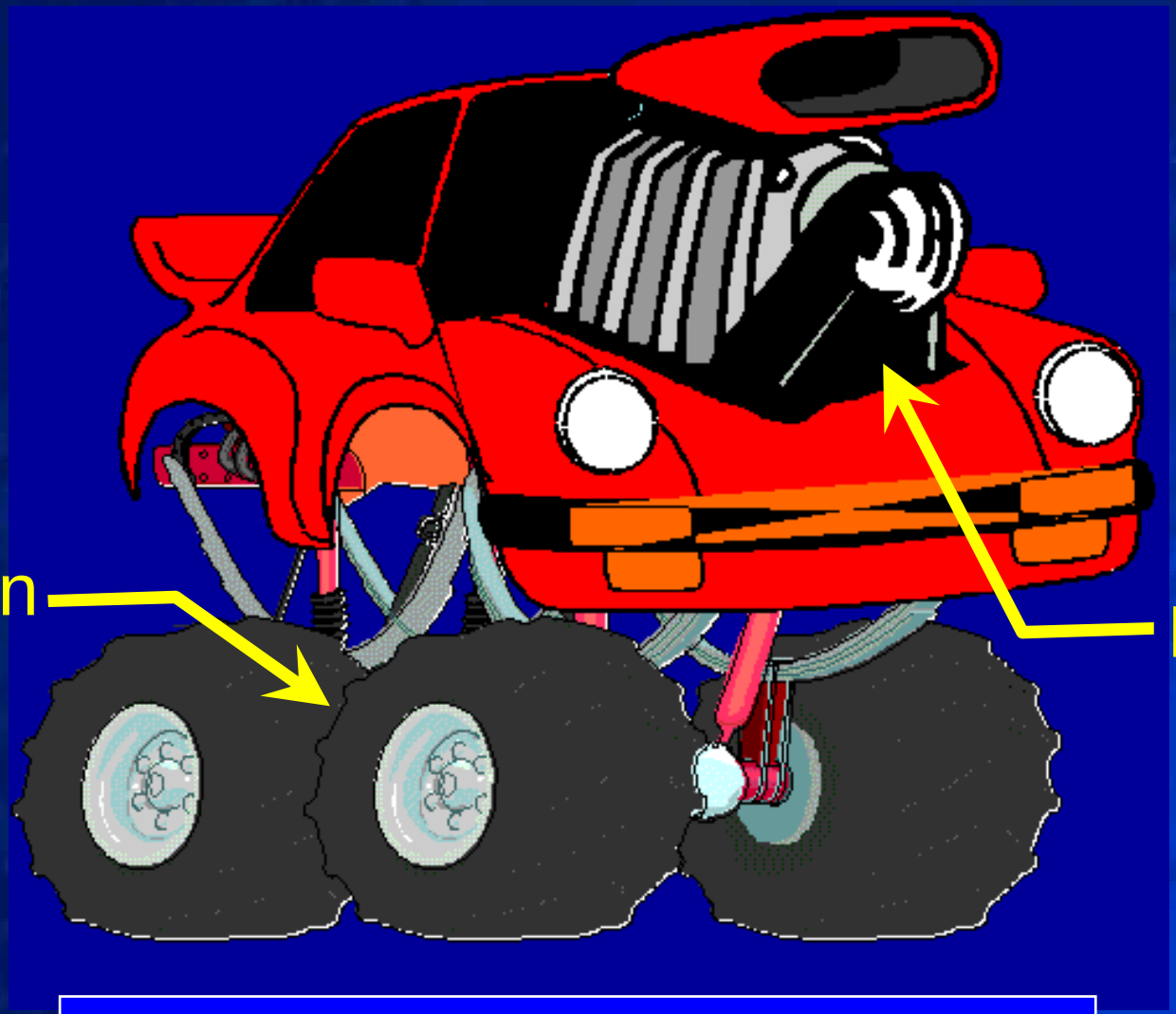
NVH Source Considerations



Two Main Sources

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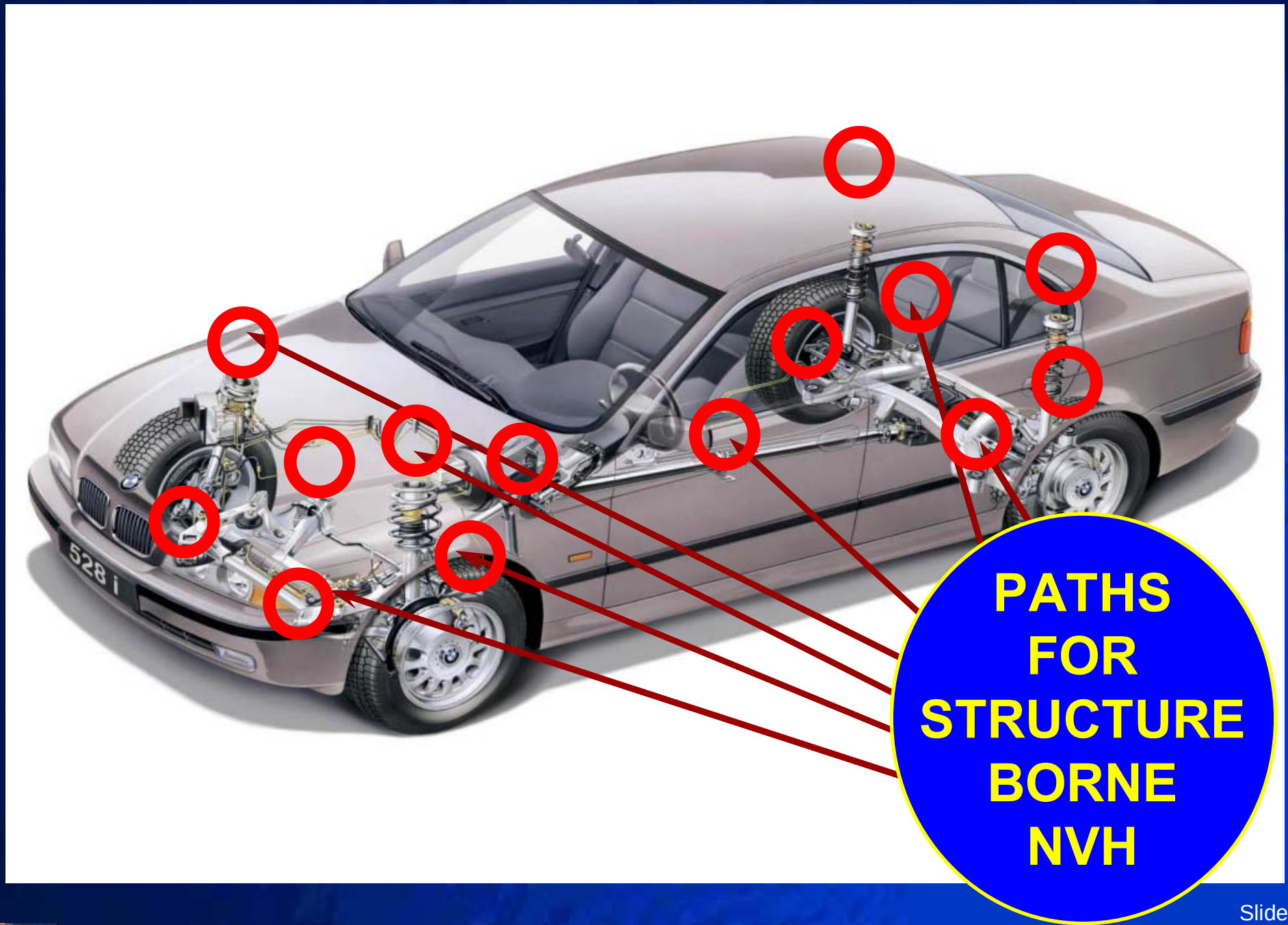
Typical NVH Pathways to the Passenger



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Structure Borne NVH Sources

	Inherent	*Process Variation*
Suspension Induced	Tar Strip Impacts	Tire/Wheel Unbalance
	Rough Road Surface	Tire Force Variation
Driveline Induced	Engine Fuel Combustion and Reciprocating Unbalance . Idle-in-Gear . Constant Speed Cruise . WOT Acceleration	Driveshaft and Halfshaft Unbalances
		Torque Converter Unbalance
		Axle Gear Mesh Variation

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Structure Borne NVH Sources

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.

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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

$$A_2 \approx {}^1/_2 \; A_1$$

Represents 1.0 Rating Change

TACTILE: 50% reduction in motion

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

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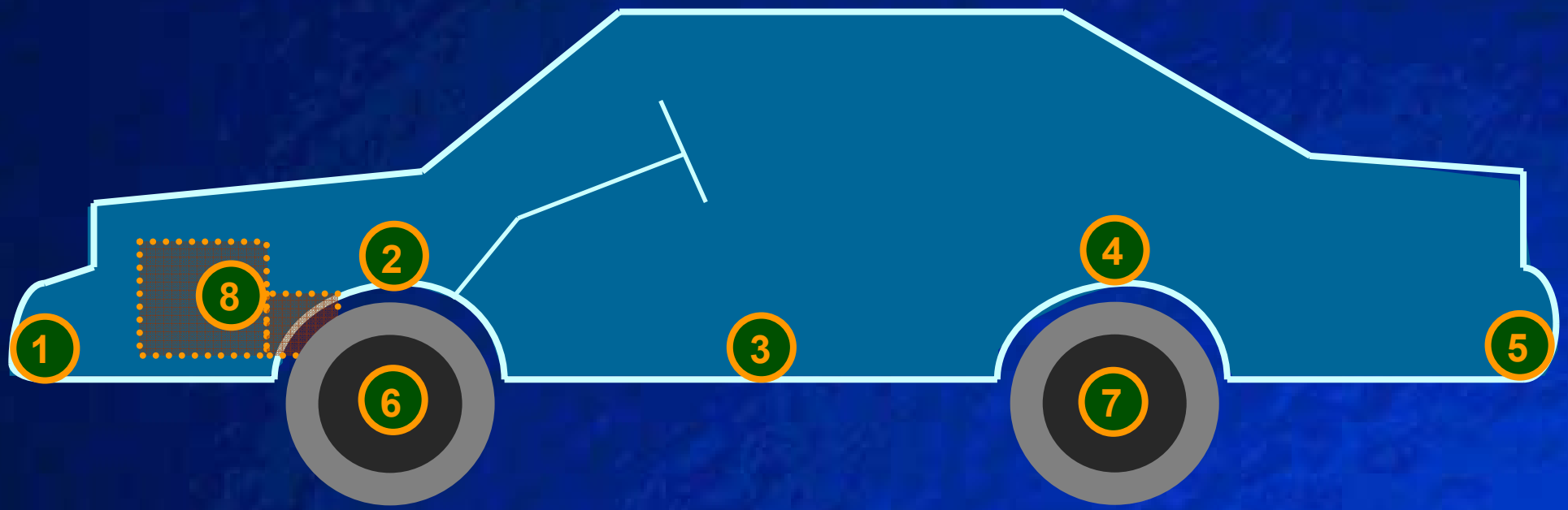
Symbolic Model of Unibody Passenger Car

8 Degrees of Freedom



Mass	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

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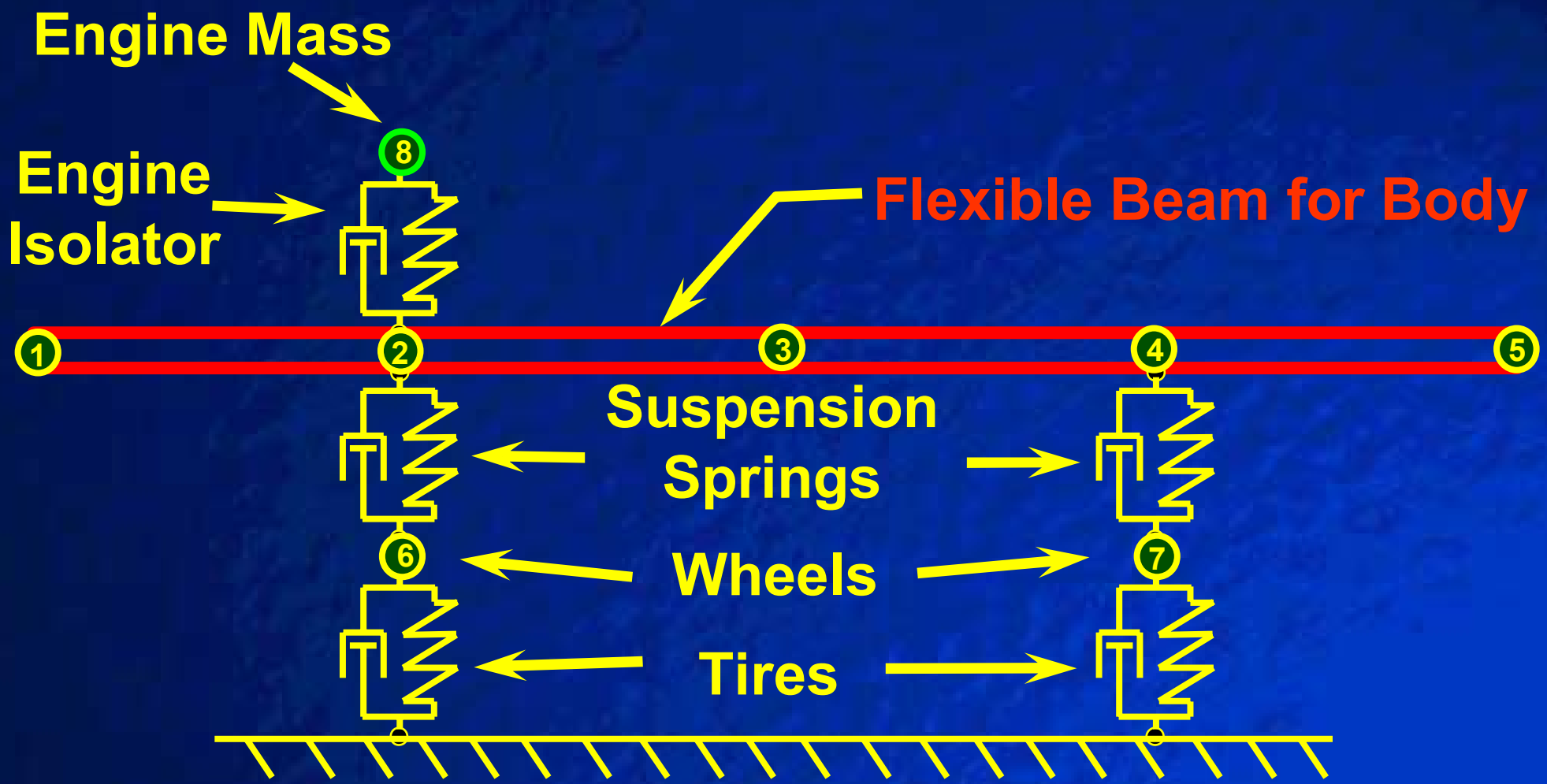
8 Degree of Freedom Vehicle NVH Model



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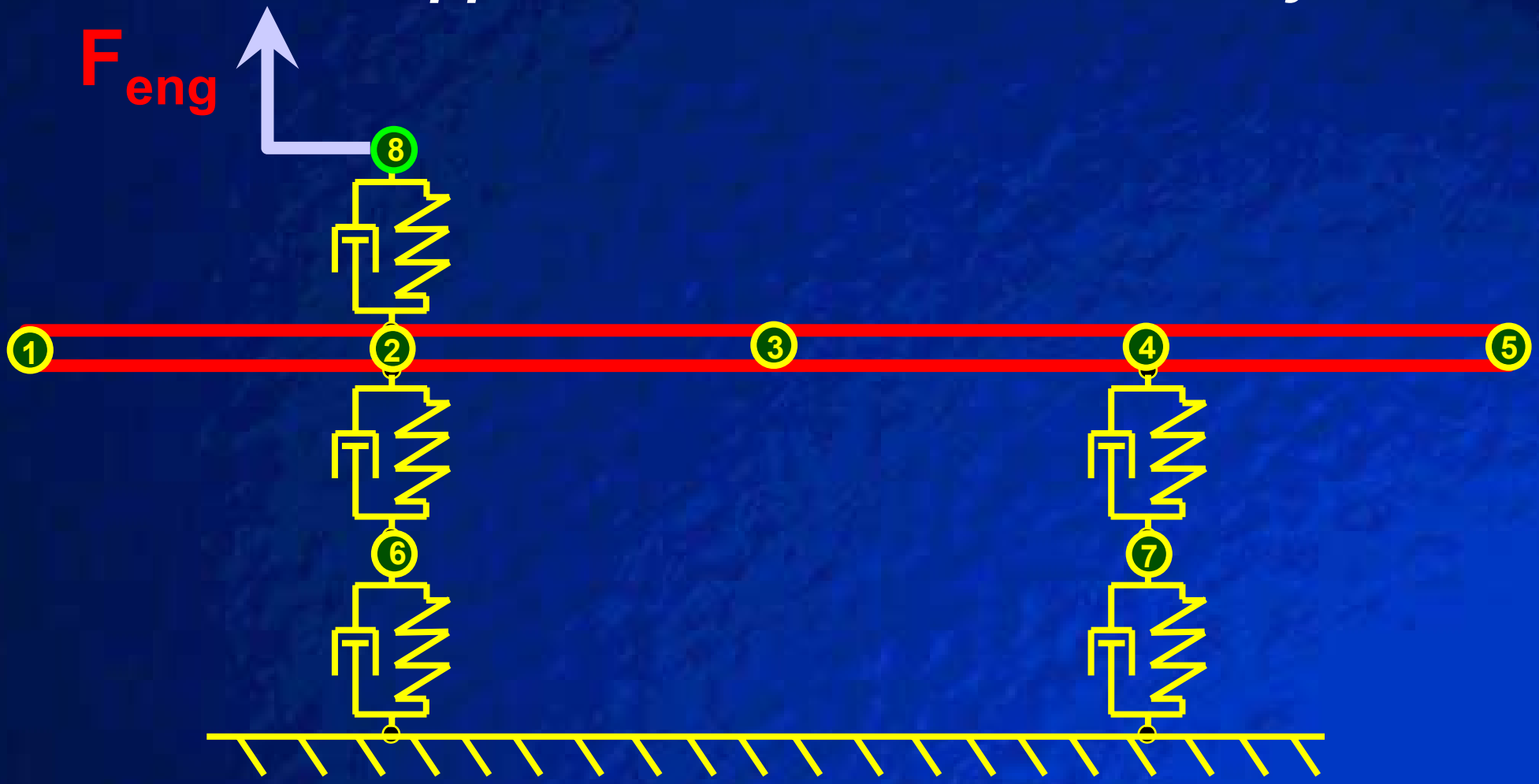
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



Forces at Powertrain could represent a First Order Rotating Imbalance

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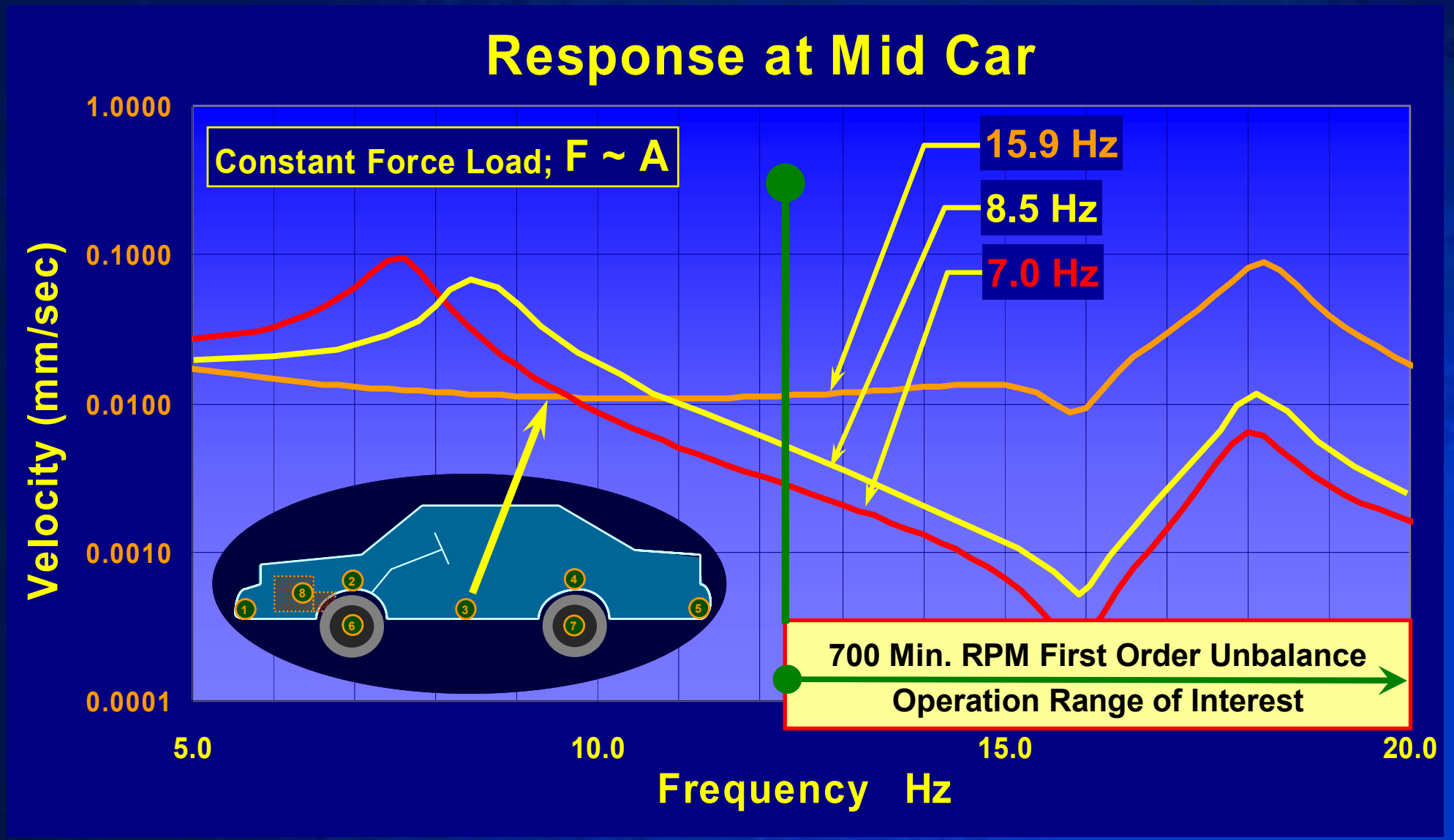
Engine Isolation Example



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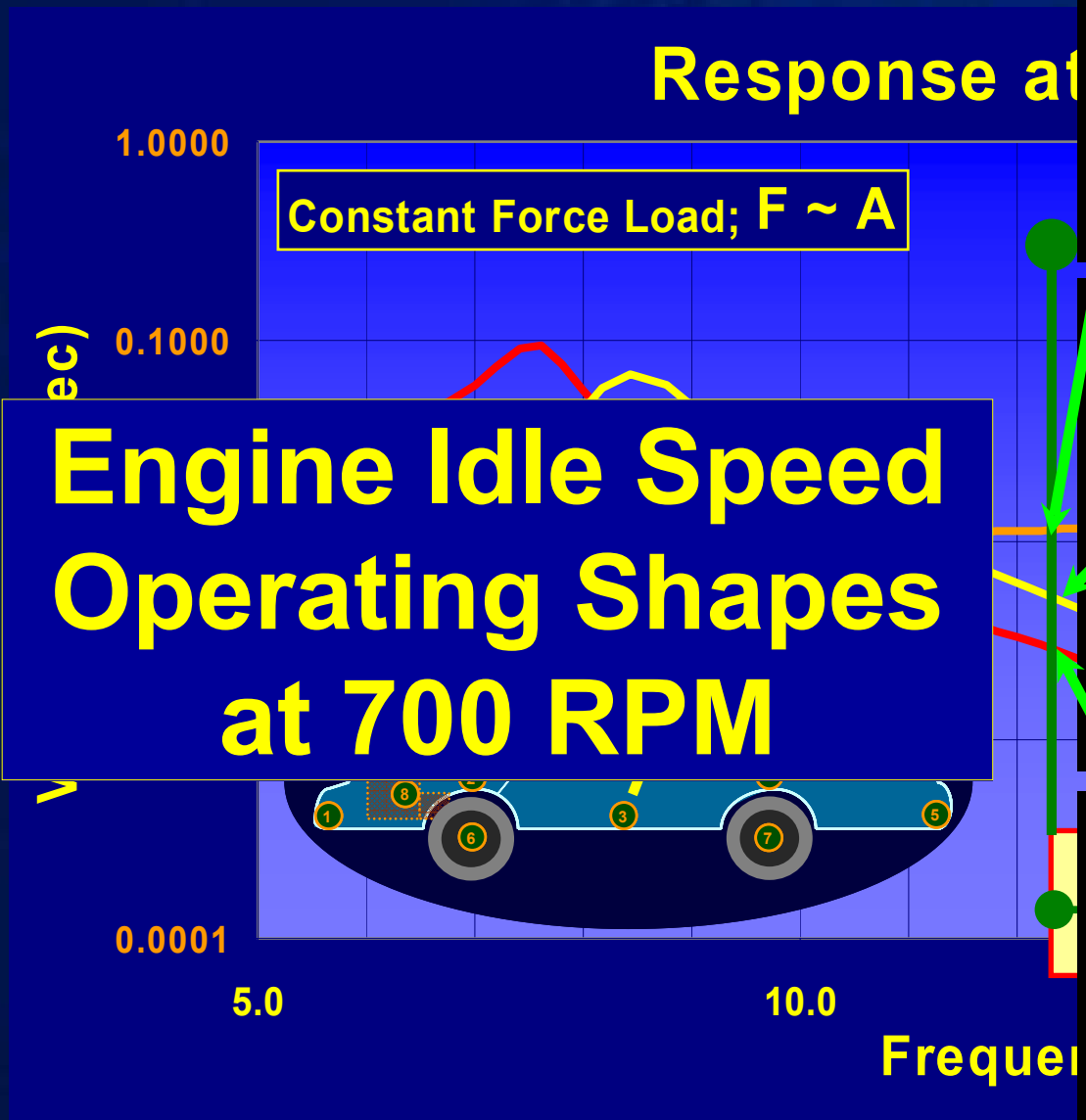
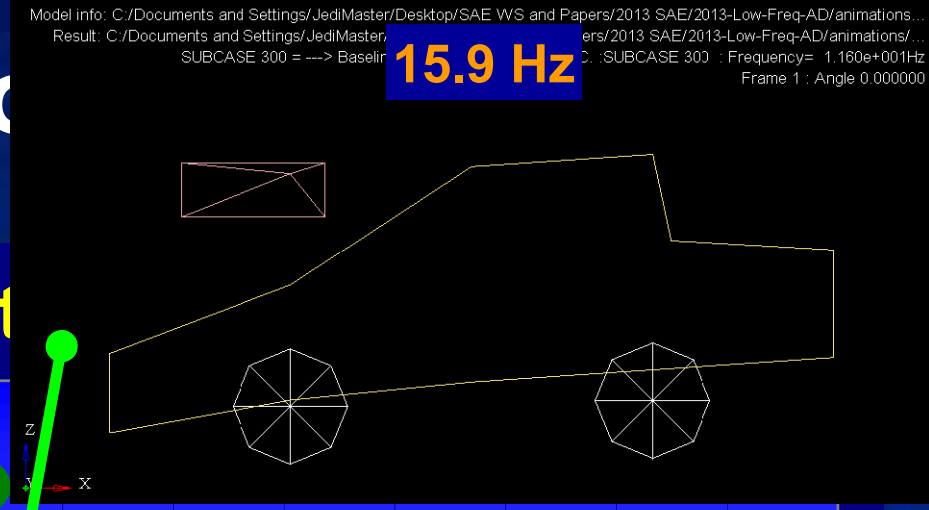
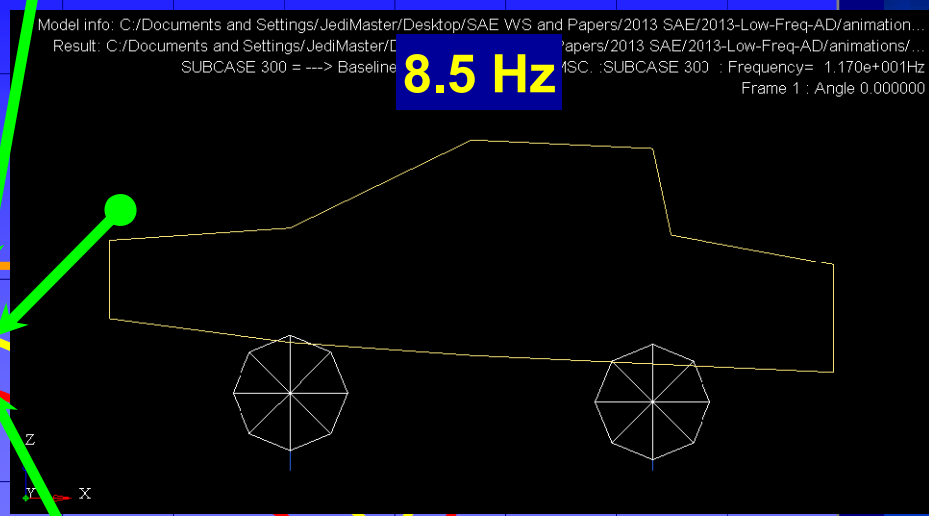
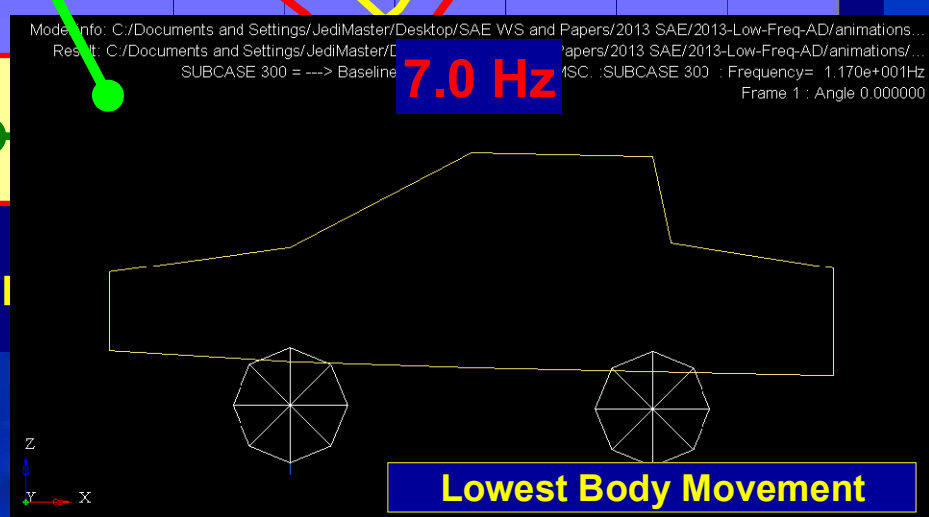
Engine Isolatio



Engine Idle Speed Operating Shapes at 700 RPM

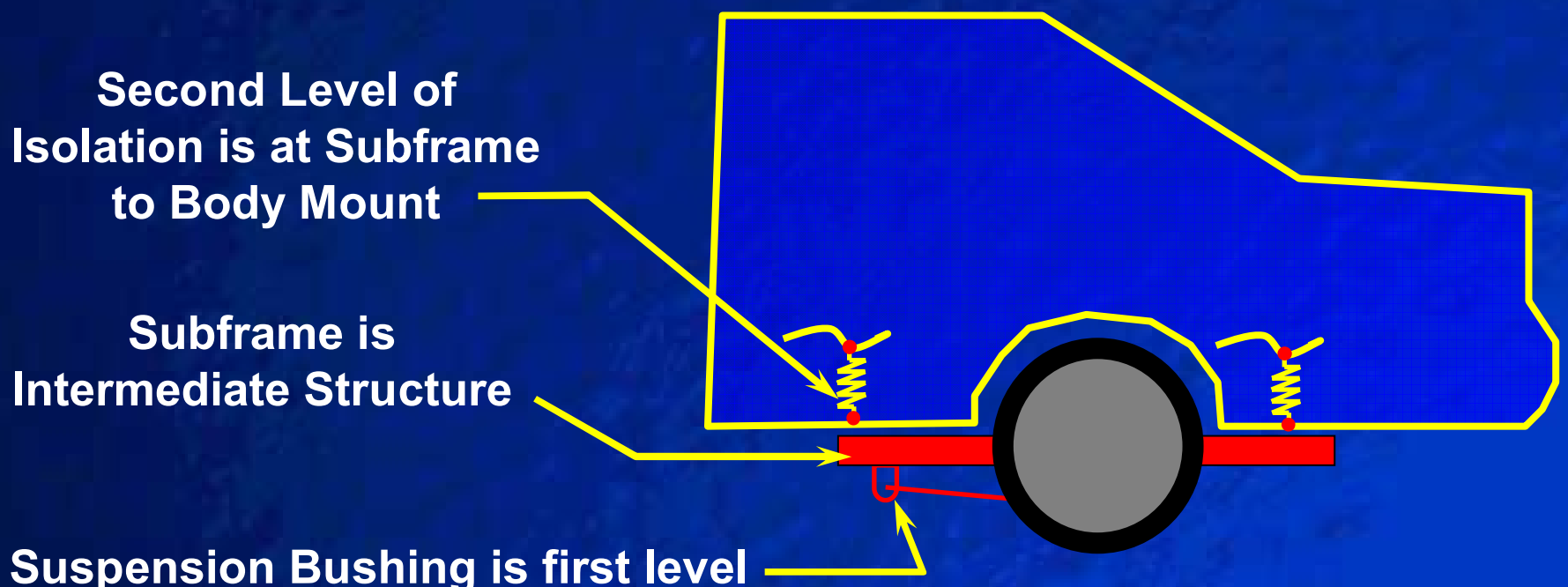






Concepts for Increased Isolation

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



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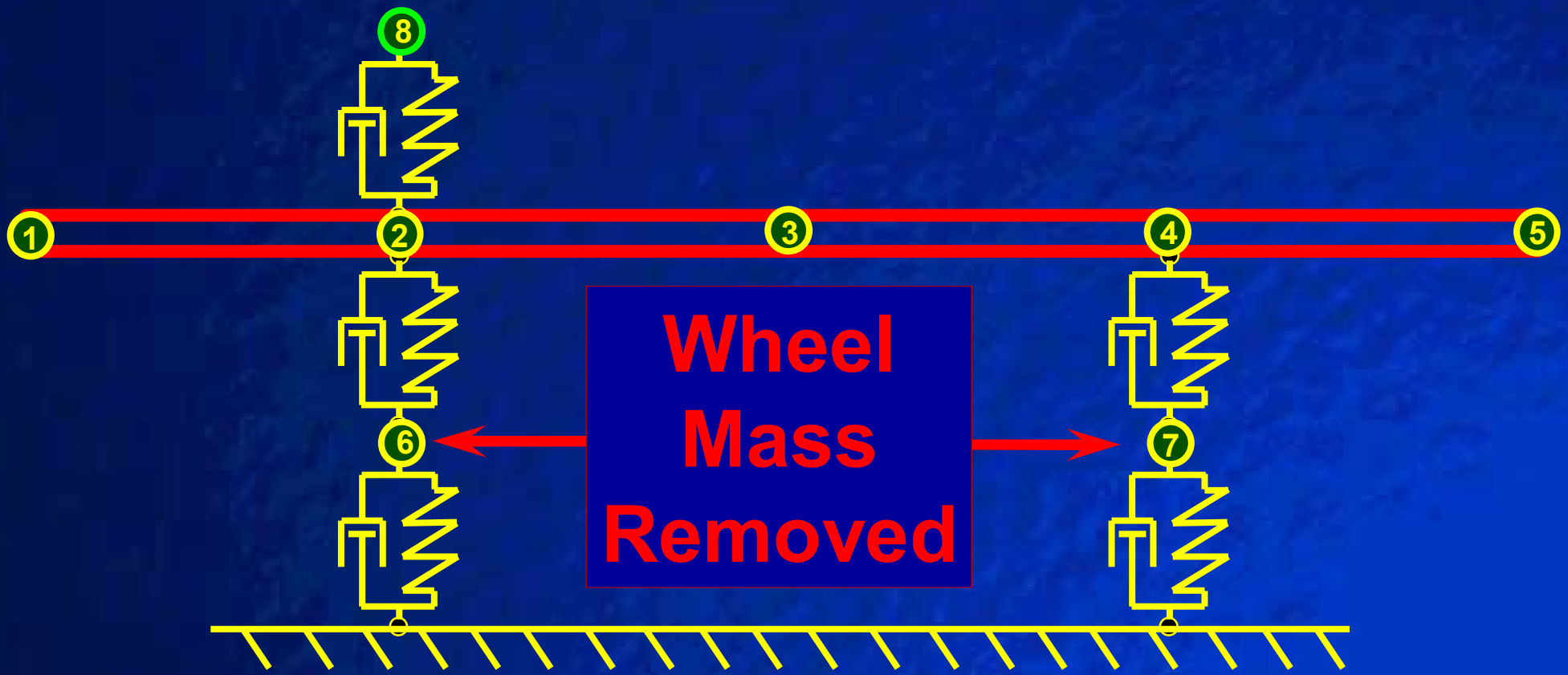
8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



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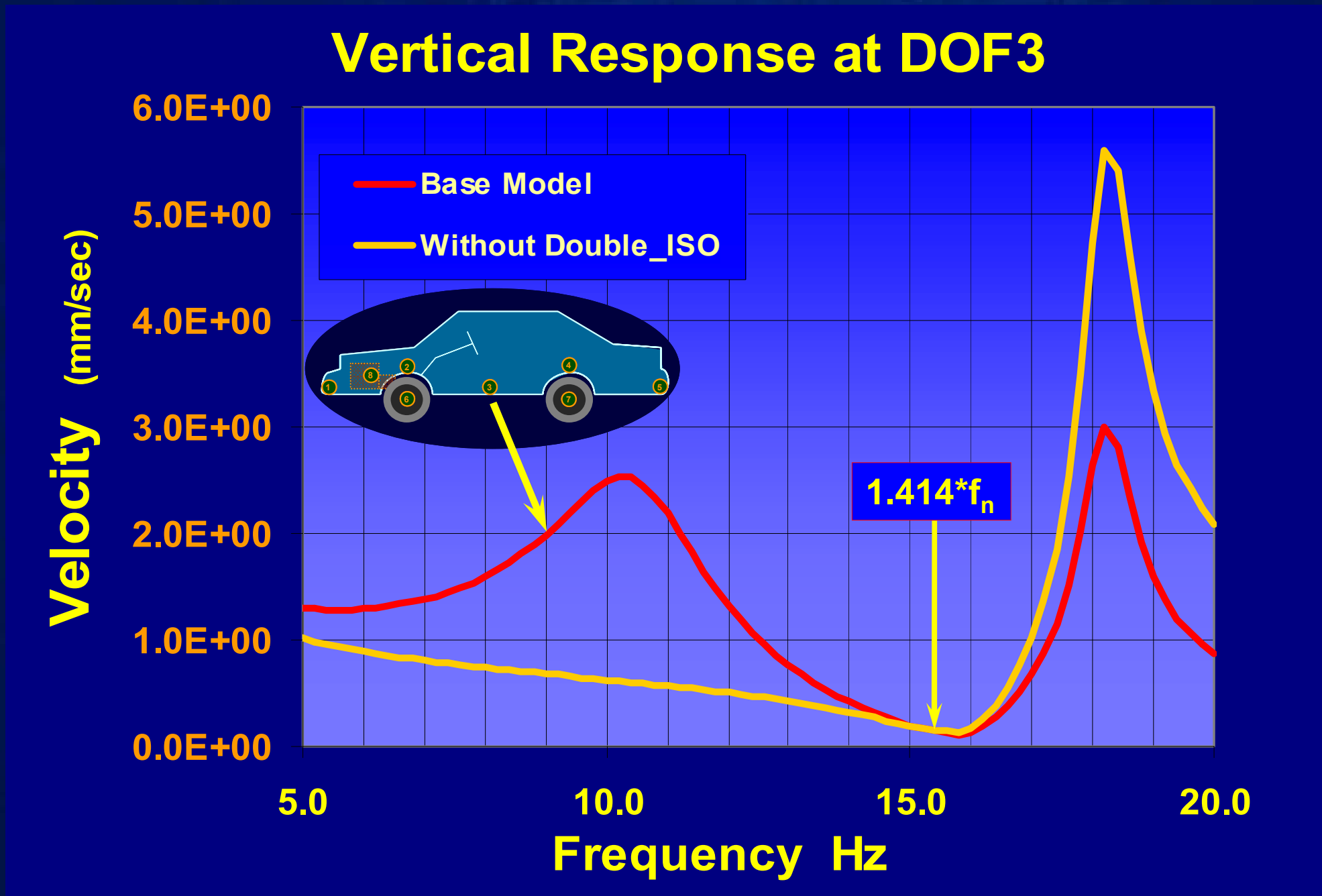
Double Isolation Example



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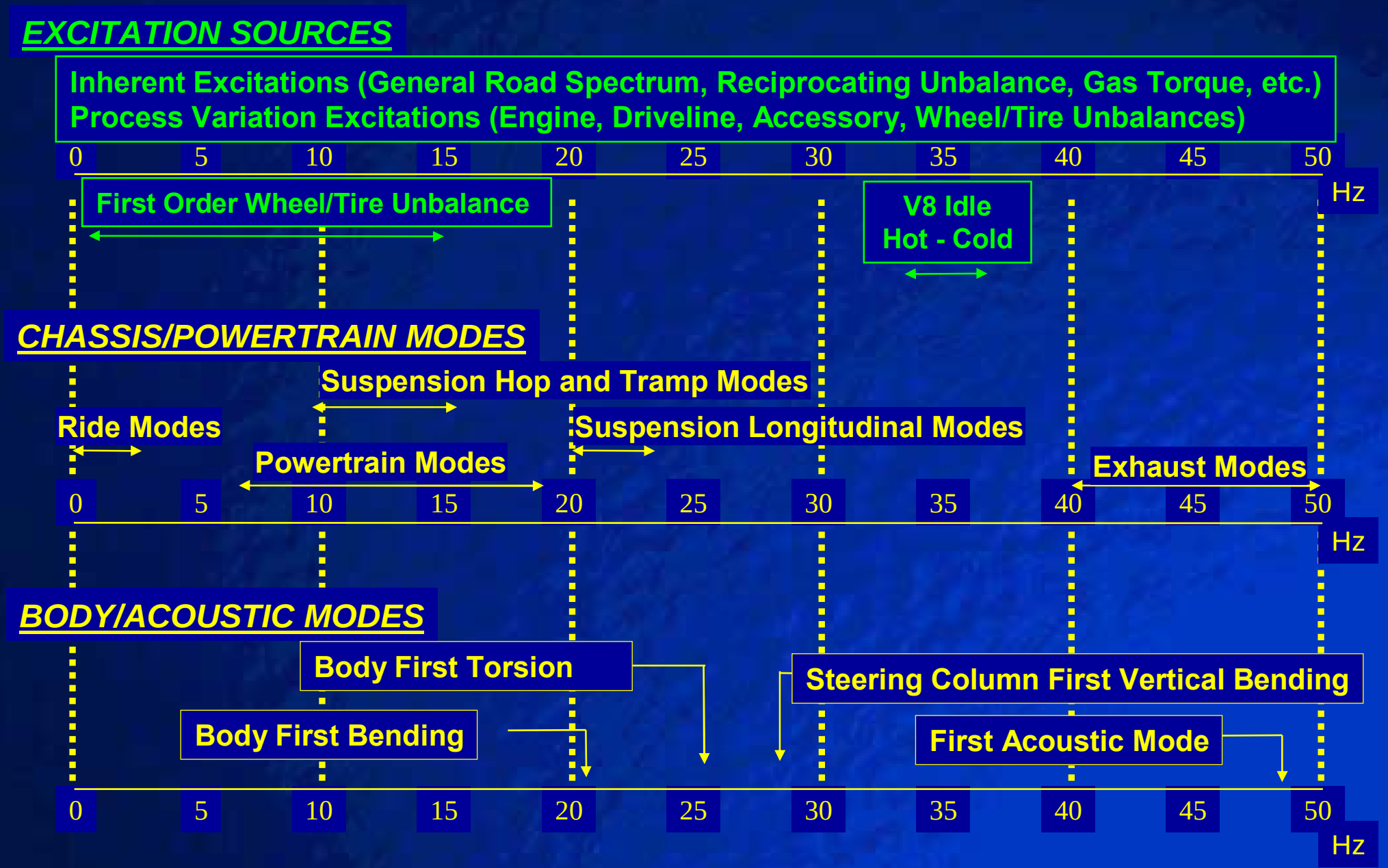
Mode Management Chart



(See Ref. 1)

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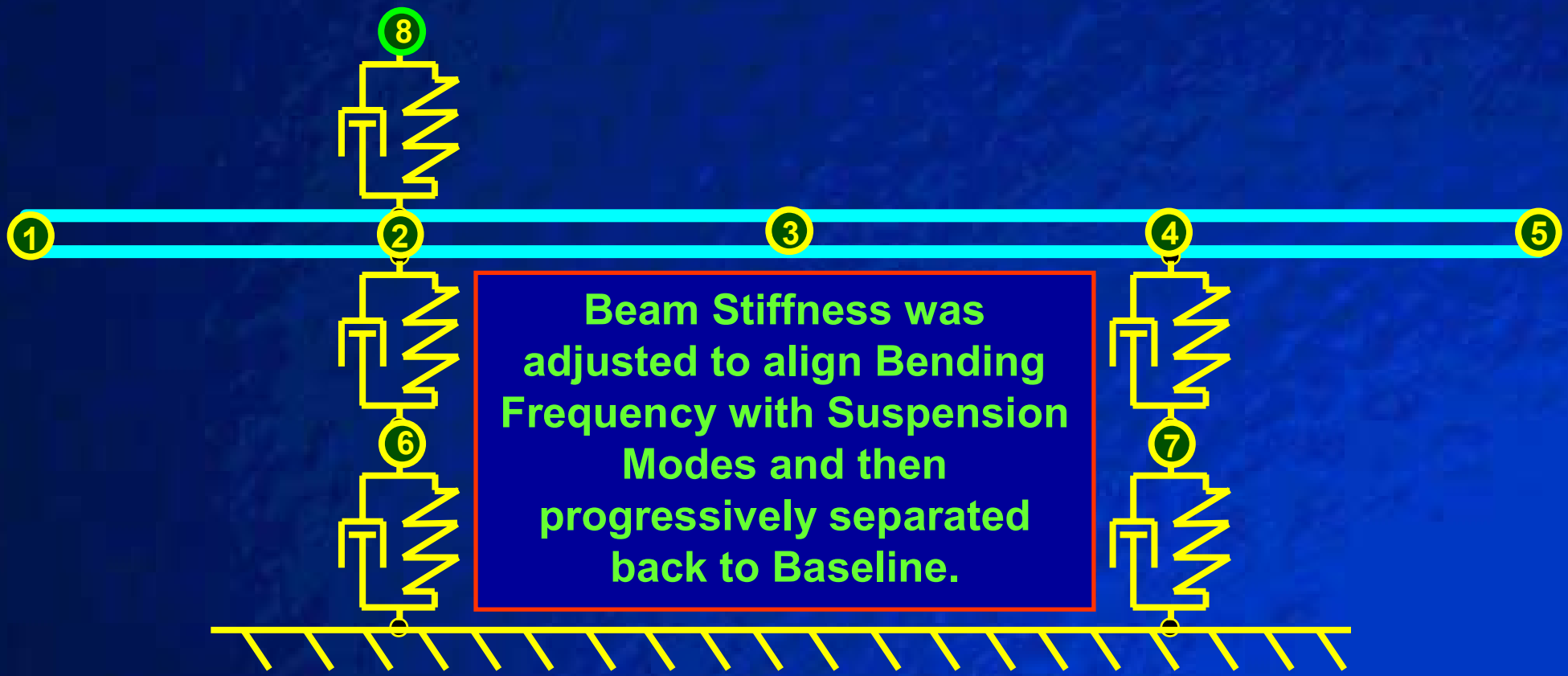
8 Degree of Freedom Vehicle NVH Model

Bending Mode Frequency Separation



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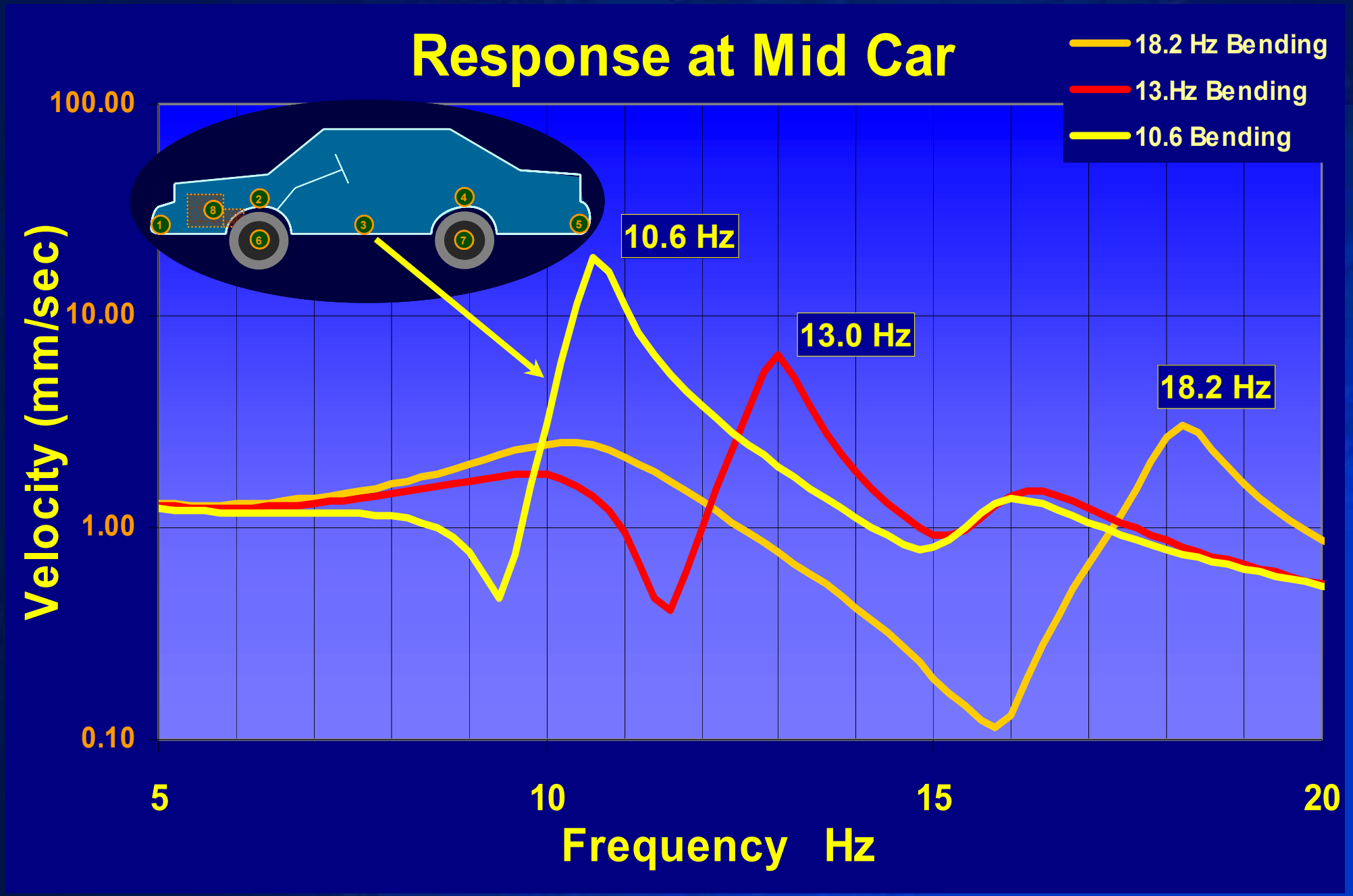
8 DOF Mode Separation Example



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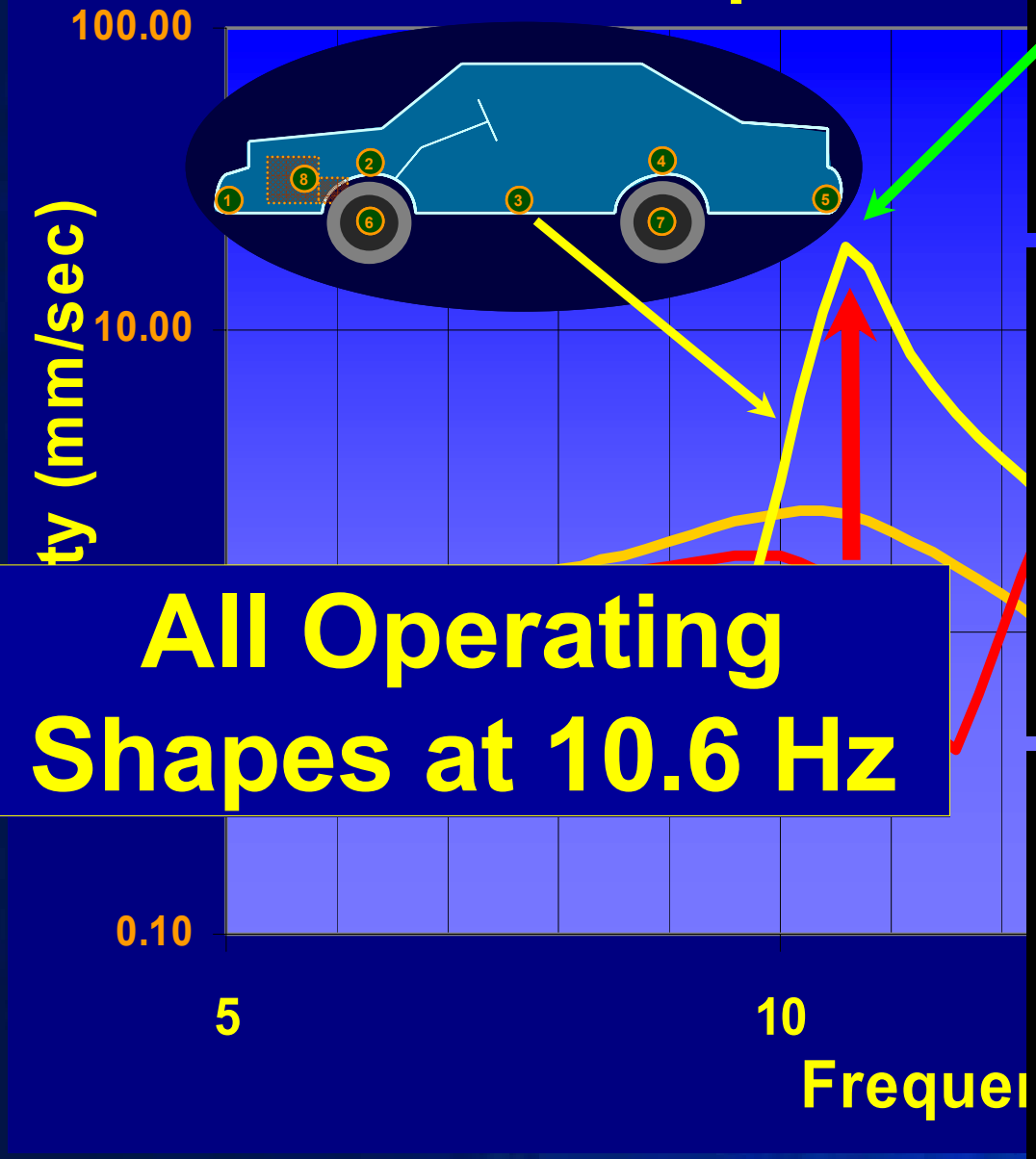
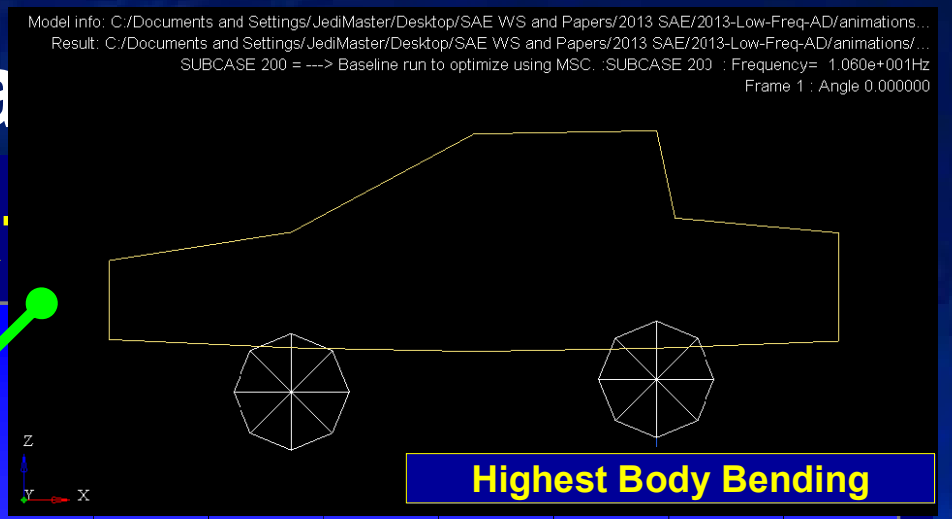
8 DOF Mode Sepa

Response a



All Operating Shapes at 10.6 Hz



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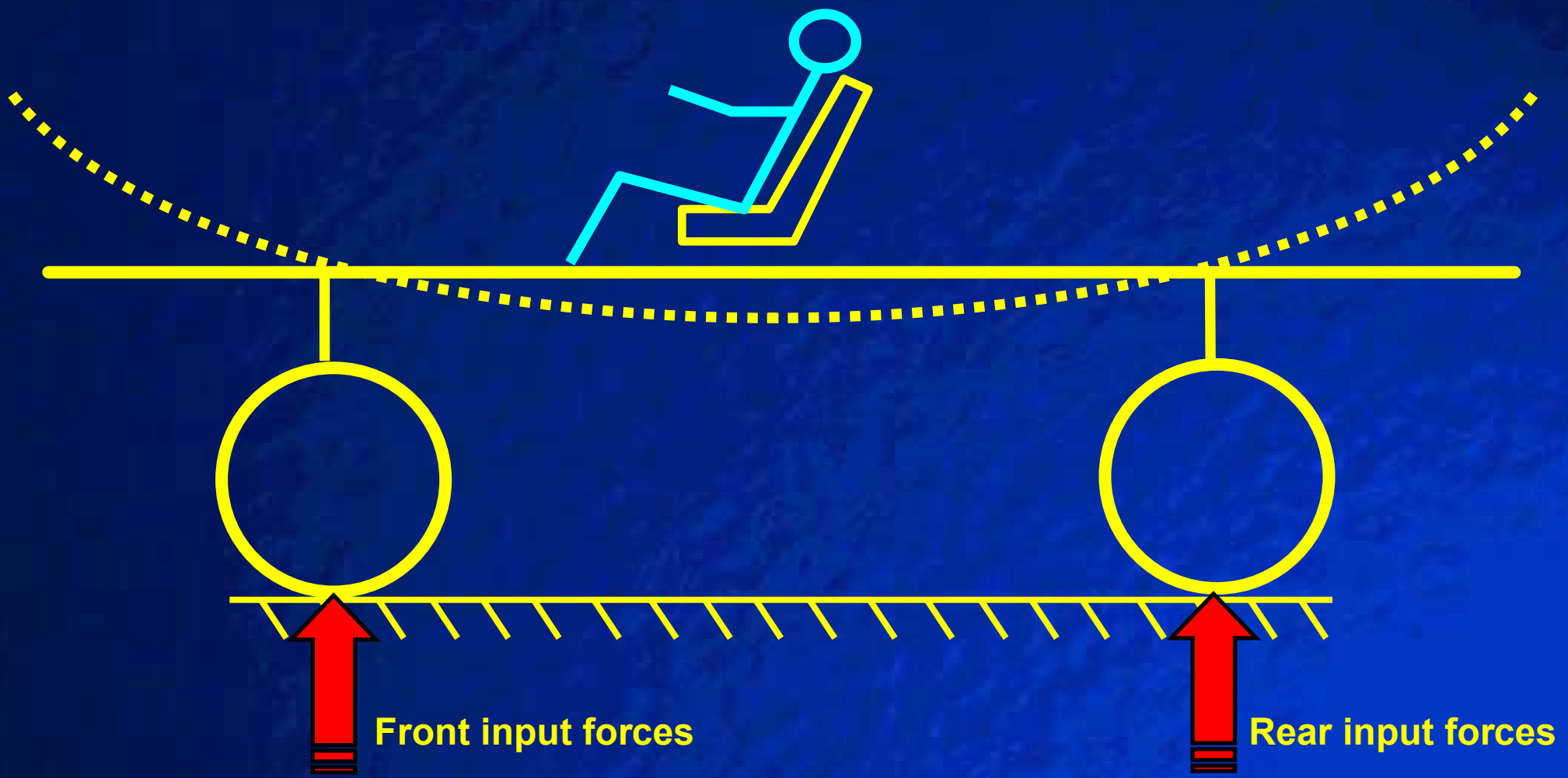
Mount at Nodal Point

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.

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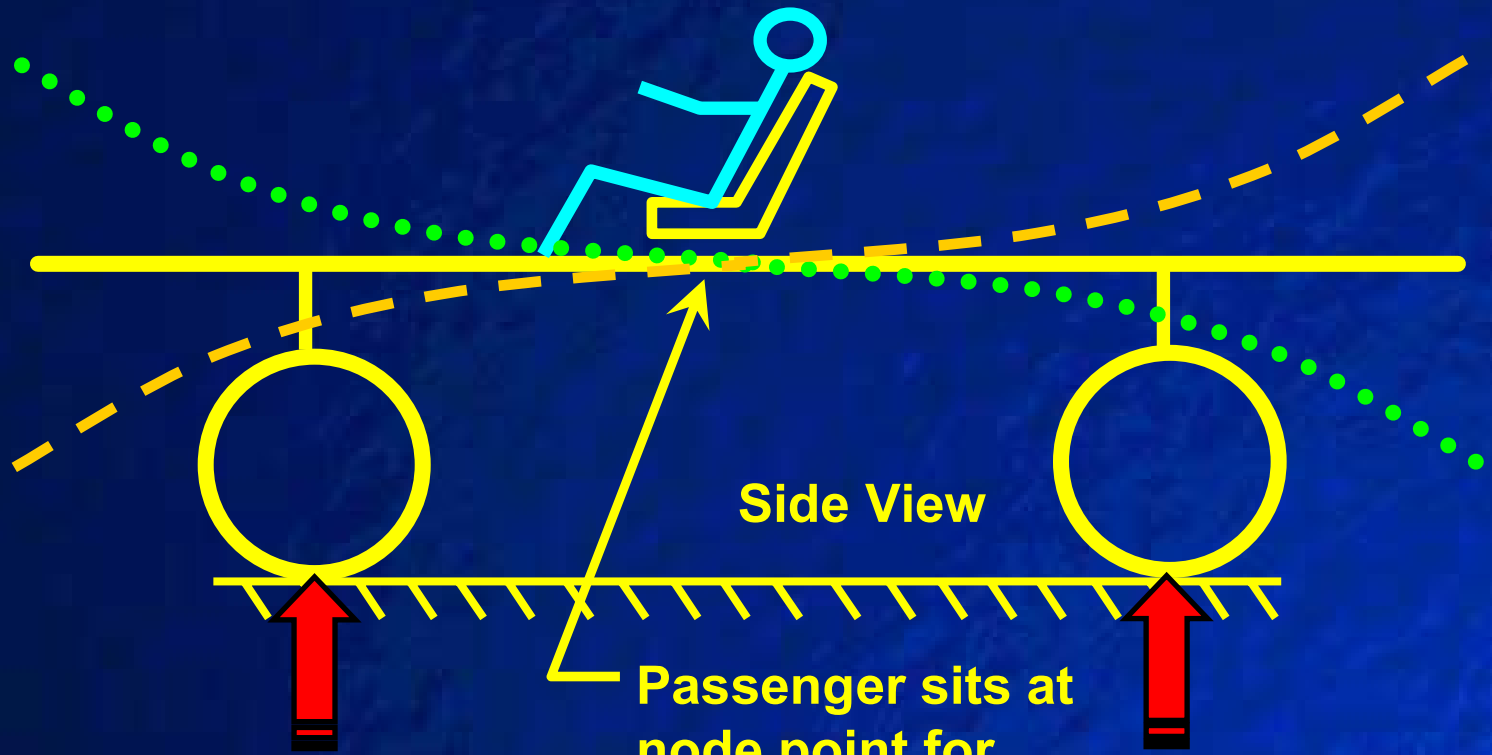
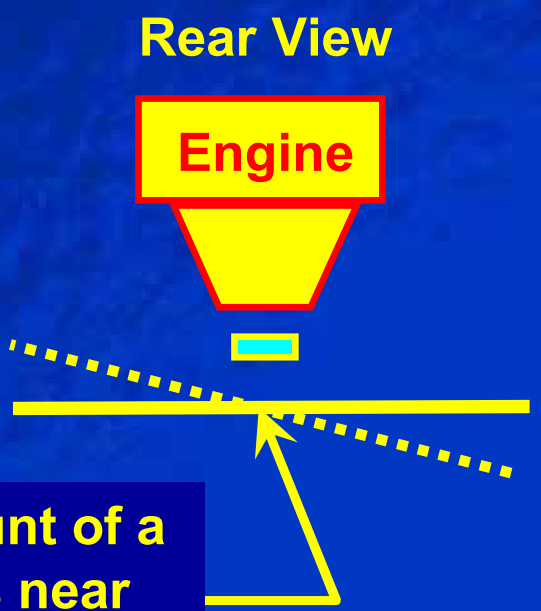
Mount at Nodal Point

First Torsion: Nodal Point Mounting Examples





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

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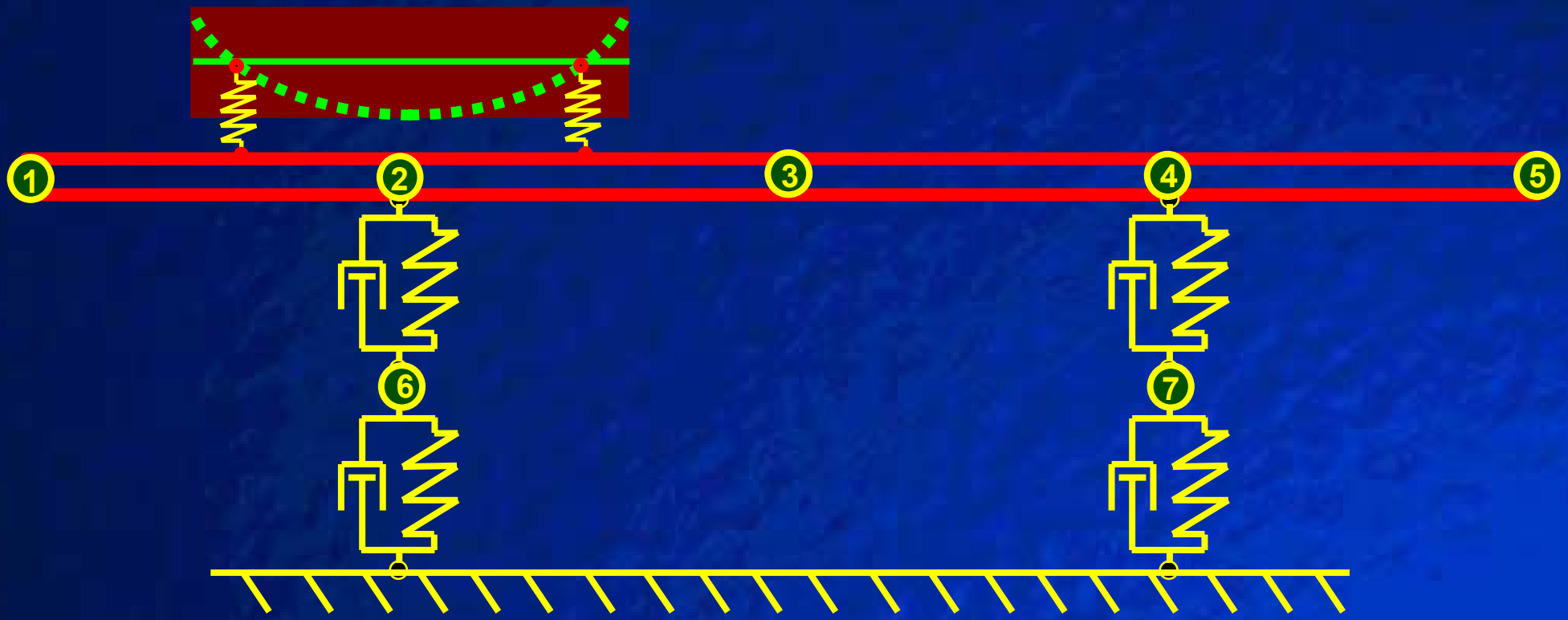
Powertrain Bending Mode Nodal Mounting



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.

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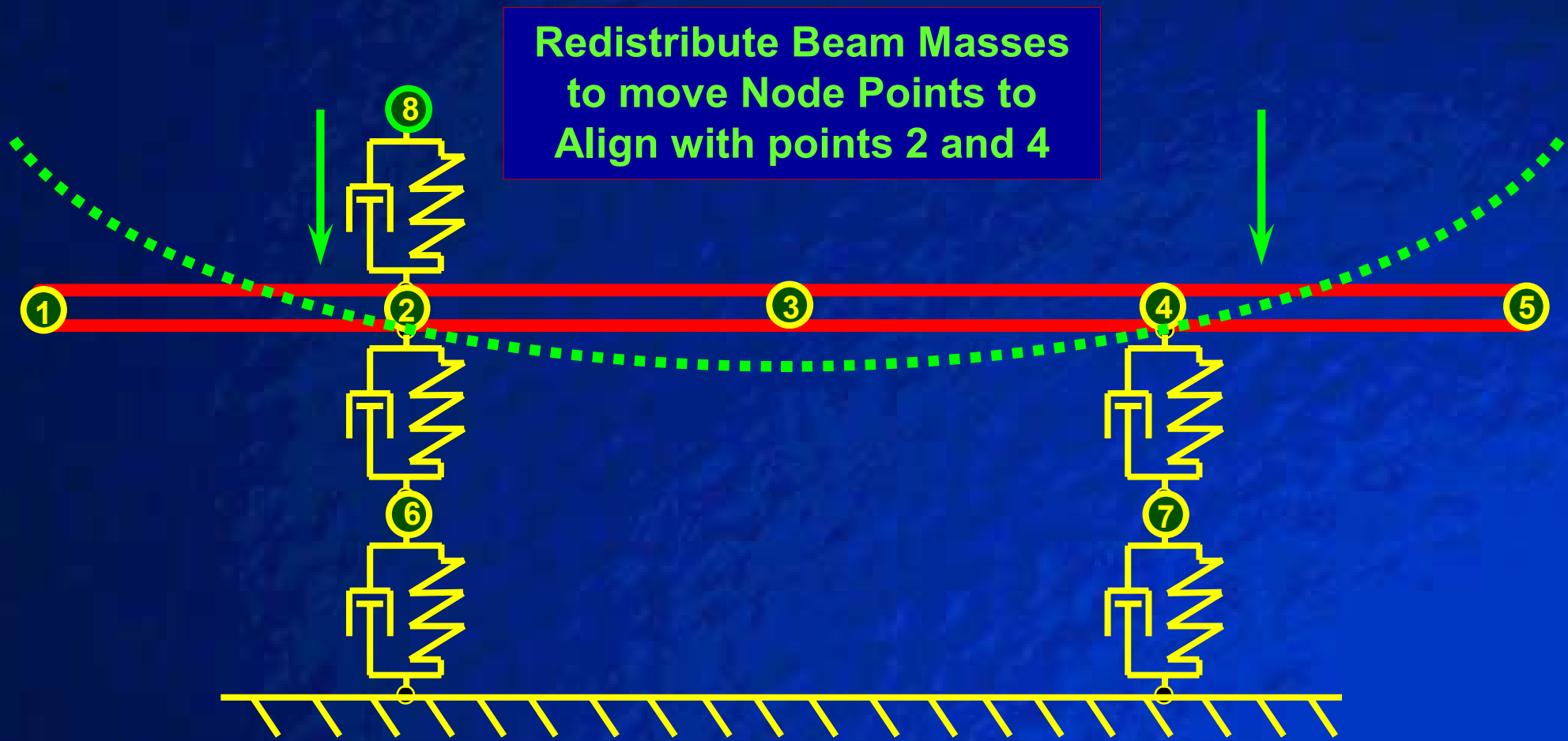
8 Degree of Freedom Vehicle NVH Model

Bending Node Alignment with Wheel Centers



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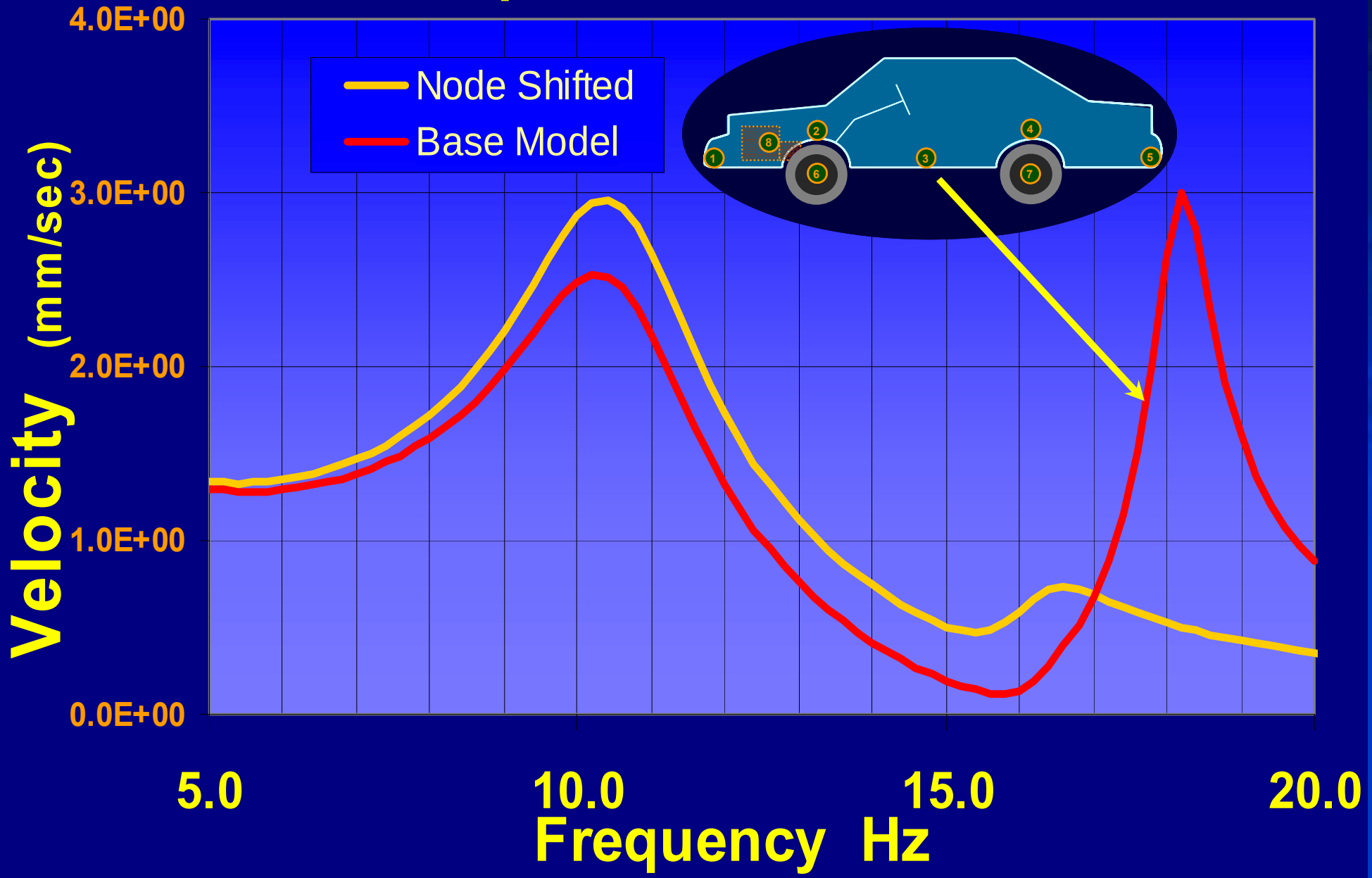
First Bending Nodal Point Alignment

Response at Mid-Car



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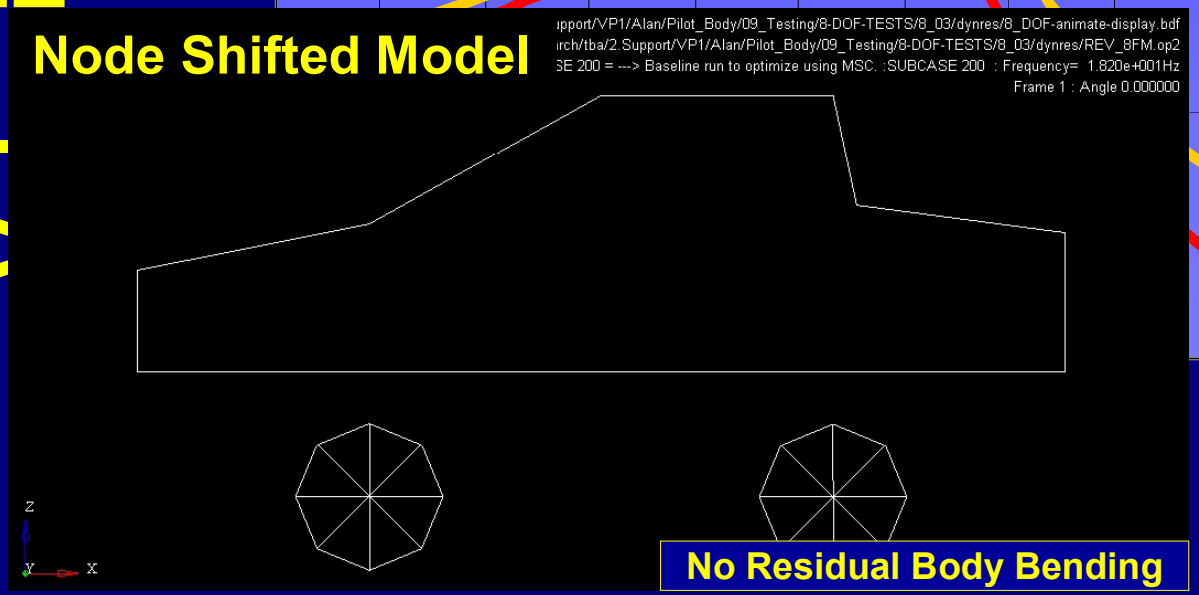
First Bending Nodal Point A

Operating Shapes at 18.2 Hz



No Residual Body Bending

2.0E+00

15.0

20.0

Hz

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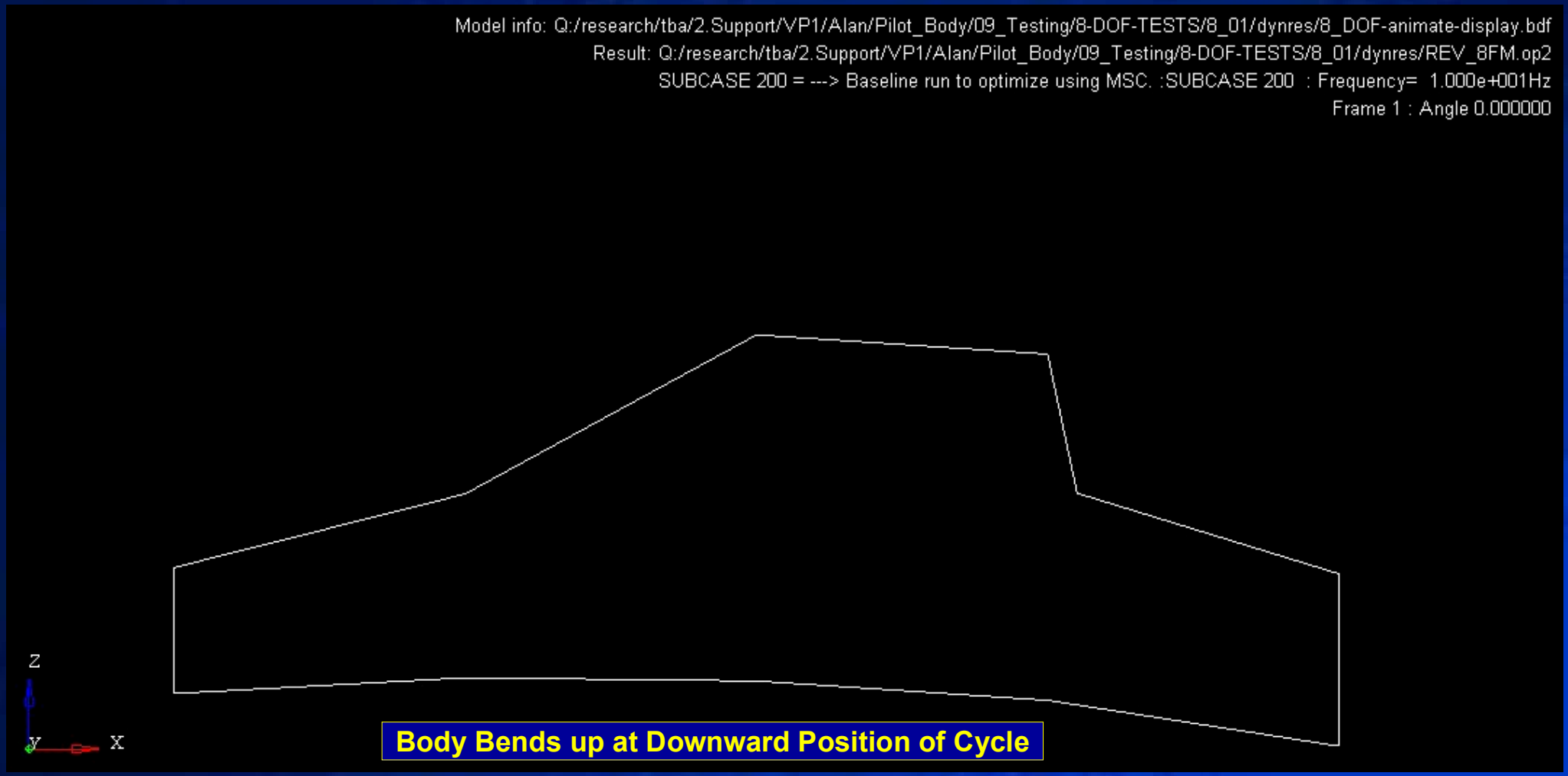
Diagnosis: Increase at 10.2 Hz



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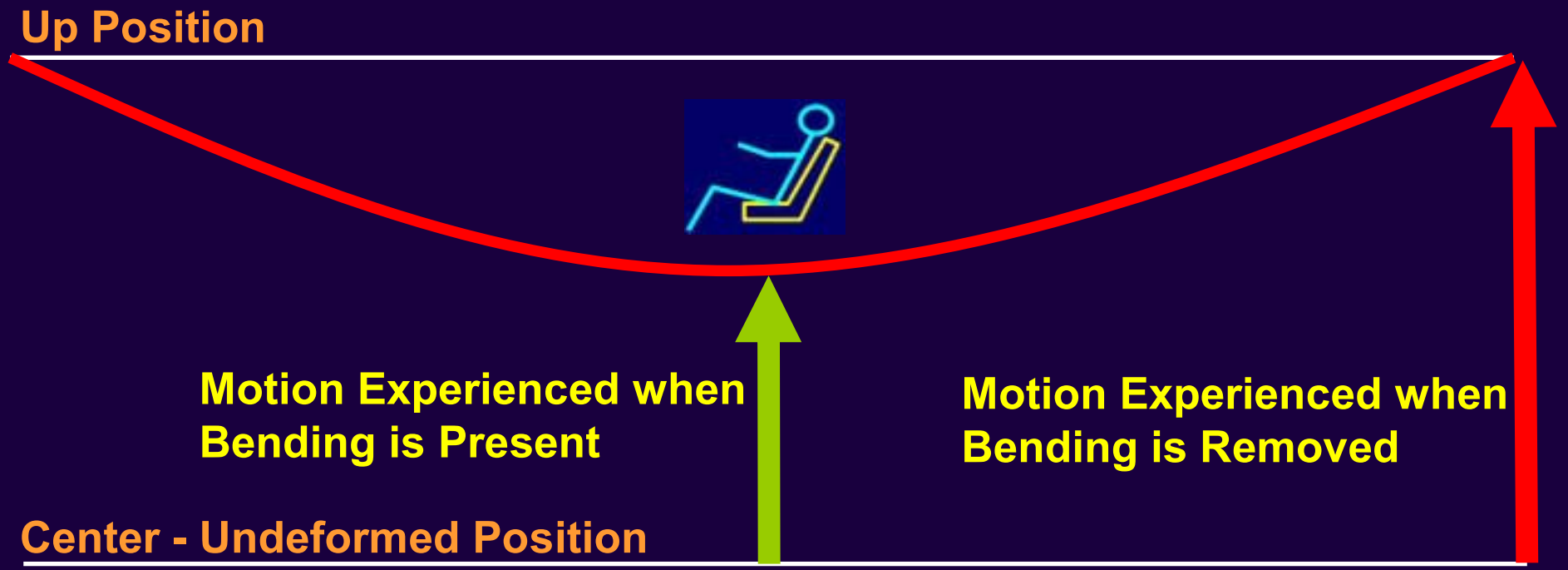
Diagnosis: Increase at 10.2 Hz



CONCLUSION:

Response Increases when a Beneficial mode is Removed.

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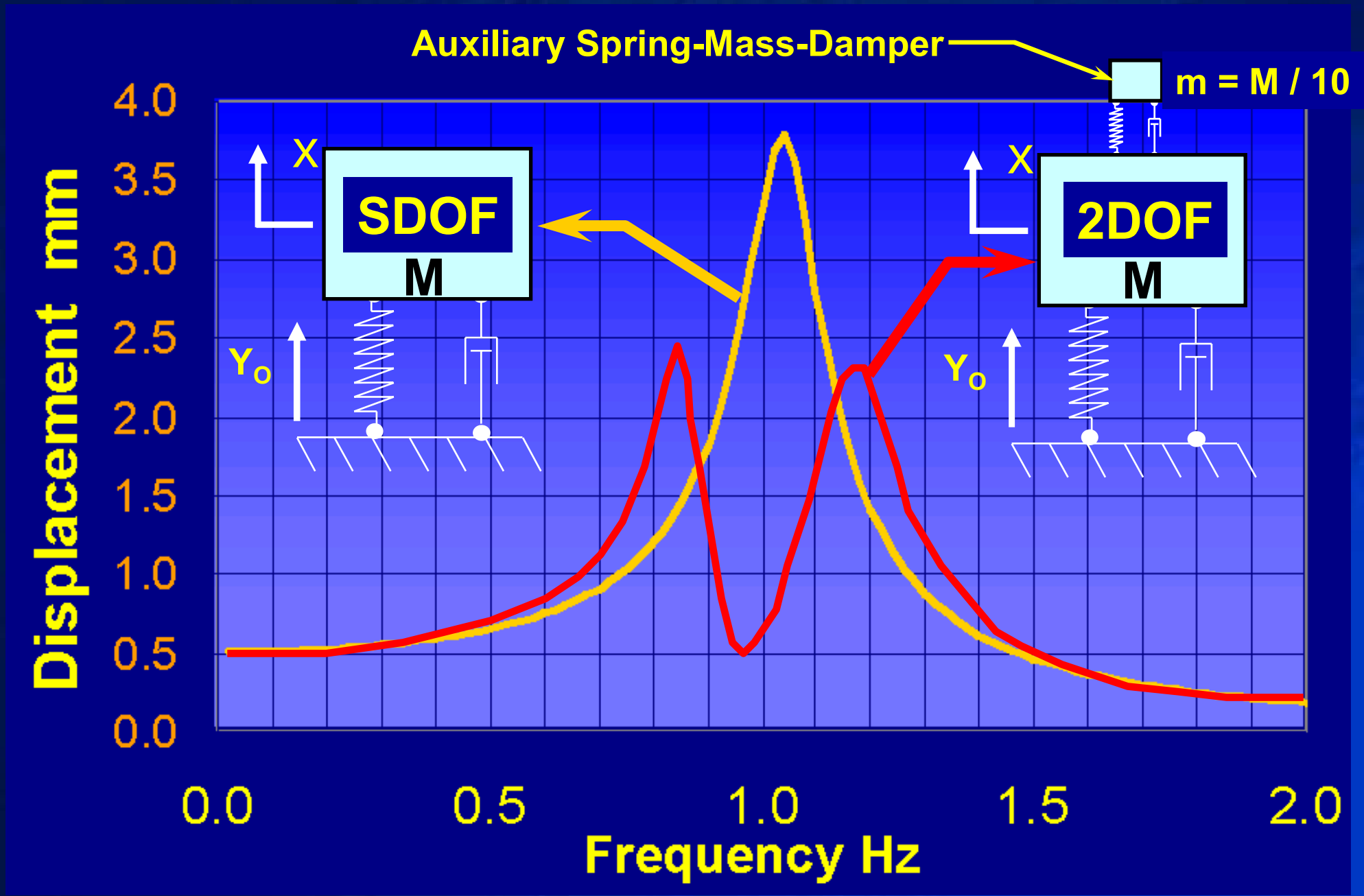
Dynamic Absorber Concept



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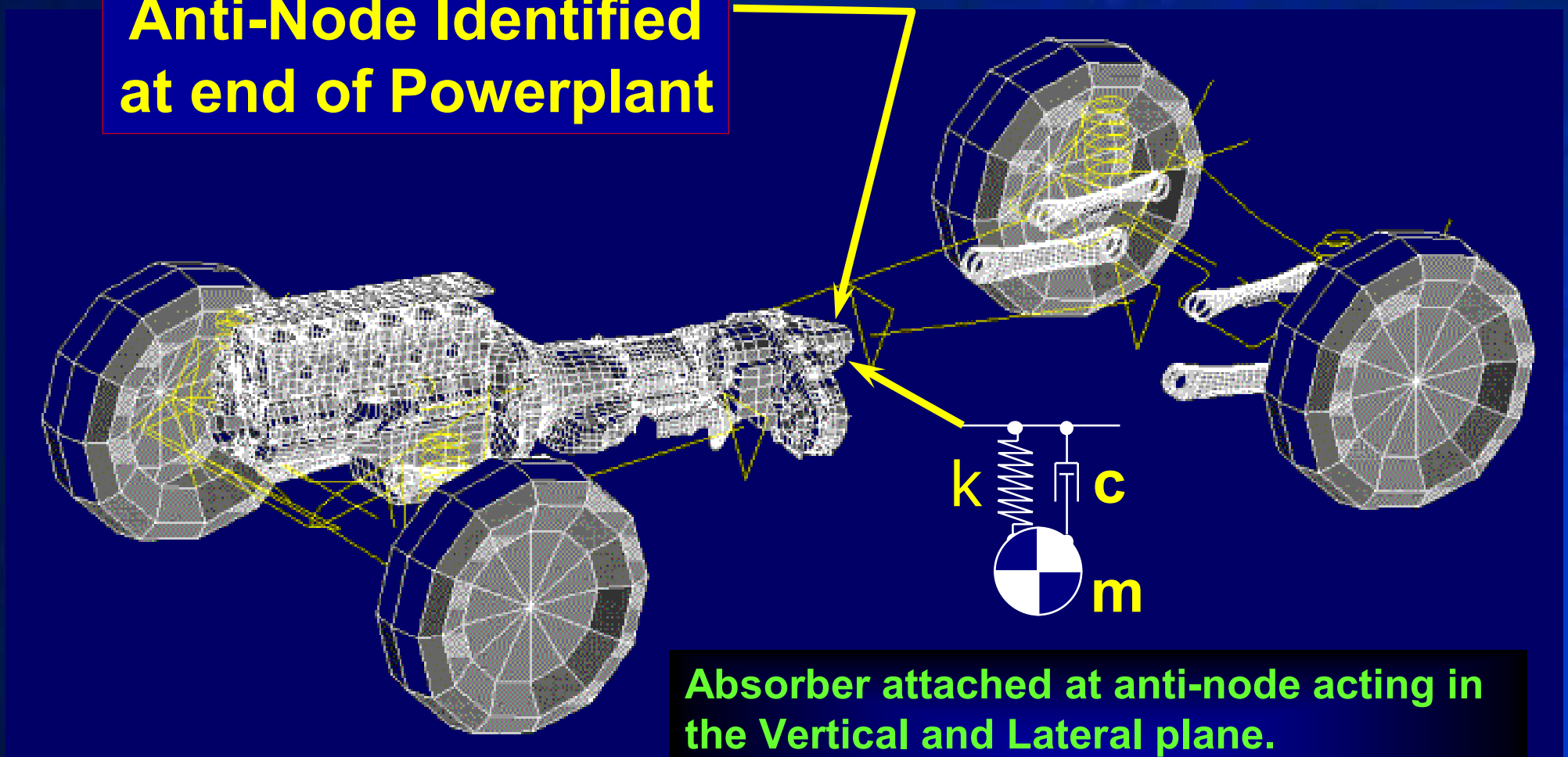
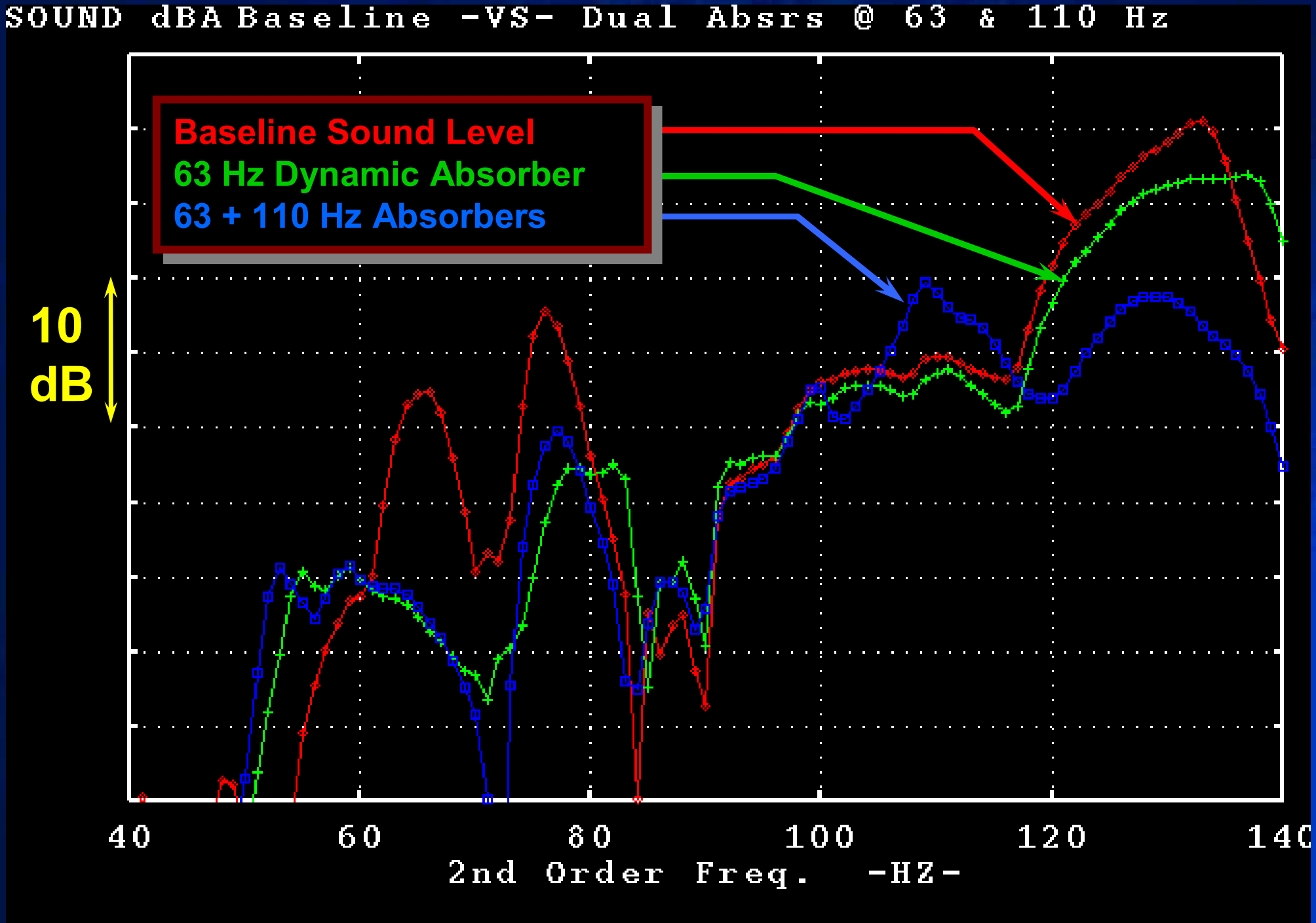
Powertrain Example of Dynamic Absorber



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

Tuning Frequency = $\sqrt{k/m}$

[Figure Courtesy of DaimlerChrysler Corporation]



[Figure Courtesy of DaimlerChrysler Corporation]

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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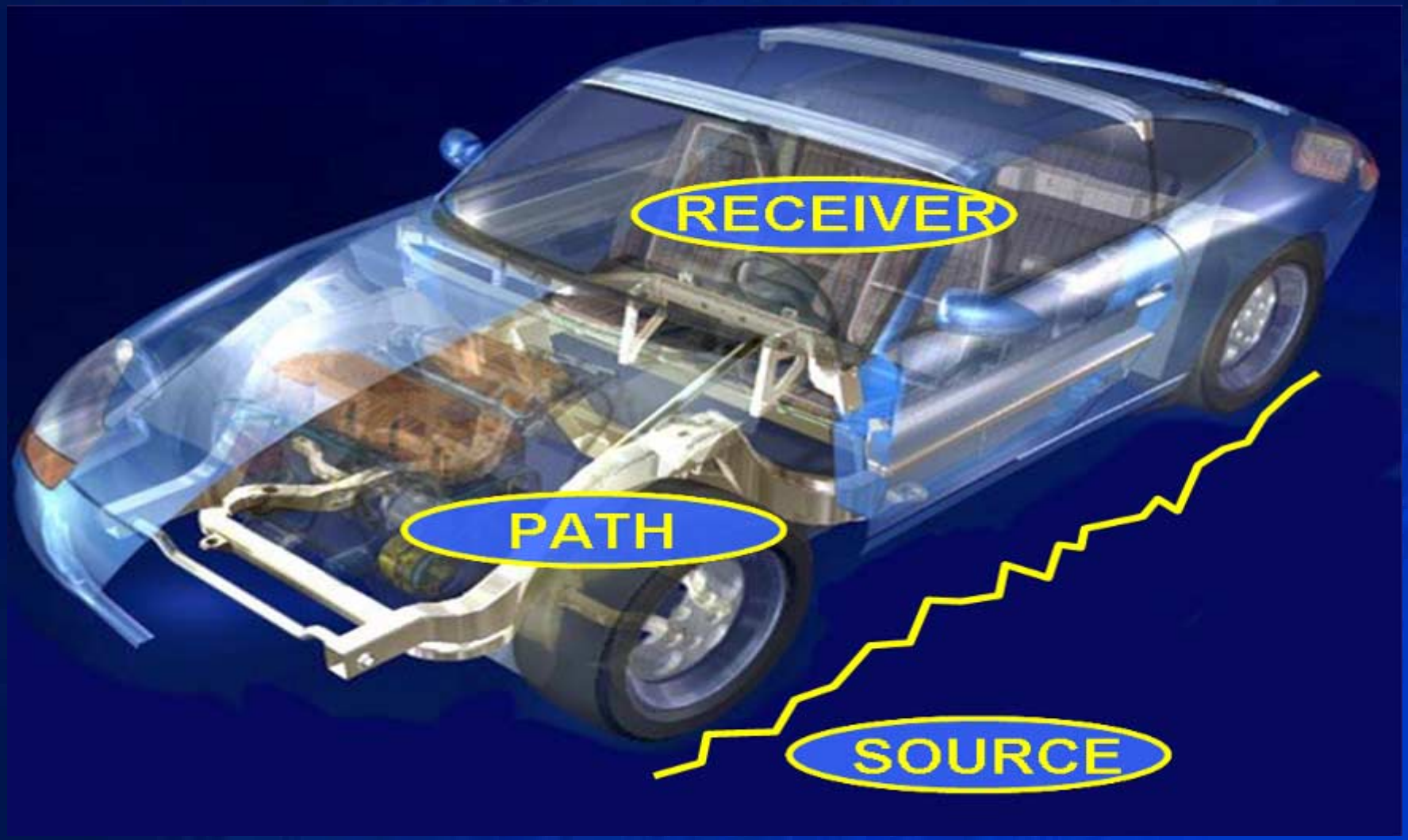
Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- **Mid Frequency Basics** **Jianmin Guan**
- *Live Noise Attenuation Demo*
- New Technology

  Uncertainty and NVH Scatter
- Closing Remarks

Mid Frequency NVH Fundamentals



This looks familiar!

Frequency Range of Interest has changed to 150 Hz to 1000 Hz

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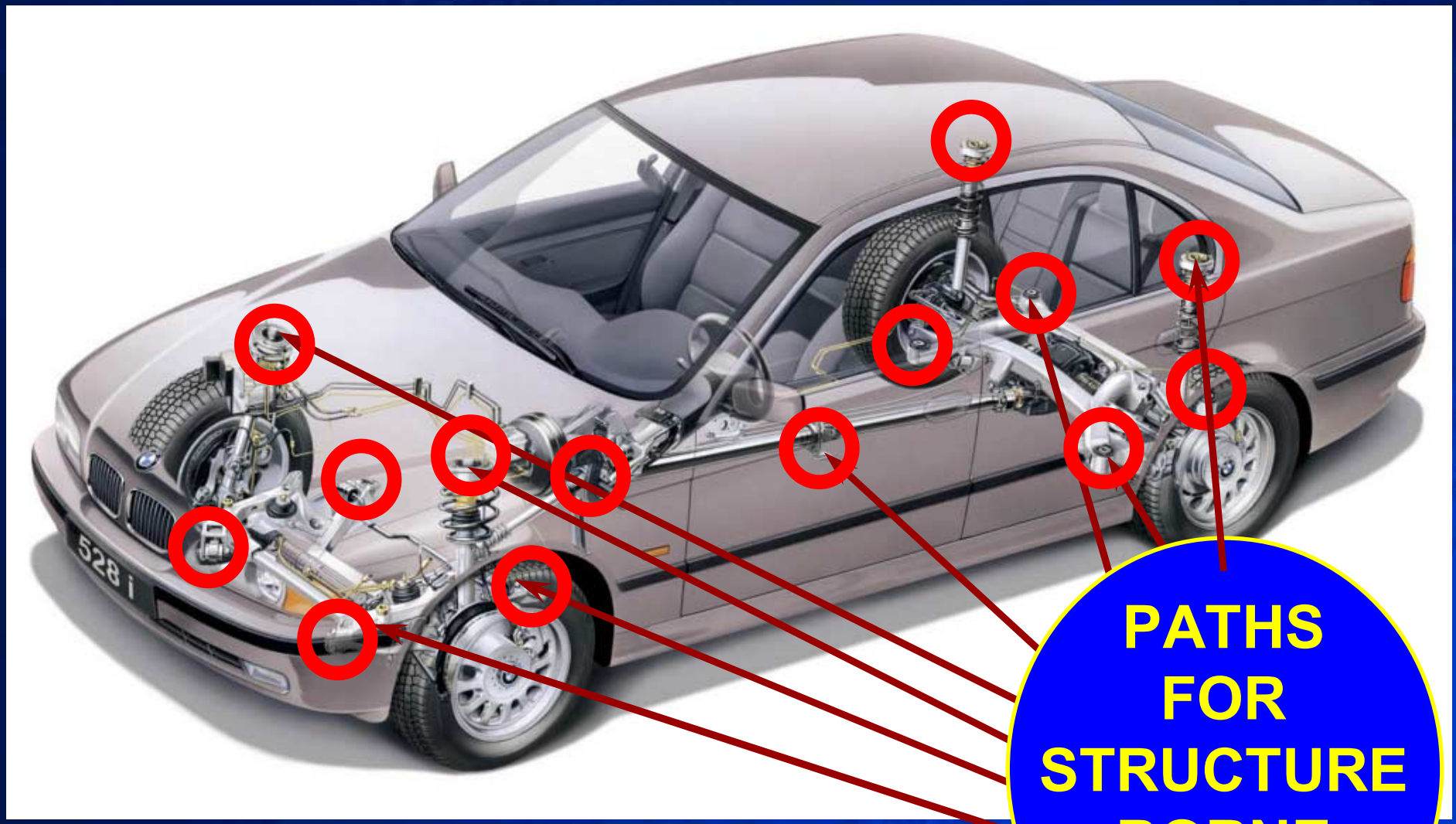
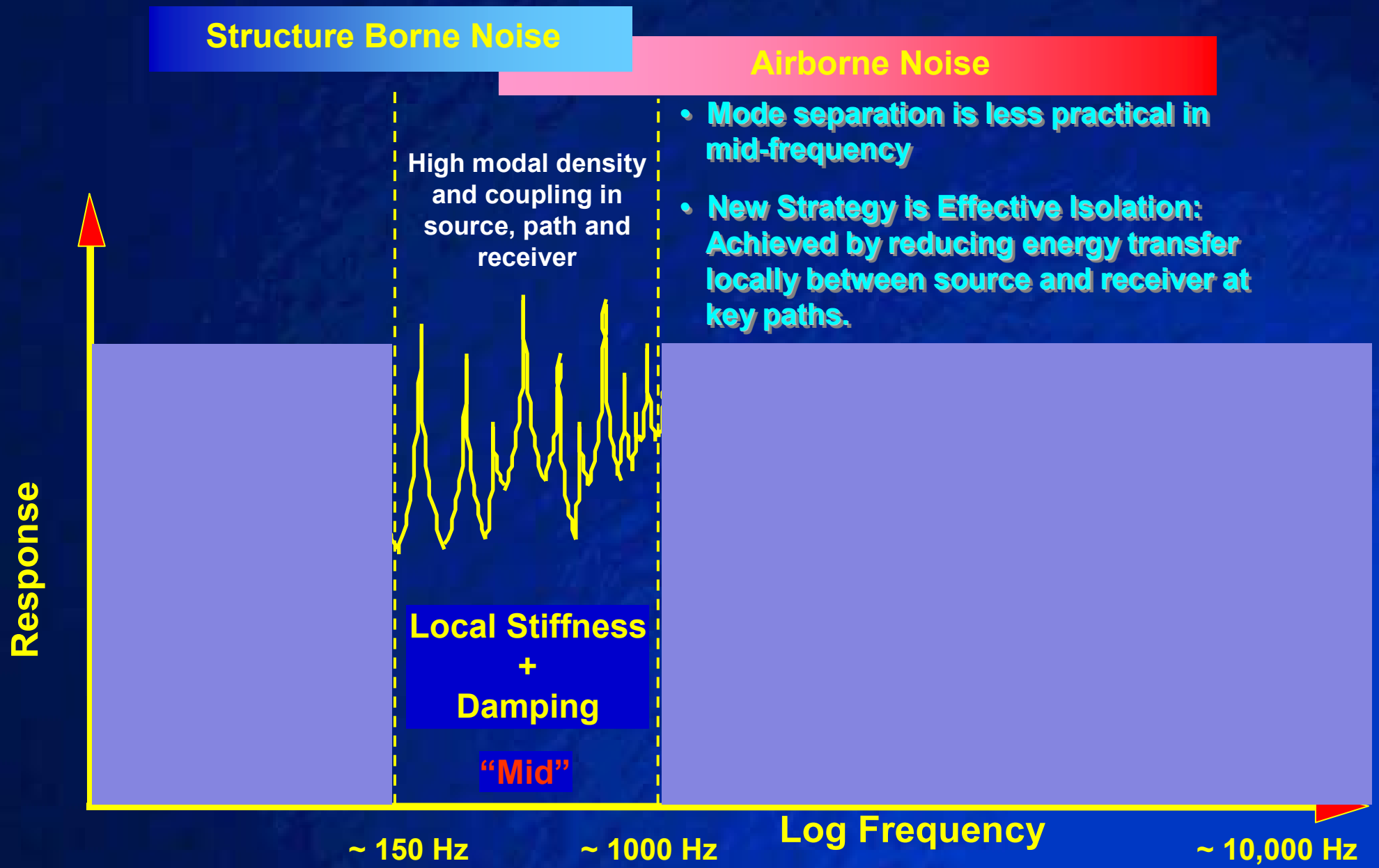
Typical NVH Pathways to the Passenger



Noise Paths are the same as Low Frequency Region

Mid-Frequency Analysis Character



- 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.

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Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths

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Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

- Effective Isolation
- Attenuation along Key Noise Paths

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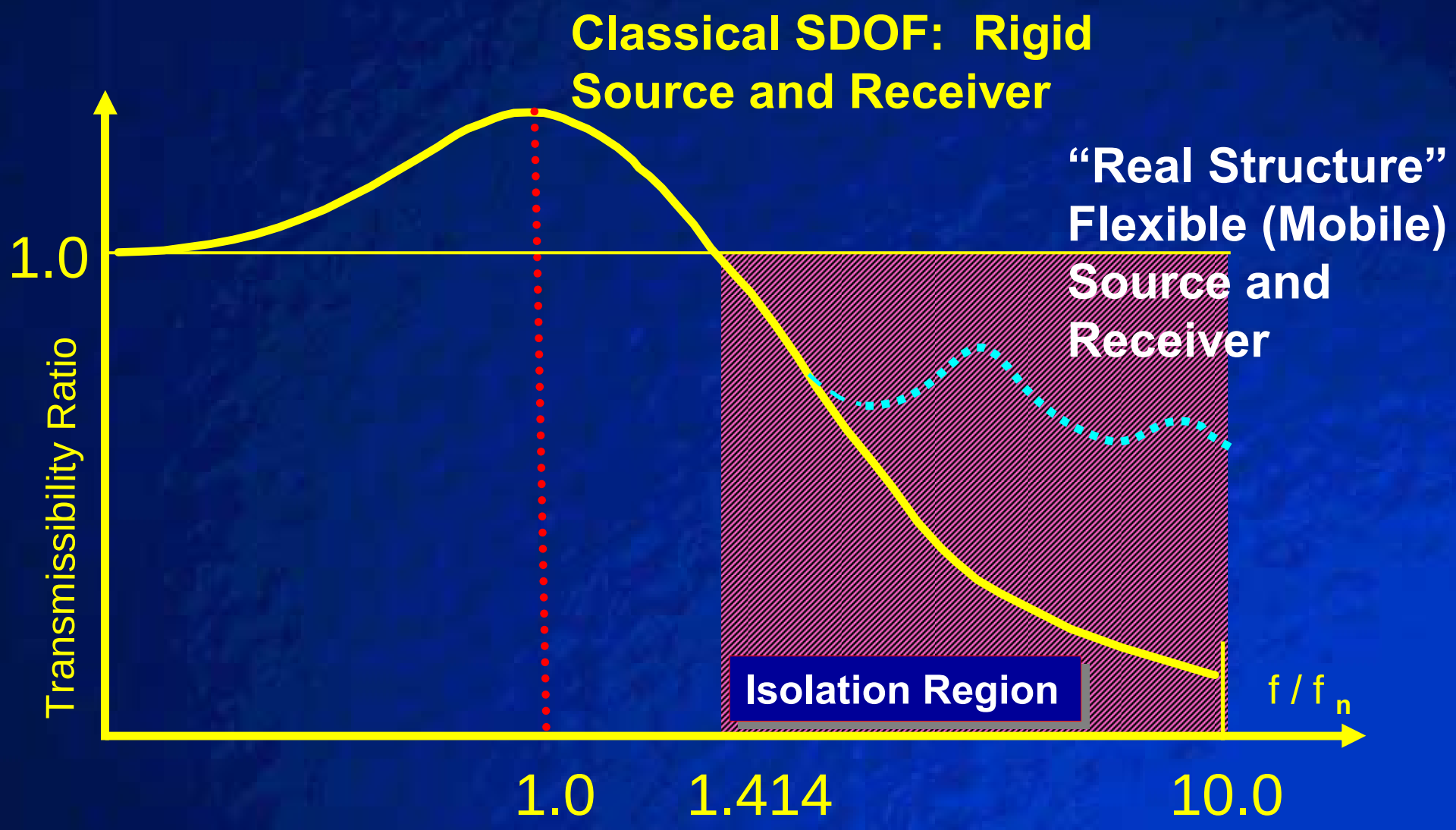
Isolation Effectiveness



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

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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**

$$\text{Mobility} = \frac{\text{Frequency} * \text{Displacement}}{\text{Force}}$$

$$= \frac{\text{Frequency}}{\text{Dynamic Stiffness}}$$

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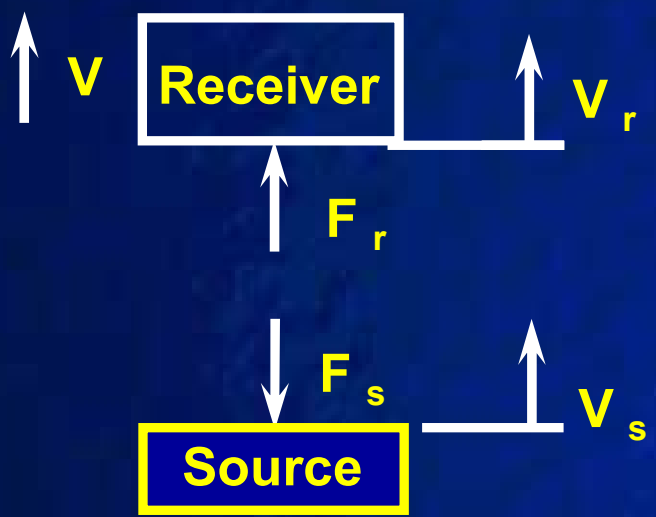
Isolation

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

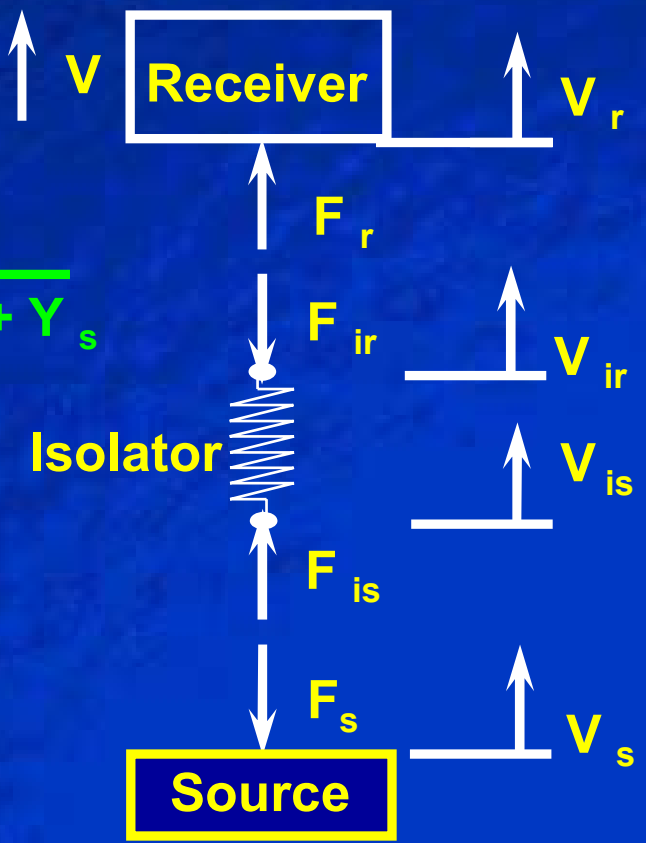
- Transmissibility ratio is used to objectively define measure of isolation

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



$$F_s = \frac{V}{Y_r + Y_s}$$

$$F_s = \frac{V}{Y_i + Y_r + Y_s}$$



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Isolation

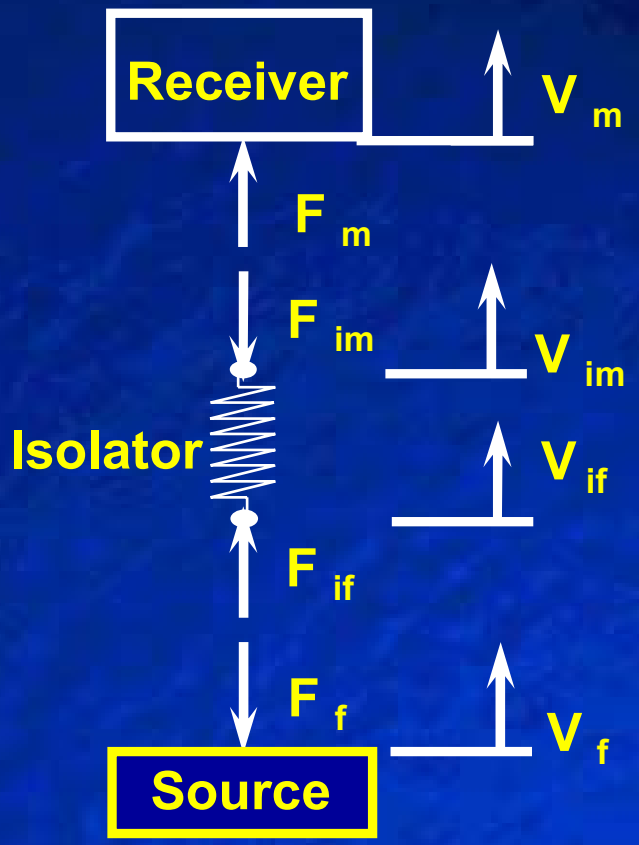
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

$$TR = \left| \left(Y_r + Y_s \right) / \left(Y_i + Y_r + Y_s \right) \right|$$

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 $Y \propto 1/K$

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Designing Noise Paths

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

$\frac{K_{body}}{K_{iso}}$ ↓ \ $\frac{K_{source}}{K_{iso}}$ →	1.0	5.0	20.0
1.0	0.67	0.55	0.51
5.0	0.55	0.29	0.20
20.0	0.51	0.20	0.09

Generic targets:
body to bushing stiffness ratio of at least 5.0
source to bushing stiffness ratio of at least 20.0

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Body-to-Bushing Stiffness Ratio Relationship to Transmissibility

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Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths

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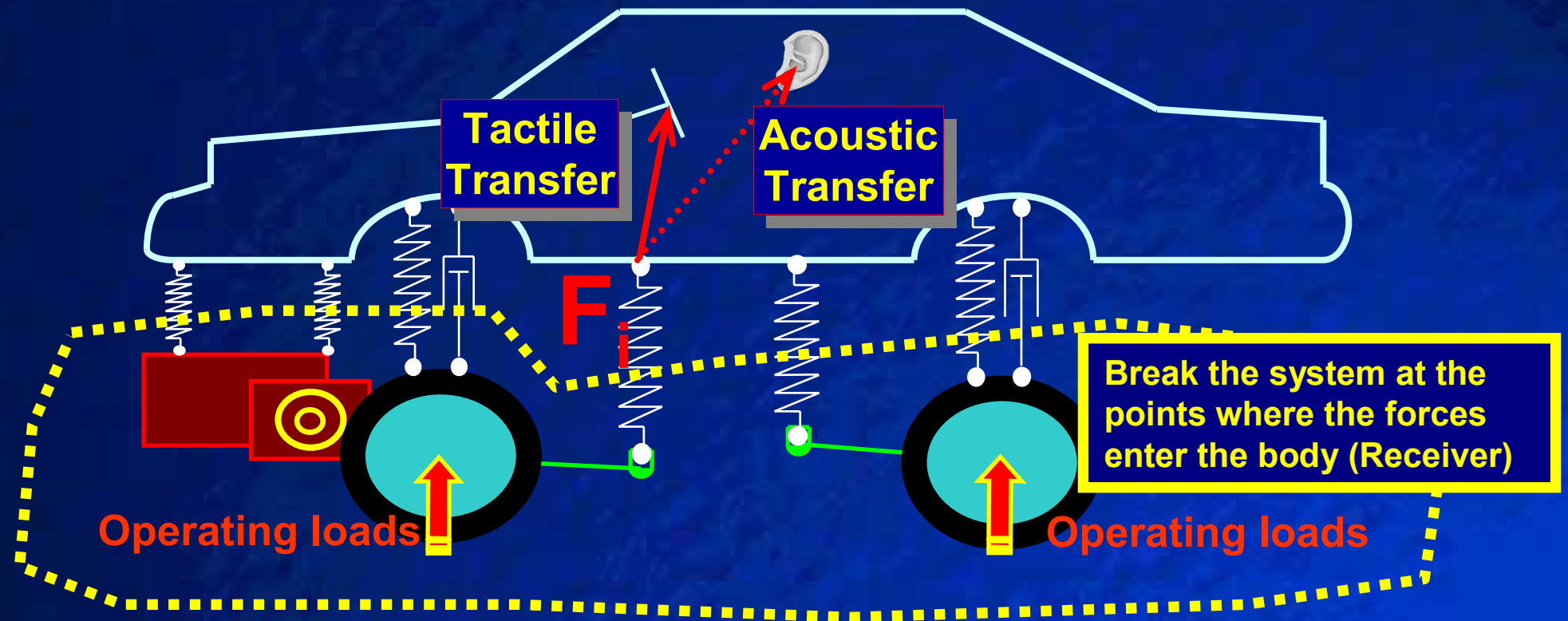
Identifying Key NVH Paths

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



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

$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [(P/F)_i * F_i]$$

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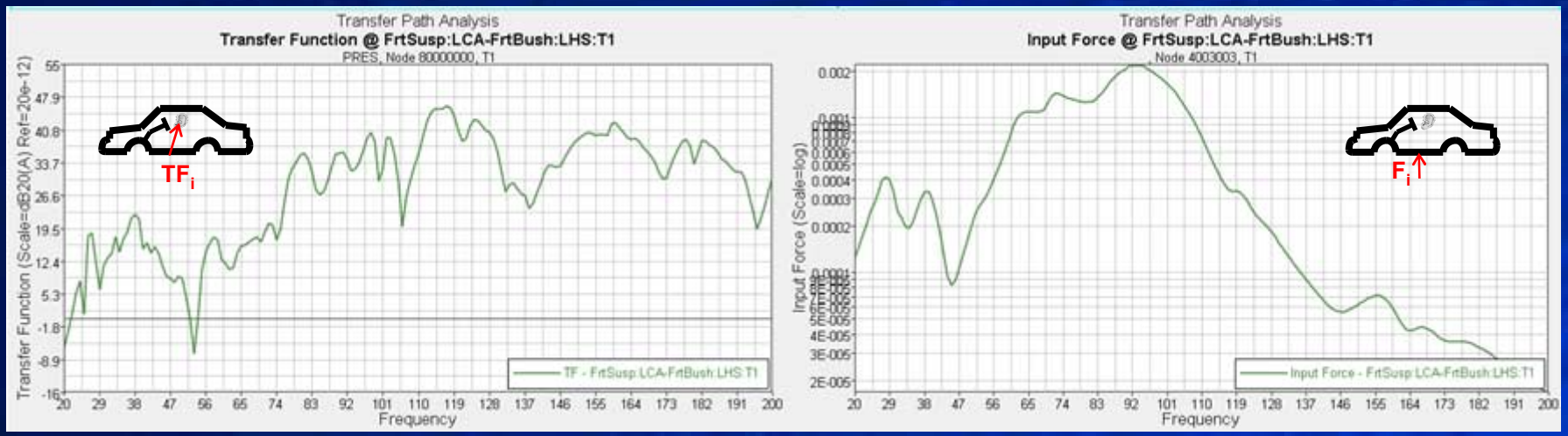
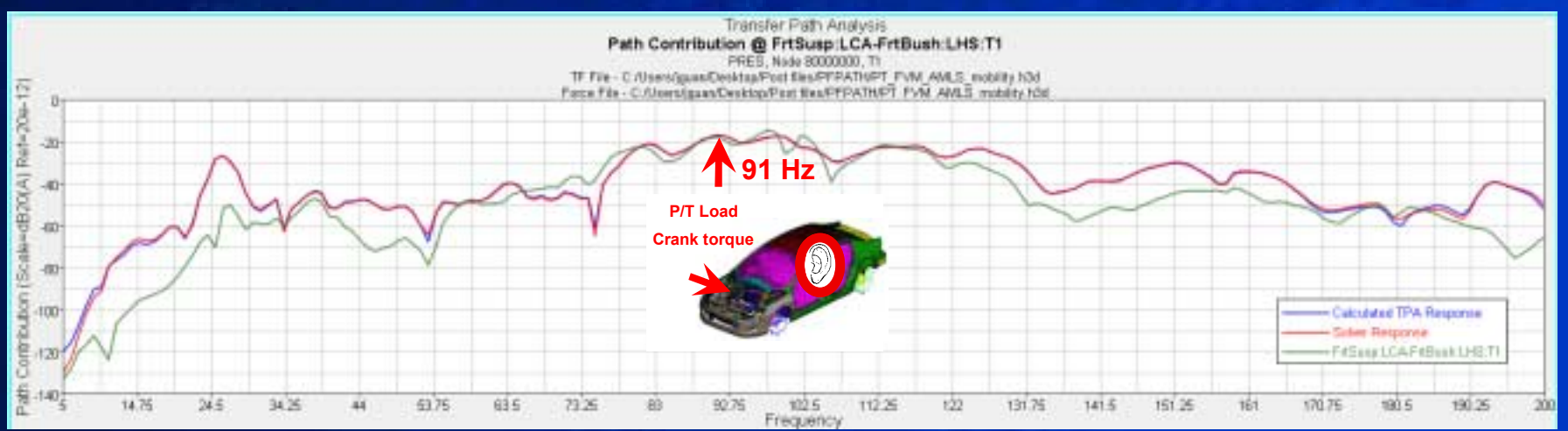
Identifying Key NVH Paths

TPA Example: Contribution at One Transfer Path

Partial response from a particular path: $P_i = (P/F)_i * F_i$





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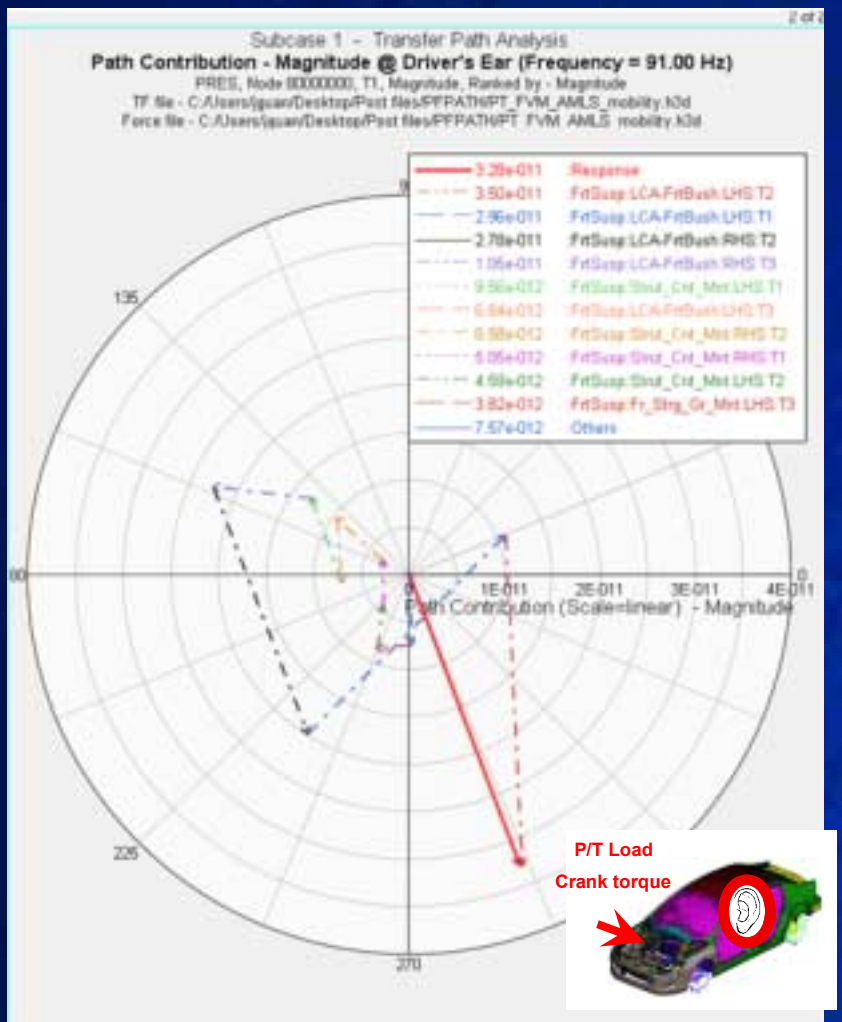
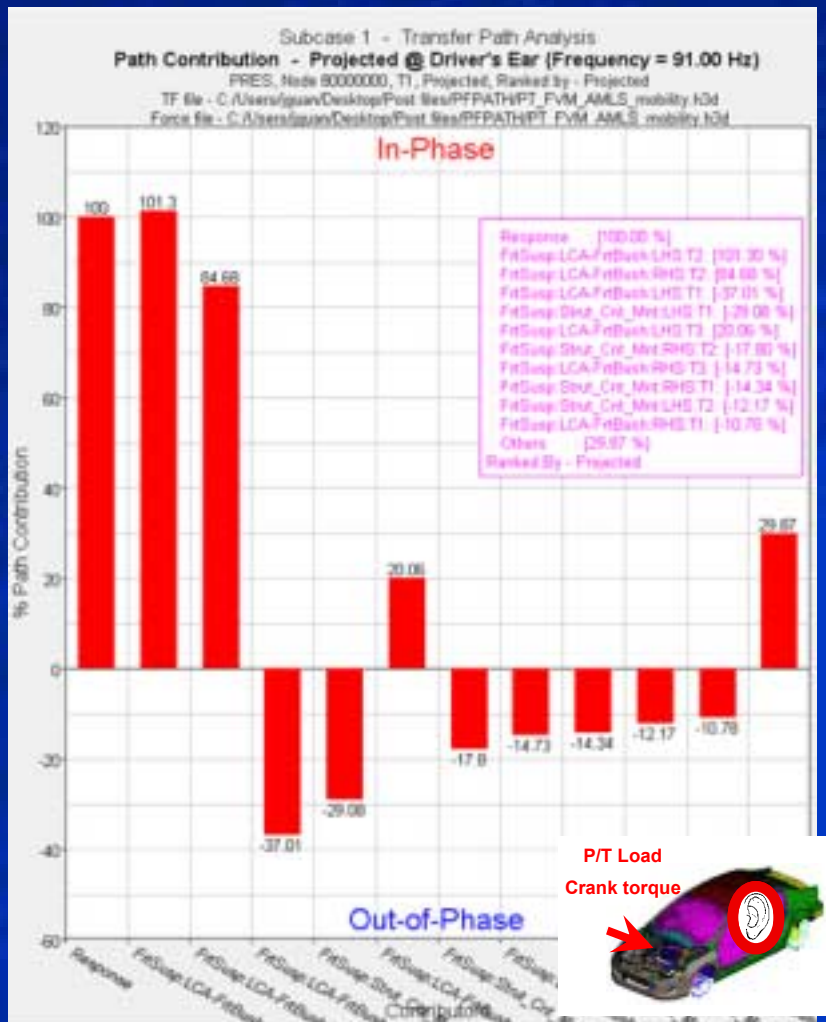
Identifying Key NVH Paths

TPA Example: Sum of Key Transfer Paths at One Peak

Total Response: $P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [(P/F)_i * F_i]$





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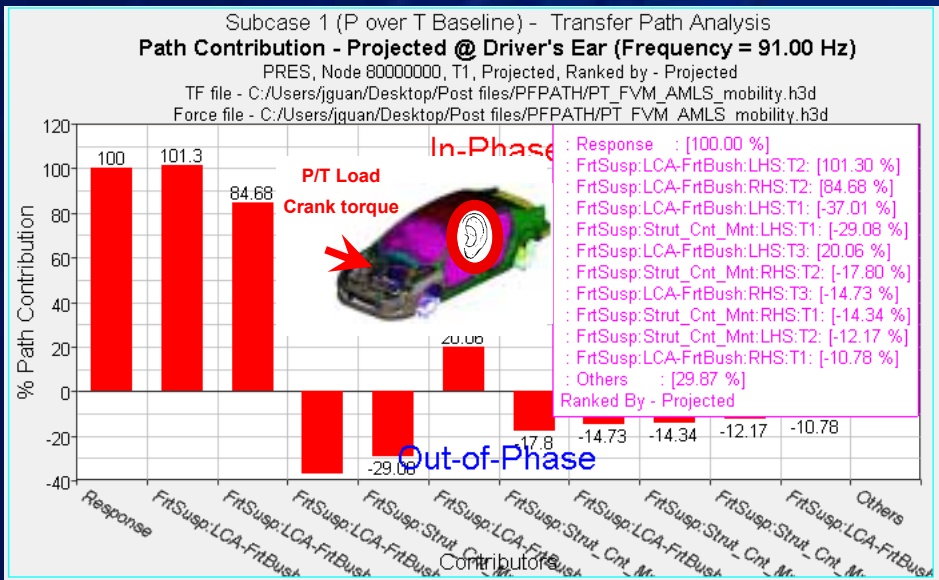
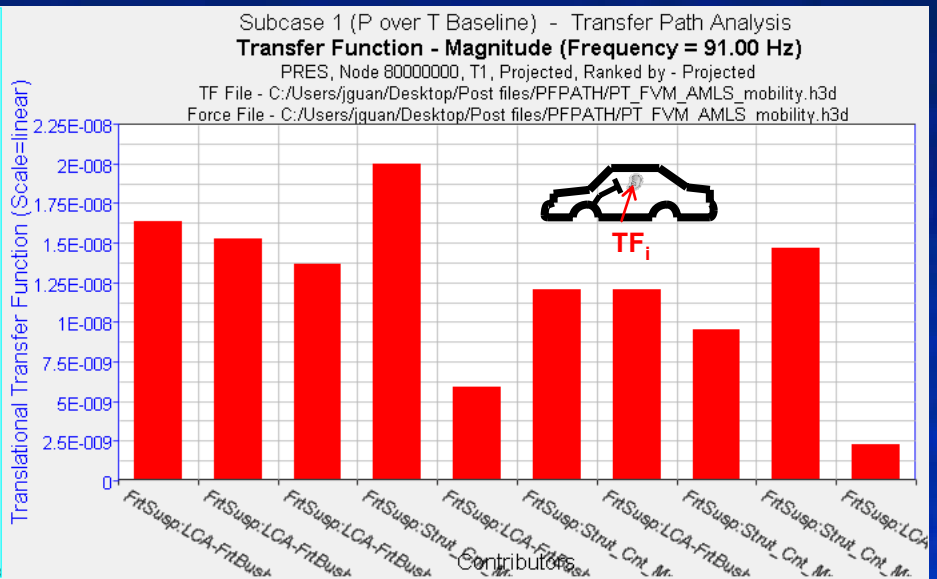
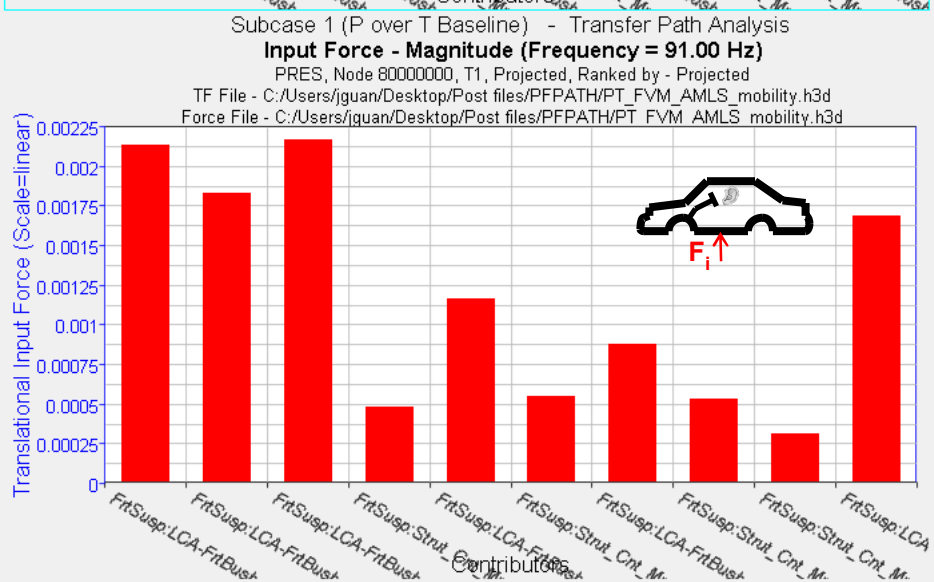
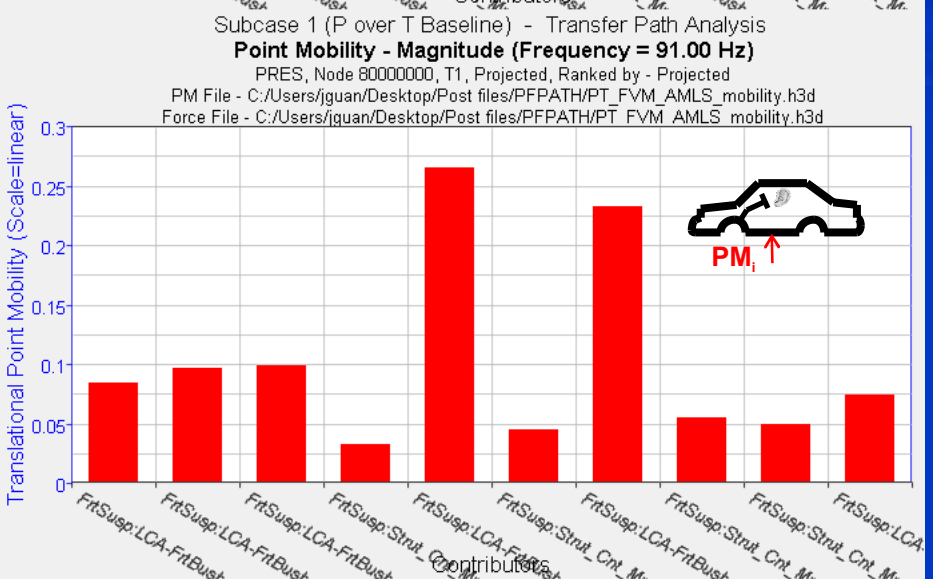
Attenuating Key NVH Paths

TPA Example: Identifying Root Cause of Dominant Paths









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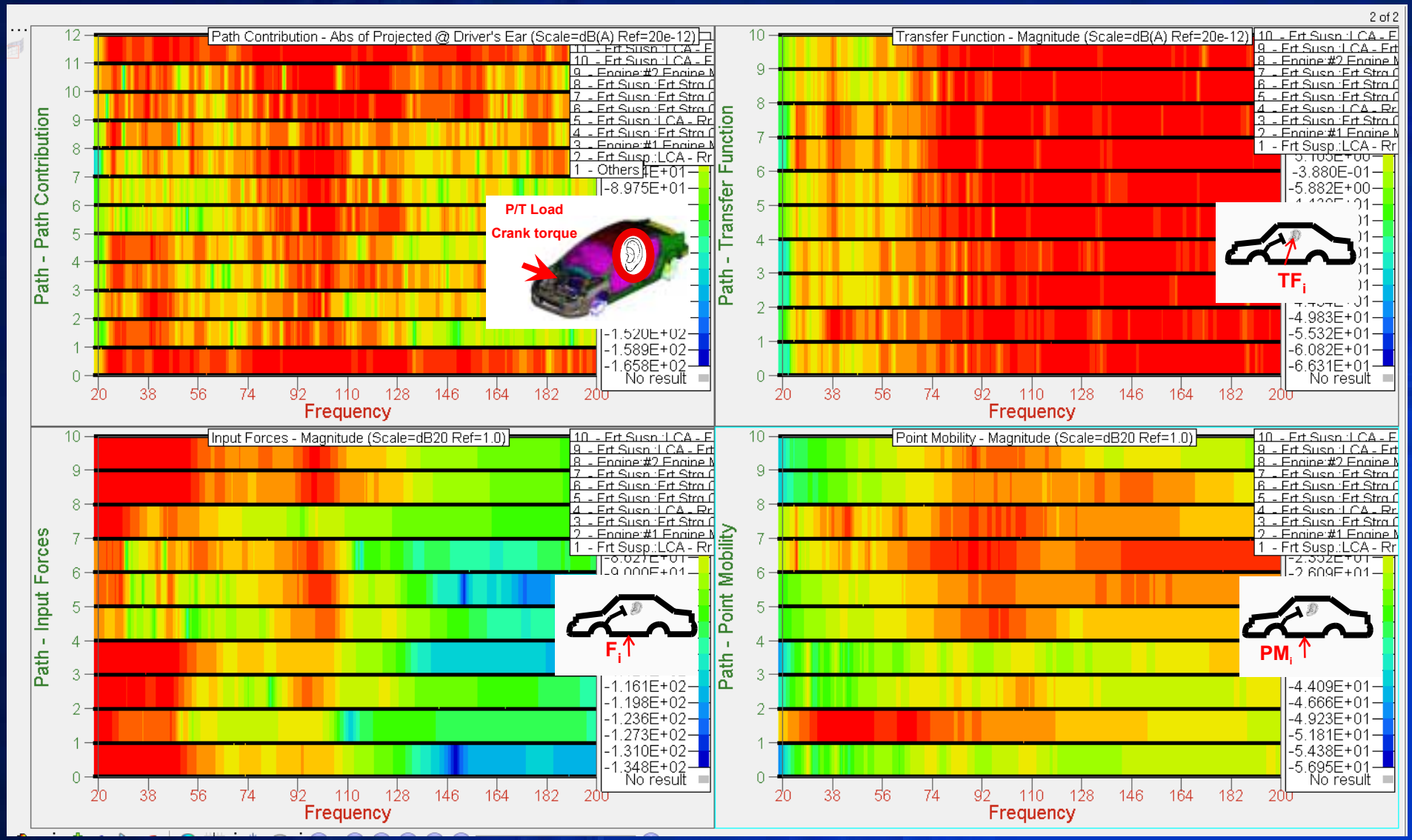
Attenuating Key NVH Paths

TPA Example: Dominant Paths over Frequencies



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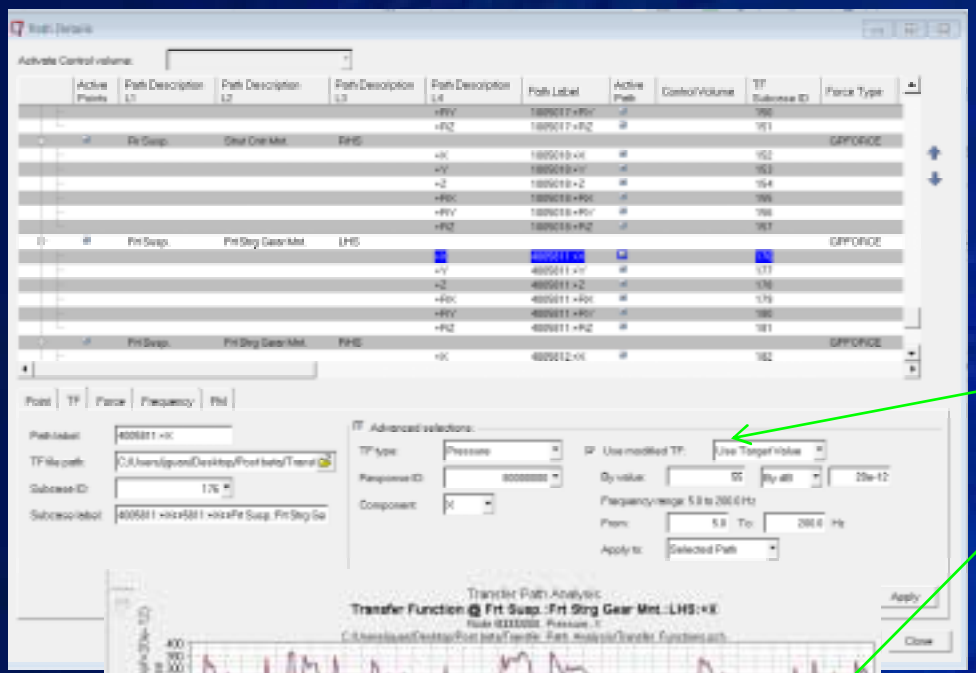
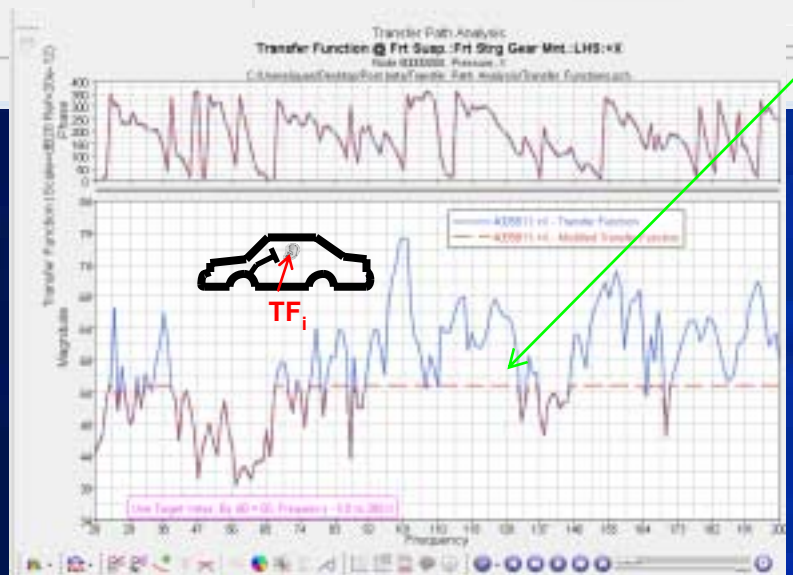
Designing Noise Paths

TPA Example: Cascading Vehicle Targets to Subsystems



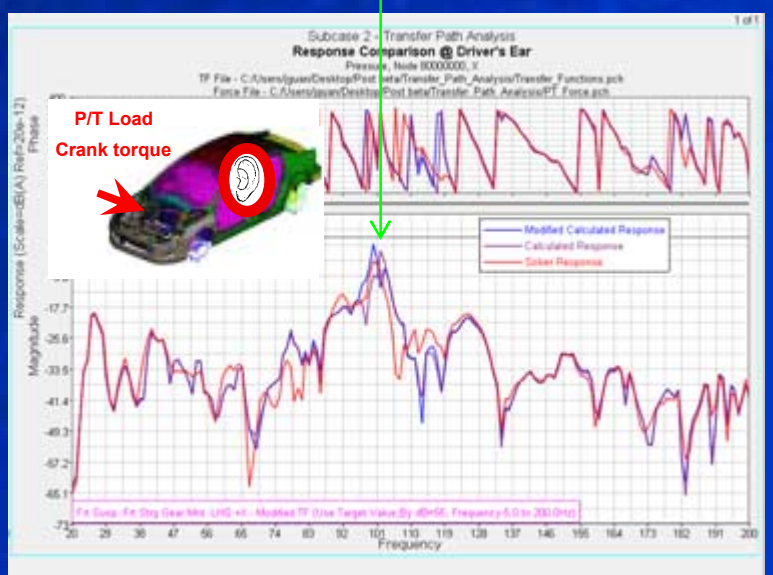


Once the dominant noise paths and root cause have been identified, the task is reduced to solving problems of:

1. **High Force**
2. **High Transfer Function**
3. **High Point Mobility**

Limit TF to a 55 dB target

See changes in response



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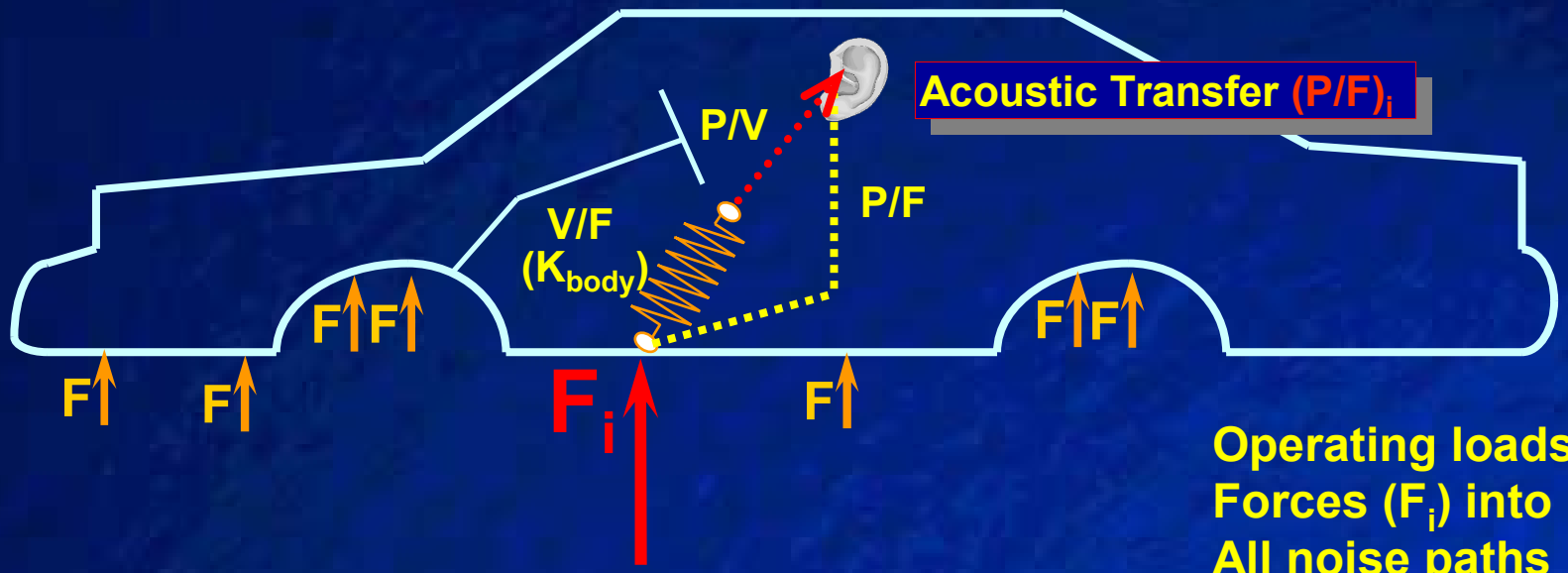
Designing Noise Paths



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

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

Measurement Parameters		*Generic Targets*
P/F	Acoustic Sensitivity	50 - 60 dBL/N
V/F	Structural Point Mobility (Receiver Side)	0.2 to 0.3 mm/sec/N

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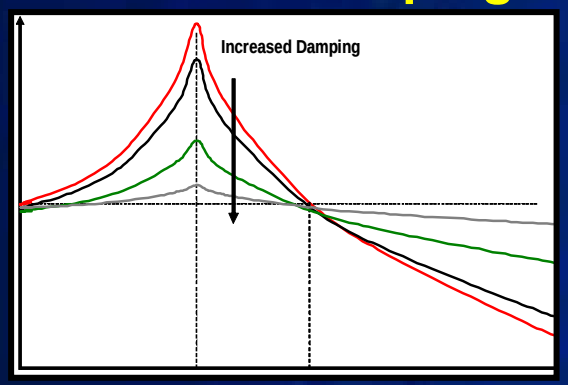
“Downstream” Effects: Body Panels

Recall for Acoustic Response P_t

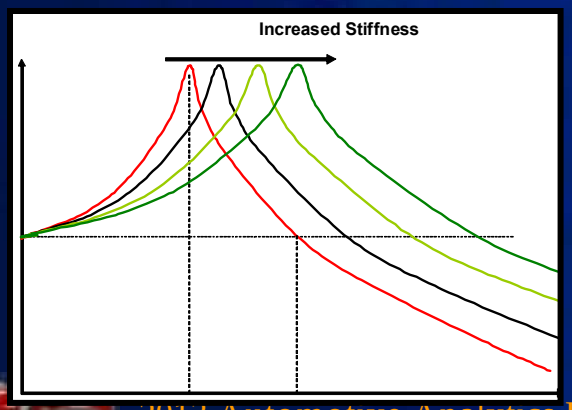
$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [F_i * (P/V)_i * (V/F)_i]$$

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

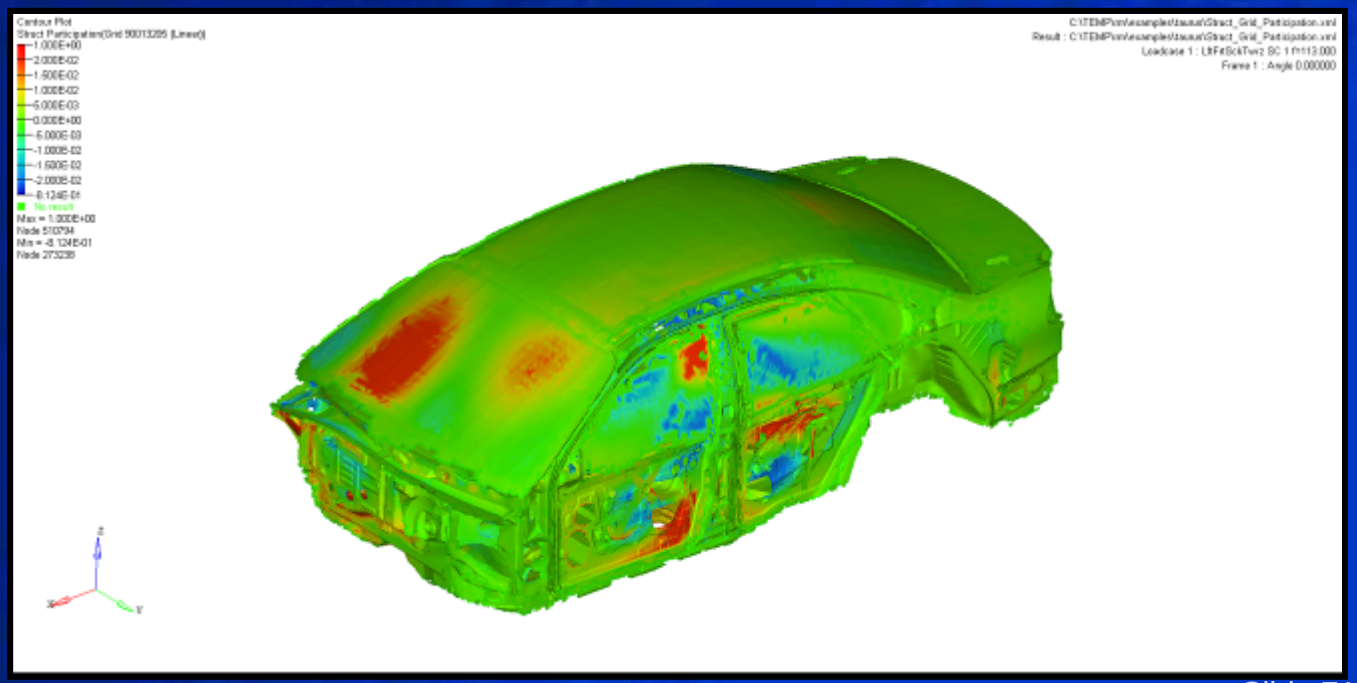
1. Panel Damping



2. Panel Stiffness



3. Panel Acoustic Contribution



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Generic Noise Path Targets

Primary: Minimize the Source Force

< 1.0 N

$$\frac{K_{body}}{K_{iso}} > 5.0$$

$$\frac{K_{source}}{K_{iso}} > 20.0$$

Structural Mobility < 0.2 to 0.3 mm/sec/N

Acoustic Sensitivity < 50 - 60 dBL/N

Panel Damping Loss Factor

> .10

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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

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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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Structure Borne NVH Workshop

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

Greg Goetchius

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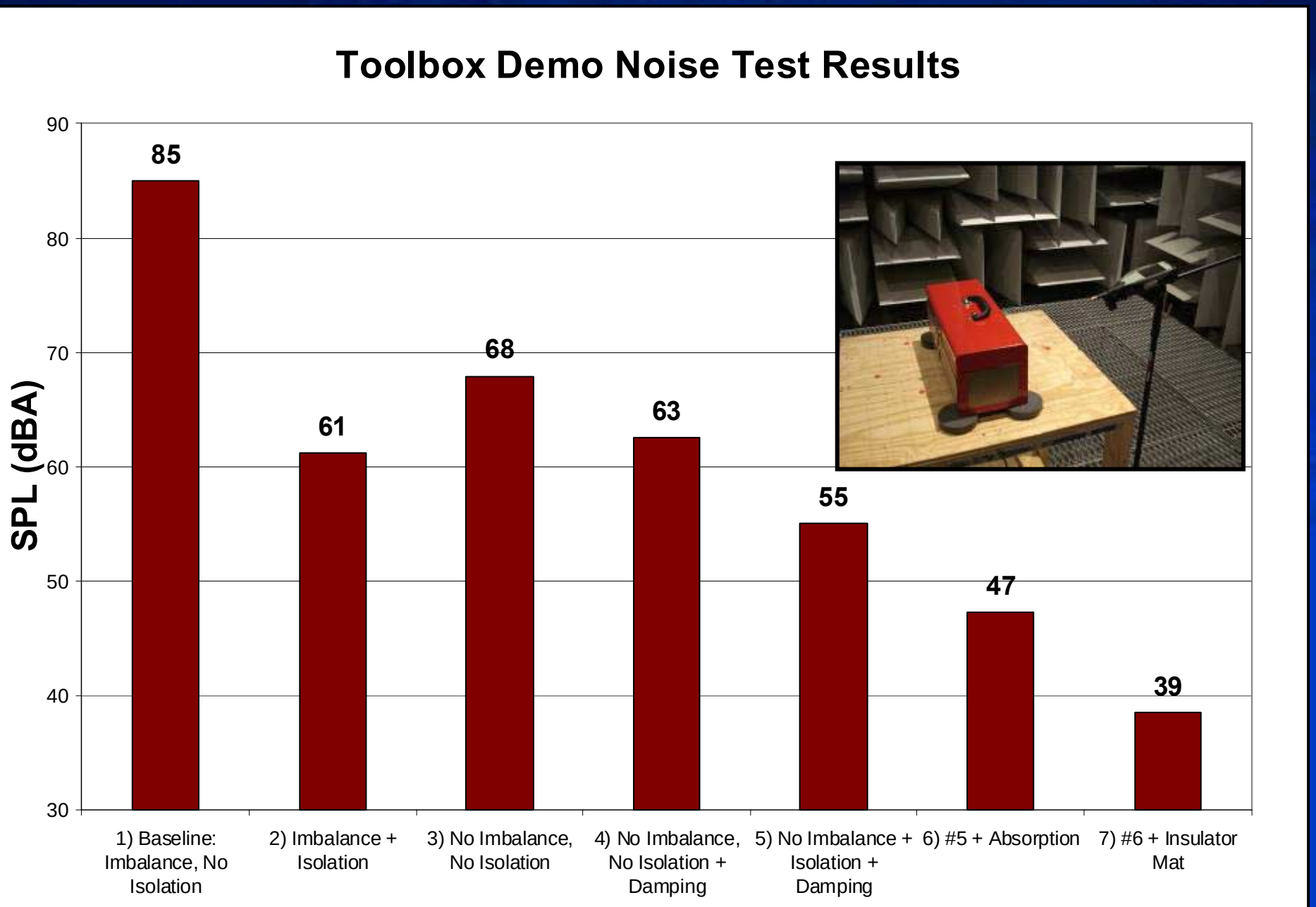
Tool Box Demo Test Results



Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- **New Technology**

  NVH Scatter and Uncertainty

  Alan Duncan

  Closing Remarks

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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**

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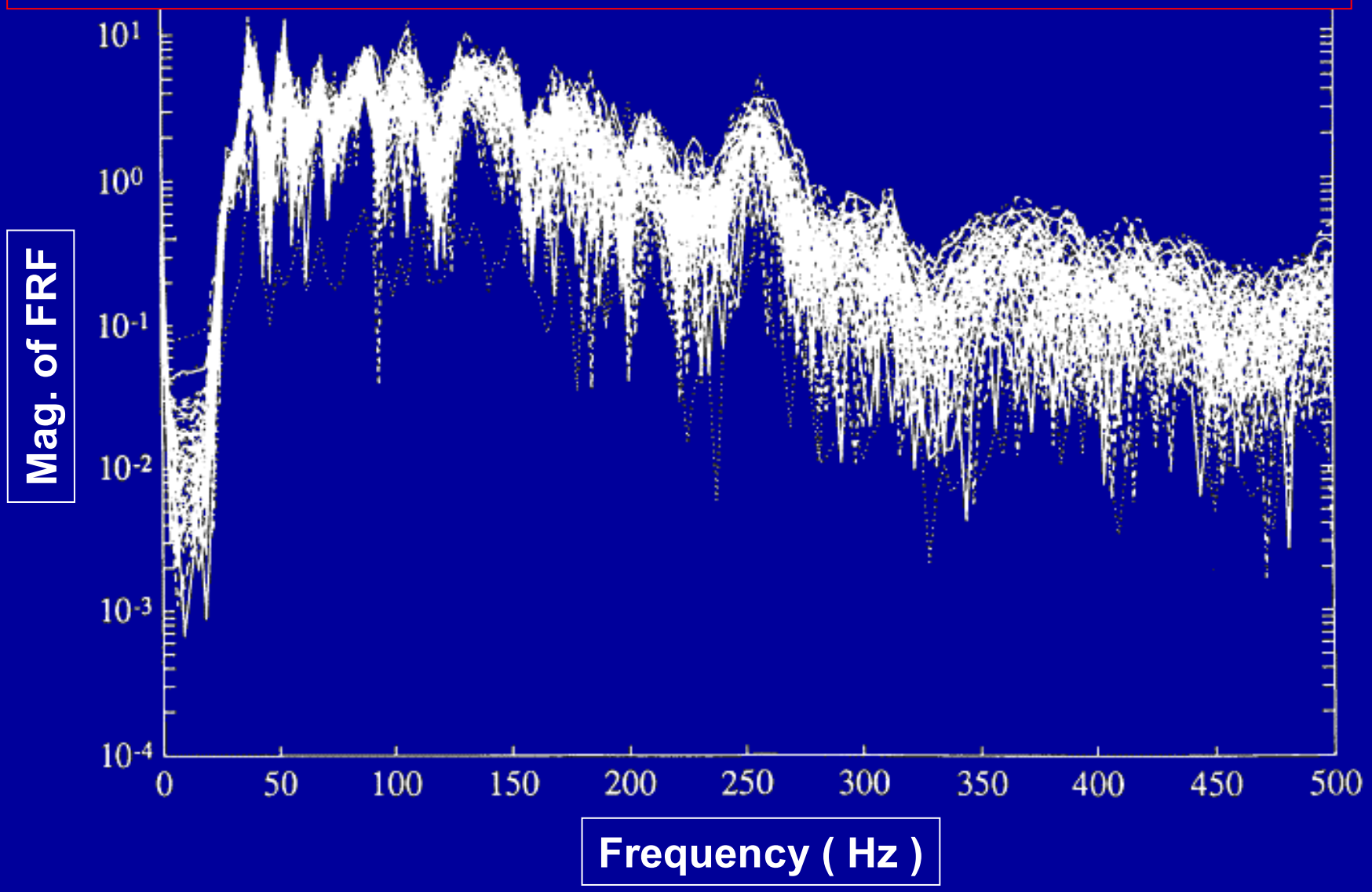
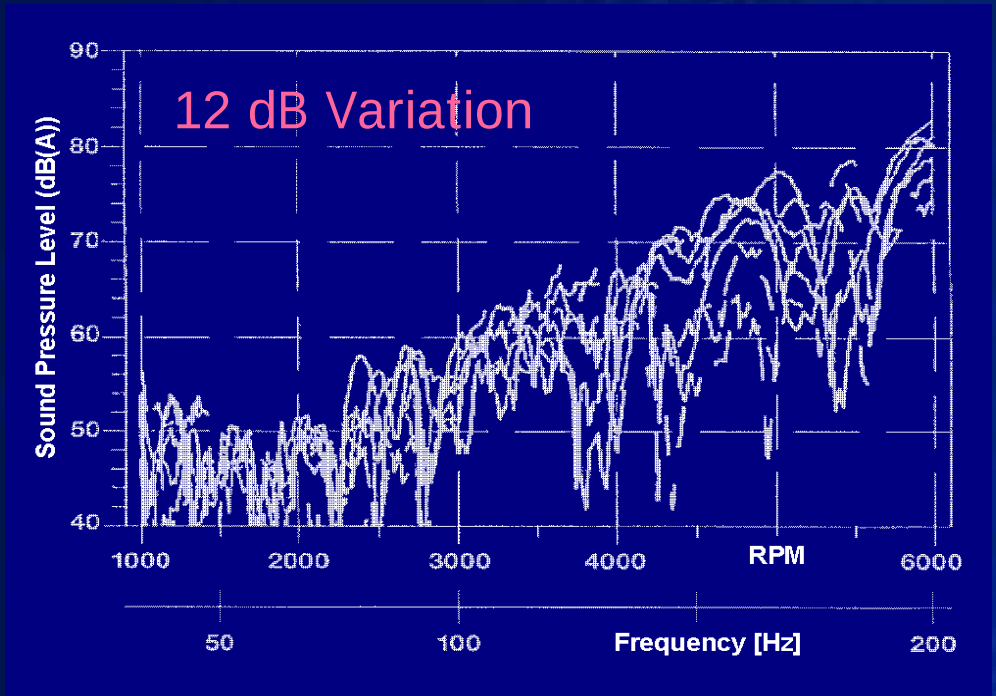
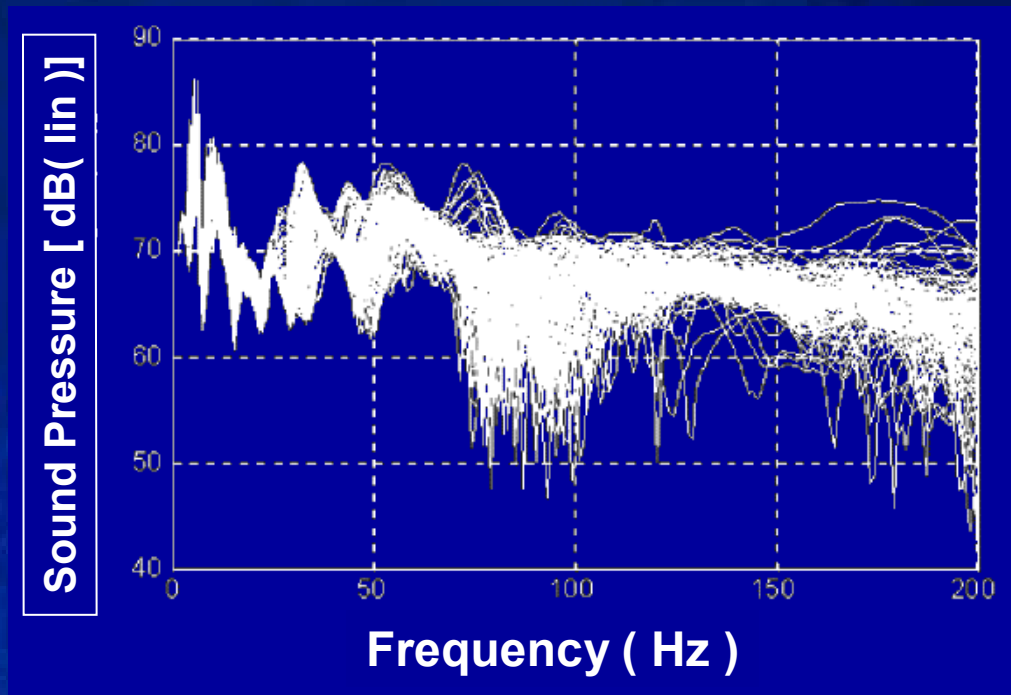
Magnitude of 99 Structure Borne Noise Transfer Functions for Rodeo's at the Driver Microphone
Measurements from Kompella and Bernhard
(Ref. 8) ©1993 Society of Automotive Engineers, Inc.



Freymann, BMW NVH Scatter Results



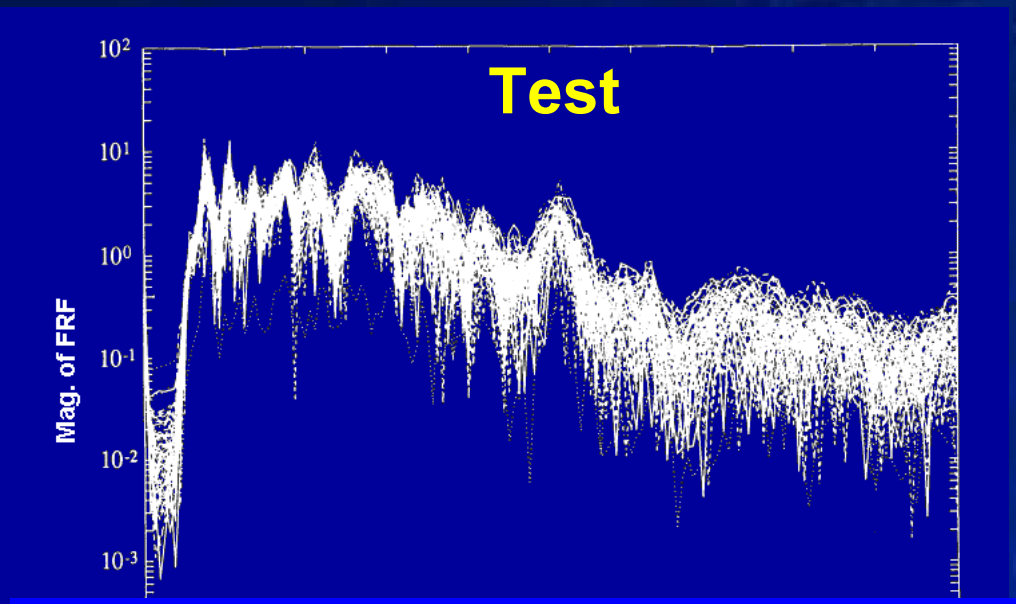


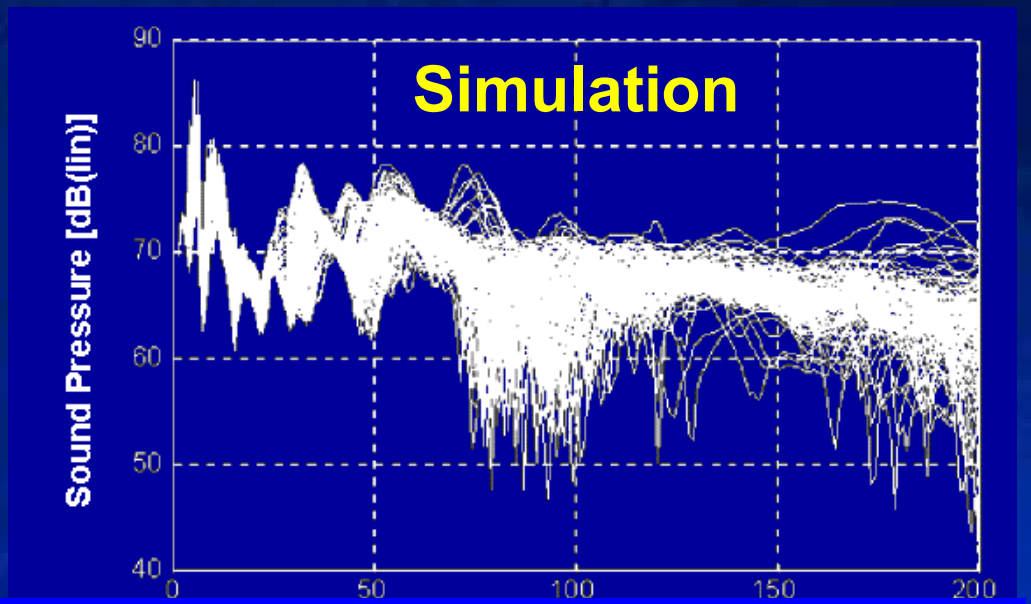
Experimentally detected Scatter in low frequency vibroacoustic behavior of production vehicles.

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

© 2000 Society of Automotive Engineers, Inc.
Reproduced with permission from paper by Freymann, et. Al. (Ref. 9)

Scatter Implications for Test (or Simulation) NVH Development





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?

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**

Scatter is the "Physics"

NVH Scatter and Uncertainty

- Scatter 20 Years Ago
- **Scatter 10 Years Ago**
- New Technology to Address Scatter

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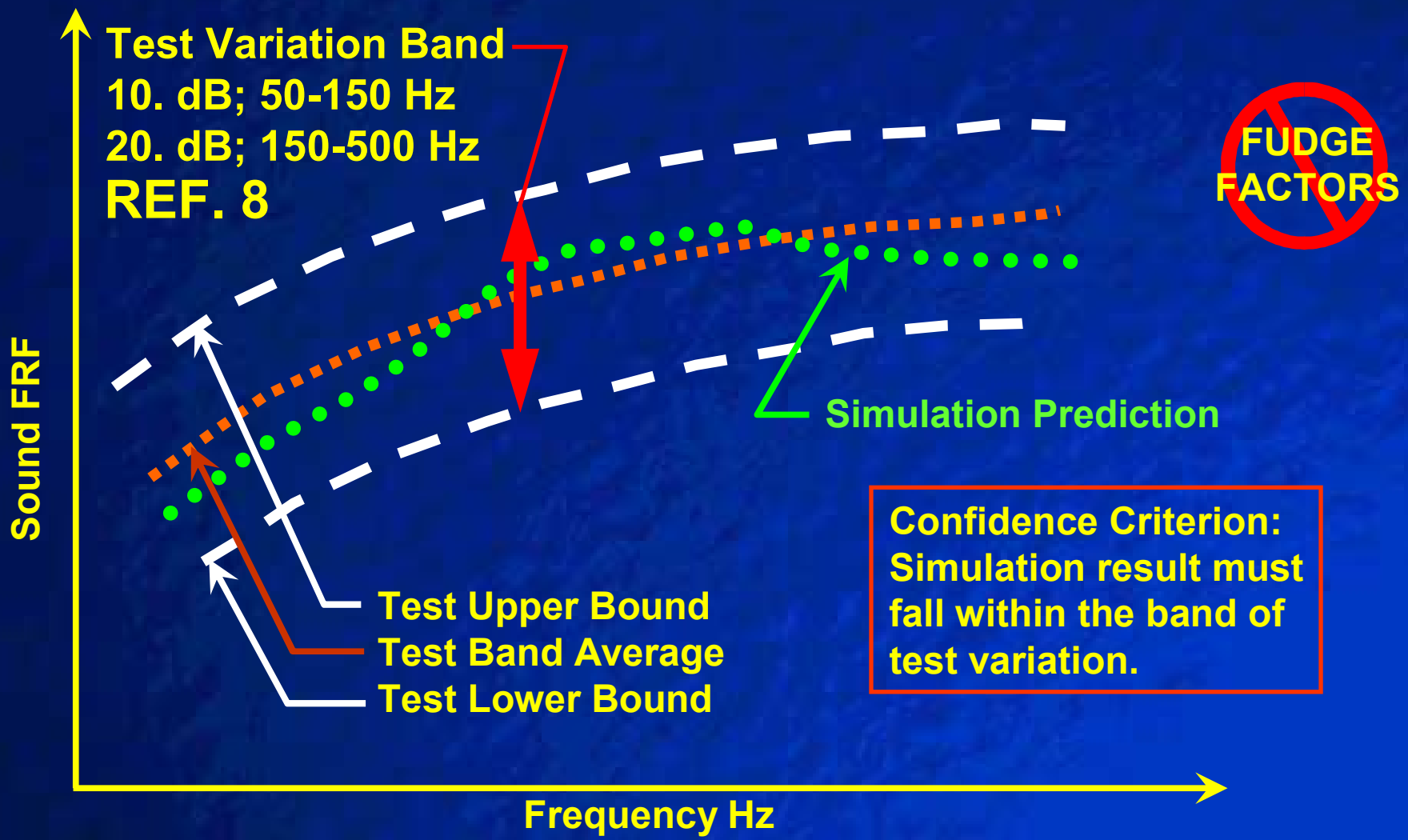
Reference Baseline Confidence Criterion

For *Operating* Response Simulations



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

2003 Workshop Confidence Criterion Lecture with Voice Track available at www.AutoAnalytics.com on DOWNLOAD page.

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FlashBack: 1995 Paper on Root Cause of Scatter

951302

Variation of Vehicle NVH Properties due to Component Eigenfrequency Shifting - Basic Limits of Predictability

Bo Gärdhagen and Juha Plunt
Chalmers University of Technology

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)

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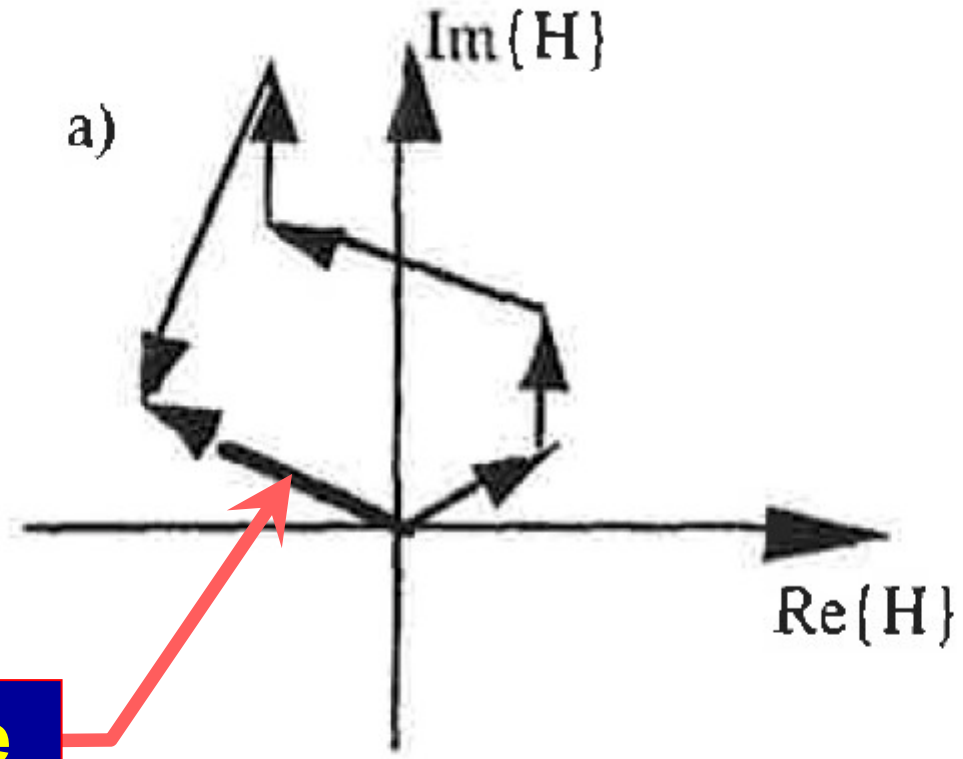
Root Cause of Scatter : Conditions



Total Response

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**

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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

2005-01-2385

Nonparametric Modeling of the Variability of Vehicle Vibroacoustic Behavior

Jean-François Durand and Laurent Gagliardini
PSA- Peugeot-Citroën- FRANCE

Christian Soize
University of Marne-La-Vallée- FRANCE

ABSTRACT

In order to improve the robustness of vibroacoustic numerical predictions, one introduces a model of random uncertainties. The random uncertainty modelling relies on a nonparametric approach providing random system realizations with a maximum entropy. This approach only requires a few uncertainty parameters but takes into account data errors as well as model errors. It appears to be well adapted to study the variability of structural-acoustic systems; the implementation of the method for this class of problem is presented here for the first time. Practically, the paper deals with a classical low frequency vibroacoustic modelling such as used for booming noise predictions. The application of the nonparametric approach to vehicle uncertainties modelling shows the sensitivity of the vibroacoustic frequency responses to structural and cavity uncertainties as well as coupling interface uncertainties. Flexible parts appear to be more sensitive to random uncertainties than stiff parts. The sensitivity of the structural modes to structural random uncertainties is also shown in a stochastic MAC table.

Uncertainties can be classified into two classes:

(i) Data uncertainties concern the parameters of the mathematical-mechanical model such as geometrical parameters, boundary conditions, material behaviour etc. Data uncertainties can be described by a parametric probabilistic approach. Nevertheless, in the case of vehicles, such an approach requires a considerable amount of statistical data which makes it inappropriate to an industrial design process.

(ii) Model uncertainties are introduced during the construction of the mathematical-mechanical model: the constructed model cannot exactly represent a structure as complex as a car body due to the introduction of approximations and simplifications which make the modelling practicable. This class of uncertainties is not relevant to the parametric approach because, by definition, the model uncertainties cannot be taken into account by the parameters of the mathematical-mechanical model under consideration. It has been seen many times that small design changes -that may be considered as model uncertainties- lead to much higher

(Ref. 12)

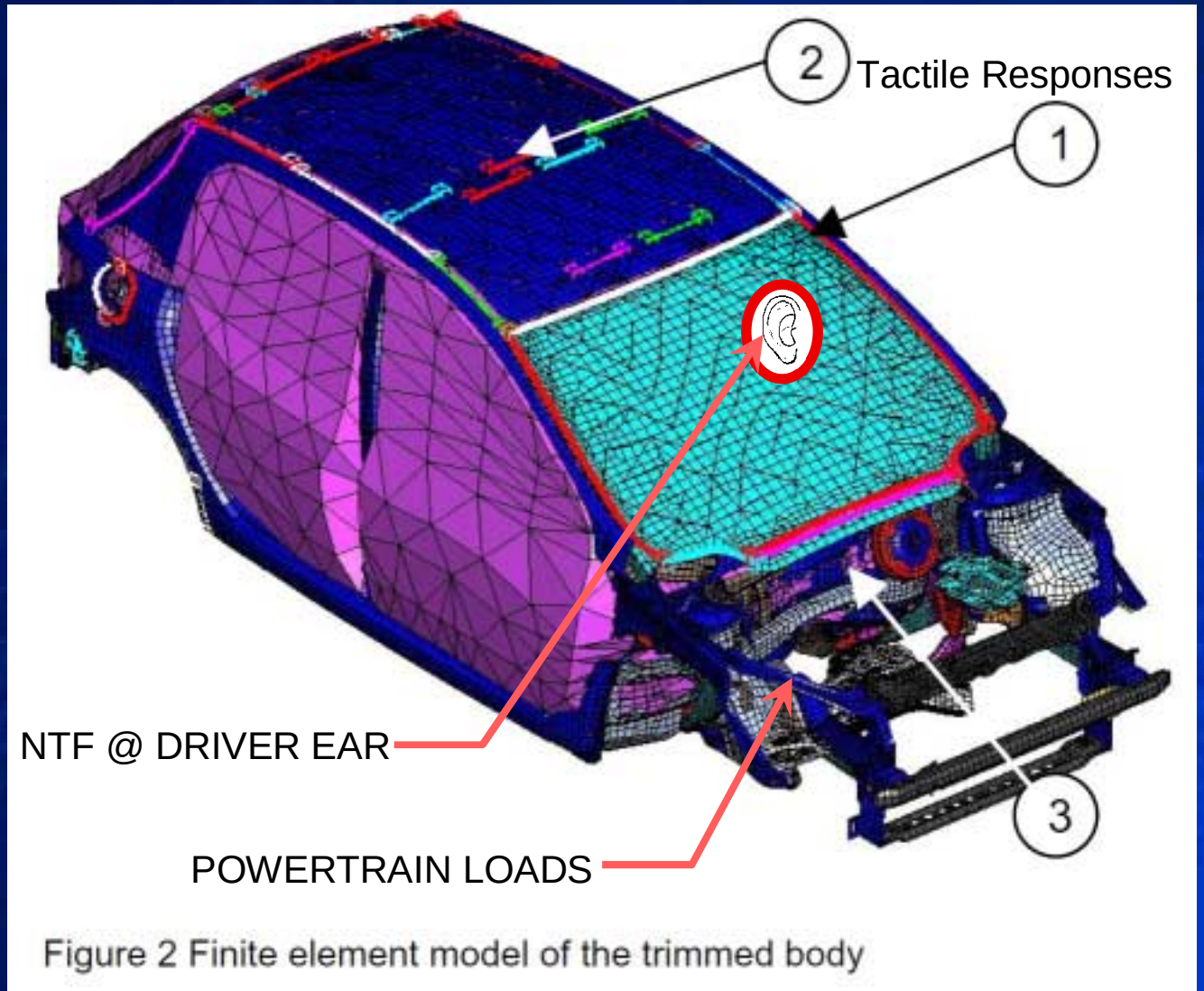

Figure 2 Finite element model of the trimmed body

(See Ref. 11 for Model Details)

Standard Eqn. for Structural-Acoustic Coupling Modal Model

Typical Derivation for creating the randomized Dynamic Matrix with Gaussian Distribution. Non-Parametric: Direct Changes in Modal Matrices.

The scatter created is a function of: Dispersion Parameter: δ Once determined, δ is a constant controlling the amplitude level of scatter.

7 Dispersion Constants are needed:

3 for Structure: M, D, K

3 for Acoustics: M, D, K

1 for Coupling: $C_{n,m}$

$$\begin{bmatrix} [\mathbf{A}^s_n(\omega)] & [\mathbf{C}_{n,m}] \\ \omega^2[\mathbf{C}_{n,m}]^T & [\mathbf{A}^a_m(\omega)] \end{bmatrix} \begin{bmatrix} \boldsymbol{Q}^s(\omega) \\ \boldsymbol{Q}^a(\omega) \end{bmatrix} = \begin{bmatrix} \underline{\mathcal{F}}^s(\omega) \\ \underline{\mathcal{F}}^a(\omega) \end{bmatrix} , \qquad (12)$$

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

Let $\boldsymbol{H}$ be the positive-definite symmetric $(\nu \times \nu)$ random matrix representing $[\boldsymbol{G}_{M^s_n}]$, $[\boldsymbol{G}_{D^s_n}]$, $[\boldsymbol{G}_{K^s_n}]$, $[\boldsymbol{G}_{M^a_m}]$, $[\boldsymbol{G}_{D^a_m}]$, $[\boldsymbol{G}_{K^a_m}]$ or $[\boldsymbol{G}_{C_{n,m}}]$. The following algebraic representation of random matrix $[\boldsymbol{G}_H]$ allows independent realizations of $[\boldsymbol{G}_H]$ to be constructed. Random matrix $[\boldsymbol{G}_H]$ is written $[\boldsymbol{G}_H] = [\boldsymbol{L}_H]^T[\boldsymbol{L}_H]$ in which $[\boldsymbol{L}_H]$ 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 $[\boldsymbol{L}_H]_{jj'}$ is written as $[\boldsymbol{L}_H]_{jj'} = \sigma_\nu U_{jj'}$ in which $\sigma_\nu = \delta_H\,(\nu+1)^{-1/2}$ and where $U_{jj'}$ is a real-valued Gaussian random variable with zero mean and variance equal to 1. The parameter δ_H controlling the dispersion level of random matrix $[\boldsymbol{H}]$ is such that $\delta_H = \sqrt{E\{\|[\boldsymbol{G}_H] - [I]\|_F^2\}/\nu}$ 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 $[\boldsymbol{L}_H]_{jj'}$ is written as $[\boldsymbol{L}_H]_{jj'} = \sigma_\nu\sqrt{2V_j}$ in which σ_ν 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

$$p_{V_j}(v) = 1_{\mathbb{R}^+}(v)\,\frac{1}{\Gamma(\frac{\nu+1}{2\delta_H^2} + \frac{1-j}{2})}\,v^{(\frac{\nu+1}{2\delta_H^2} - \frac{1-j}{2})}\,e^{-v} \quad .$$

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Proof of Process Convergence

CONCLUSIONS: Converged at 200 REALIZATIONS and with Struct to 400 and Fluid to 350 Hz

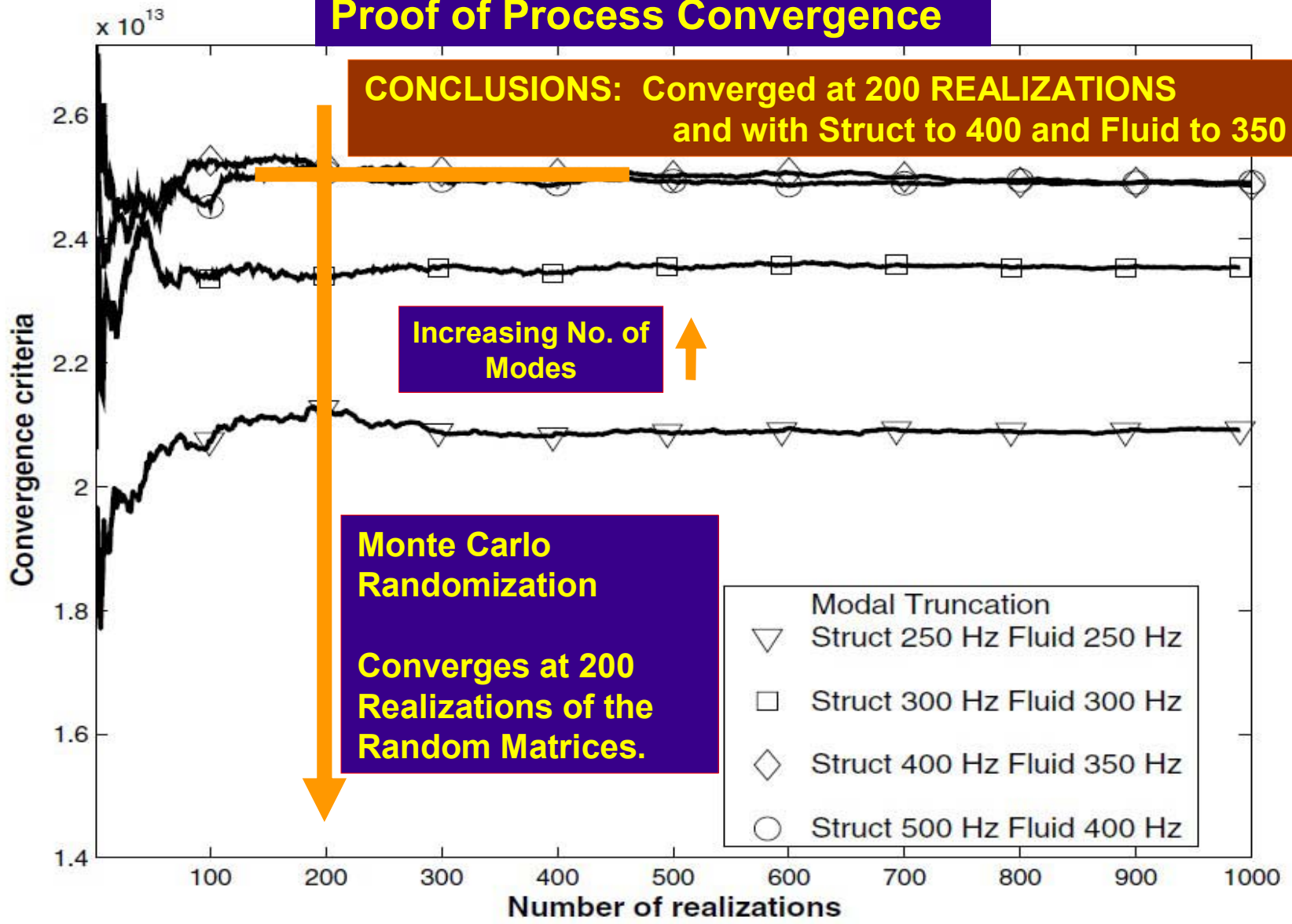

Figure 4 Graph of the function $n_r \rightarrow conv^F(nr)$ related to the convergence of the random acoustic generalized coordinates.

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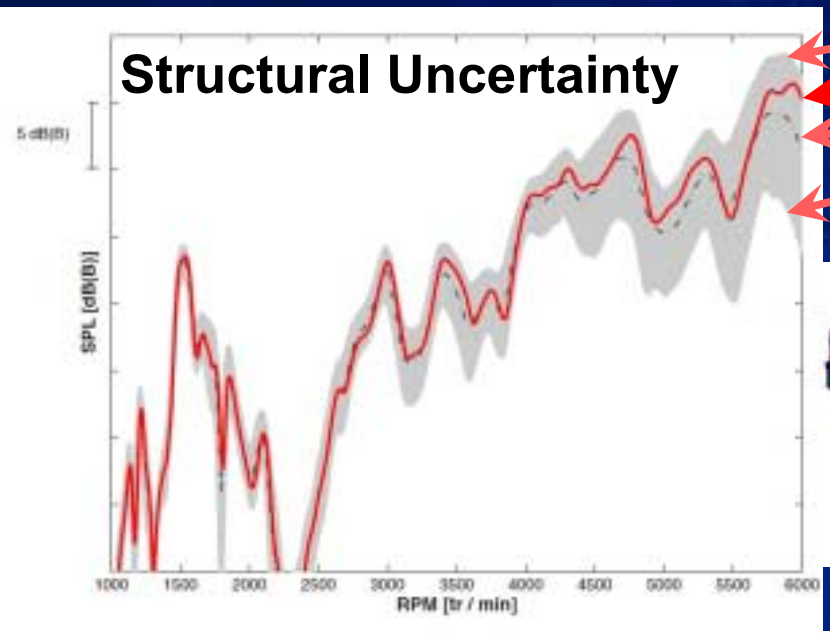

95% Upper Confidence Band
Nominal / Original Model Response
50% Confidence Band (Mean Stochastic Model)
95% Lower Confidence Band

Figure 8 Confidence region of the random frequency response for the internal acoustic pressure at a point located at the driver's ear for structural uncertainties. Mean model prediction (thick solid line), mean response of the stochastic model (dashed line), confidence region (grey region).

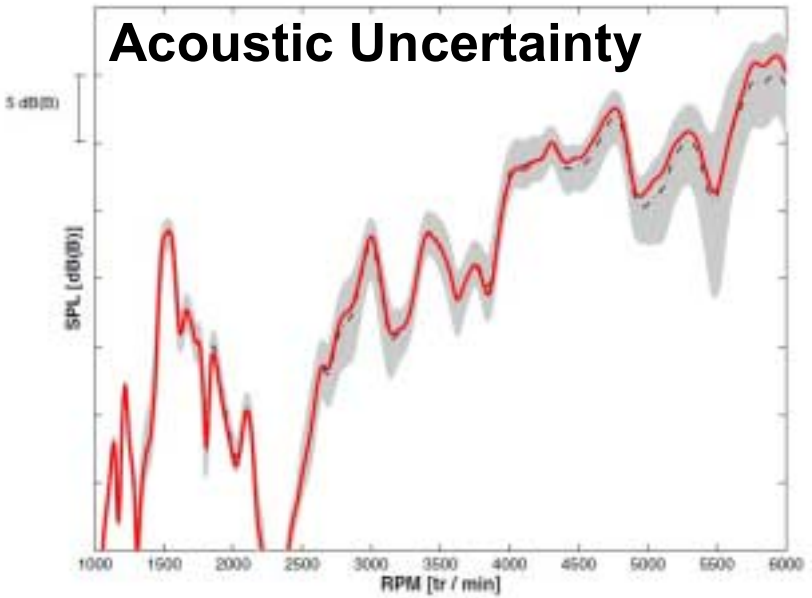

Figure 13 Confidence region of the random frequency response for the internal acoustic pressure at a point located at the driver's ear for acoustic cavity uncertainties. Mean model prediction (thick solid line), mean response of the stochastic model (dashed line), confidence region (grey region).

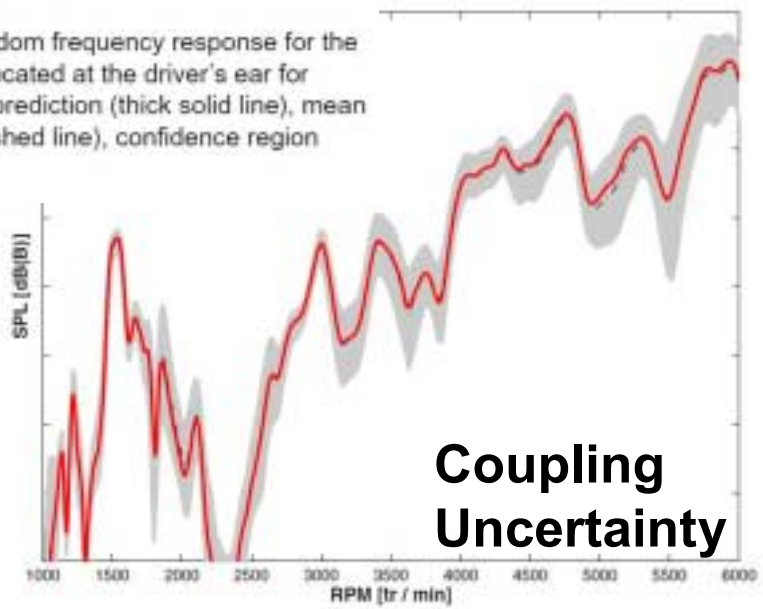

Figure 11 Confidence region of the random frequency response for the internal acoustic pressure at a point located at the driver's ear for vibroacoustic coupling uncertainties. Mean model prediction (thick solid line), mean response of the stochastic model (dashed line), confidence region (grey region).

CONCLUSION:
• The Model shows 95% Confidence Bands with increasing Scatter at higher frequency similar to K-B Study.

Main Paper - Development is Extensive Includes Test Data Comparison

Structural-acoustic modeling of automotive vehicles in presence of uncertainties and experimental identification and validation

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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 Bernhard, 1996). Nevertheless, these two experimental databases cannot easily be used because there are no available compu-

(Ref. 13)

Full Vehicle System – Body and Chassis Structural-Acoustic Model Similar Detail Level as first paper

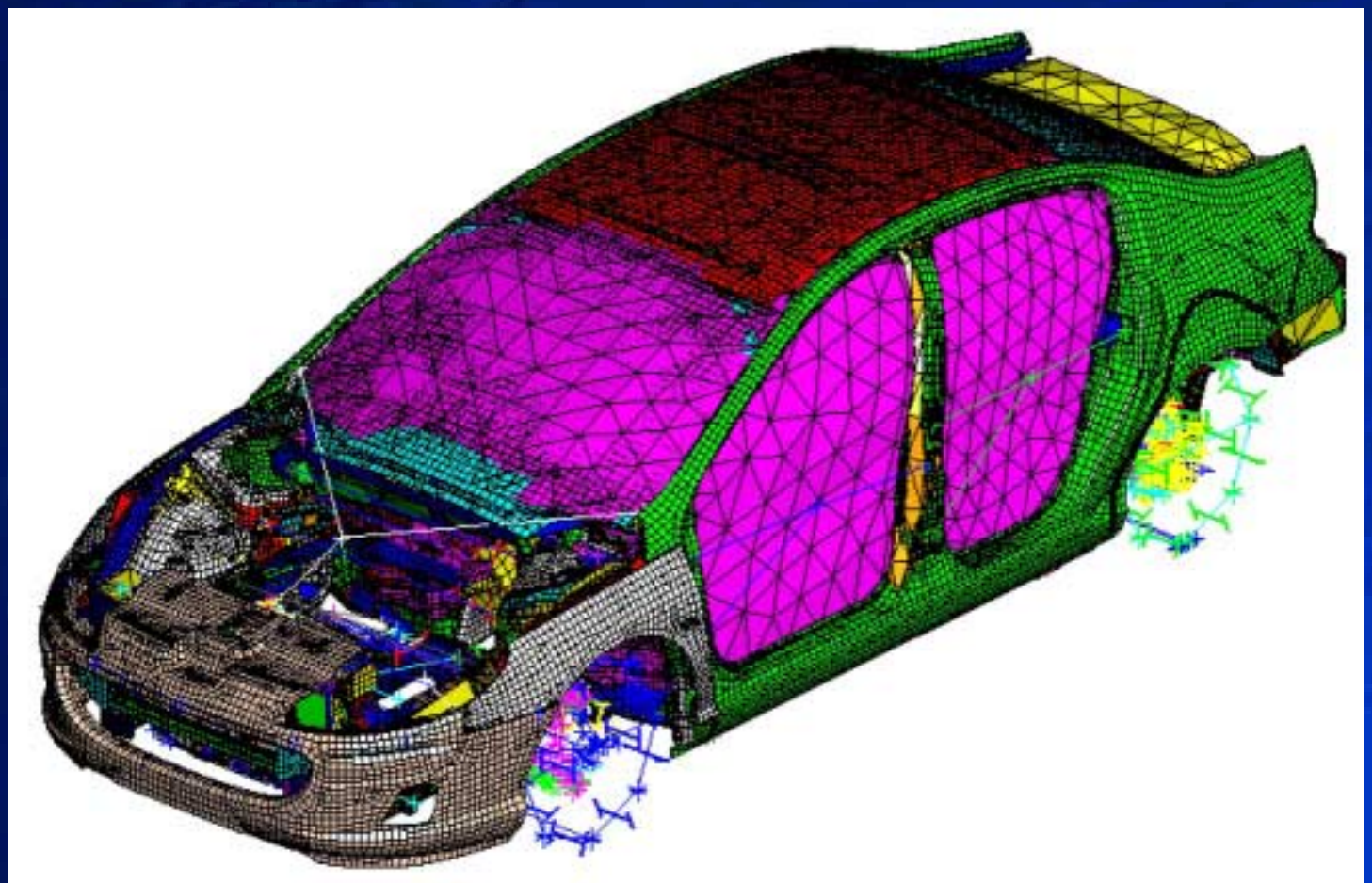

FIG. 10. Finite element mesh of the computational structural acoustic model.

(See Ref. 11 for Model Details)

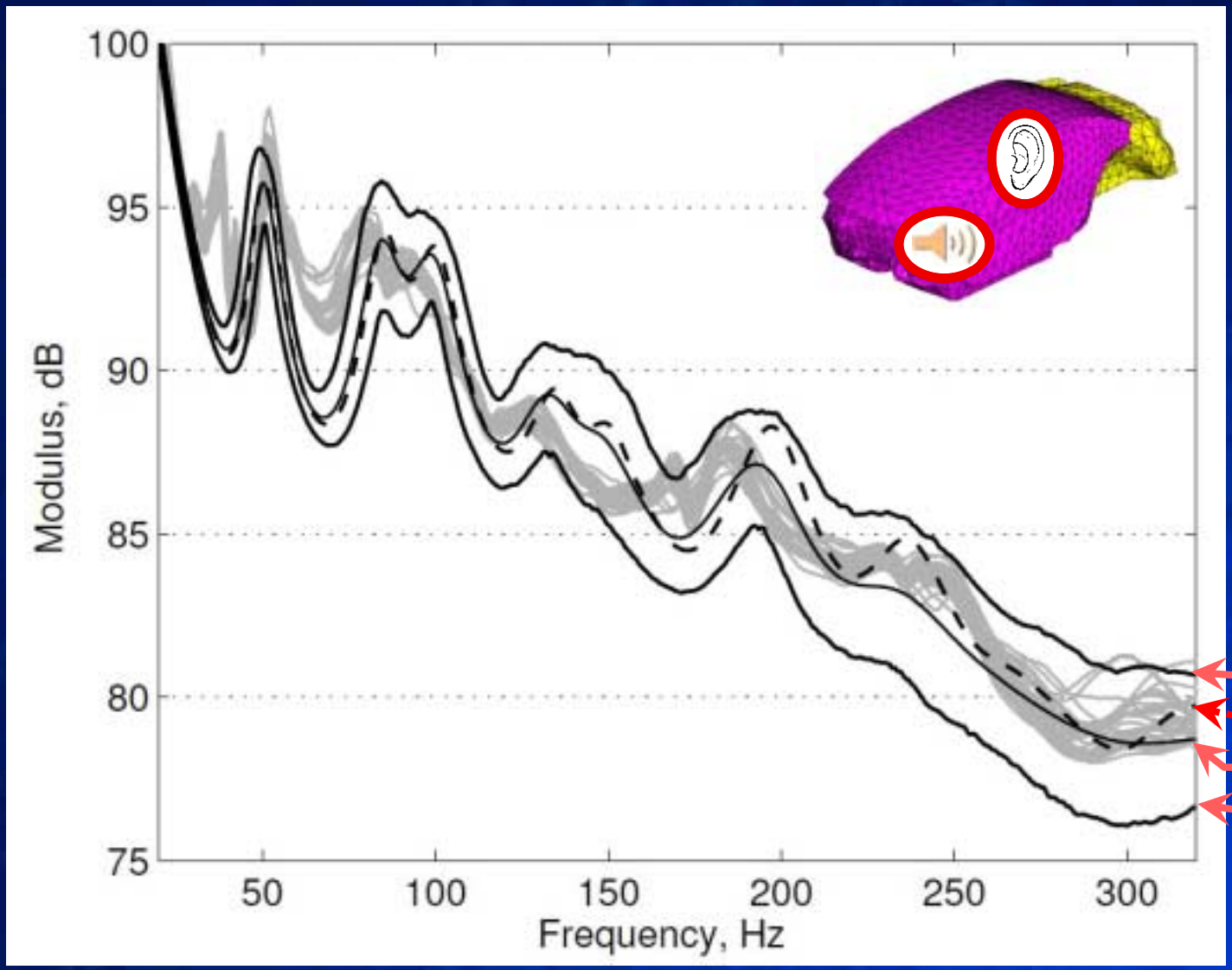

FIG. 16. Comparisons of the stochastic computational model results with the experiments. Graphs of the root mean square of the acoustic pressures averaged in the cavity in dB scale: experiments for the 30 configurations (gray lines); Mean computational model (dashed line); mean value of the random response (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.

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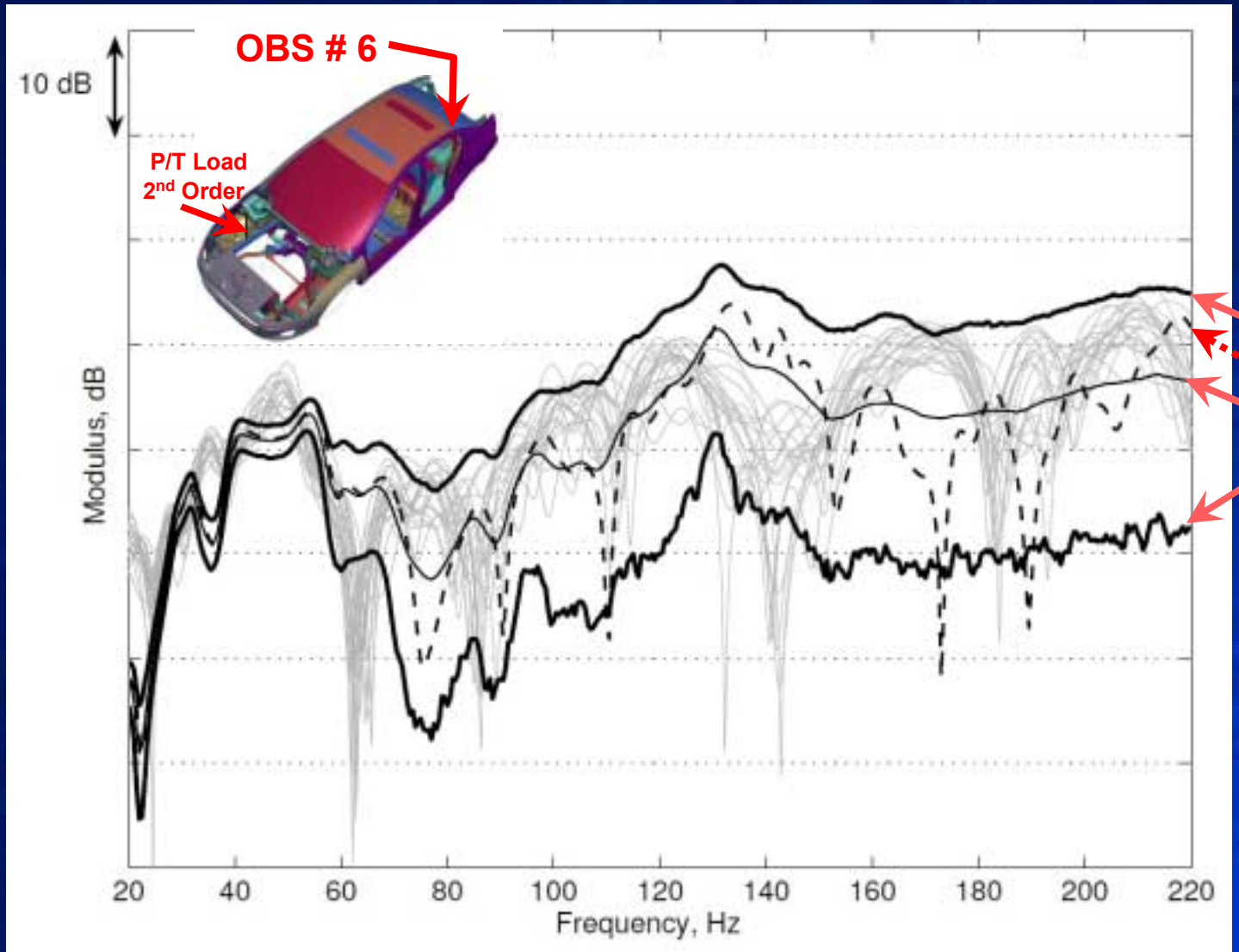

FIG. 19. Comparisons of the stochastic computational model results with the experiments for observation Obs6. Graphs of the moduli of the FRFs in dB scale: experiments for the 20 cars (gray lines); Mean computational model (dashed line); mean value of the random response (mid thin solid line); confidence 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.

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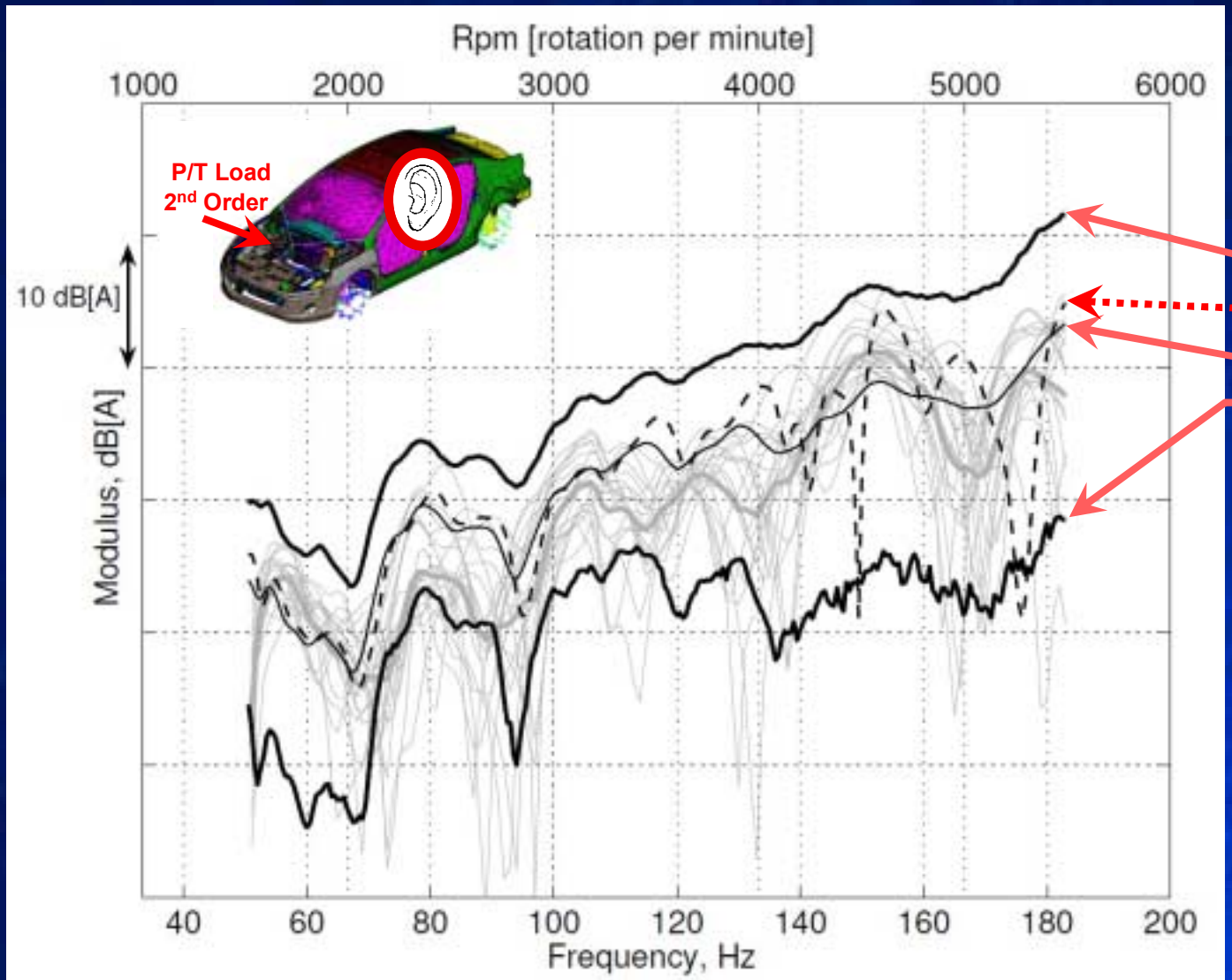

FIG. 20. Comparisons of the stochastic computational model results with the experiments for the booming noise. Graphs of the moduli of the FRFs in dB(A) scale: experiments for the 20 cars (thin gray lines); mean value of the experiments (thick gray line); Mean computational model (dashed line); Mean 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.

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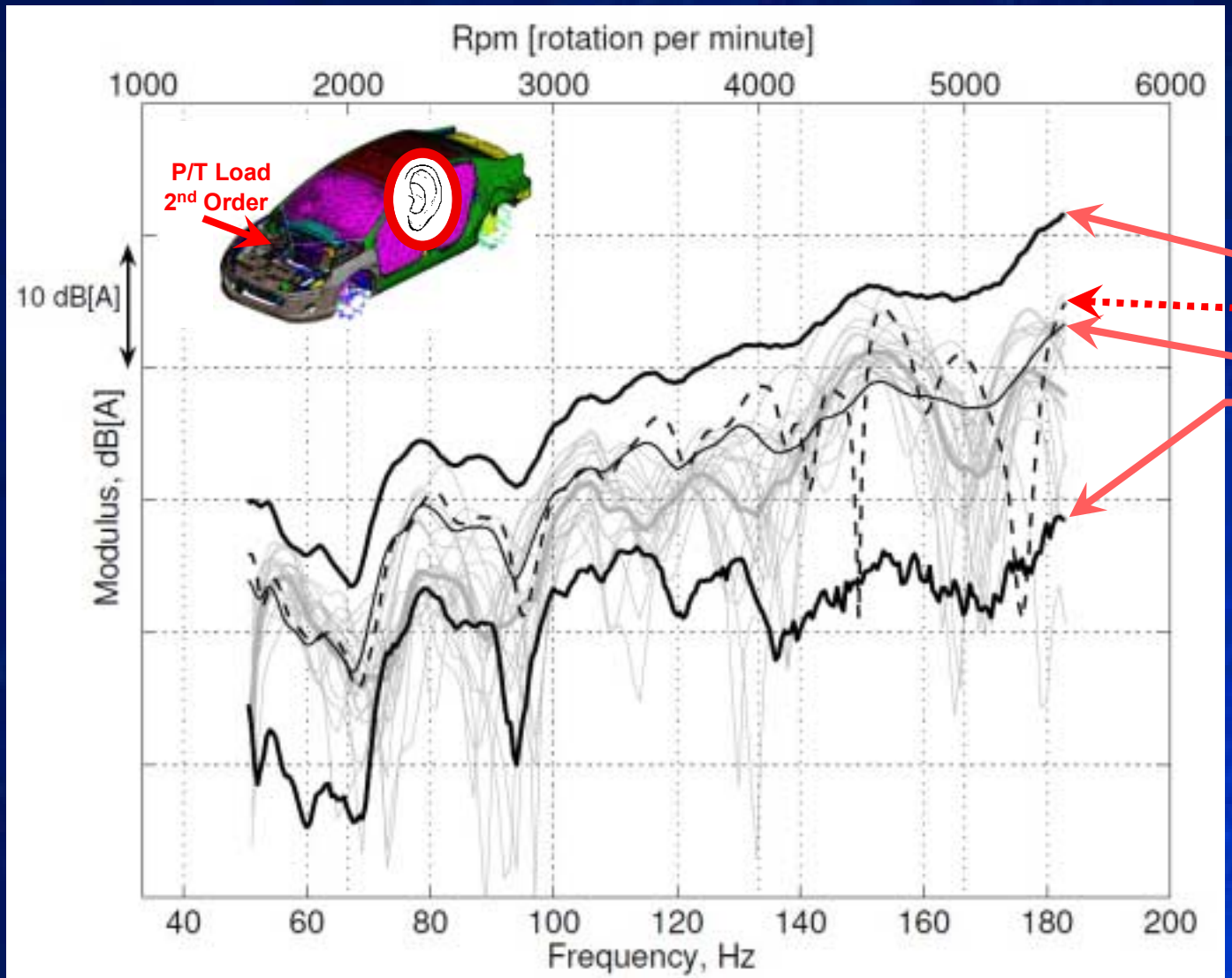

CONCLUSIONS:

. 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.

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AN INDUSTRIAL IMPLEMENTATION OF NON-PARAMETRIC STOCHASTIC MODELLING OF VEHICLE VIBROACOUSTIC RESPONSE

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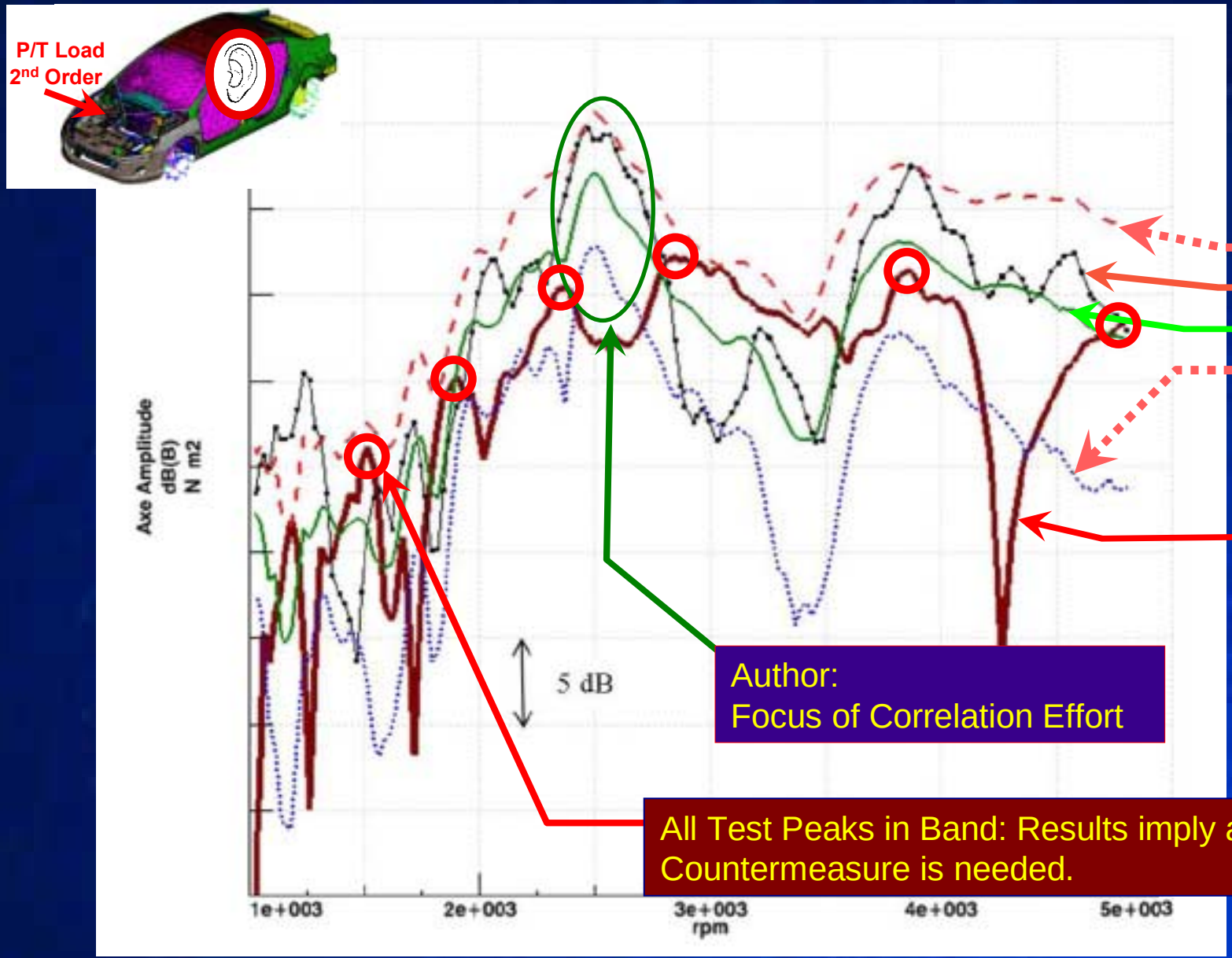

FIG. 4: Computed booming noise inside a car vs. rpm, using only structure-borne excitation compared with the measured value (brown bold). Thin dotted black: deterministic computation; green: median value; thin dotted blue: lower bound with a 95% probability; dotted red: upper bound with a 95% probability.

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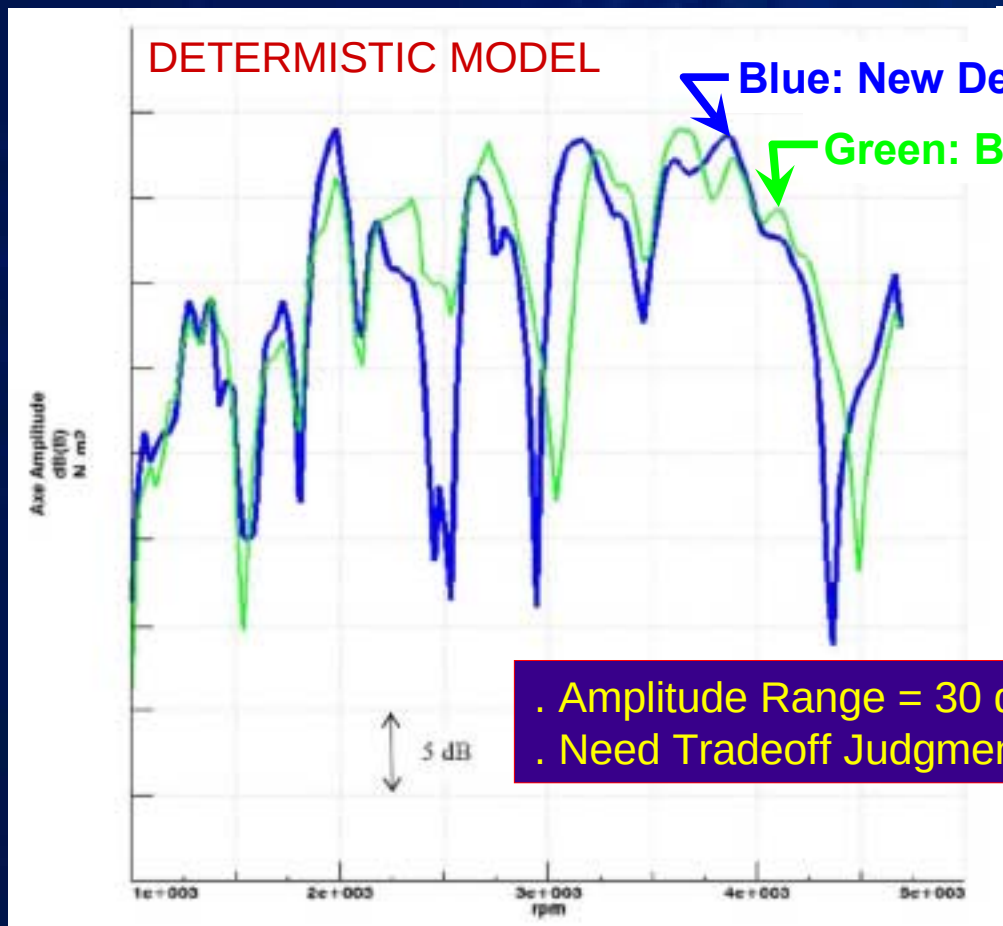
A-B Design Decision using Deterministic Model - vs - Mean Stochastic Model



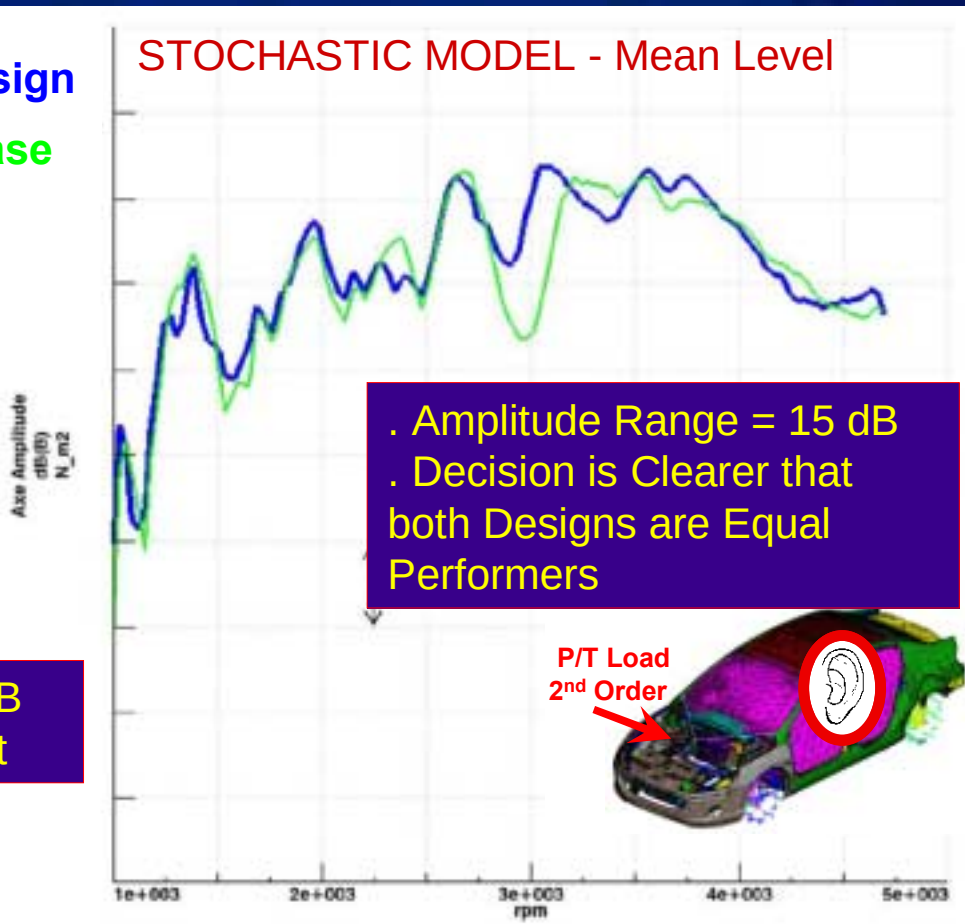


FIG. 6: Booming noise inside a car vs rpm. Deterministic models. Comparison of a baseline configuration (green) with a modification set (blue bold).

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)

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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.

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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

SAE 2013 NVH Conference
Structure Borne NVH Workshop

Thank You for Your Time!

Q & A

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Structure Borne NVH Workshop - on Internet

At SAE www.sae.org/events/nvc/specialevents.htm

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