SAE 2009 NVH Conference
Structure Borne NVH Workshop

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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 Real World Application Example.

Intended Audience –

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

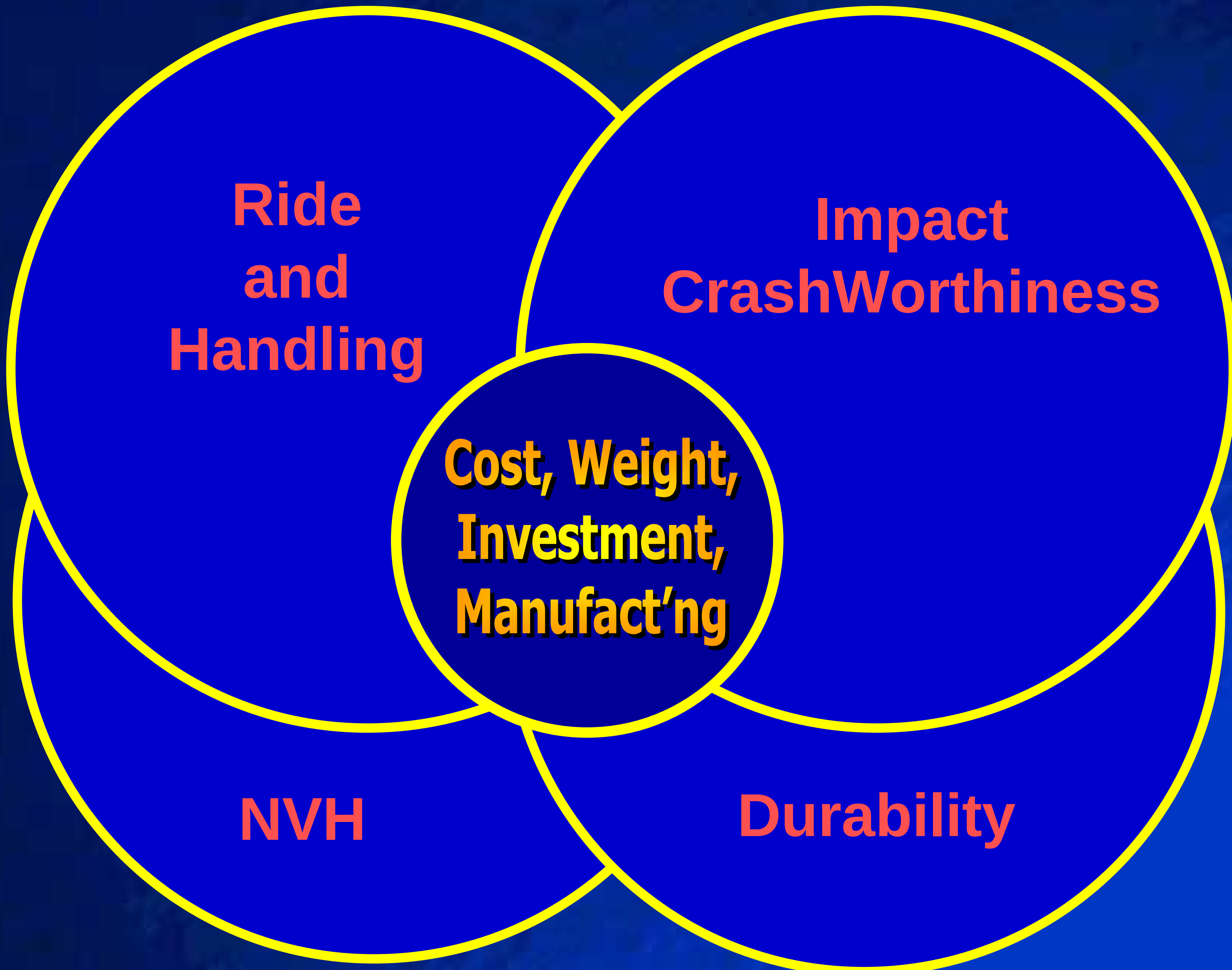
Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- ***Live Noise Attenuation Demo***
- Real World Application Example
- Closing Remarks

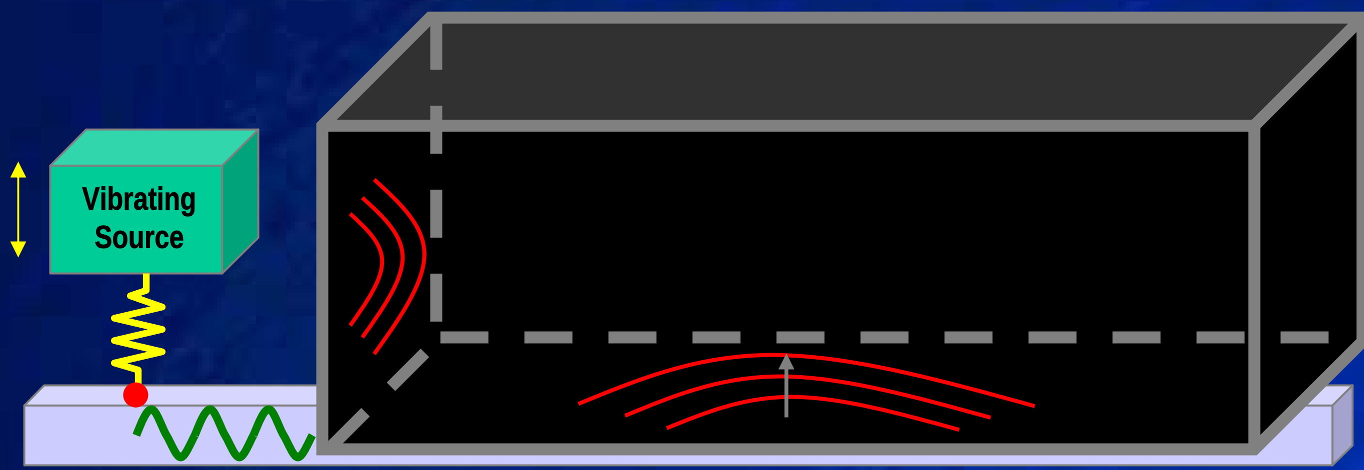
Structure Borne NVH Workshop

- **Introduction**
- **Low Frequency Basics**
- **Mid Frequency Basics**
- ***Live Noise Attenuation Demo***
- **Real World Application Example**
- **Closing Remarks**

Competing Vehicle Design Disciplines



Structure Borne Noise and Vibration

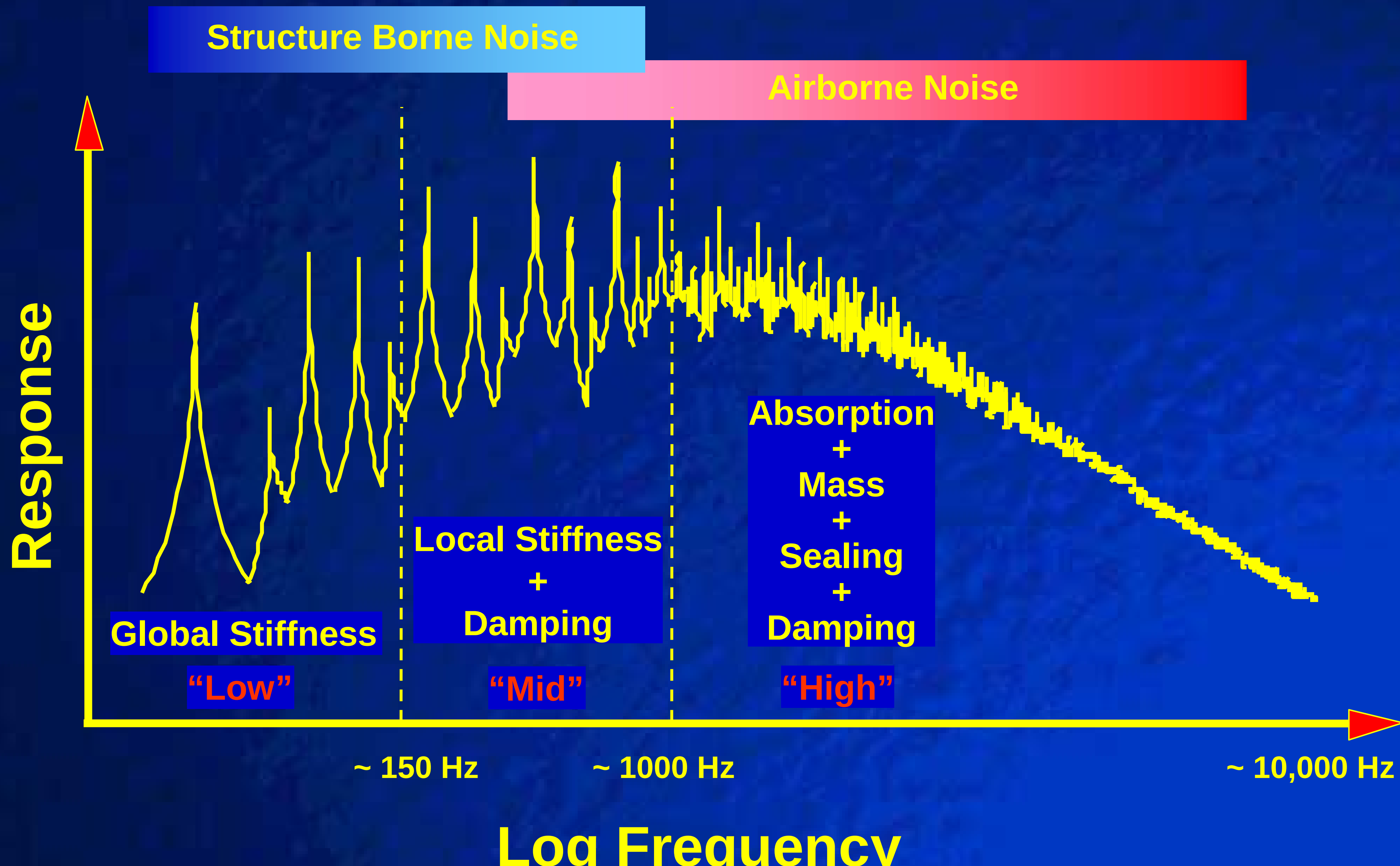


Frequency Range: up to 1000 Hz

System Characterization

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

Automotive NVH Frequency Range



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- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- Real World Application Example
- Closing Remarks

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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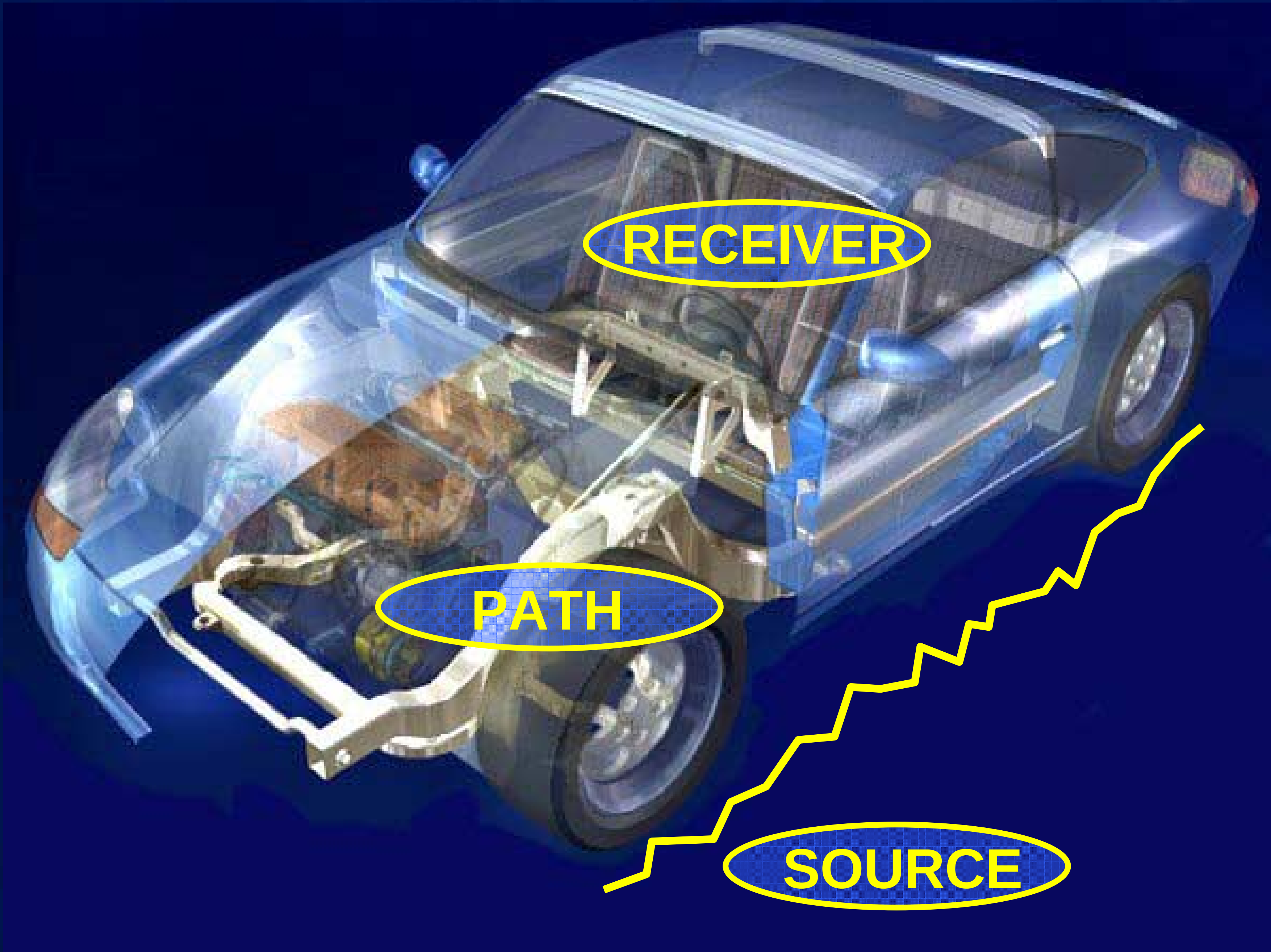
Structure Borne NVH Basics



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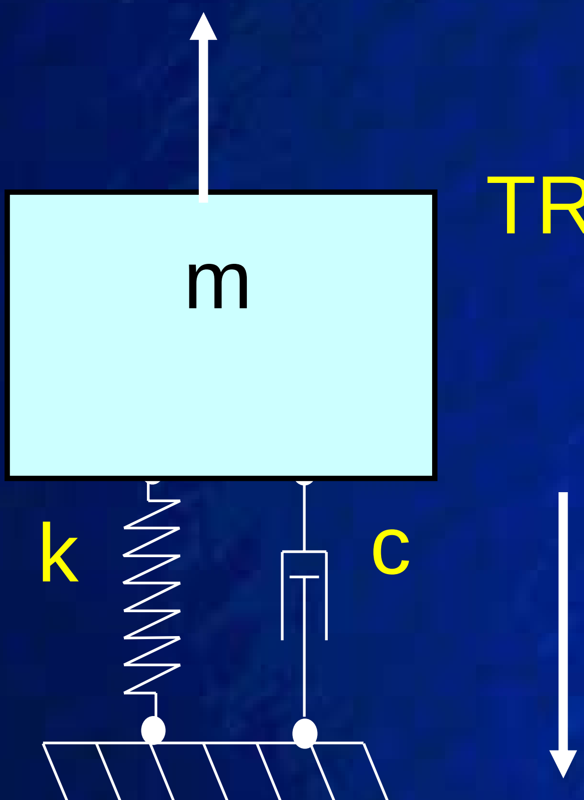
Single Degree of Freedom Vibration

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

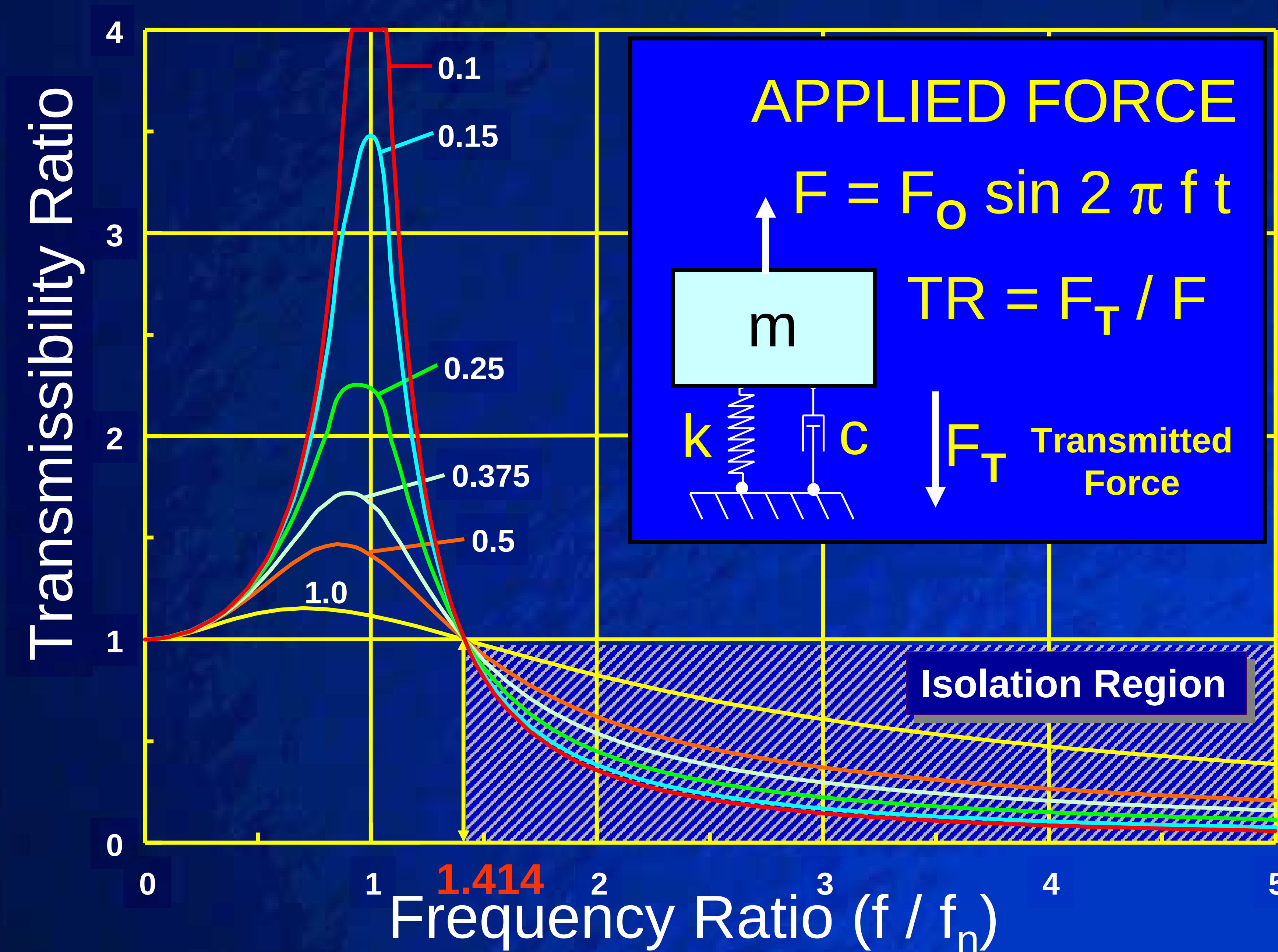
F_T

ζ = fraction of critical damping

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

f = operating frequency

Vibration Isolation Principle



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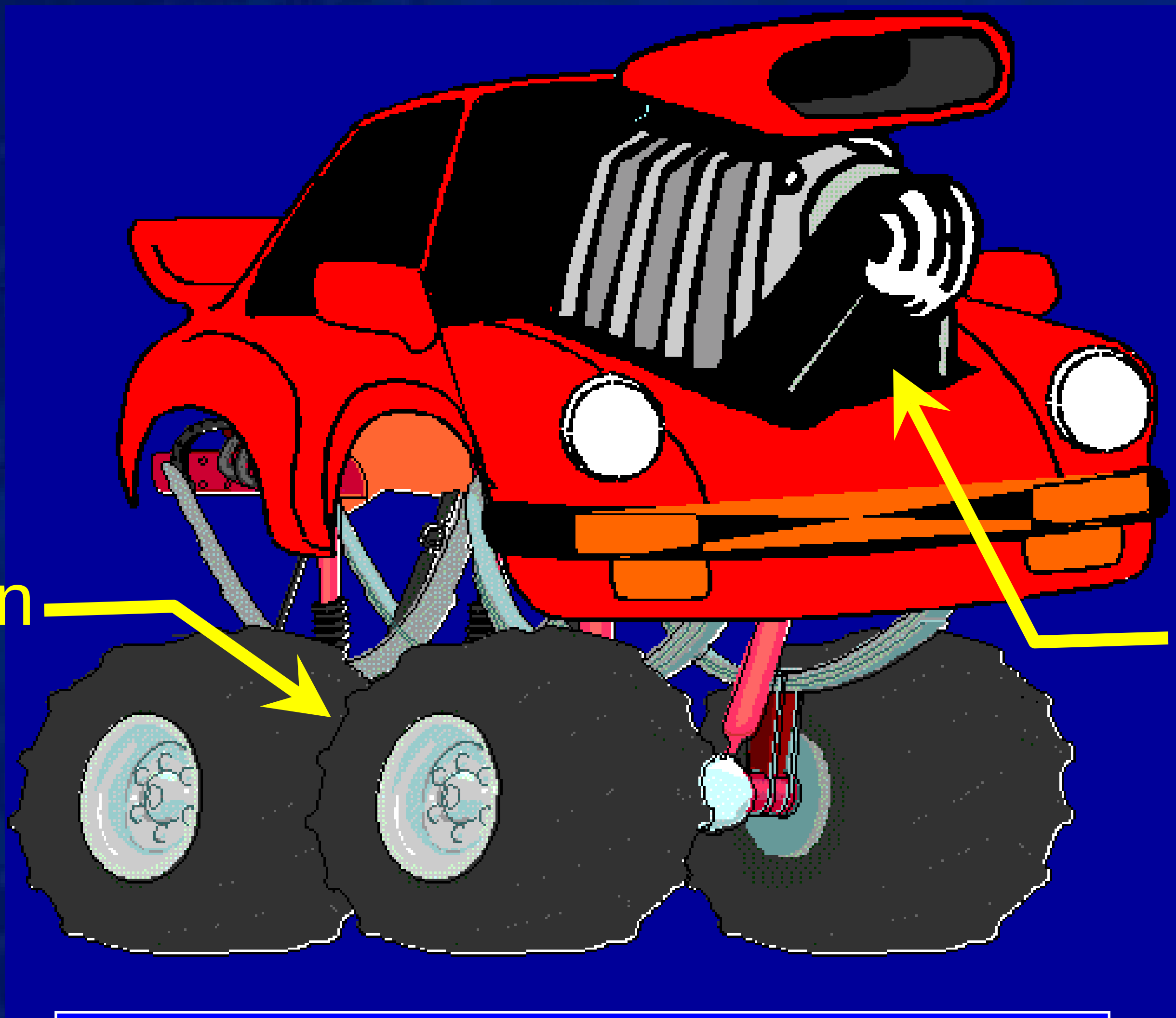
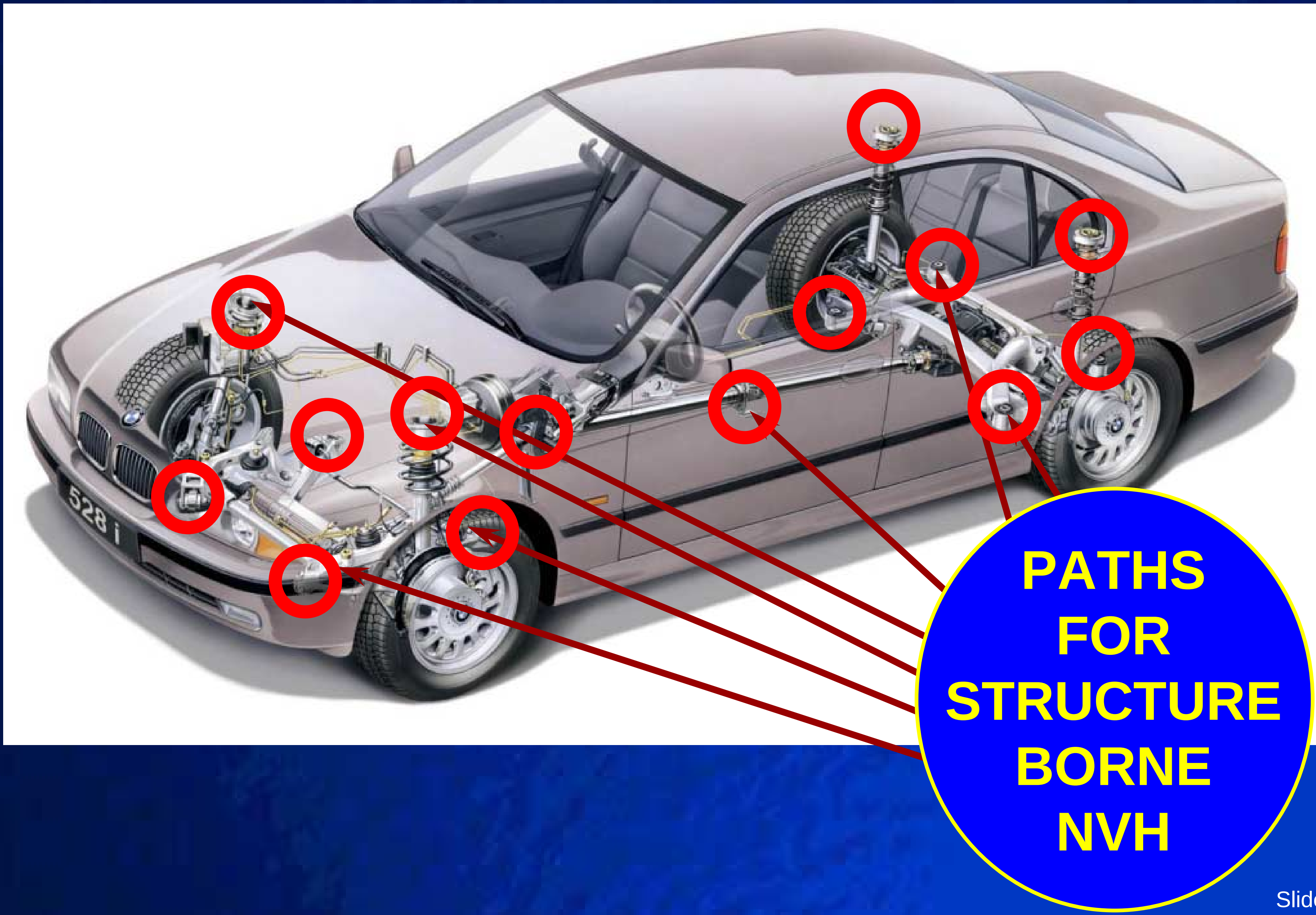
NVH Source Considerations



Two Main Sources

Typical NVH Pathways to the Passenger



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

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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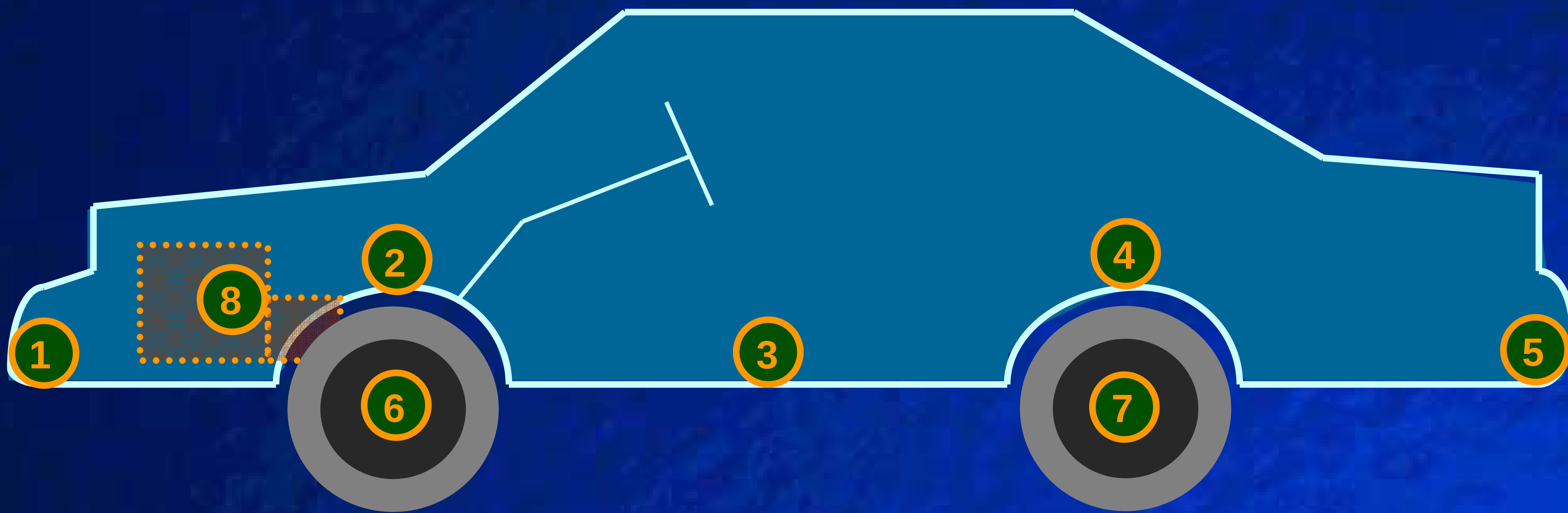
Symbolic Model of Unibody Passenger Car

8 Degrees of Freedom



	<u>Total</u>	<u>2178.2 Kg</u>	(4800LBS)
Mass	Sprung	1996.7 Kg	
	<u>Unsprung</u>	181.5 Kg	(8.33% of Total)
	Powertrain	181.5 Kg	

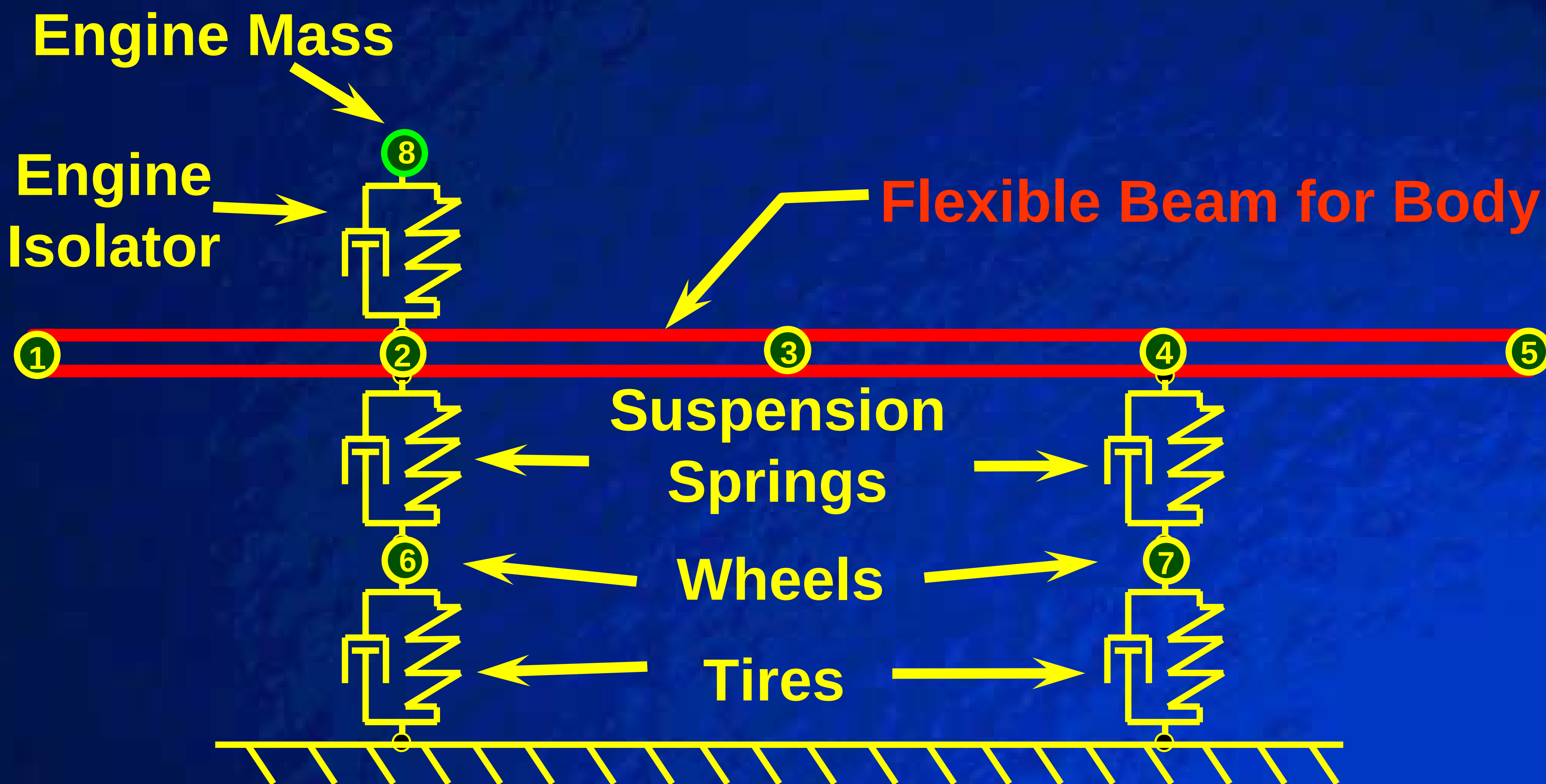
Tires	350.3	N/mm
KF	43.8	N/mm
KR	63.1	N /mm

Beam mass lumped on grids like a beam

M2,3,4 =2 * M1,5

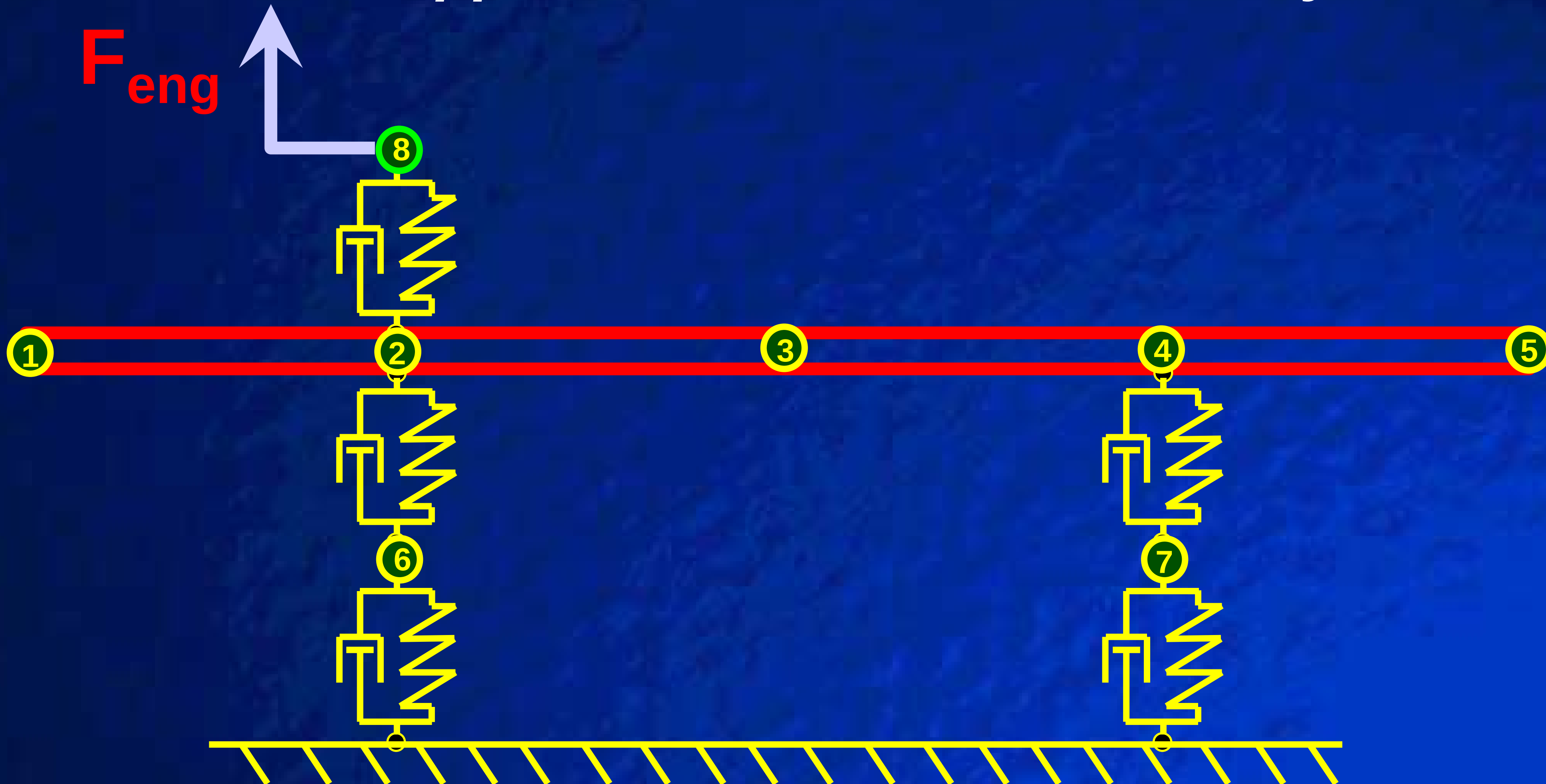
From Reference 6

8 Degree of Freedom Vehicle NVH Model



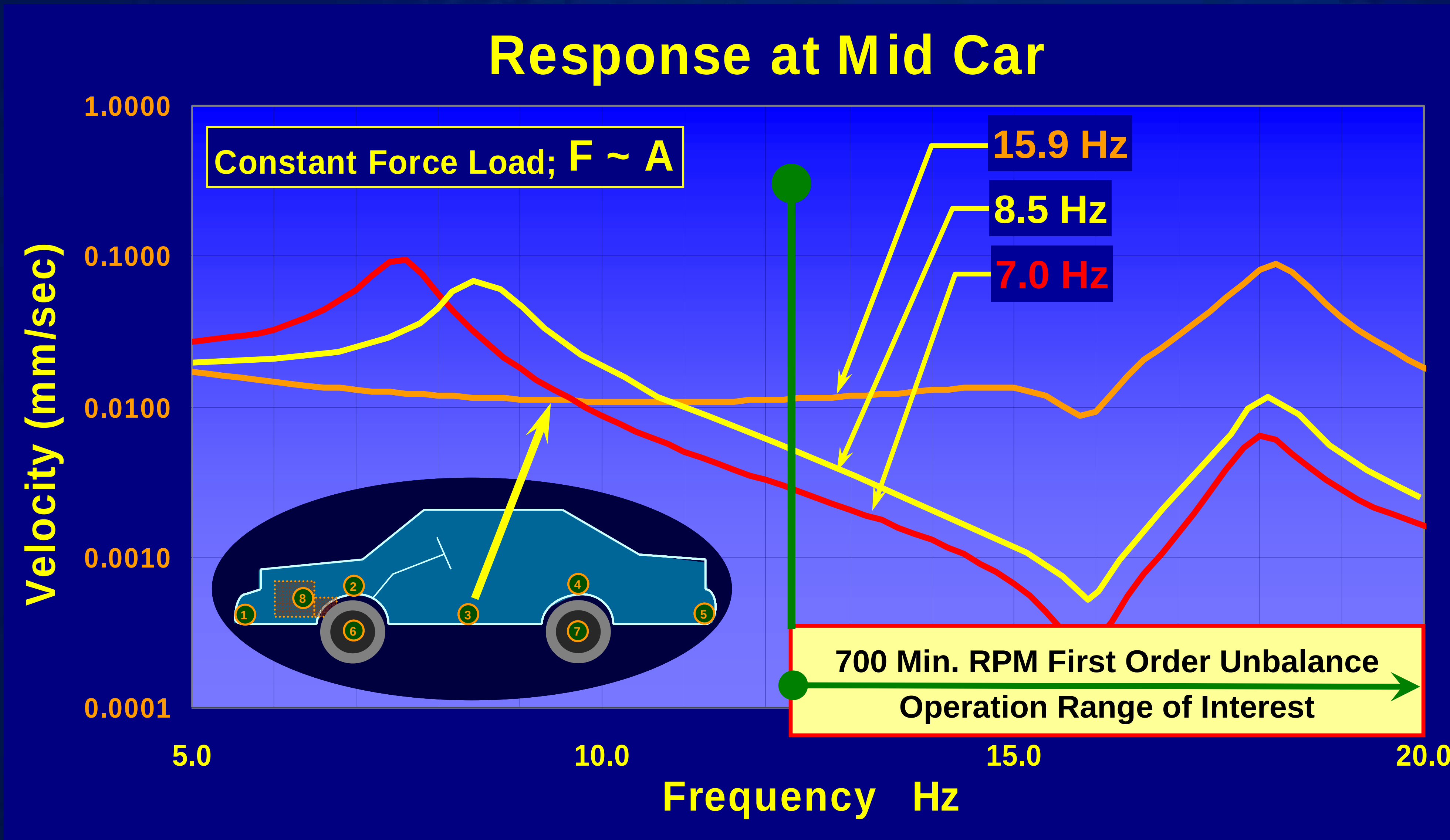
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



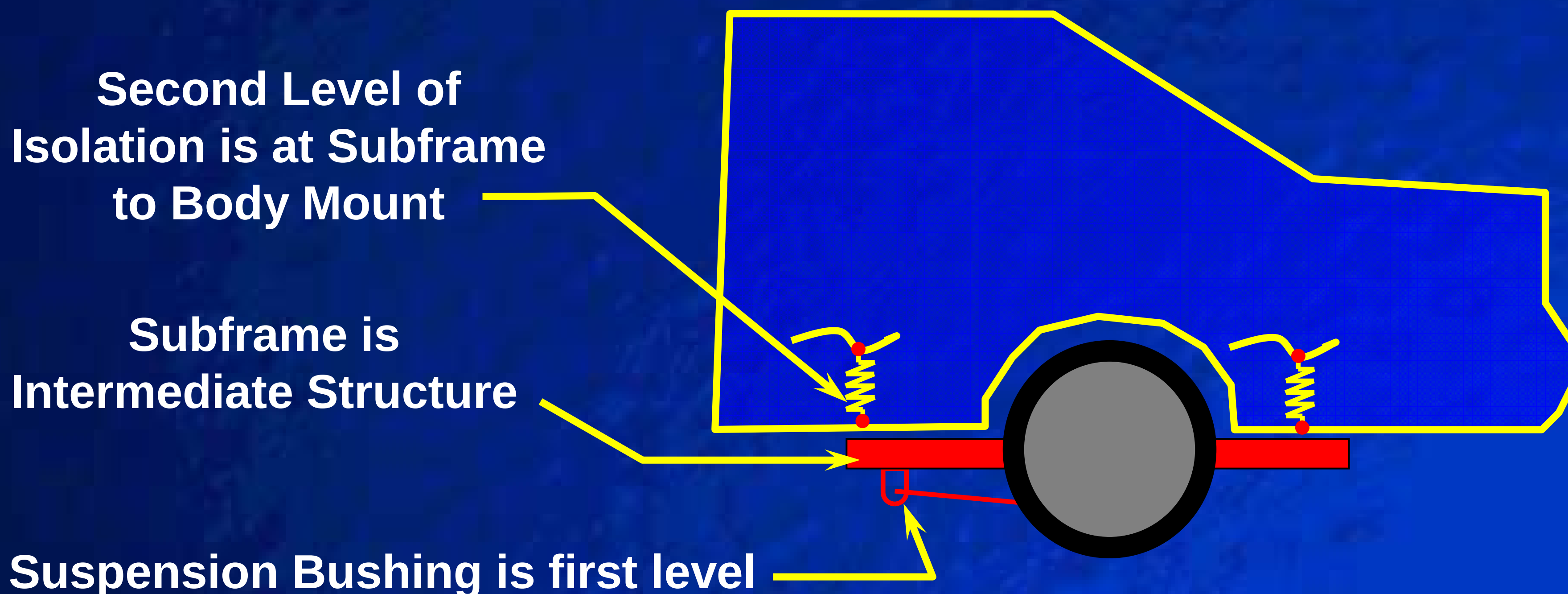
Forces at Powertrain could represent a First Order Rotating Imbalance

Engine Isolation Example



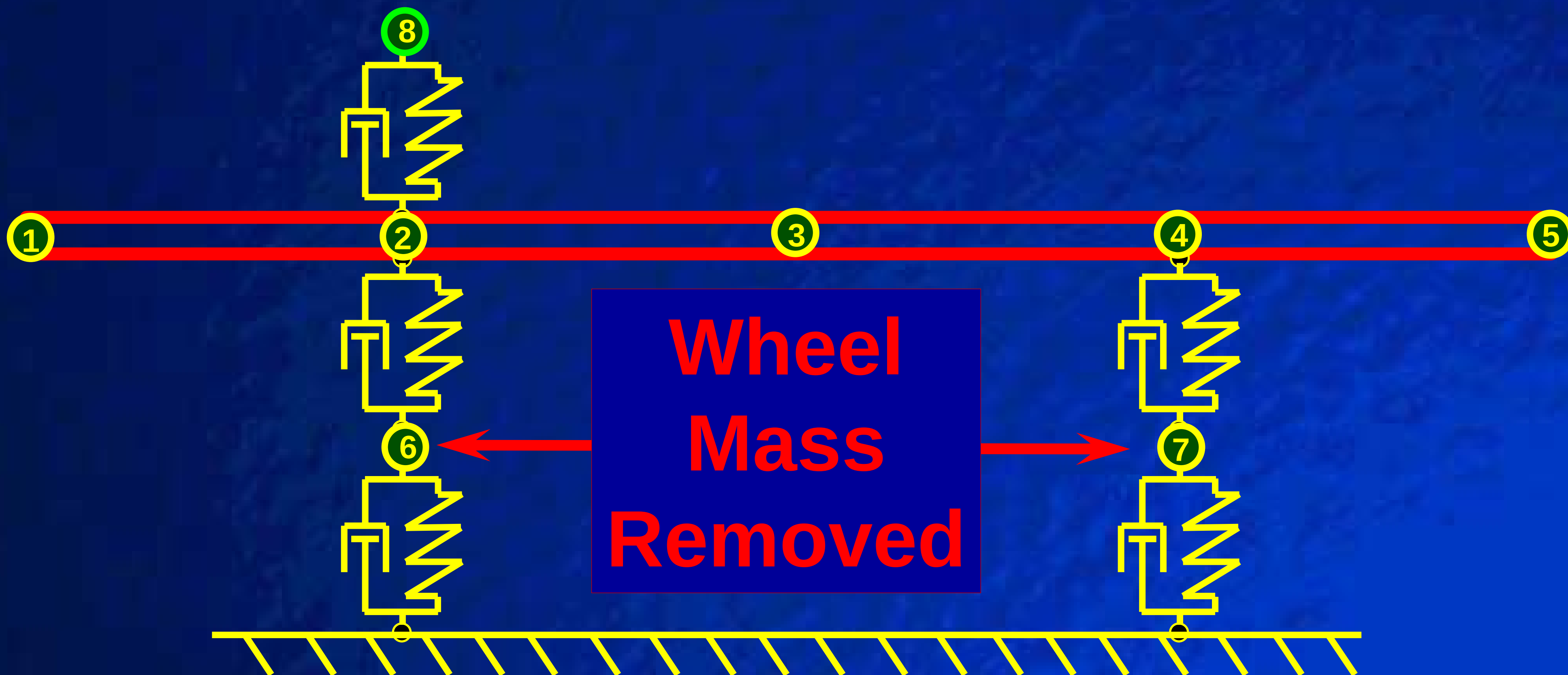
Concepts for Increased Isolation

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

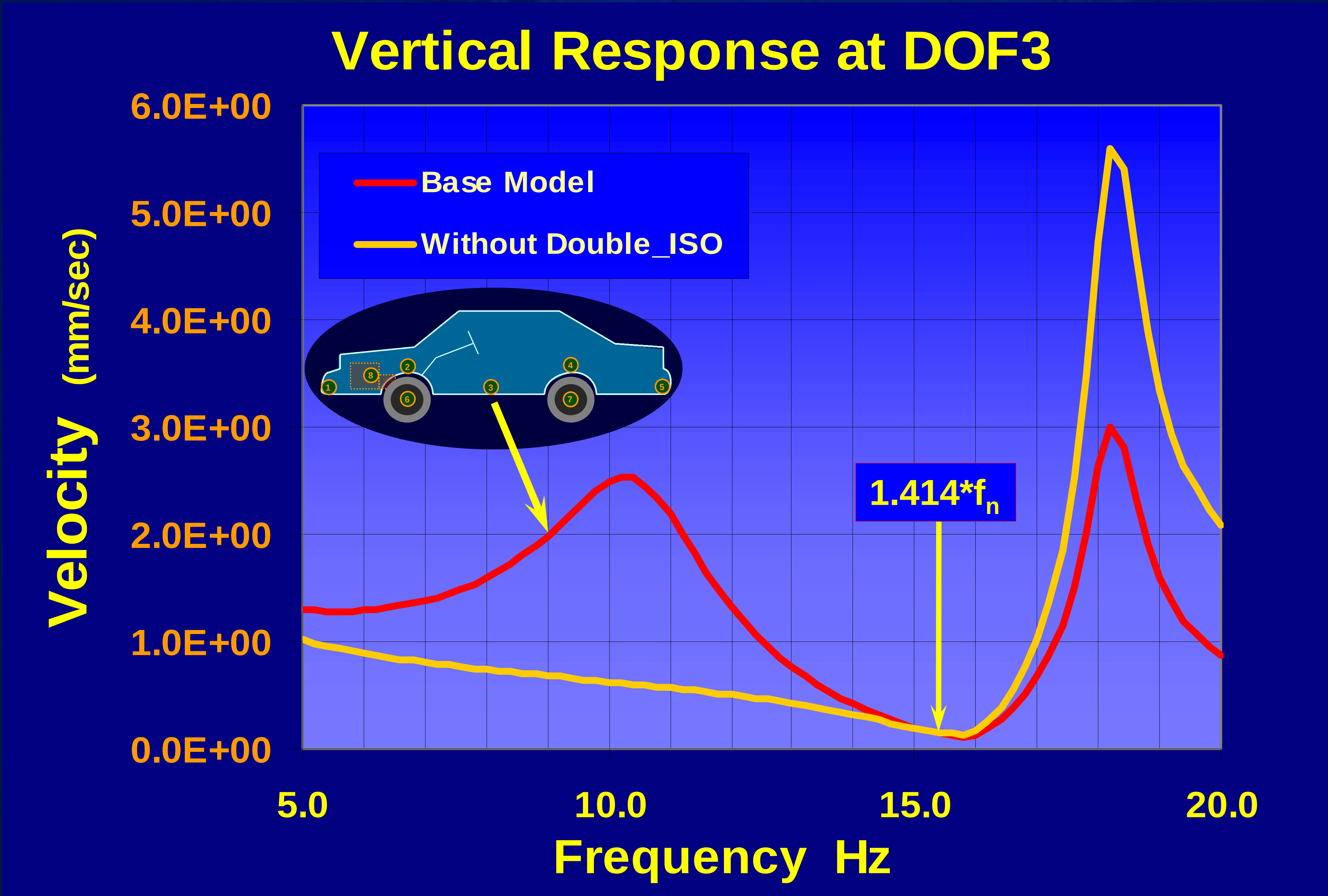


8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



Double Isolation Example



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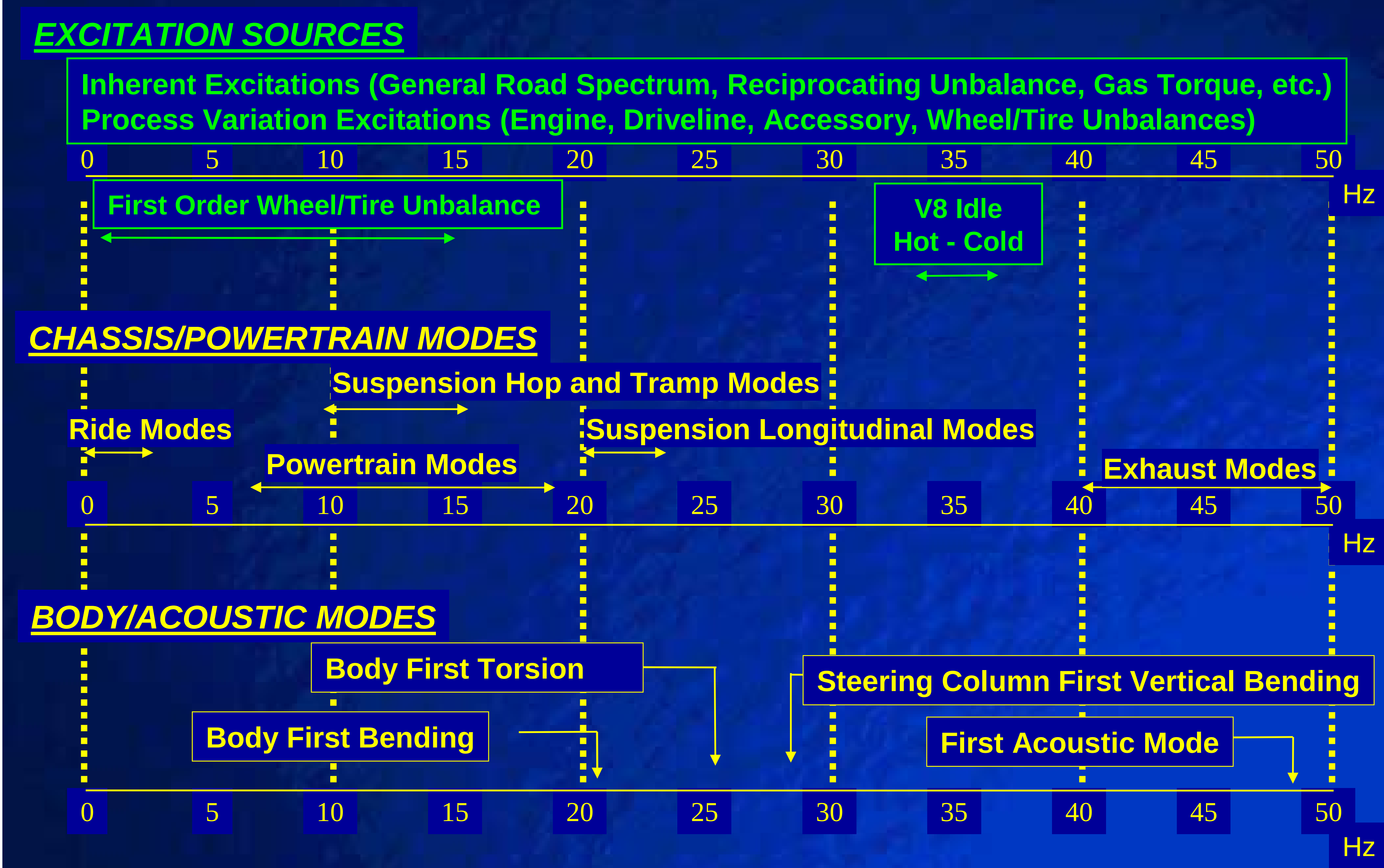
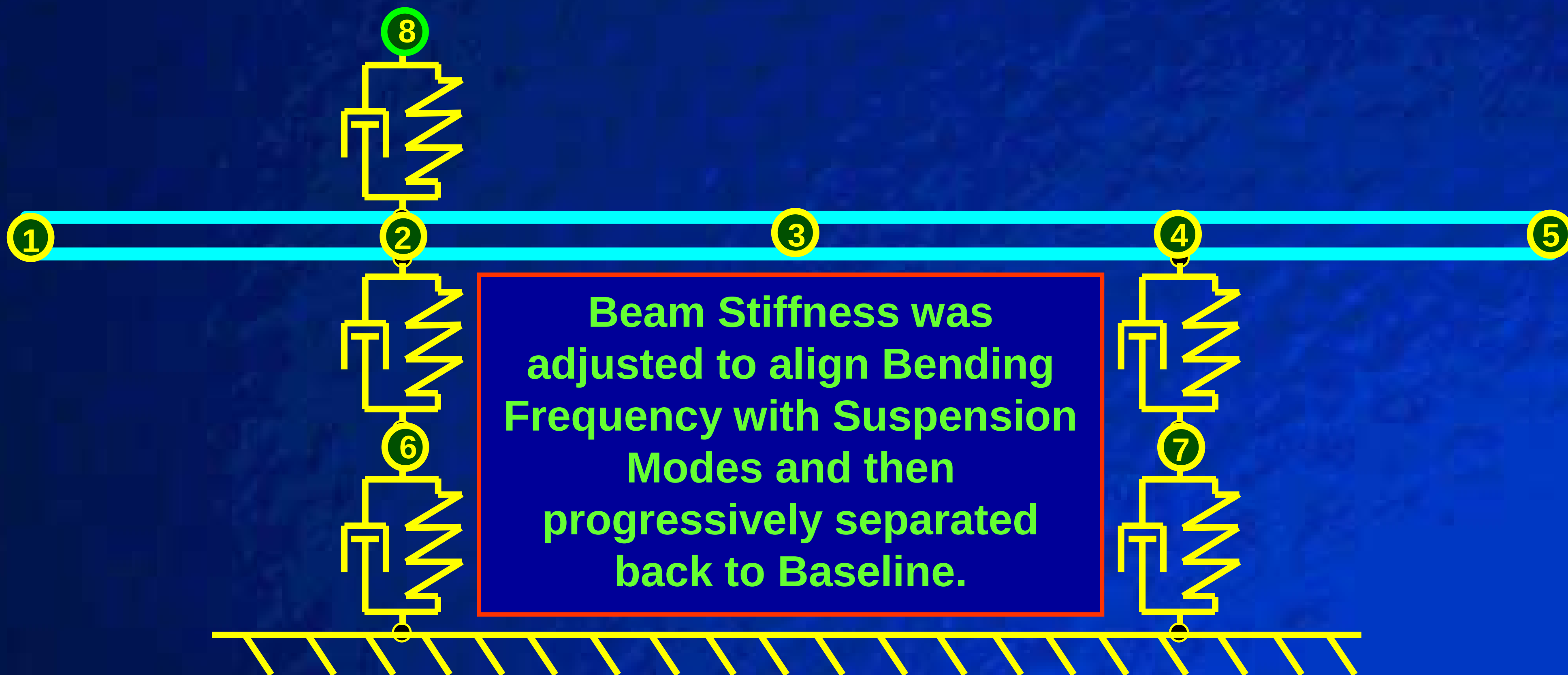
Mode Management Chart



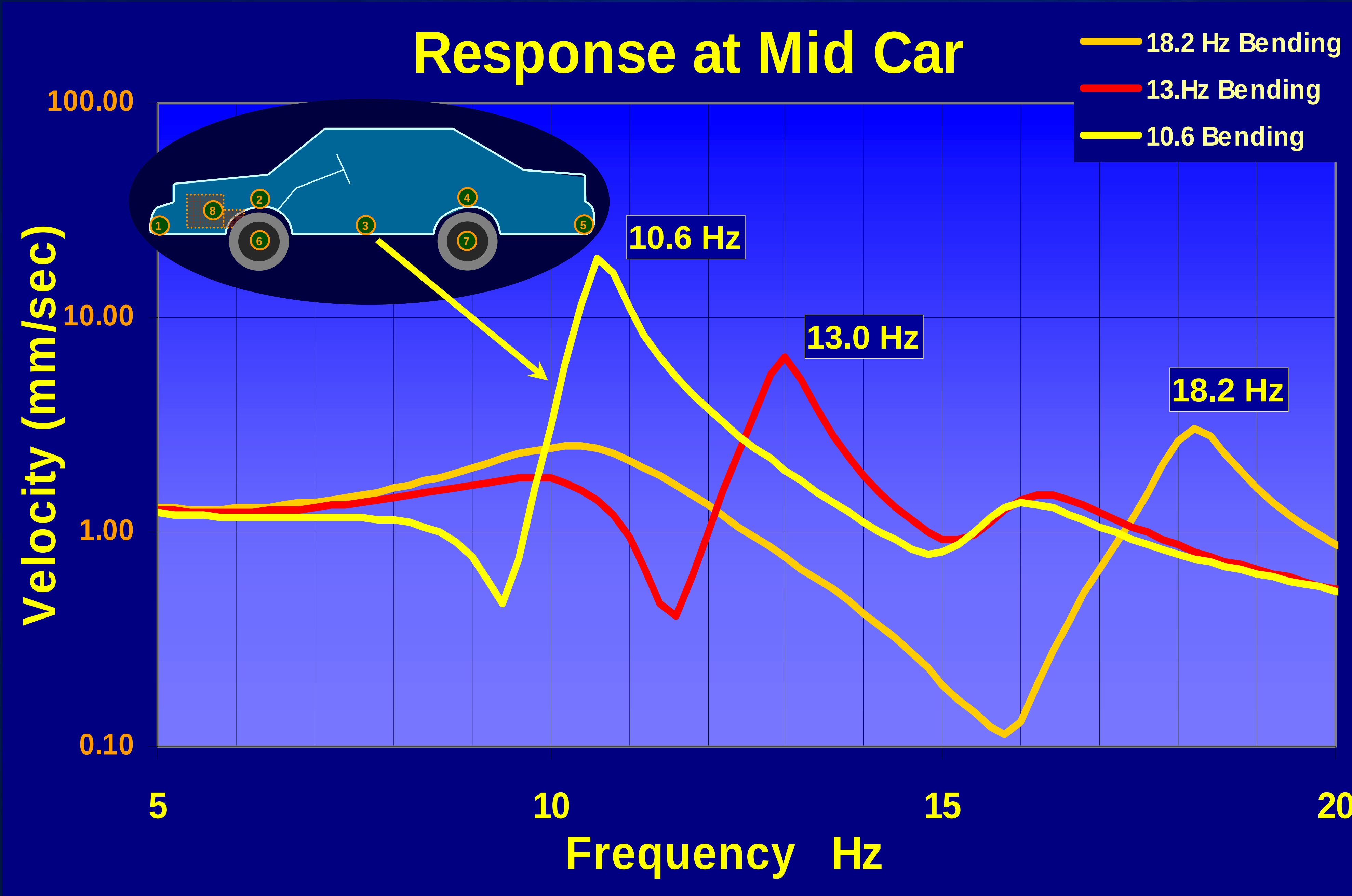
(See Ref. 1)

8 Degree of Freedom Vehicle NVH Model

Bending Mode Frequency Separation



8 DOF Mode Separation Example



Slide 33

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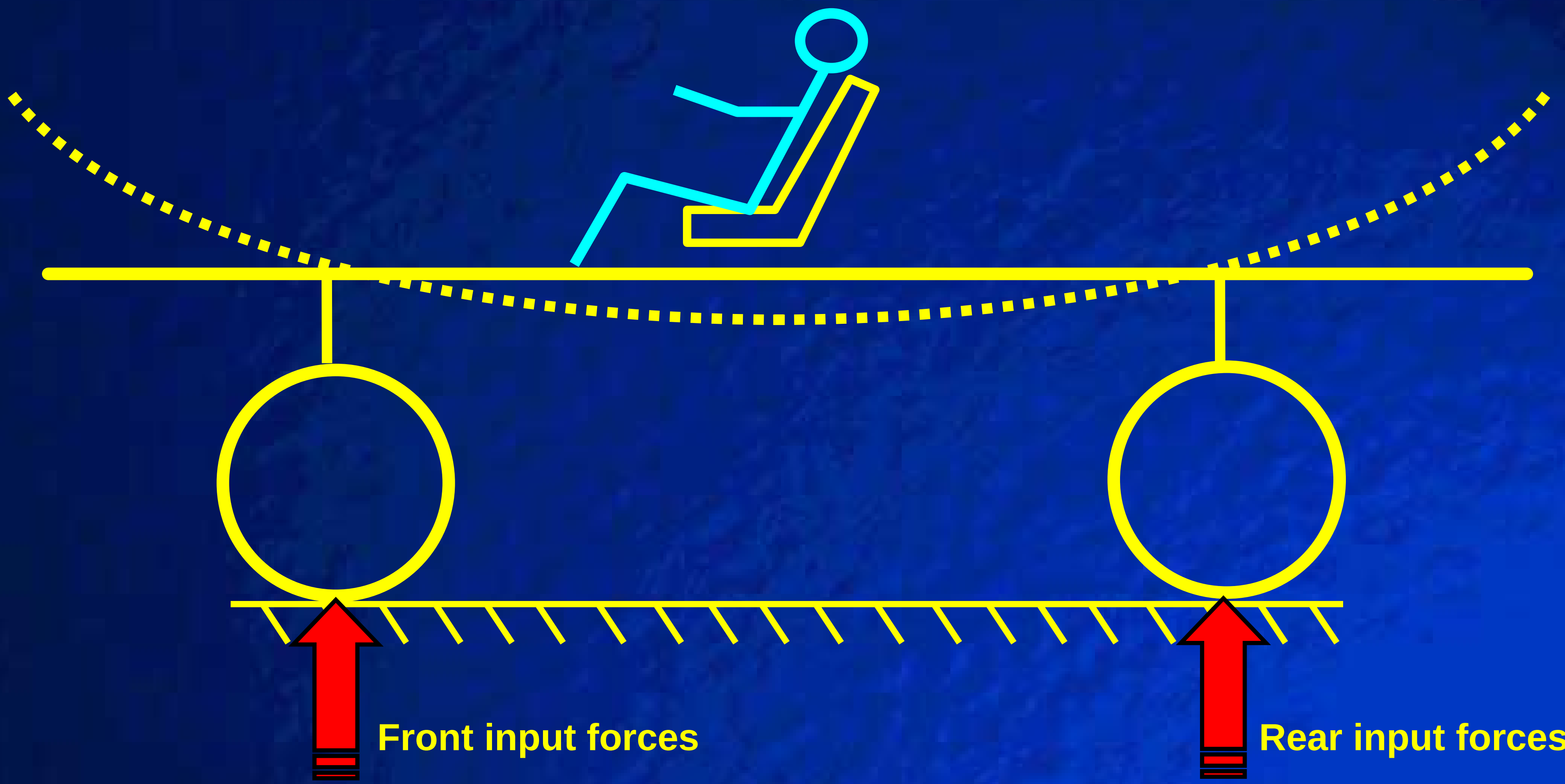
Mount at Nodal Point

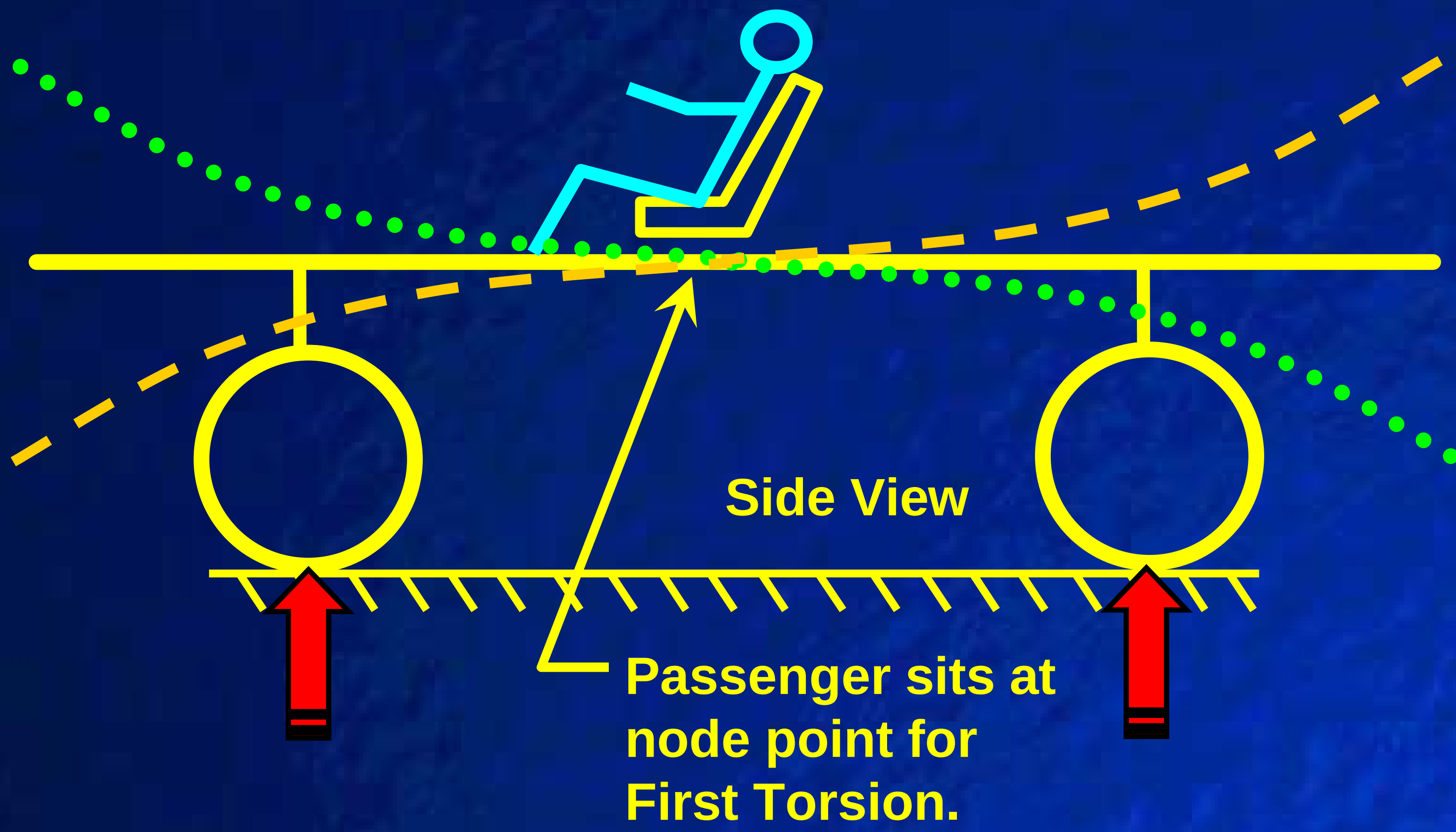
First Bending: Nodal Point Mounting Example

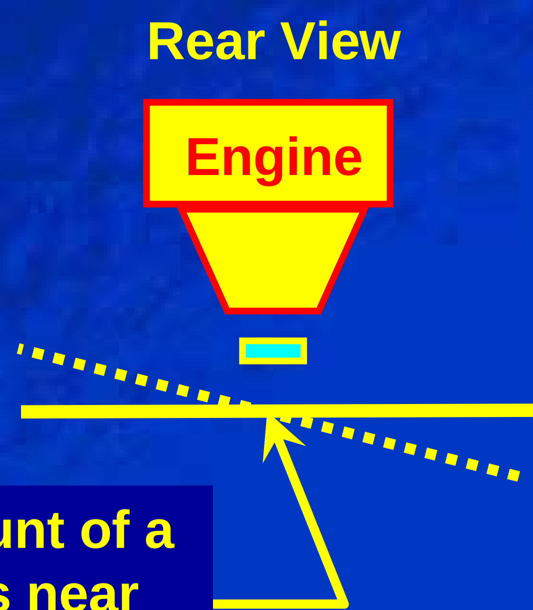


Locate wheel centers at node points of the first bending modeshape to prevent excitation coming from suspension input motion.

Mount at Nodal Point

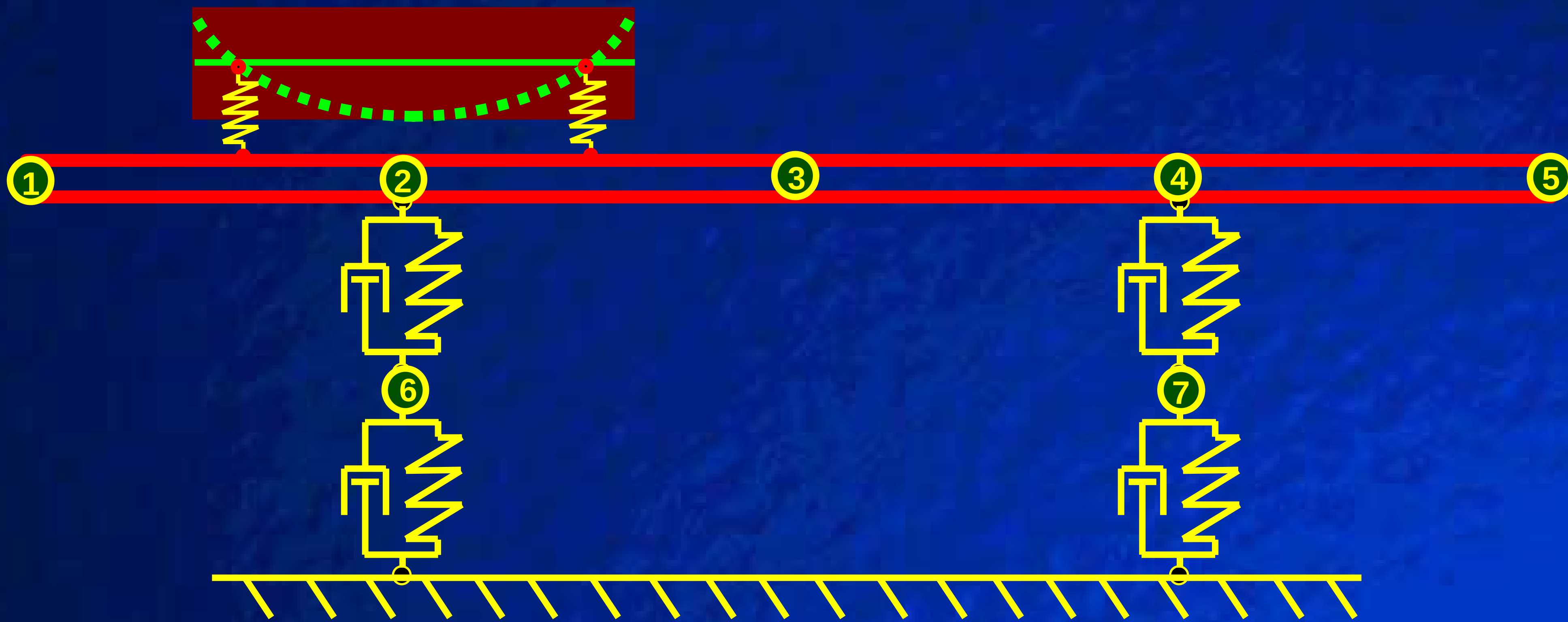
First Torsion: Nodal Point Mounting Examples





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

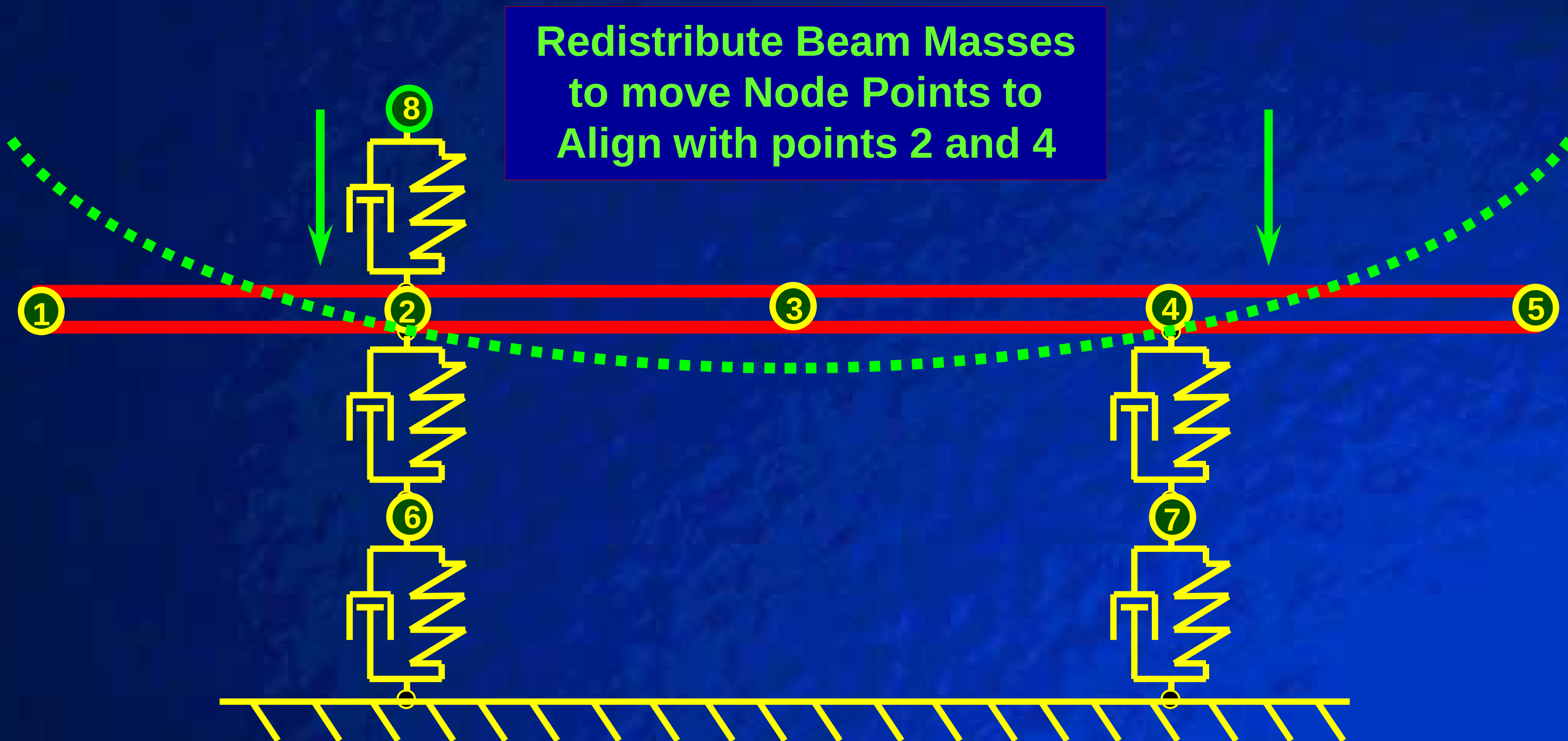
Powertrain Bending Mode Nodal Mounting



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.

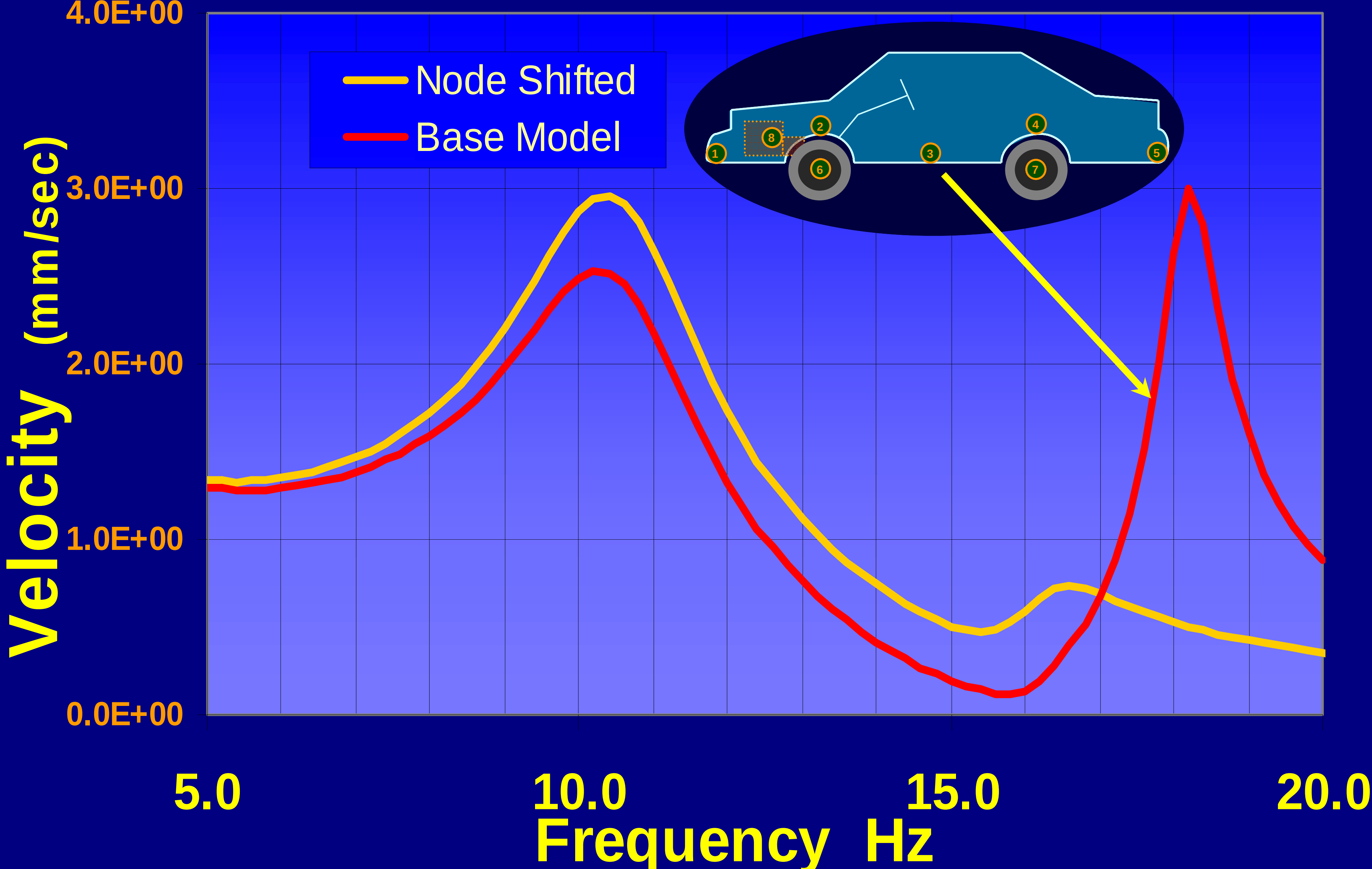
8 Degree of Freedom Vehicle NVH Model

Bending Node Alignment with Wheel Centers



First Bending Nodal Point Alignment

Response at Mid-Car



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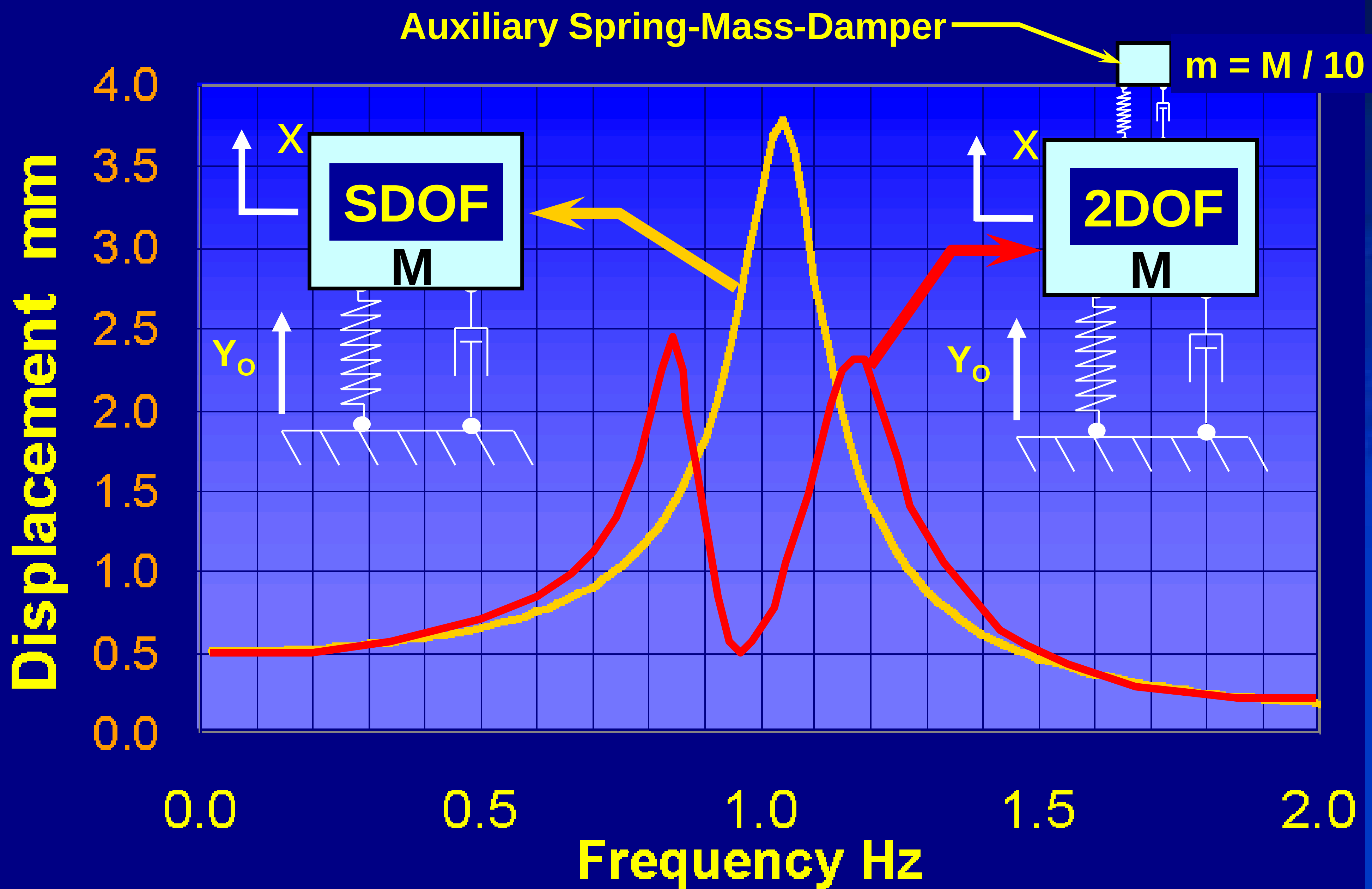
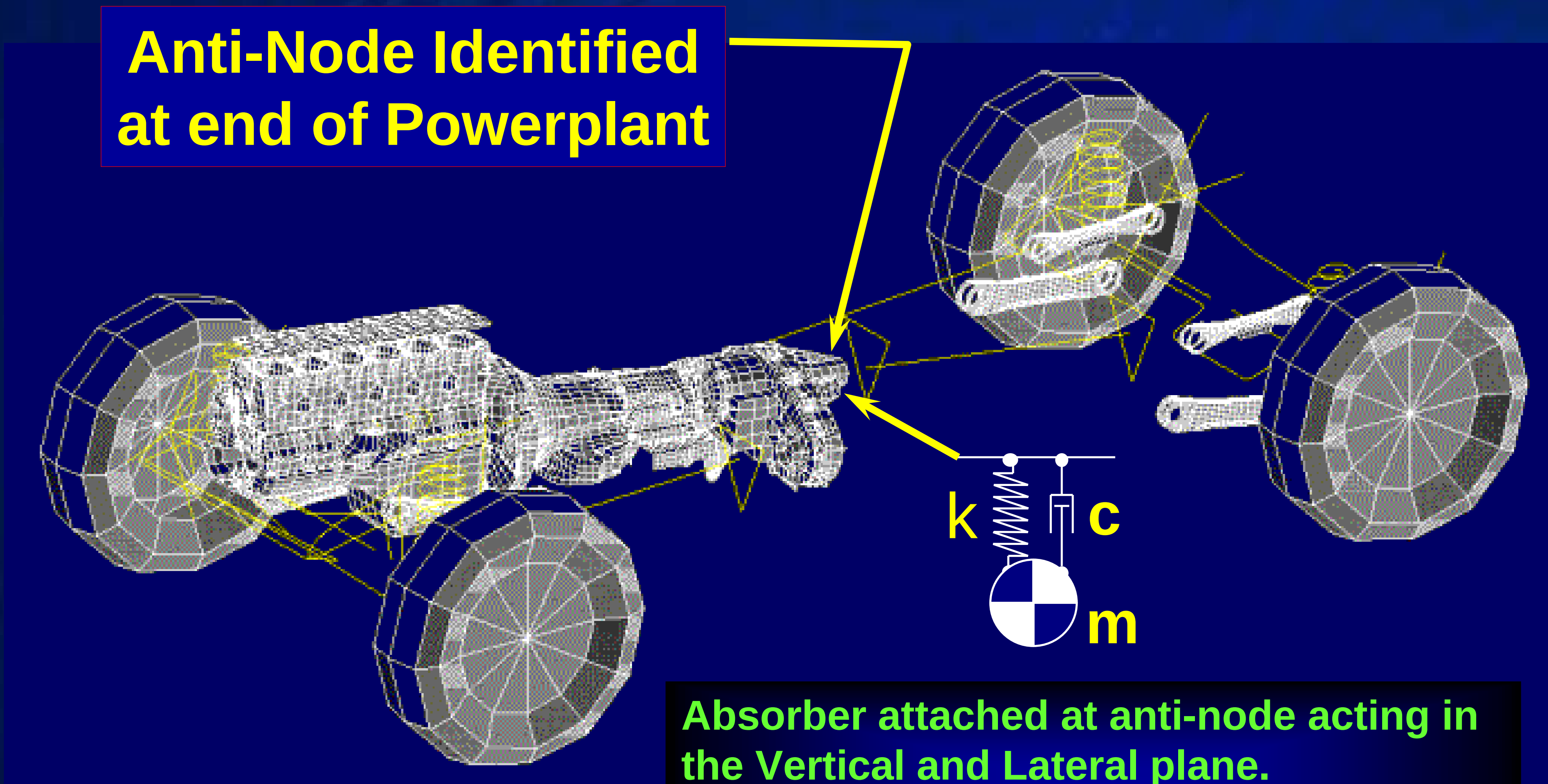
Dynamic Absorber Concept



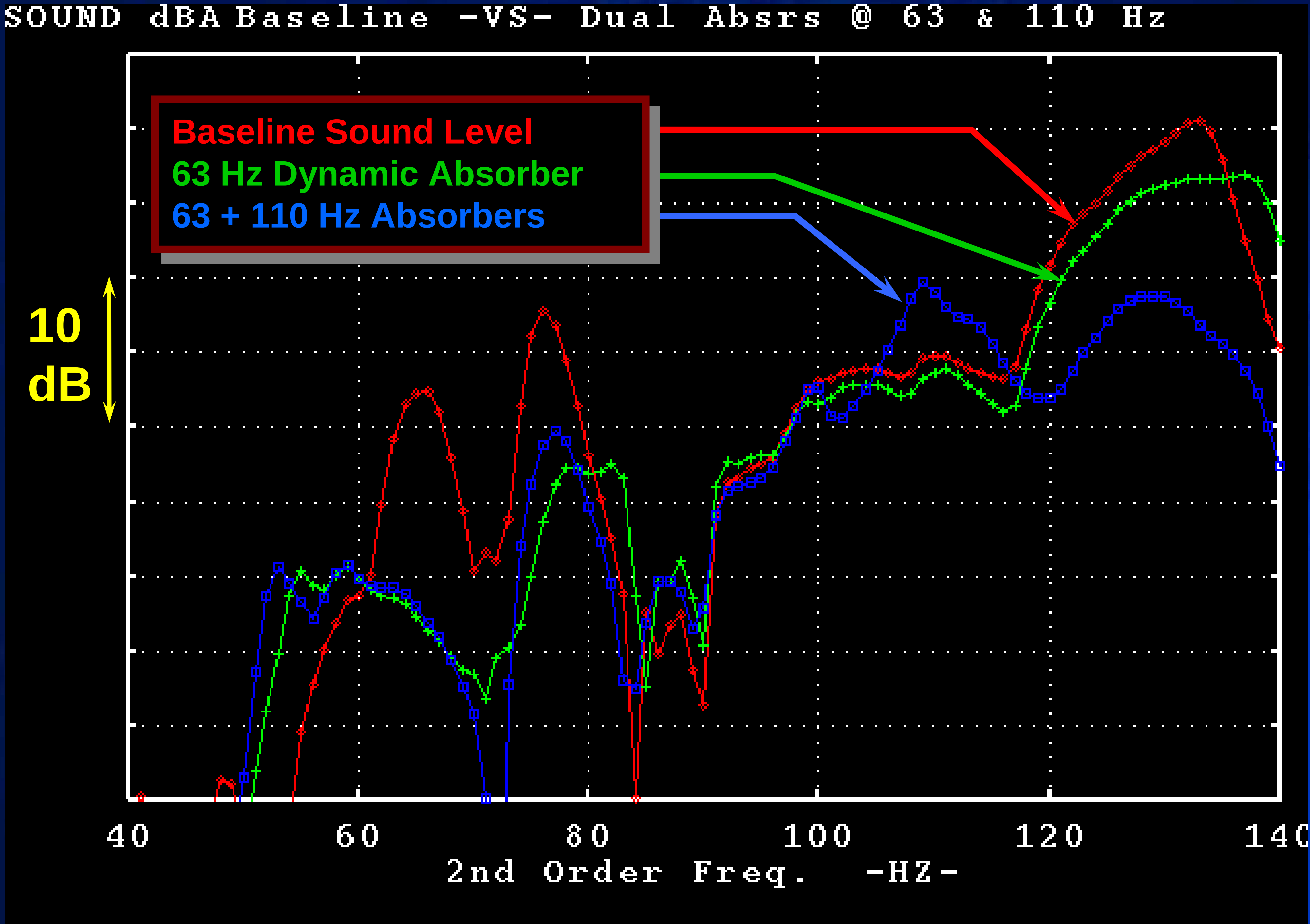
Powertrain Example of Dynamic Absorber



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
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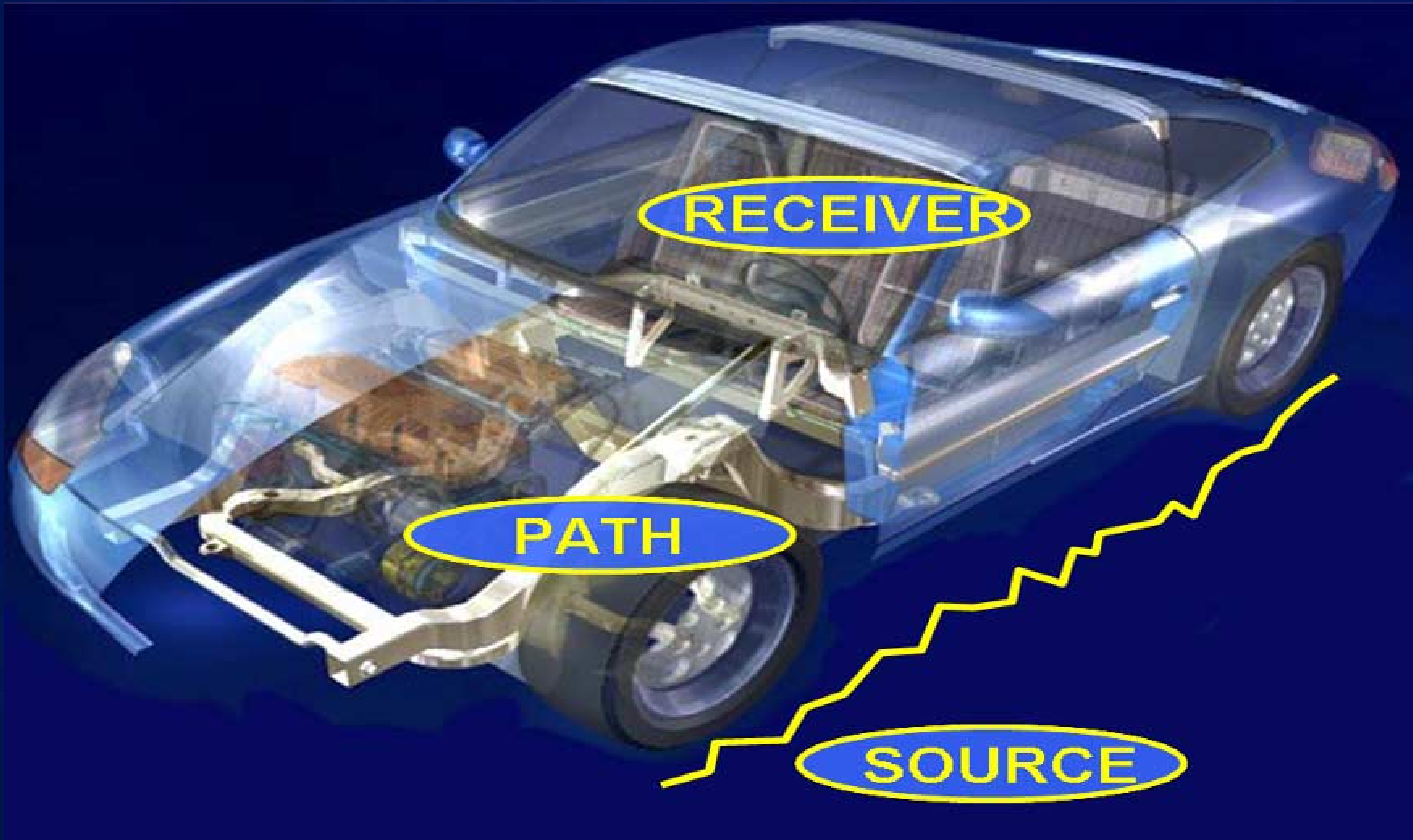
Greg Goetchius

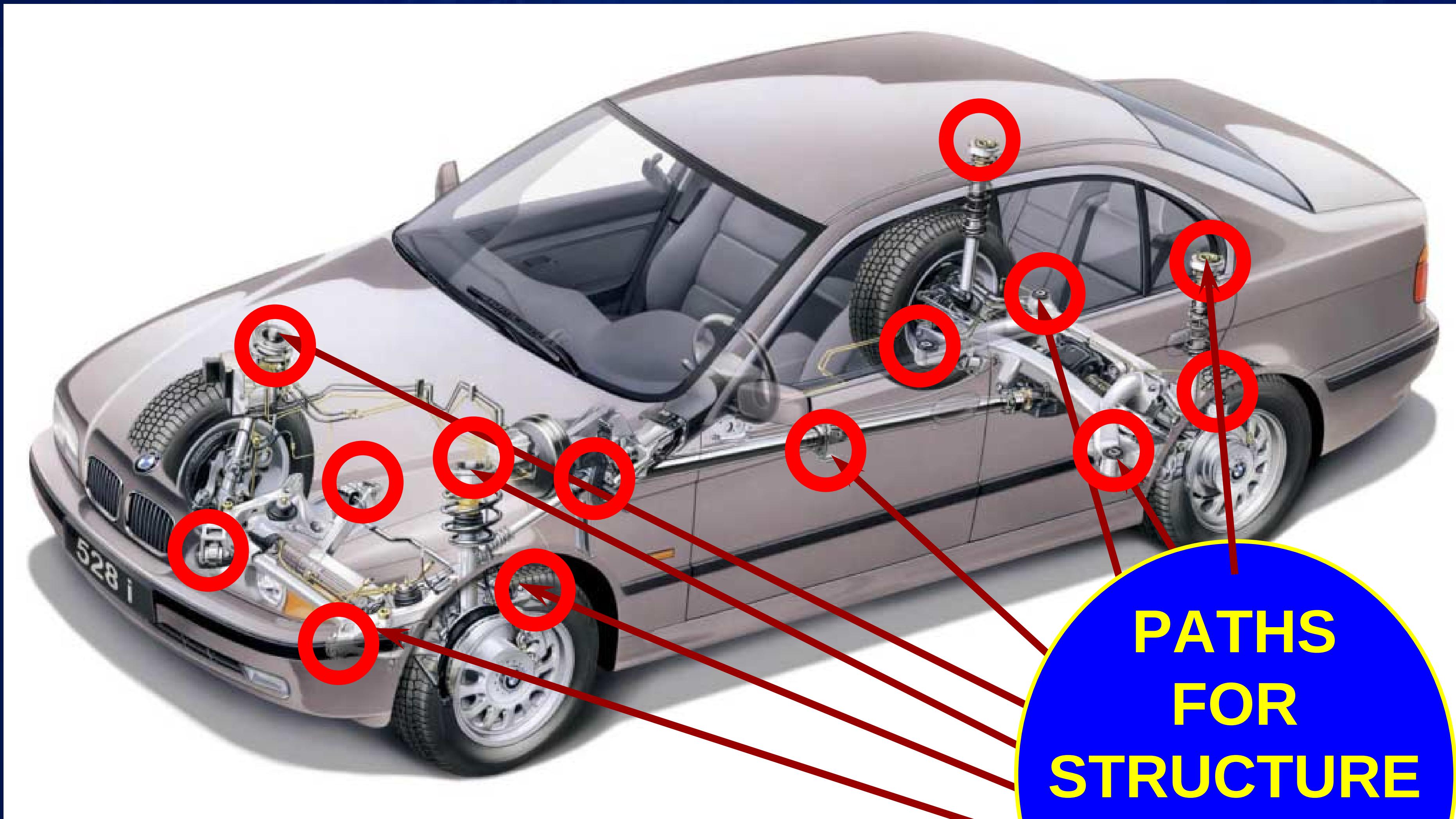
Mid Frequency NVH Fundamentals



This looks familiar!

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

Typical NVH Pathways to the Passenger

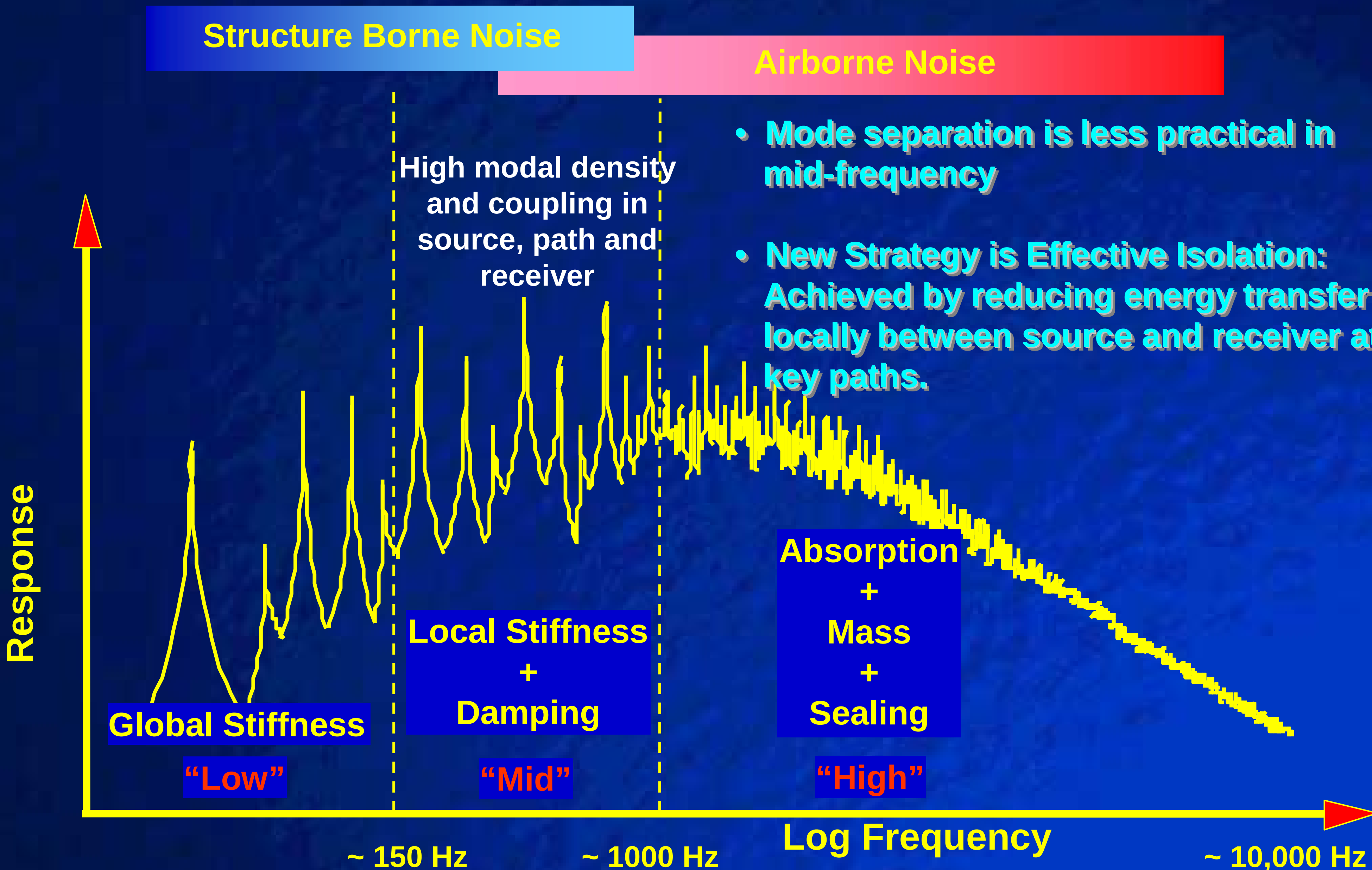


PATHS FOR STRUCTURE BORNE NVH

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.



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

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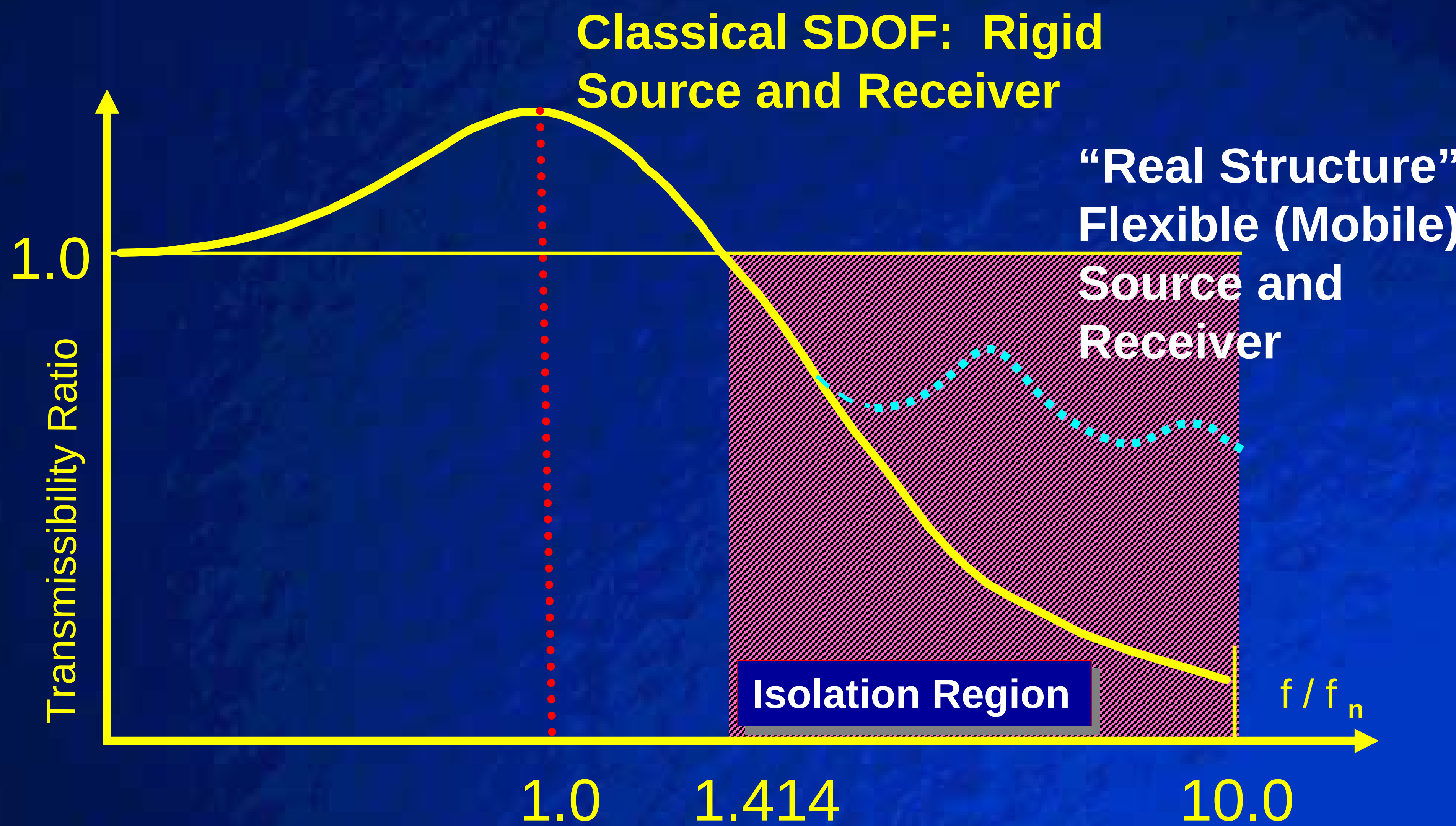
Attenuation along Key Noise Paths

Isolation Effectiveness



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

Mobility

- **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**

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

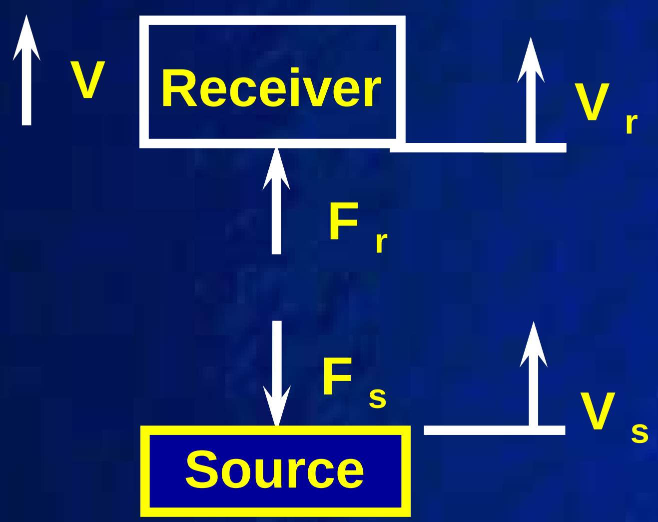
$$= \frac{\text{Frequency}}{\text{Dynamic Stiffness}}$$

Isolation

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

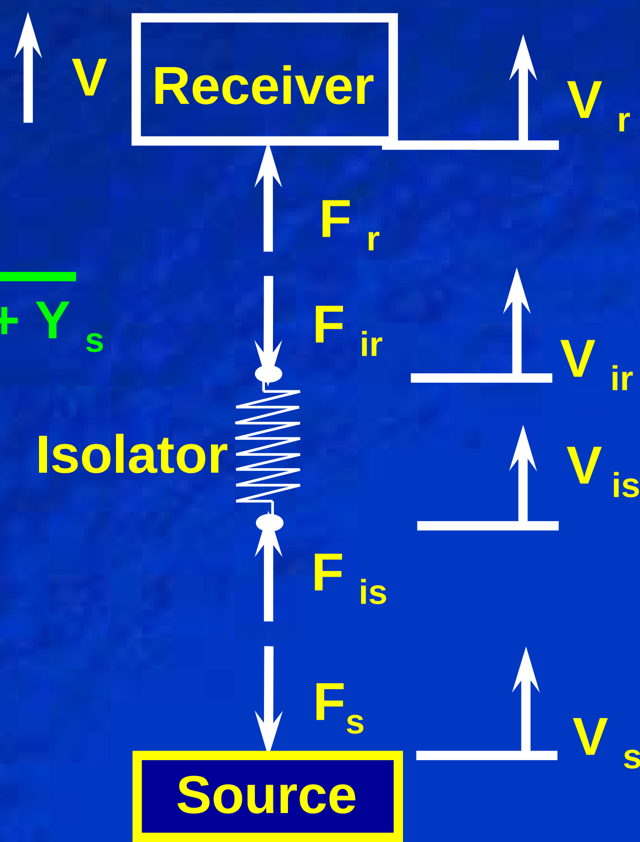
- Transmissibility ratio is used to objectively define measure of isolation

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



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

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



Isolation

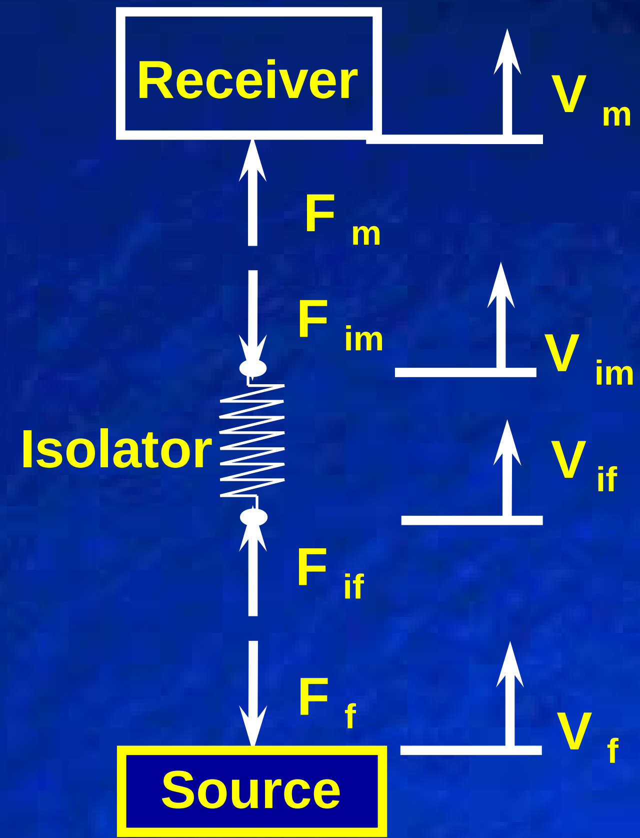
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

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

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 $Y \propto {}^{1}\!/_{K}$

Designing Noise Paths

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

$\frac{K_{body}}{K_{iso}}$ ↓ \ $\frac{K_{source}}{K_{iso}}$ →	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

Body-to-Bushing Stiffness Ratio Relationship to Transmissibility

Mid-Frequency Analysis Character

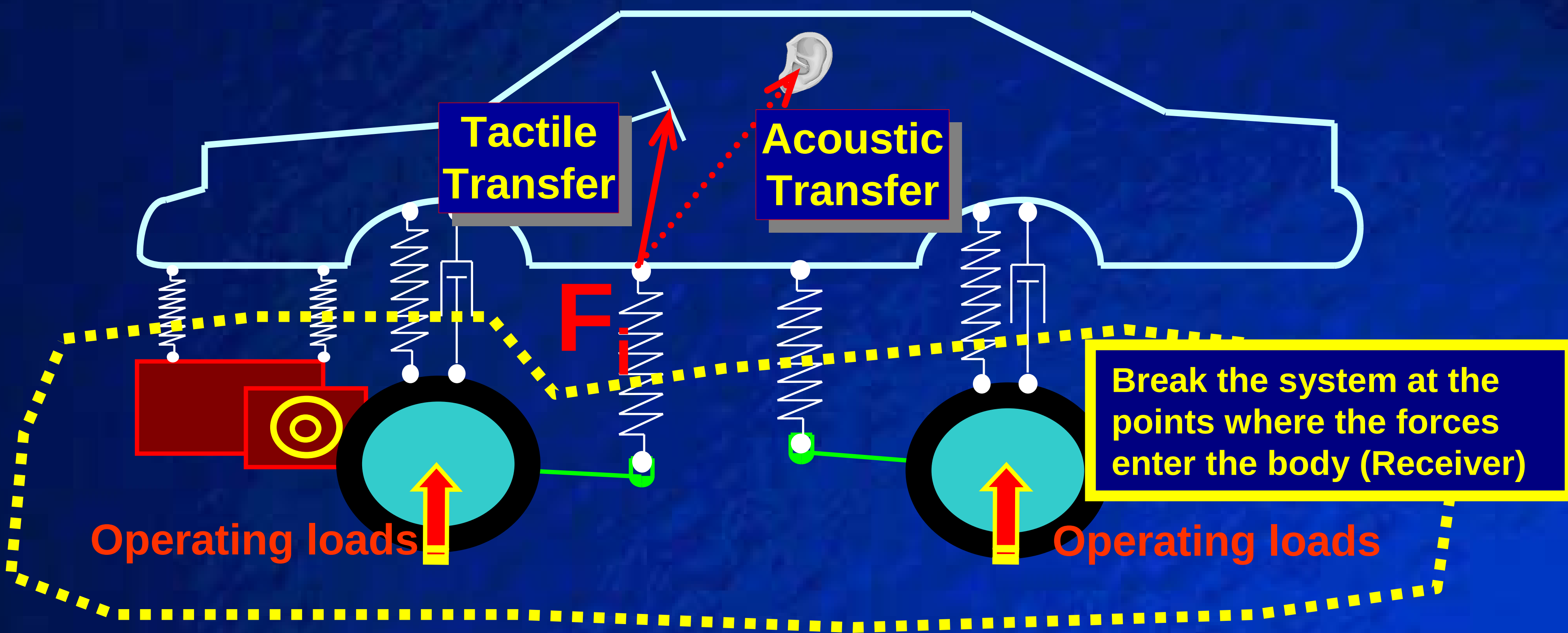
Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths

Identifying Key NVH Paths

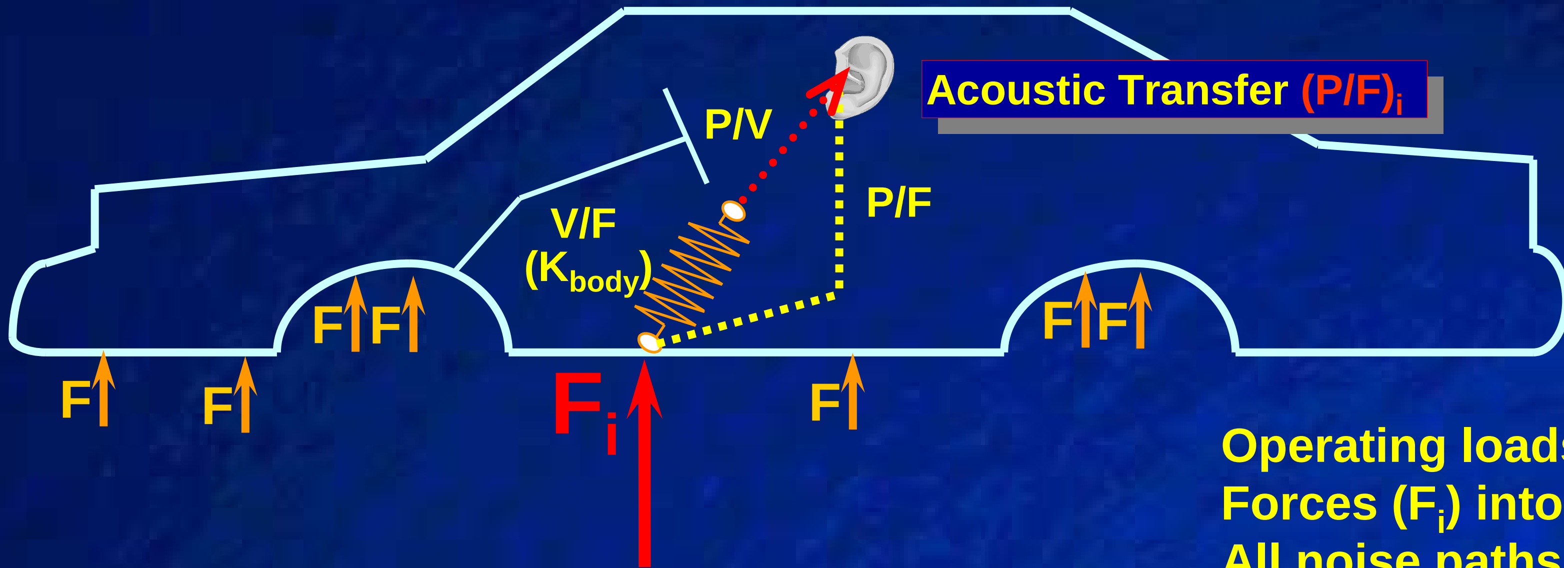
Key NVH paths are identified by Transfer Path Analysis (TPA)



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

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

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

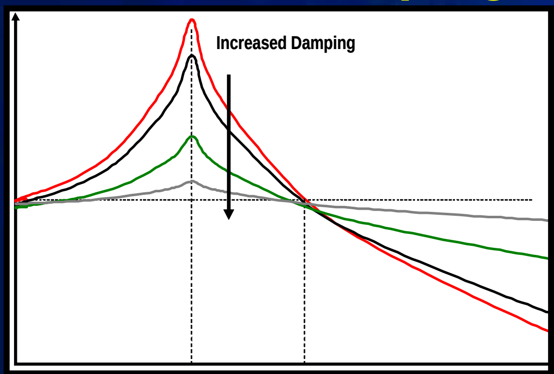
"Downstream" Effects: Body Panels

Recall for Acoustic Response P_t

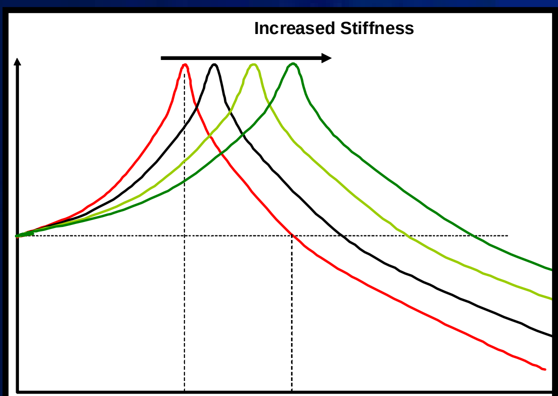
$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [F_i * (P/V)_i * (V/F)_i]$$

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

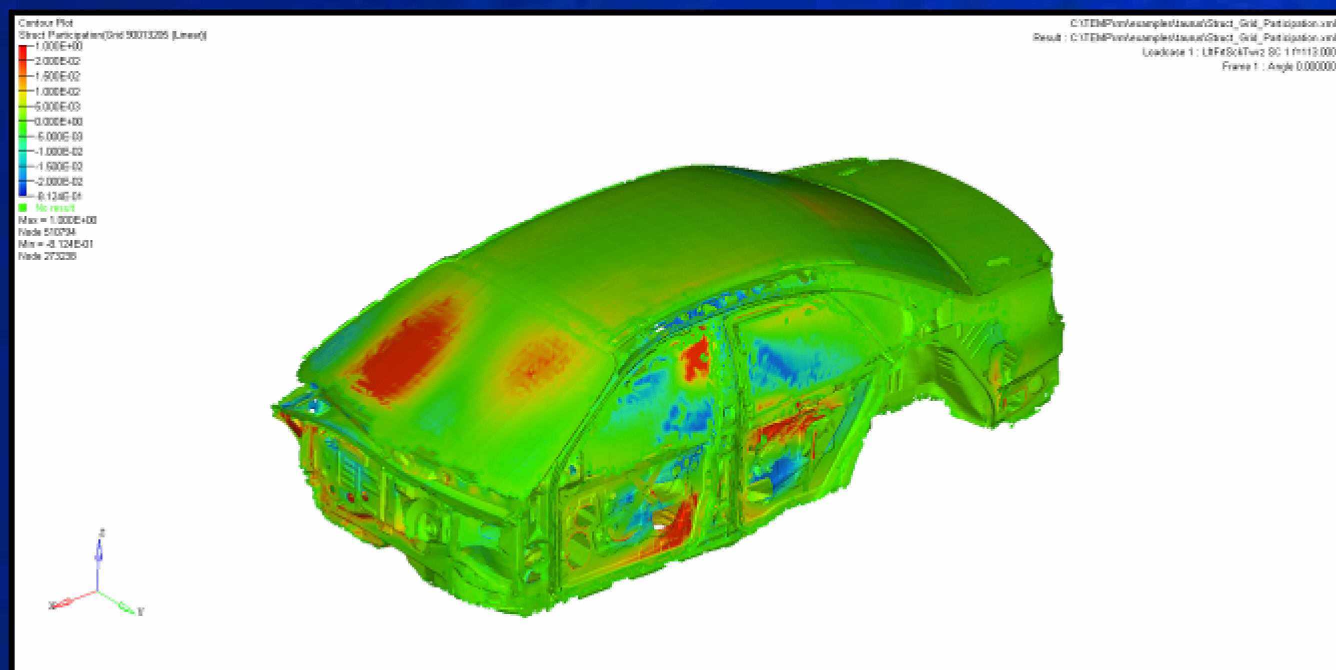
1. Panel Damping



2. Panel Stiffness



3. Panel Acoustic Contribution



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

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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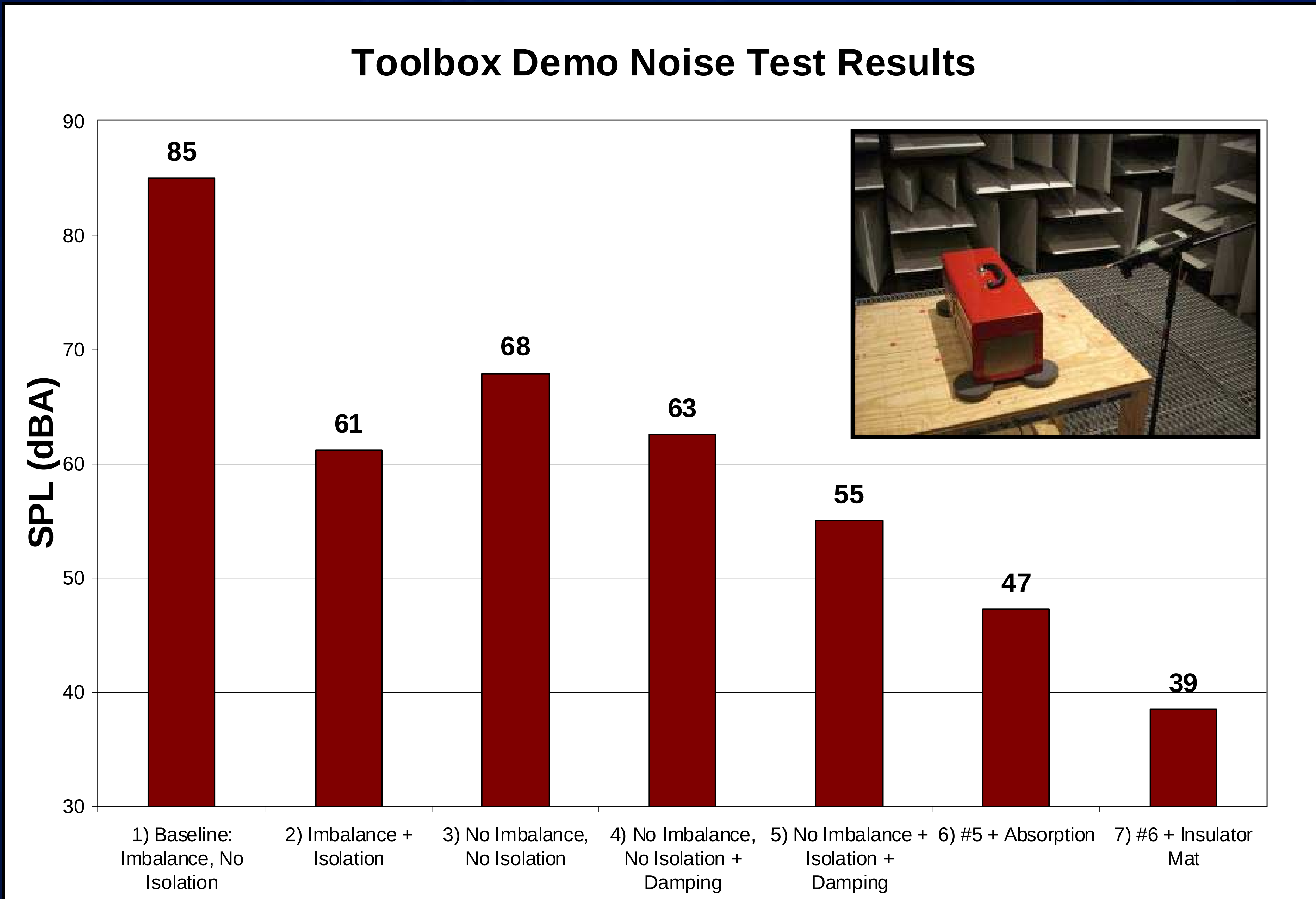
Tool Box Demo Test Results



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Jianmin Guan

Real World Application Example Introduction

2007-01-2232

Noise and Vibration Reduction Technology in the Development of Hybrid Luxury Sedan with Series/Parallel Hybrid System

Naoto Kawabata, Masashi Komada and Takayoshi Yoshioka
Toyota Motor Corporation

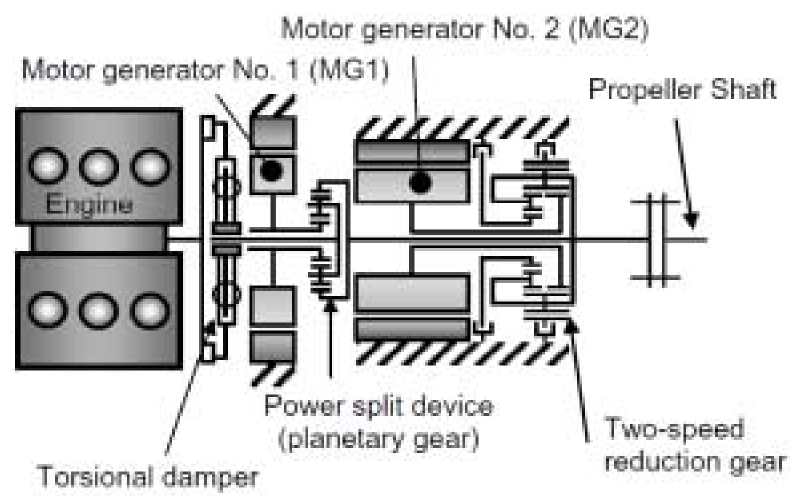
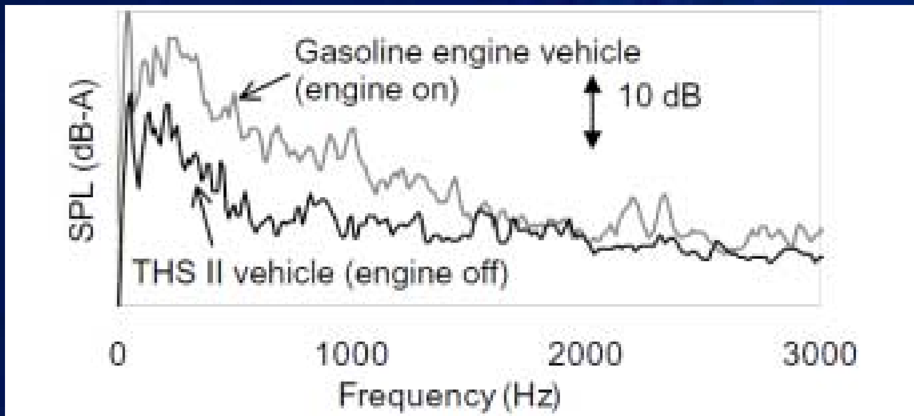


Fig.1 Longitudinal Power Train Configuration

Quietness during Idle and Electric-Vehicle Operation

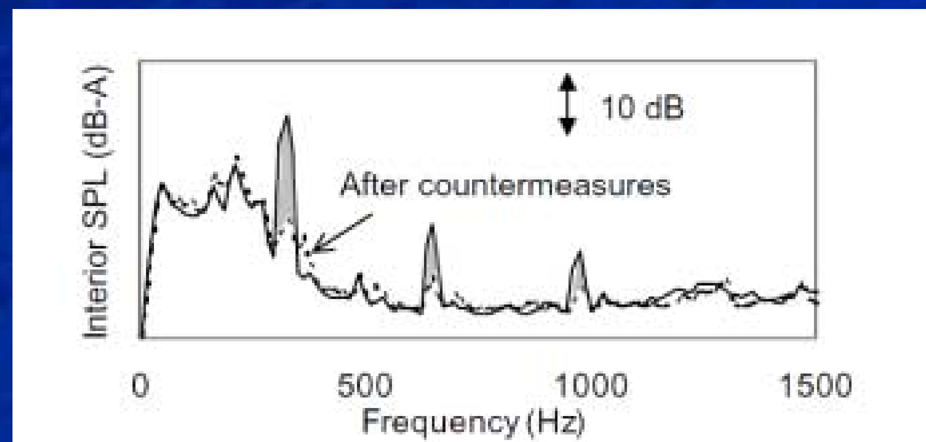


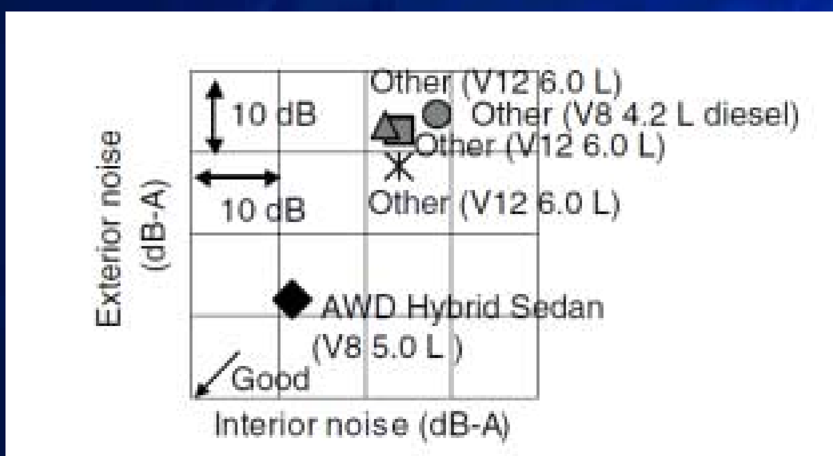
Root Cause Diagnostics:

1. Water pump in the inverter cooling system
2. Electromagnetic noise of the motor, the inverter, and other units

Reduce Inverter Water Pump Loads:

1. Reduce pump impeller imbalance
2. Redesign bearing structure
3. Changing motor structure

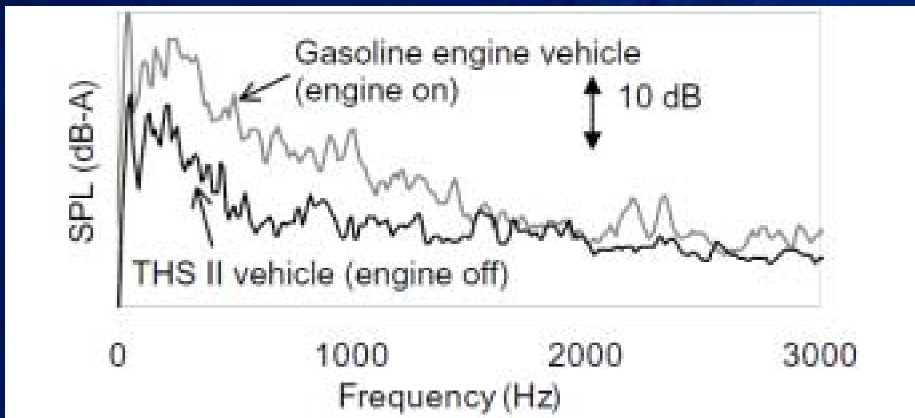




Improve Isolation from Body:

1. Install rubber isolator
2. Increase mounting bracket rigidity
3. Improve Inverter case

Quietness during Idle and Electric-Vehicle Operation



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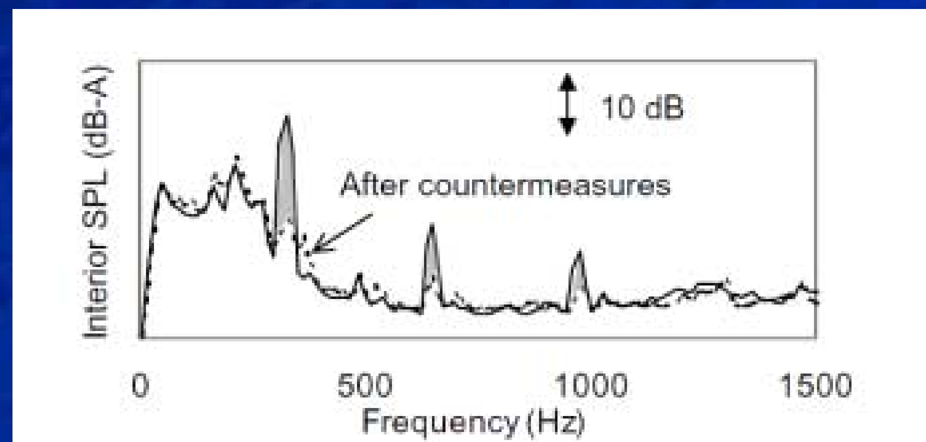
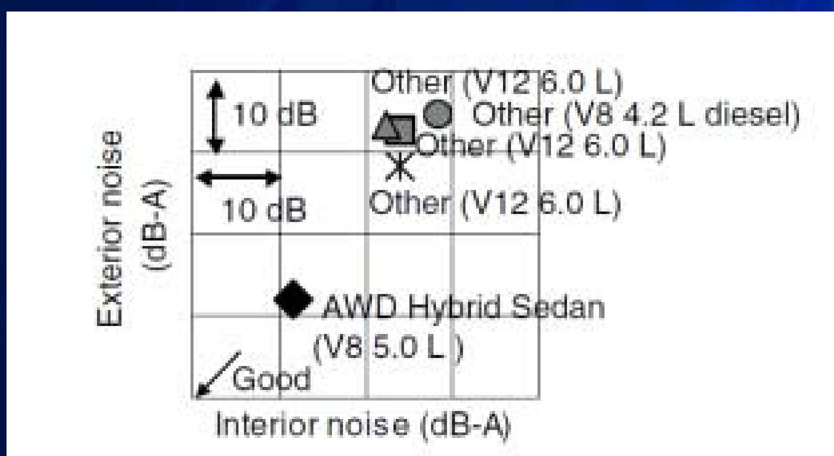


Reduce Inverter Water Pump Loads:

1. Reduce pump impeller imbalance
2. Redesign bearing structure
3. Changing motor structure

Reduce Source



Effective Isolation

Improve Isolation from Body:

1. Install rubber isolator
2. Increase mounting bracket rigidity
3. Improve Inverter case

Attach. Stiffness

Downstream

Engine Start Vibration

Root Cause Diagnostics:

1. Engine torque fluctuations
2. Engine torque reaction forces

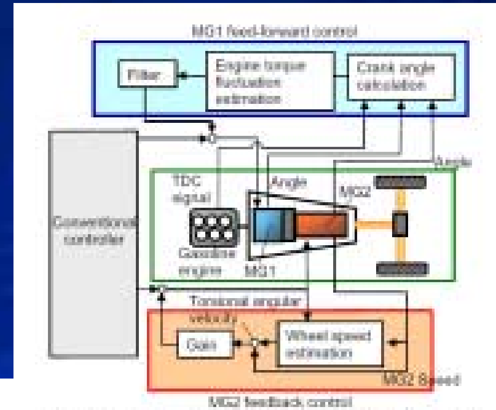
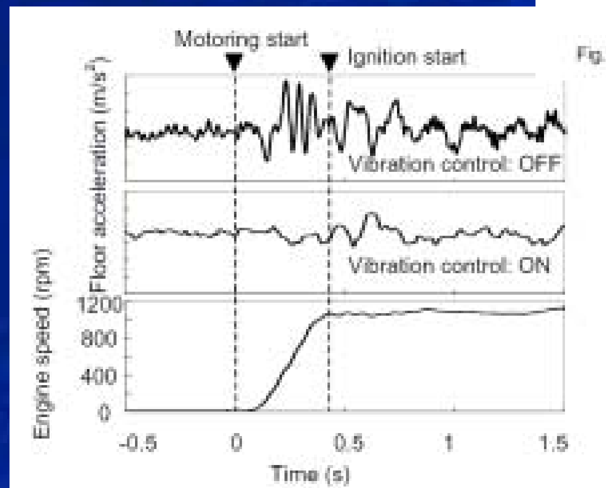


Fig. 7 Motor Control System for Vibration Reduction

Reduce Torque Fluctuation:

1. Change intake valve closing timing
2. Control piston stop position
3. Adjust injected fuel volume and ignition timing



Reduce Effect of Engine Loads:

1. Operating MG1 at high torque during engine start
2. Implement vibration-reducing motor control
3. Use two stage hysteretic torsional damper
4. Shorten distance between principal elastic axis and center of gravity of power plant

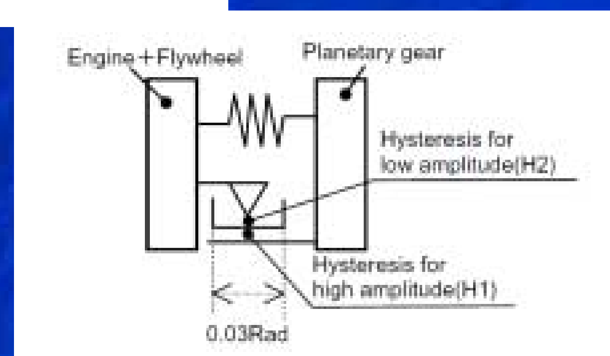


Fig. 11 Tosional Damper Concept Chart

Engine Start Vibration

Root Cause Diagnostics:

1. Engine torque fluctuations
2. Engine torque reaction forces

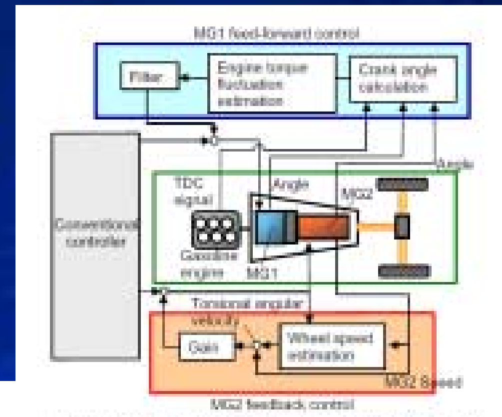


Fig. 7 Motor Control System for Vibration Reduction

Reduce Torque Fluctuation:

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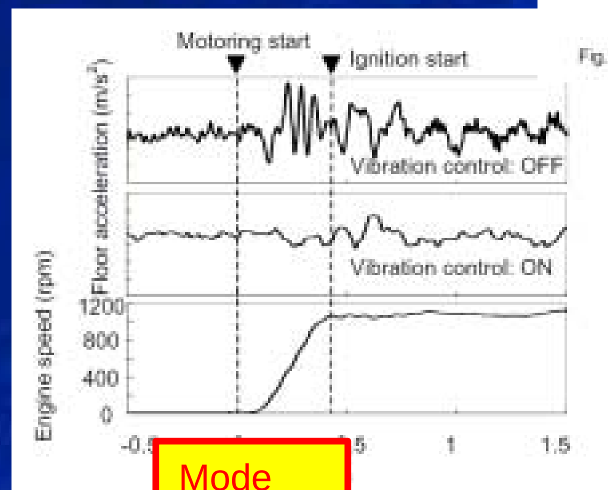
3. Adjust injected fuel volume and ignition timing

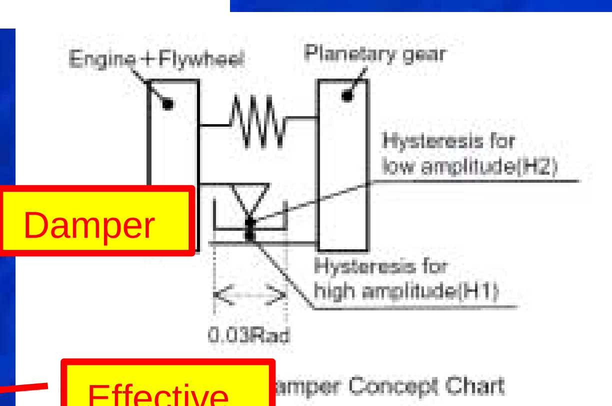
Reduce Source



Mode Manage.

Reduce Effect of Engine Loads:

1. Operating MG1 at high torque during engine start
2. Implement vibration-reducing motor control
3. Use two stage hysteretic torsional damper
4. Shorten distance between principal elastic axis and center of gravity of power plant



Damper

Effective Isolation

2nd Order Engine Induced Boom

Root Cause Diagnostics:

1. 2nd order couple of the reciprocating inertia of piston
2. THS II Trans 50 mm longer and 35 kg heavier
3. Lower power plant bending mode
4. Requires 1.5X higher mount rates

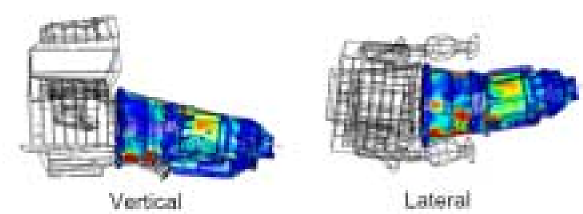
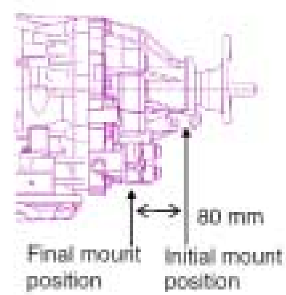
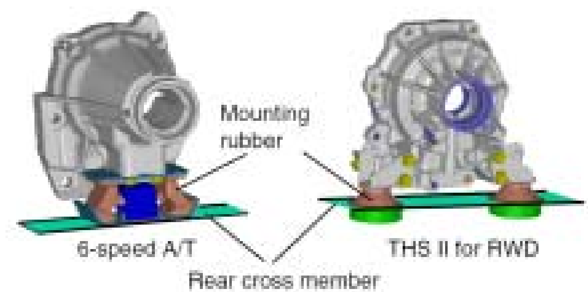


Fig. 14 Vibration Mode of Power Plant Bending Resonance





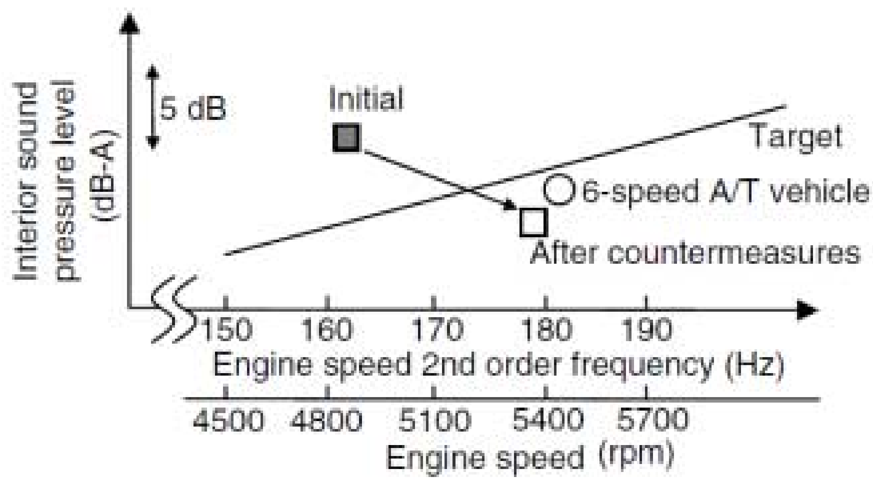


Fig. 17 Interior Booming Noise Level Caused by the Second Order Component of Engine Speed and Power Plant Bending Resonance Frequency

Reduce Effect of 2nd order Couple:

1. Increase power plant bending mode
2. Move mount to a nodal point
3. Embed mounts inside cross member
4. Reduces distance from principle elastic axis to CG
5. Optimized vertical to lateral rate ratio

2nd Order Engine Induced Boom

Root Cause Diagnostics:

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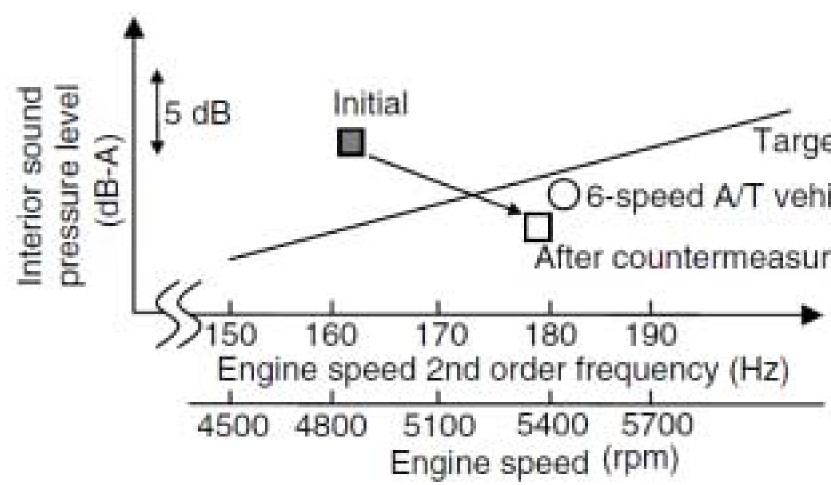
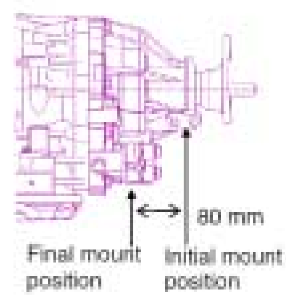


Fig. 17 Interior Booming Noise Level Caused by the Second Order Component of Engine Speed and Power Plant Bending Resonance Frequency



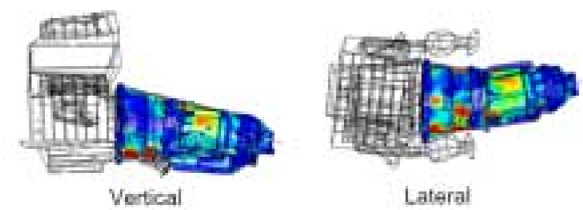
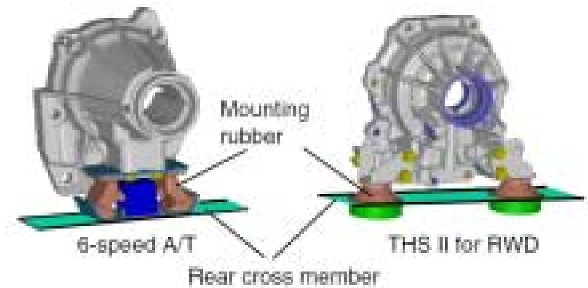


Fig. 14 Vibration Mode of Power Plant Bending Resonance



Reduce Effect of 2nd order

1. Increase power plant bending mo
2. Move mount to a nodal point
3. Embed mounts inside cross mem
4. Reduces distance from principle elastic axis to CG
5. Optimized vertical to lateral rate ratio

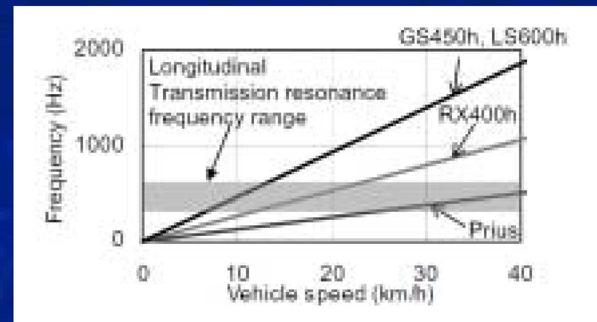
Mode Manage.

Nodal Mounting

Effective Isolation

Engine Radiated Noise

Root Cause Diagnostics:

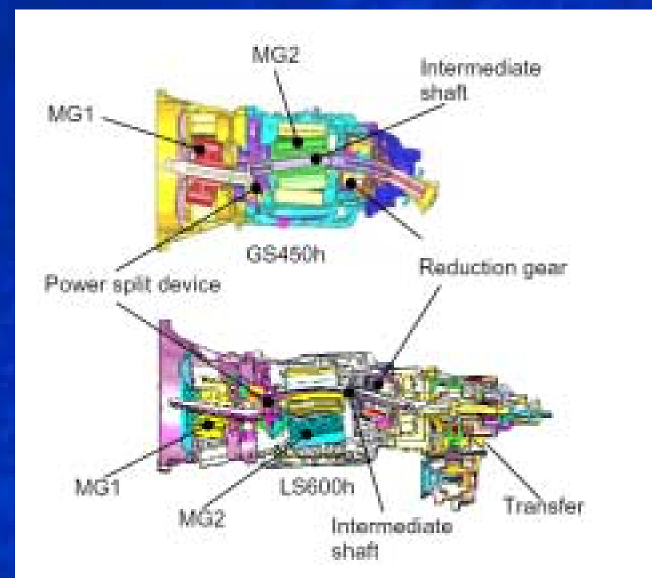


1. 24th MG2 order excitation
2. Lower MG2 reduction gear ratio
3. Lower transmission bending mode
   - two key modes identified

Reduce 24th order Loads:

Arranged permanent magnets in V shape with optimized angle

Reduce Effect of 24th order Loads:

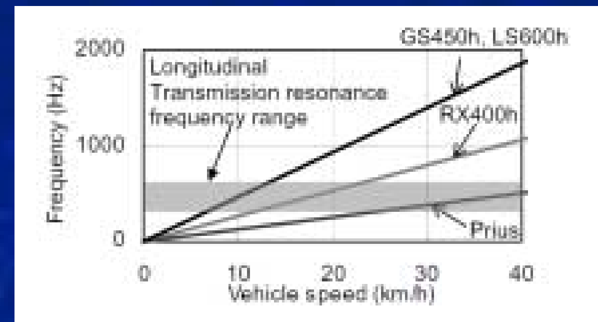


1. Modified Trans case to improve modes
2. Added dynamic damper at a high amp. point
   on Trans case
3. Added ribs in high radiating area of Trans case

Engine Radiated Noise

Root Cause Diagnostics:

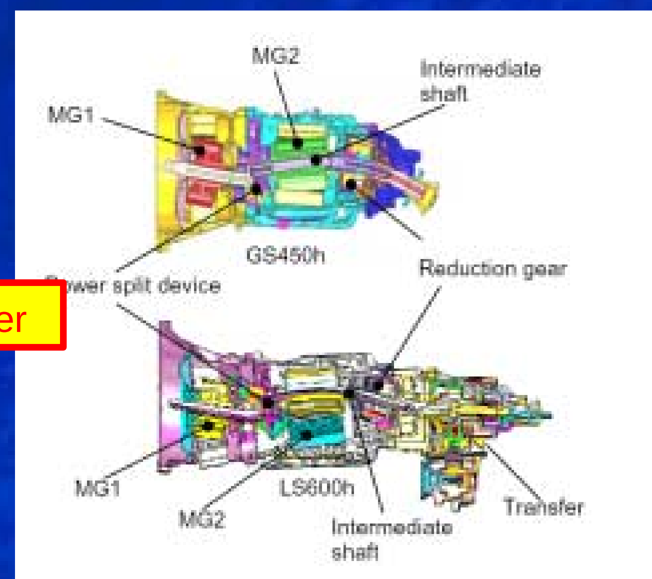
1. 24th MG2 order excitation
2. Lower MG2 reduction gear ratio
3. Lower transmission bending mode
   - two key modes identified



Reduce 24th order Loads:

Reduce Source

Arranged permanent magnets in V shape with optimized angle



Reduce Effect of 24th order Loads:

Mode Manage.

1. Modified Trans case to improve modes
2. Added dynamic damper at a high amp. point on Trans case
3. Added ribs in high radiating area of Trans case

Damper

Downstream

Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- Real World Application Example
- Closing Remarks

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**

SAE 2009 NVH Conference Structure Borne NVH Workshop

Thank You for Your Time!

Q & A

Structure Borne NVH References

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2. A. E. Duncan, et. al., “MSC/NVH_Manager Helps Chrysler Make Quieter Vibration-free Vehicles”, Chrysler PR Article, March 1998.

3. B. Dong, et. al., “Process to Achieve NVH Goals: Subsystem Targets via ‘Digital Prototype’ Simulations”, SAE 1999-01-1692, NVH Conference Proceedings, May 1999.

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Structure Borne NVH Workshop - on Internet

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

WS + Refs. at www.AutoAnalytics.com/papers.html

Structure Borne NVH References

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