SAE 2011 NVH Conference
Structure Borne NVH Workshop

Presenters:

Alan Duncan — **Altair Engineering @ Honda**
Contact Email: aeduncan@autoanalytics.com
Web: www.autoanalytics.com

Greg Goetchius — **Tesla Motors**

Kiran Govindswamy — **FEV Inc.**

Jianmin Guan — **Altair Engineering**

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***
- Case Studies
  - Interior Noise Diagnosis using VINS
  - NVH Principles: Applications
- Closing Remarks

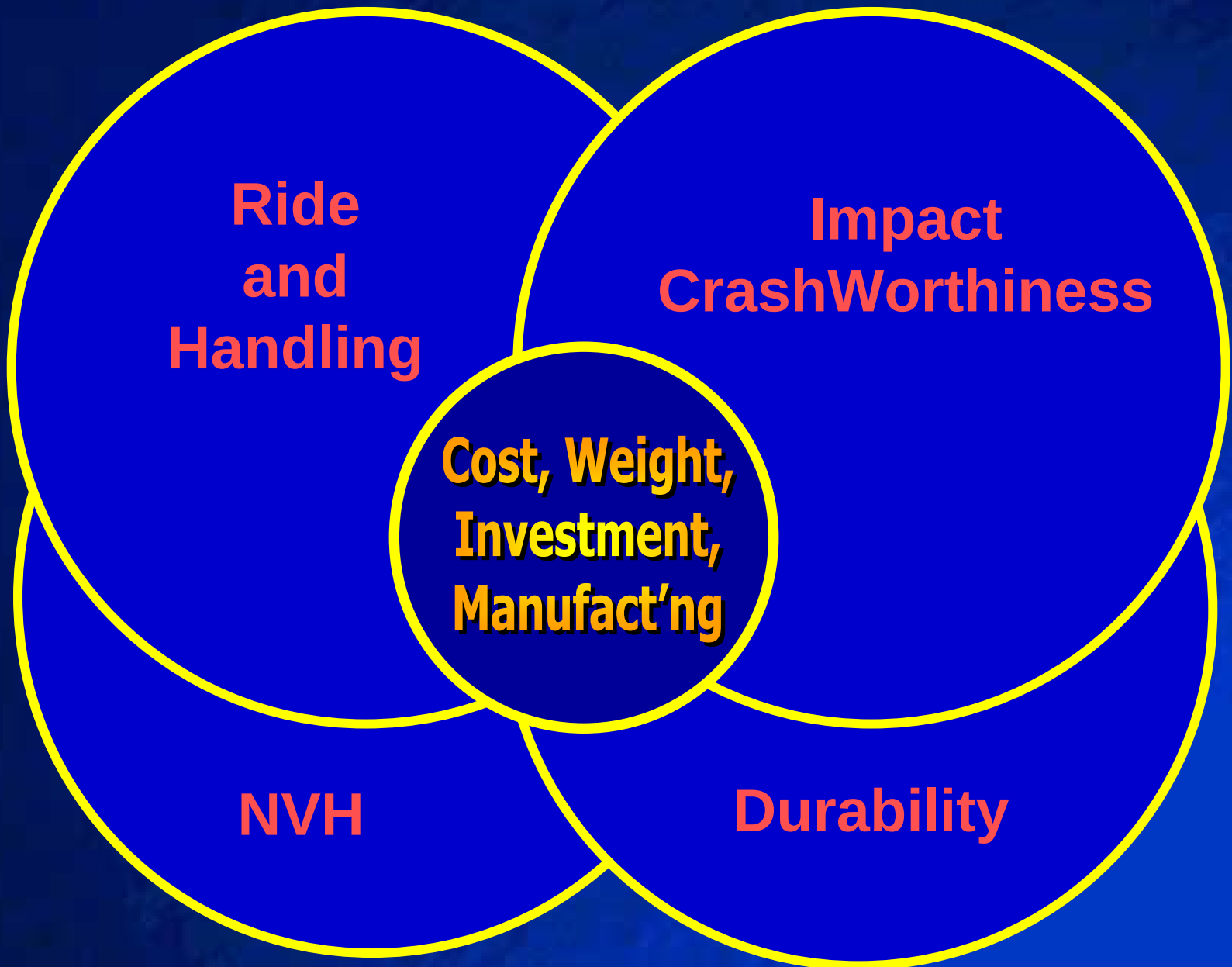
Structure Borne NVH Workshop

- **Introduction** **Greg Goetchius**
- **Low Frequency Basics**
- **Mid Frequency Basics**
- ***Live Noise Attenuation Demo***
- **Case Studies**
  - **Interior Noise Diagnosis using VINS**
  - **NVH Principles: Applications**
- **Closing Remarks**

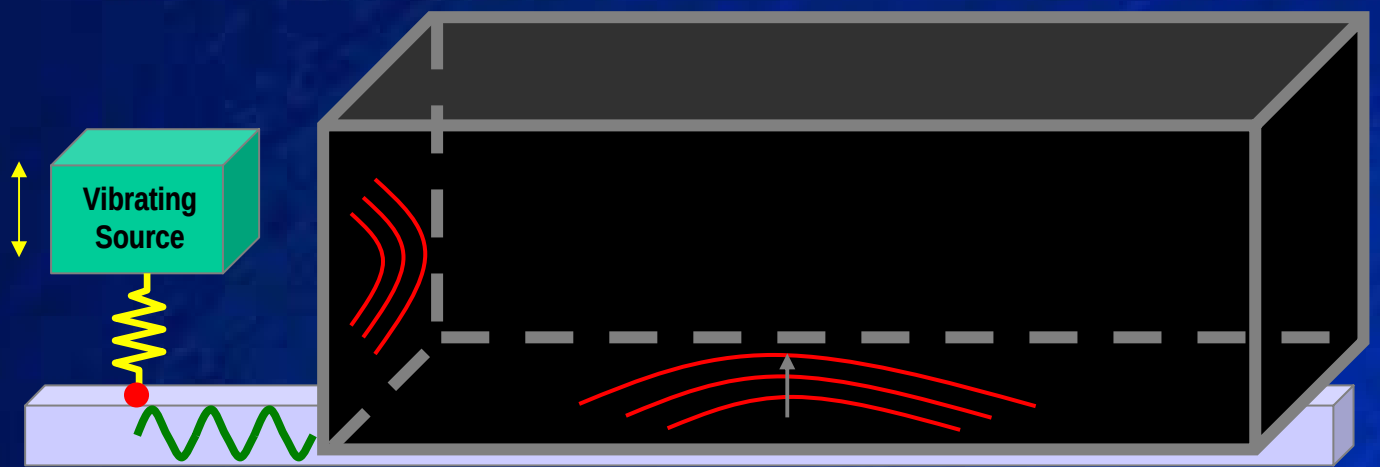
Structure Borne NVH Workshop

- **Introduction**
- **Low Frequency Basics**
- **Mid Frequency Basics**
- ***Live Noise Attenuation Demo***
- **Case Studies**
  - **Interior Noise Diagnosis using VINS**
  - **NVH Principles: Applications**
- **Closing Remarks**

Competing Vehicle Design Disciplines



Structure Borne Noise and Vibration

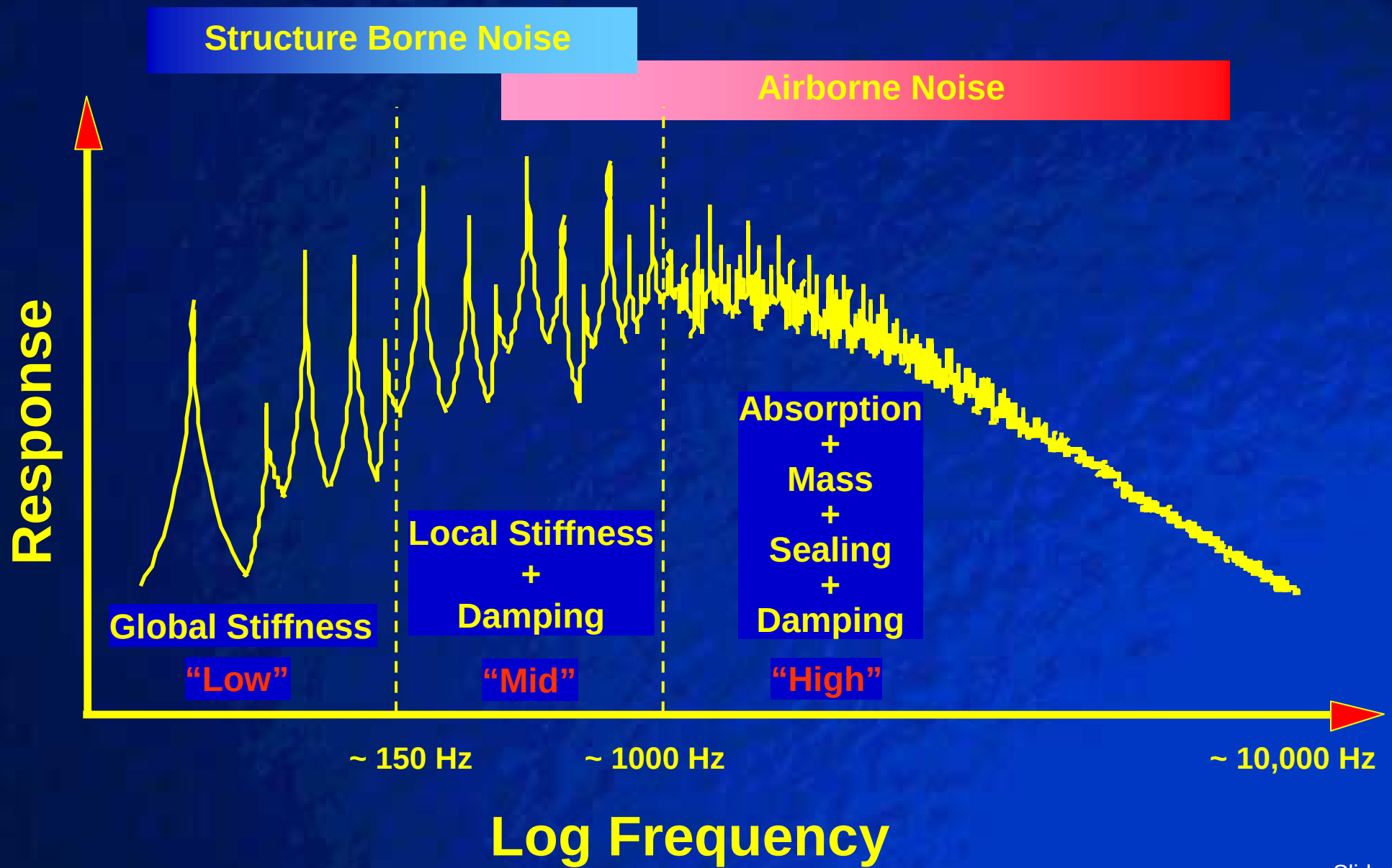


Frequency Range: up to 1000 Hz

System Characterization

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

Automotive NVH Frequency Range



Structure Borne NVH Workshop

- Introduction
- **Low Frequency Basics**
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- Case Studies
  - Vehicle Interior Noise Simulation
  - NVH Principles: Applications
- Closing Remarks

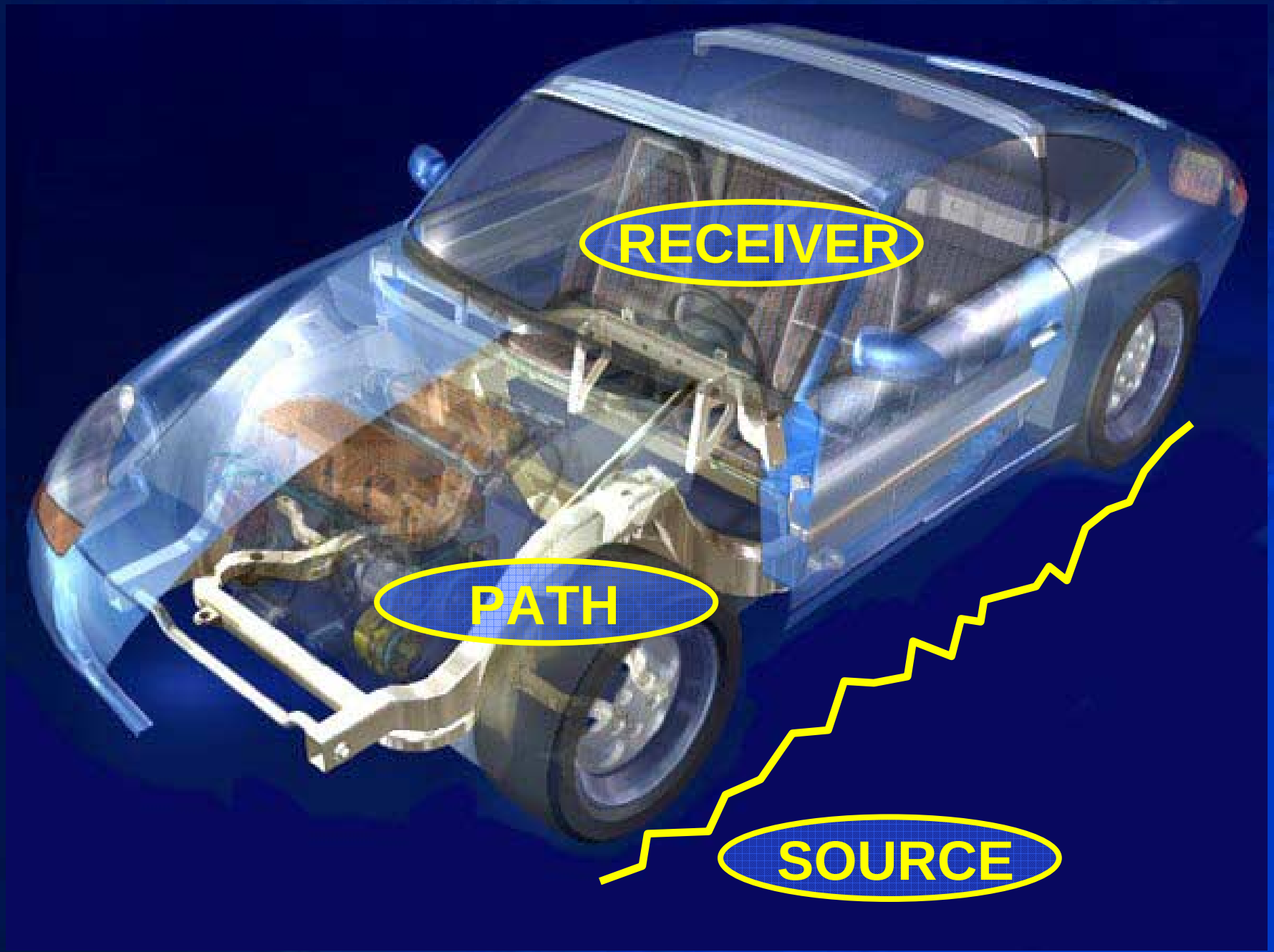
Low Frequency Basics

- 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

Low Frequency Basics

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

Structure Borne NVH Basics



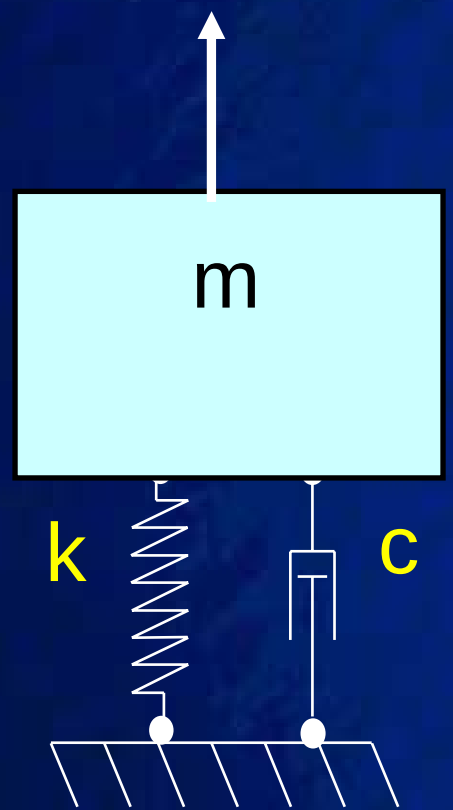
Low Frequency Basics

- 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

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

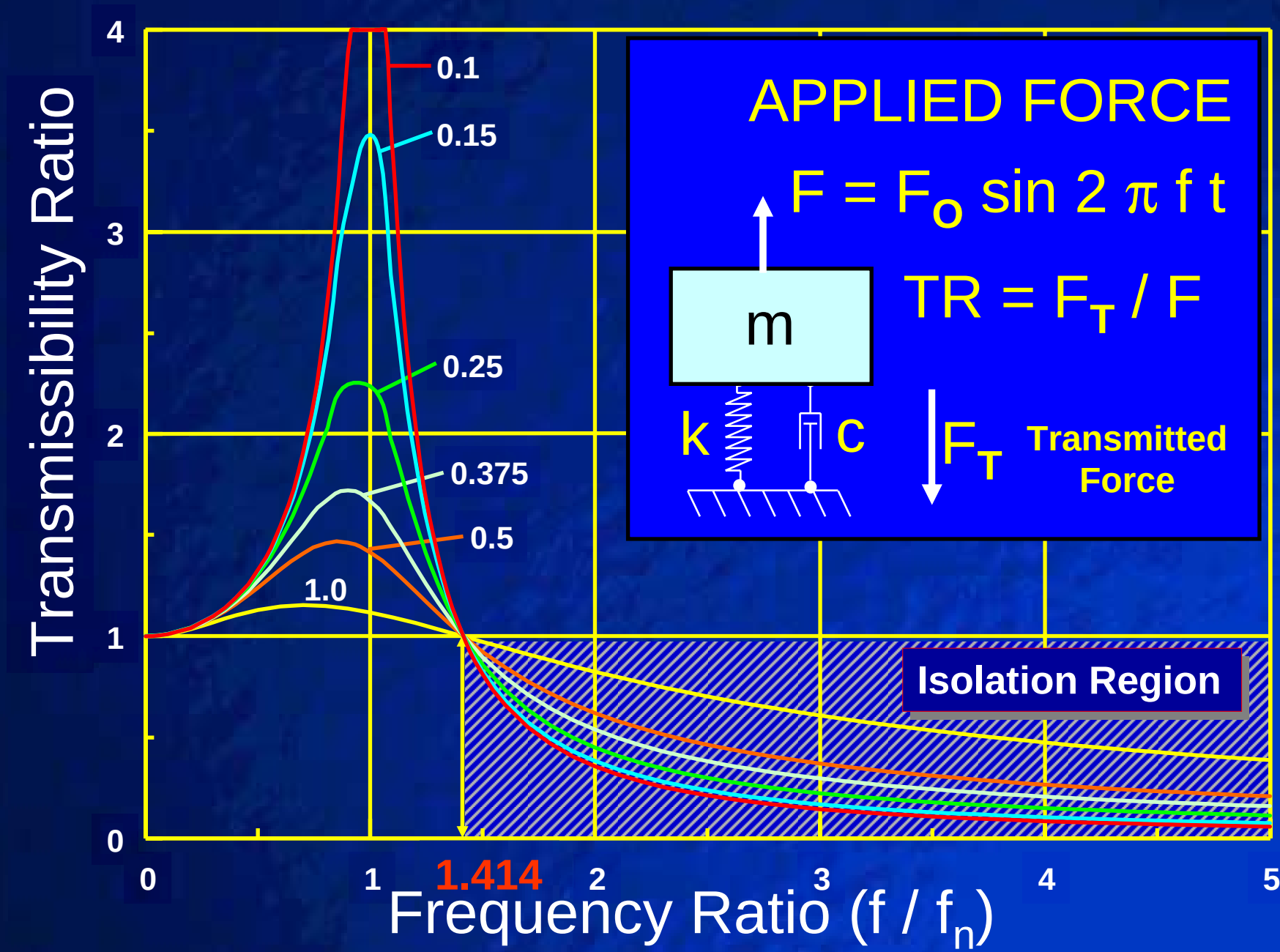
F_T

ζ = fraction of critical damping

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

f = operating frequency

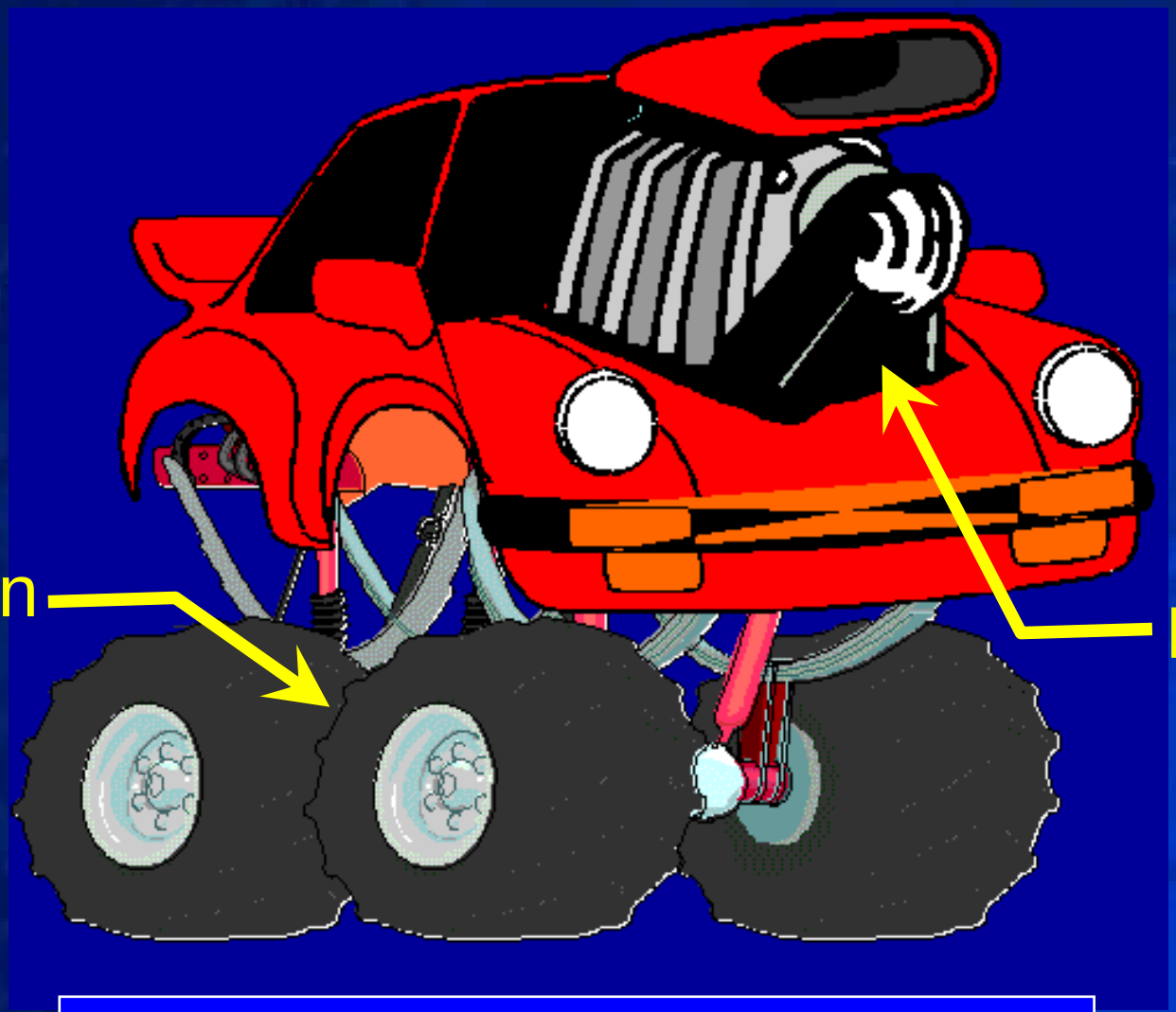
Vibration Isolation Principle



Low Frequency Basics

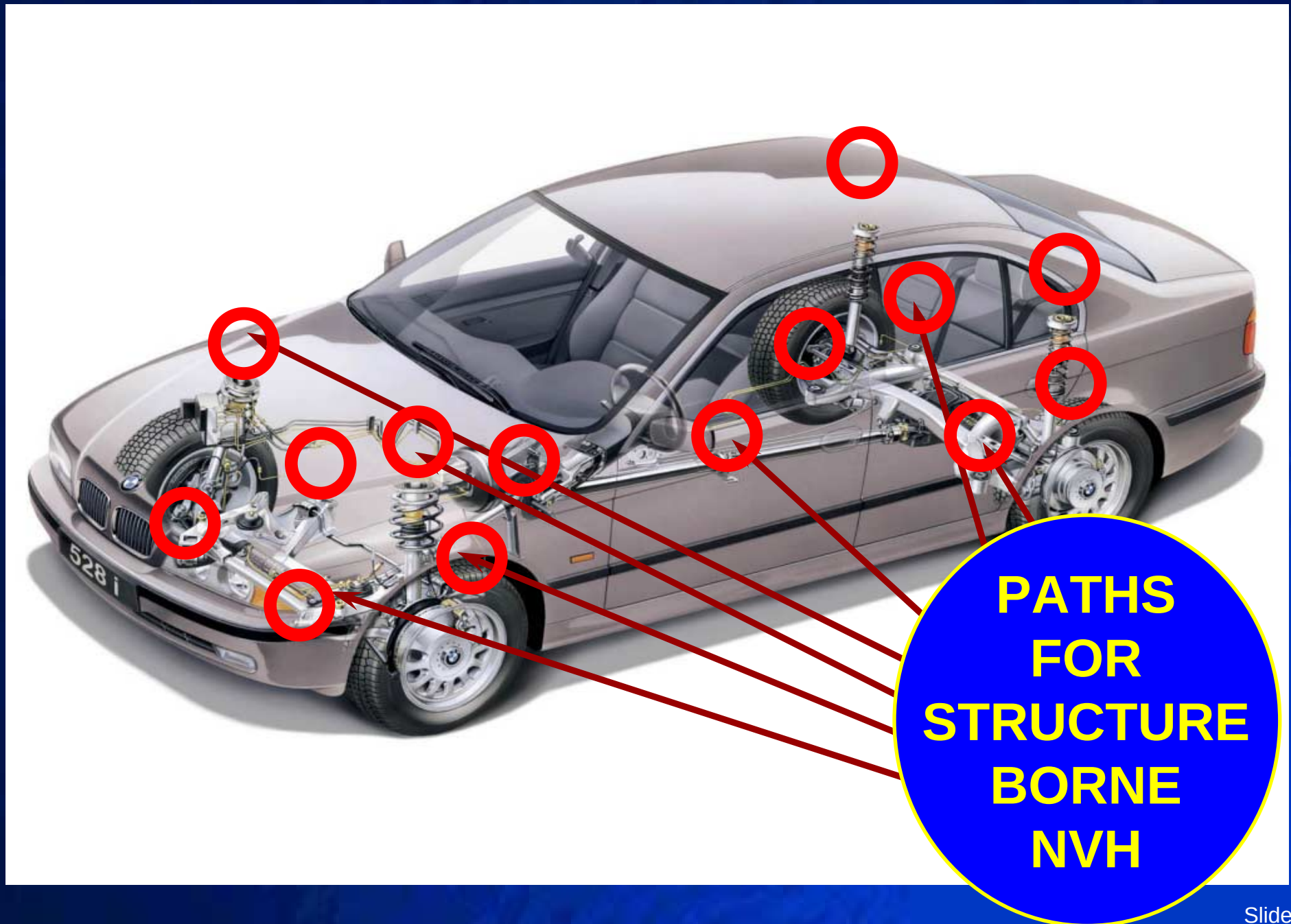
- Source-Path-Receiver Concept
- Single DOF System Vibration
- **NVH Source Considerations**
- Receiver Considerations
- Vibration Attenuation Strategies
  - Provide Improved Isolation
  - Mode Management
  - Nodal Point Mounting
  - Dynamic Absorbers

NVH Source Considerations



Two Main Sources

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.

Low Frequency Basics

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

Receiver Considerations

Subjective to Objective Conversions

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

Receiver Sensitivity is a Key Consideration

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

Represents 1.0 Rating Change

TACTILE: 50% reduction in motion

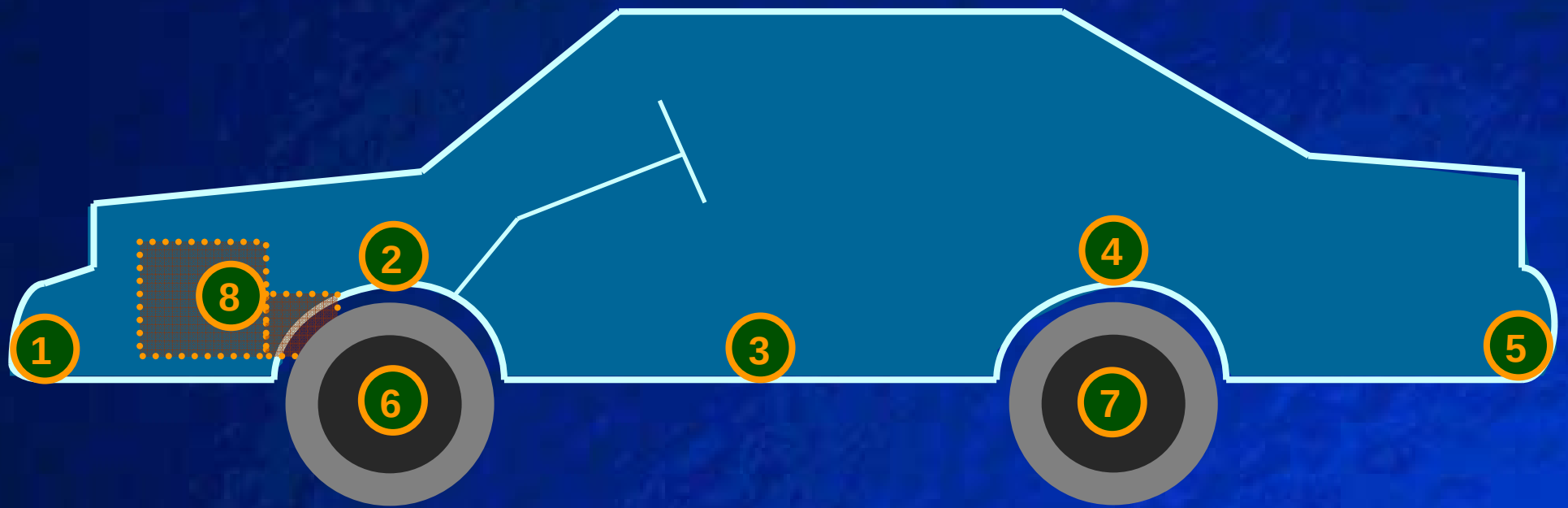
SOUND : 6.dB reduction in sound pressure level (long standing rule of thumb)

Low Frequency Basics

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

Symbolic Model of Unibody Passenger Car

8 Degrees of Freedom



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

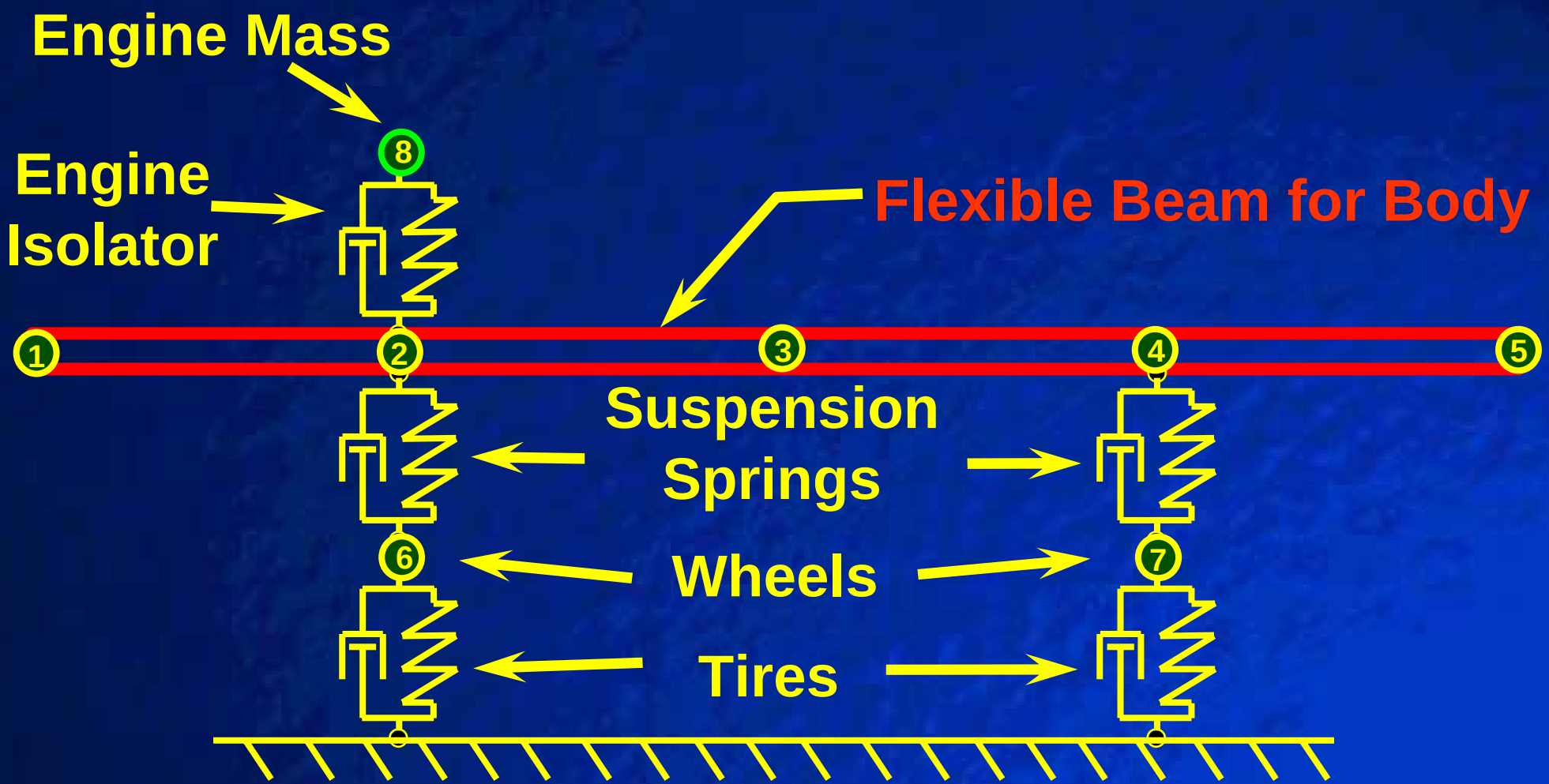
Tires	350.3	N/mm
KF	43.8	N/mm
KR	63.1	N /mm

Beam mass lumped on grids like a beam

M2,3,4 =2 * M1,5

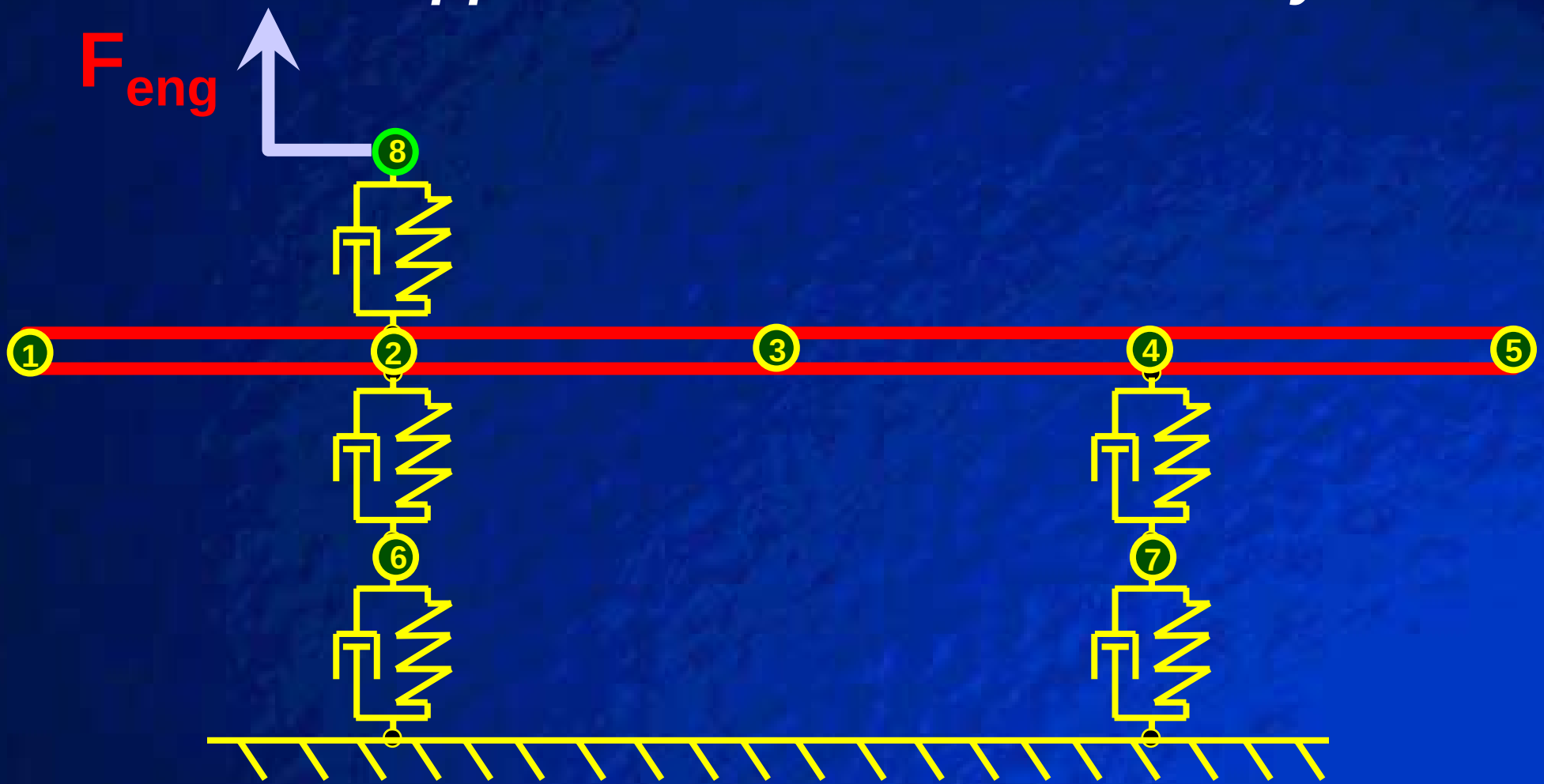
From Reference 6

8 Degree of Freedom Vehicle NVH Model



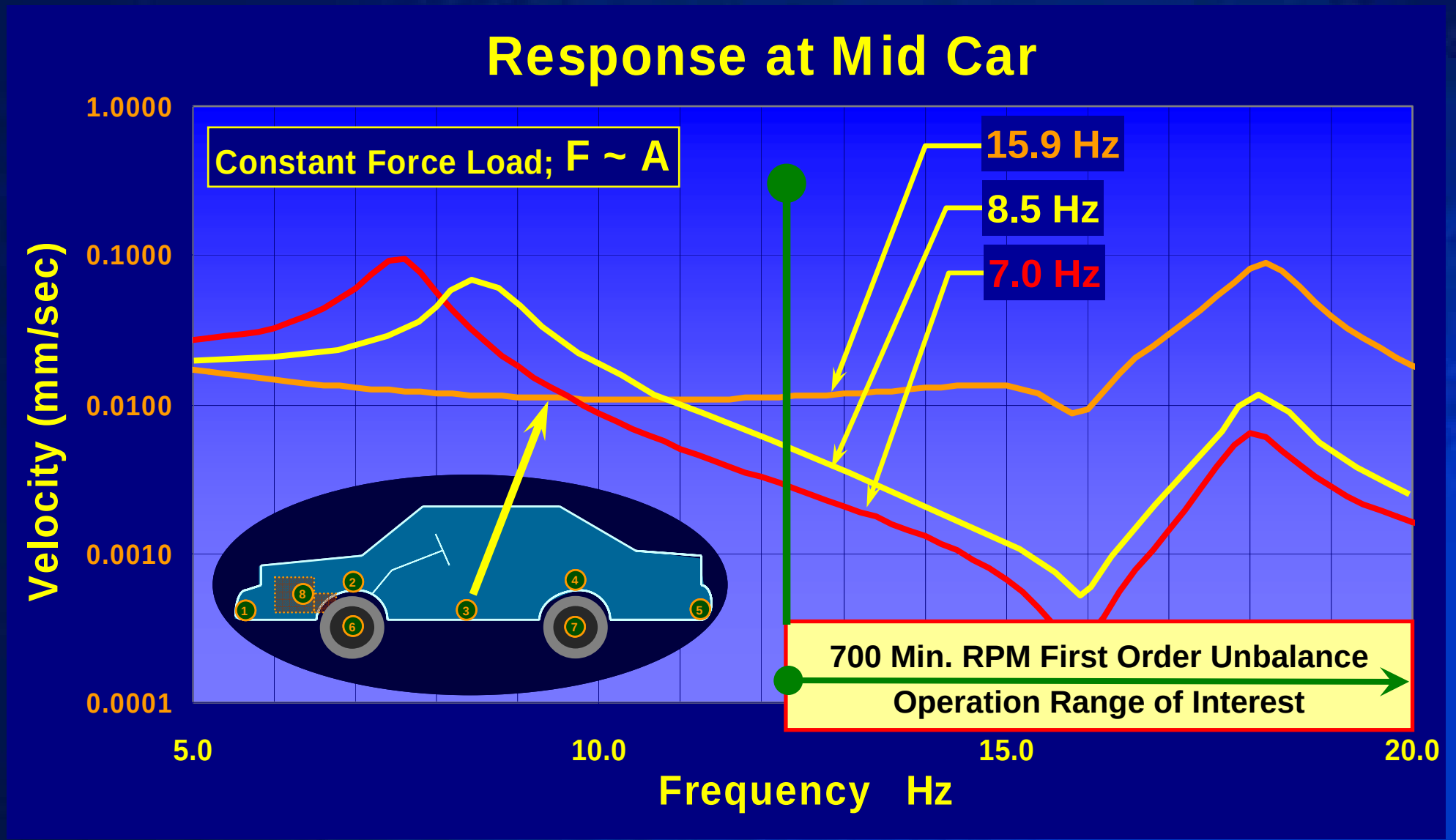
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



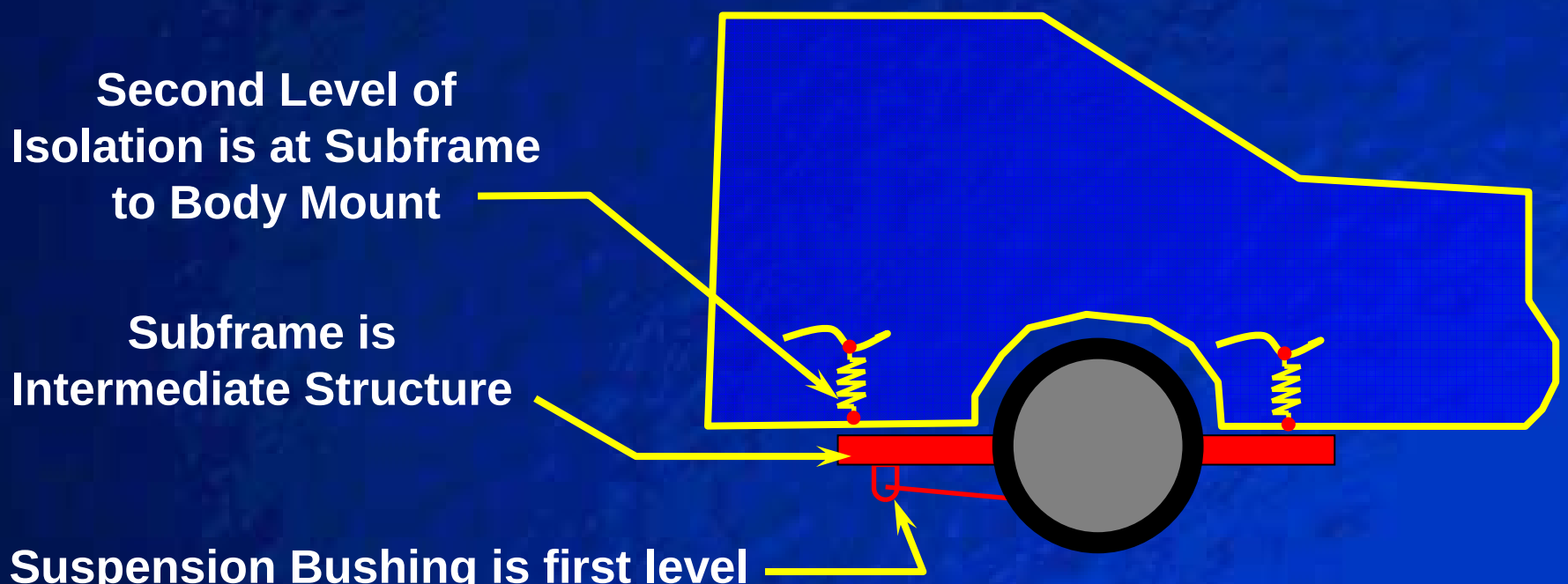
Forces at Powertrain could represent a First Order Rotating Imbalance

Engine Isolation Example



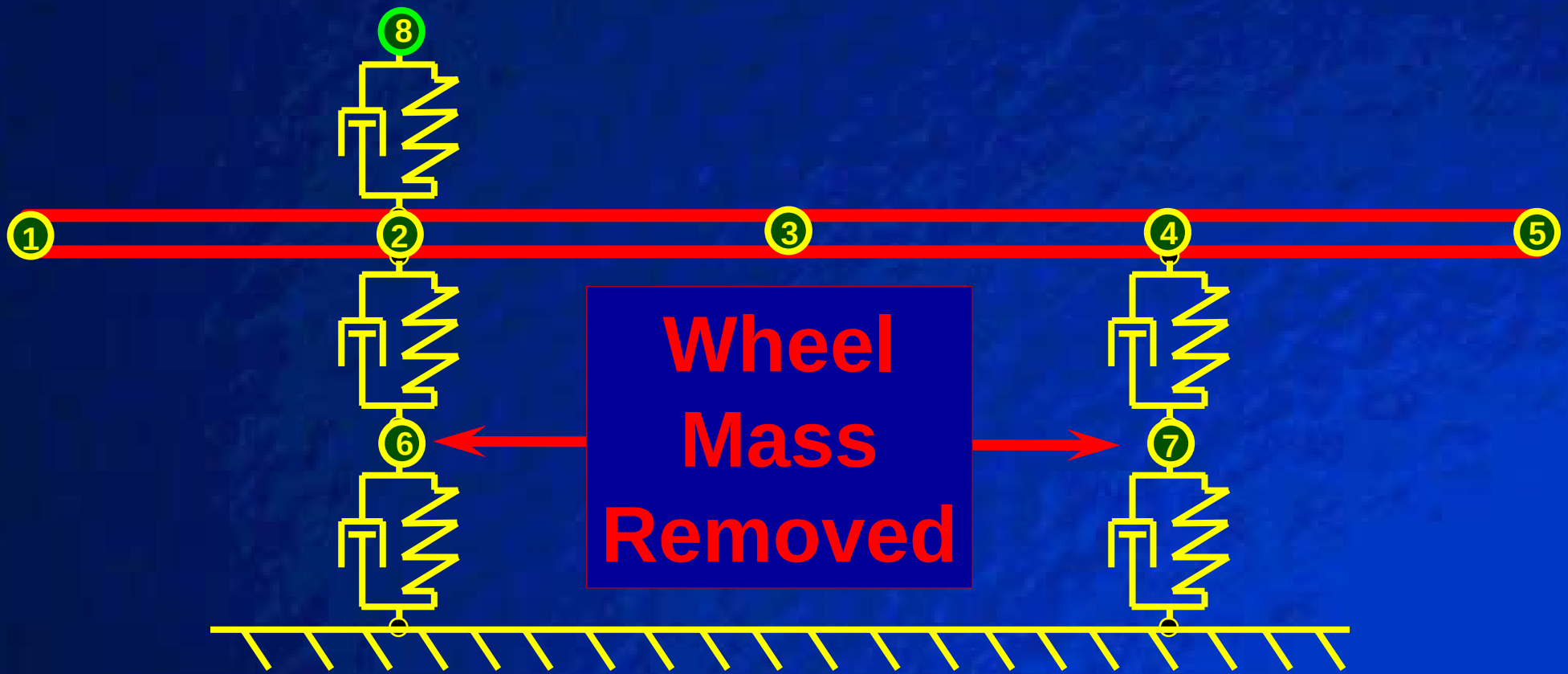
Concepts for Increased Isolation

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

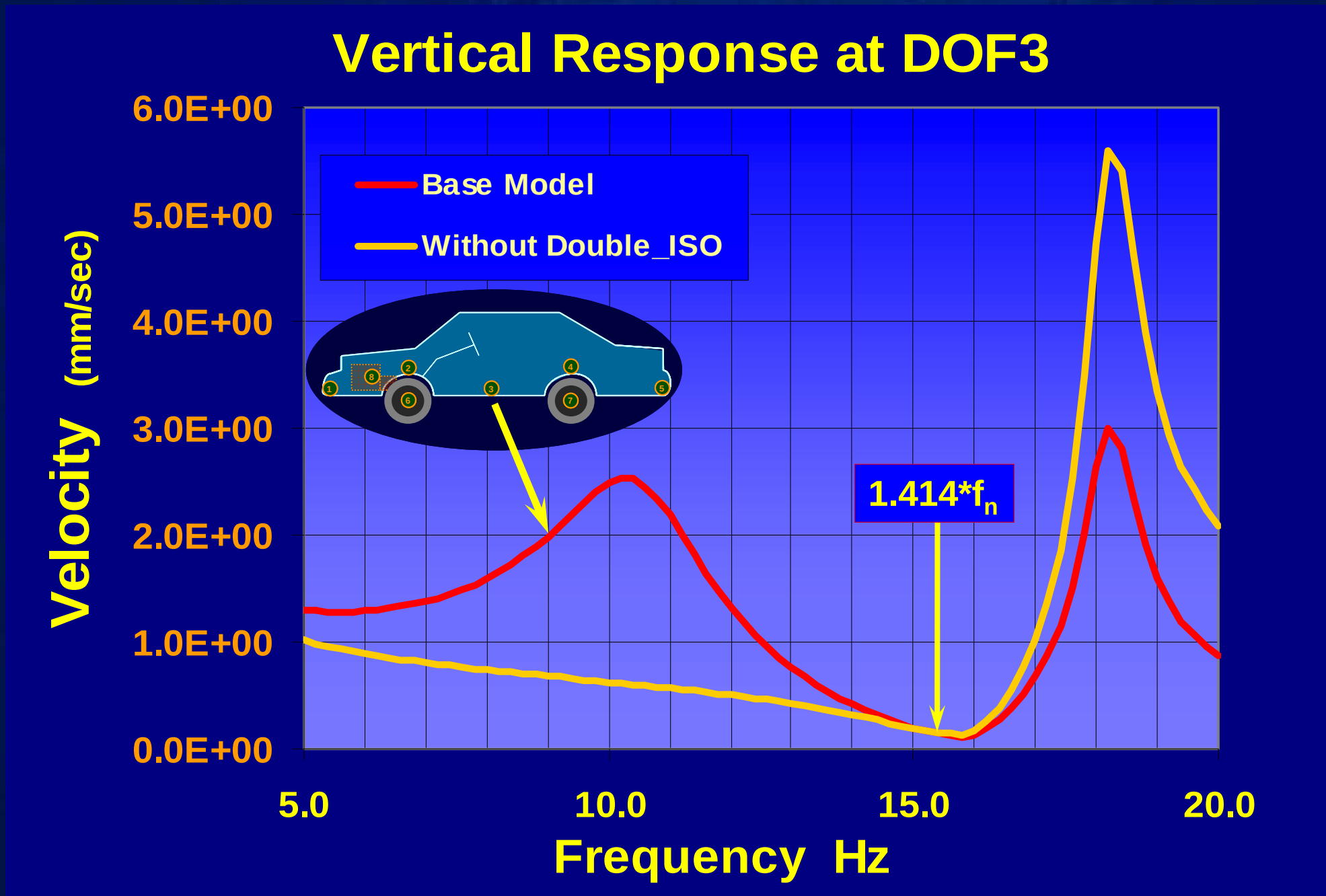


8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



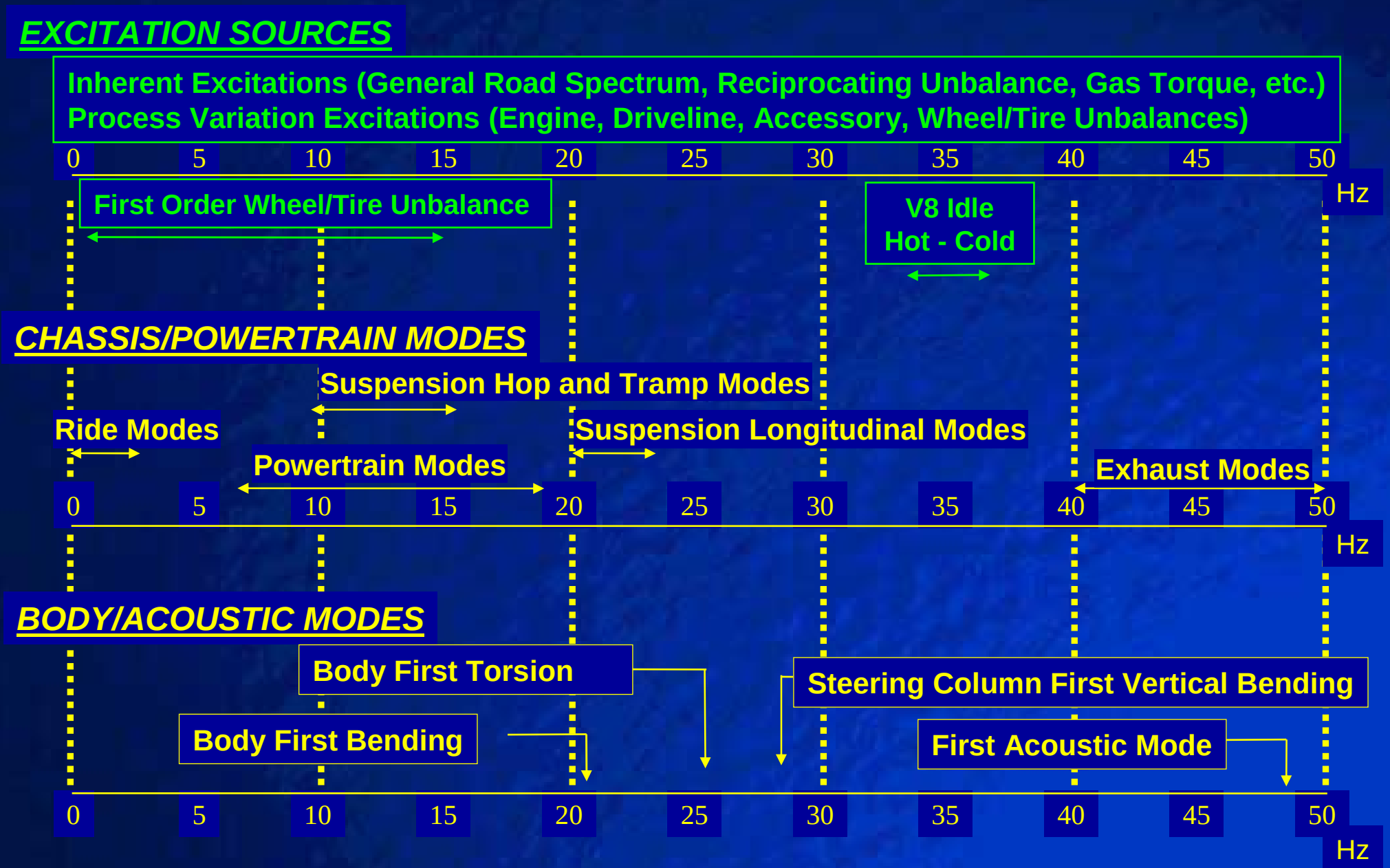
Double Isolation Example



Low Frequency Basics

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

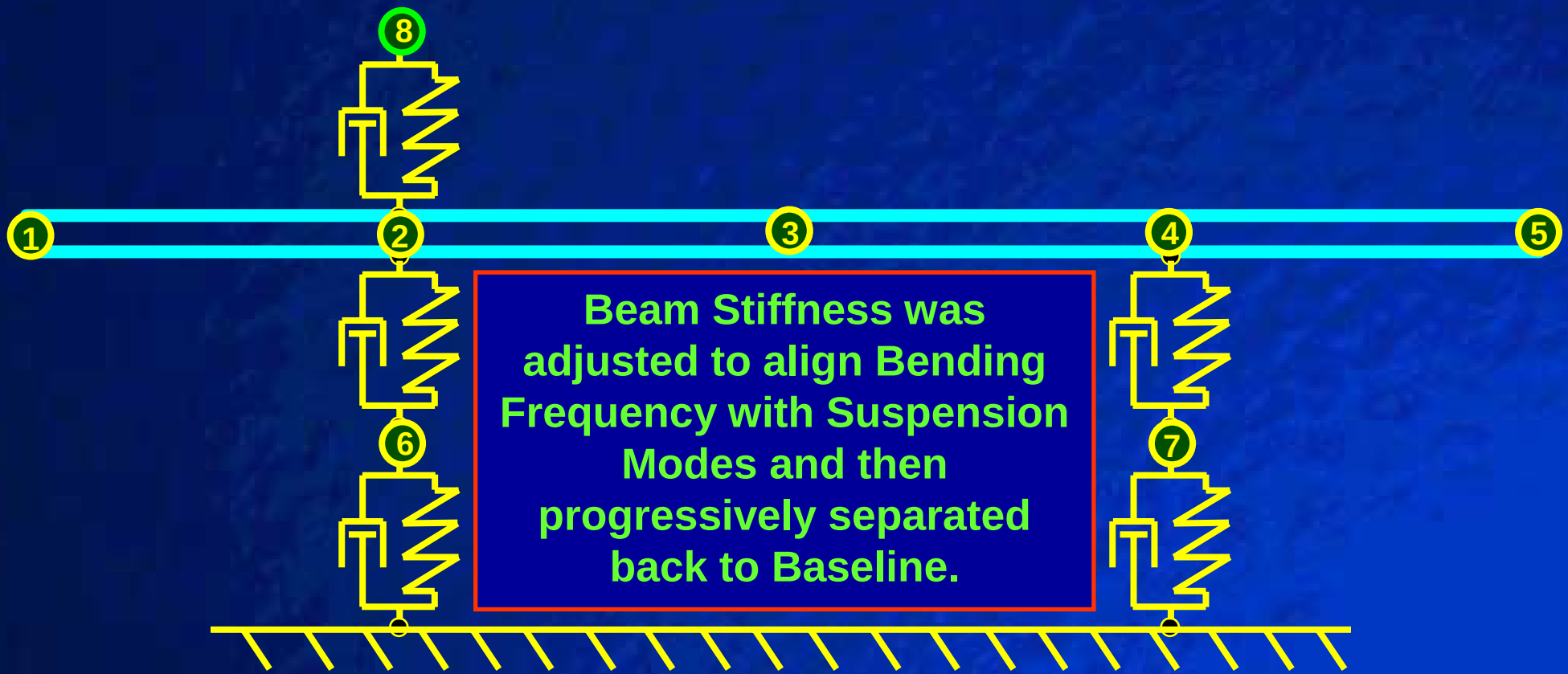
Mode Management Chart



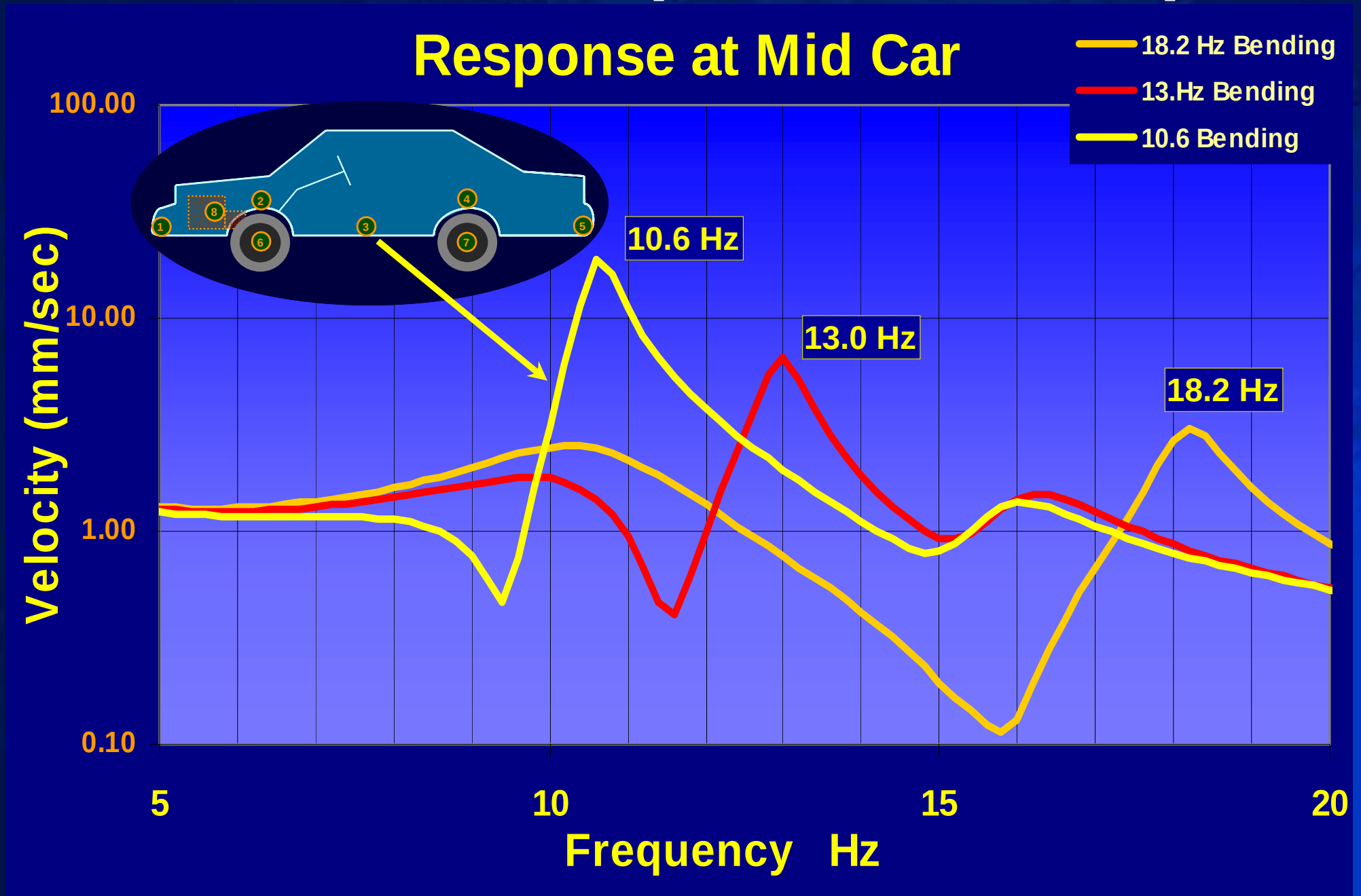
(See Ref. 1)

8 Degree of Freedom Vehicle NVH Model

Bending Mode Frequency Separation



8 DOF Mode Separation Example

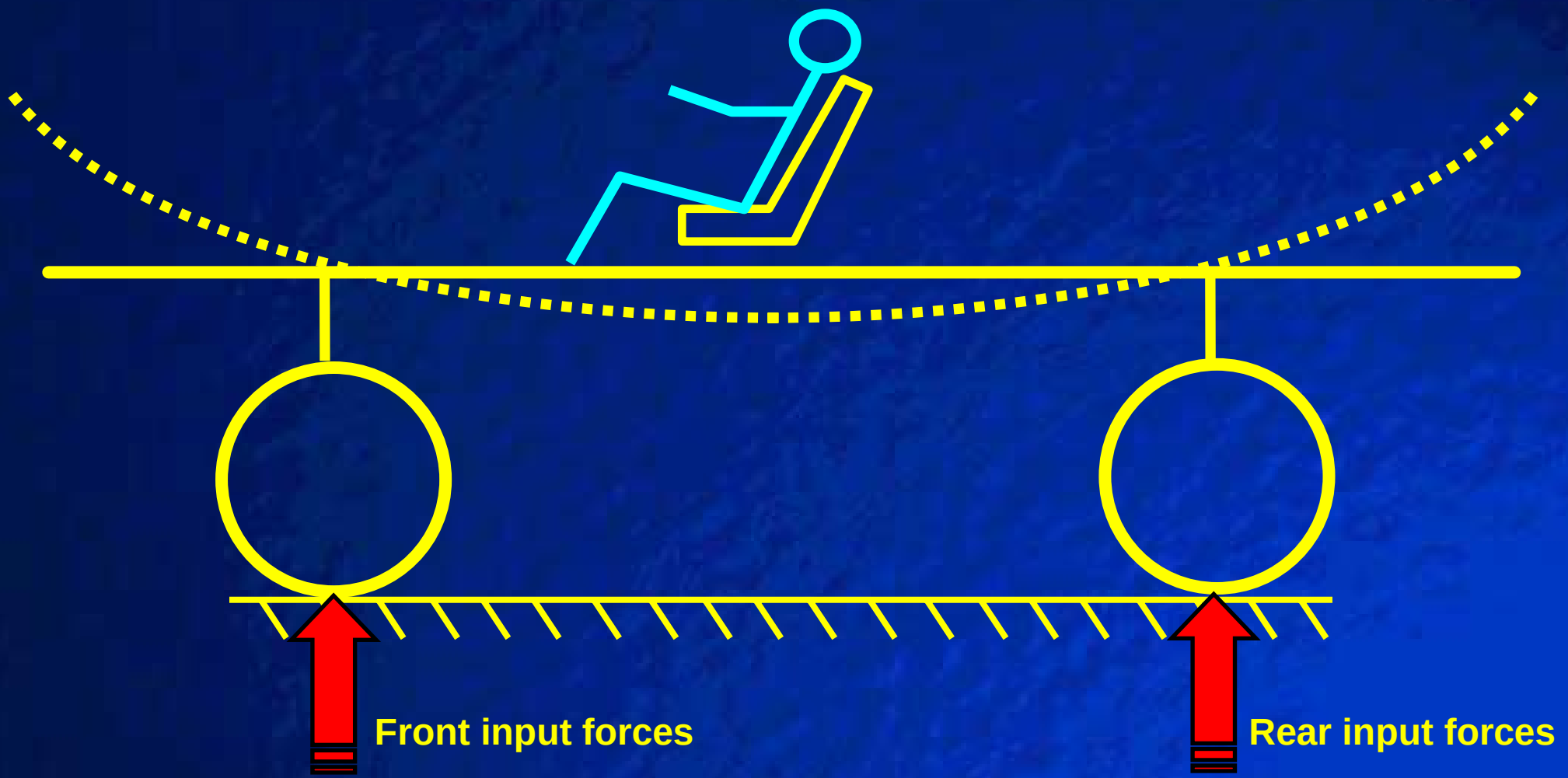


Low Frequency Basics

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

Mount at Nodal Point

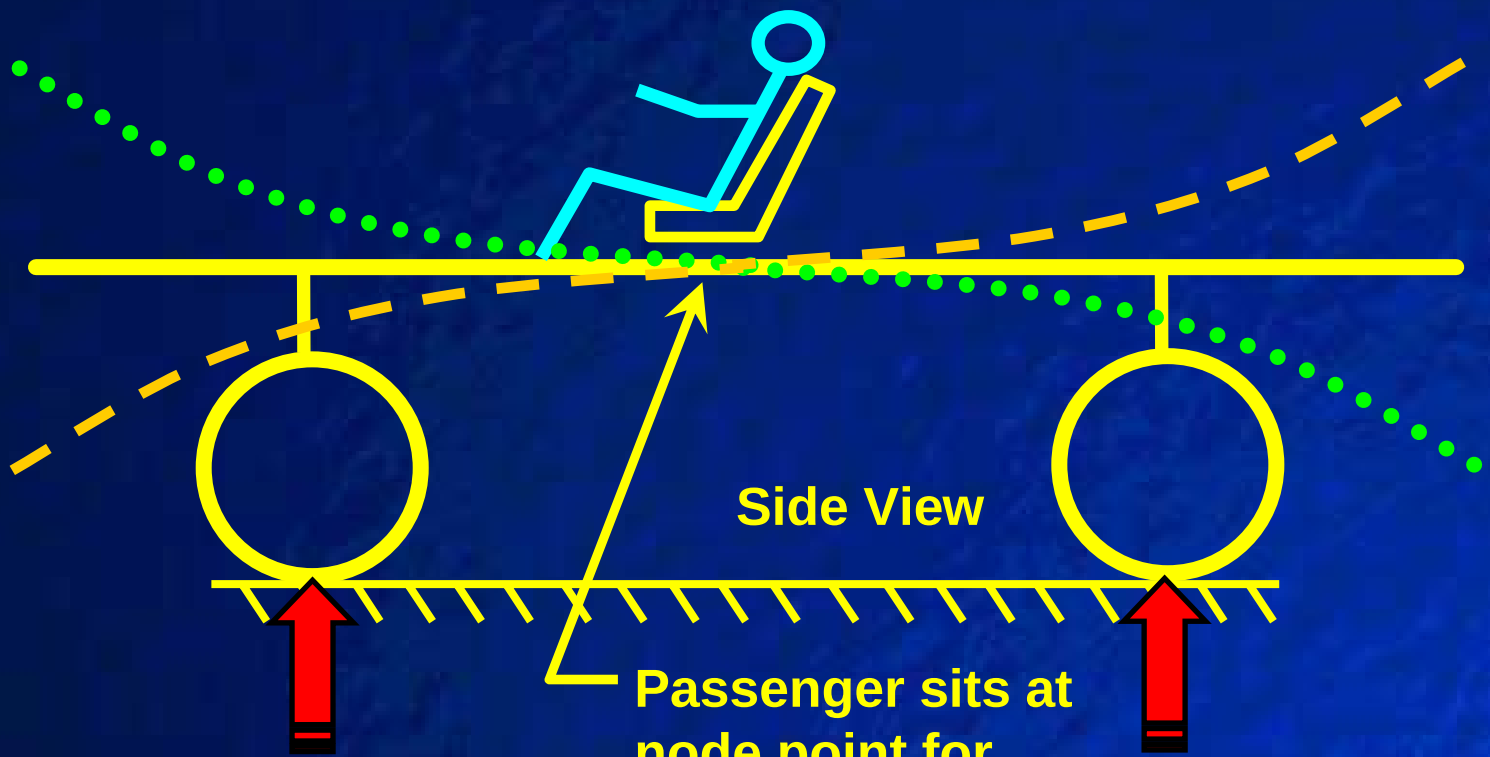
First Bending: Nodal Point Mounting Example

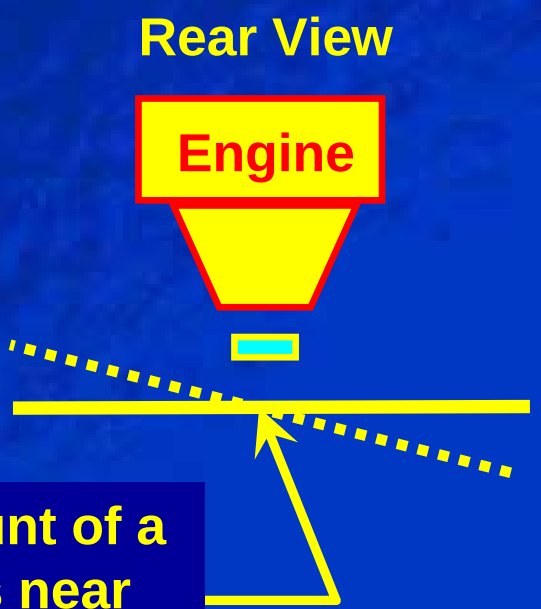


Locate wheel centers at node points of the first bending modeshape to prevent excitation coming from suspension input motion.

Mount at Nodal Point

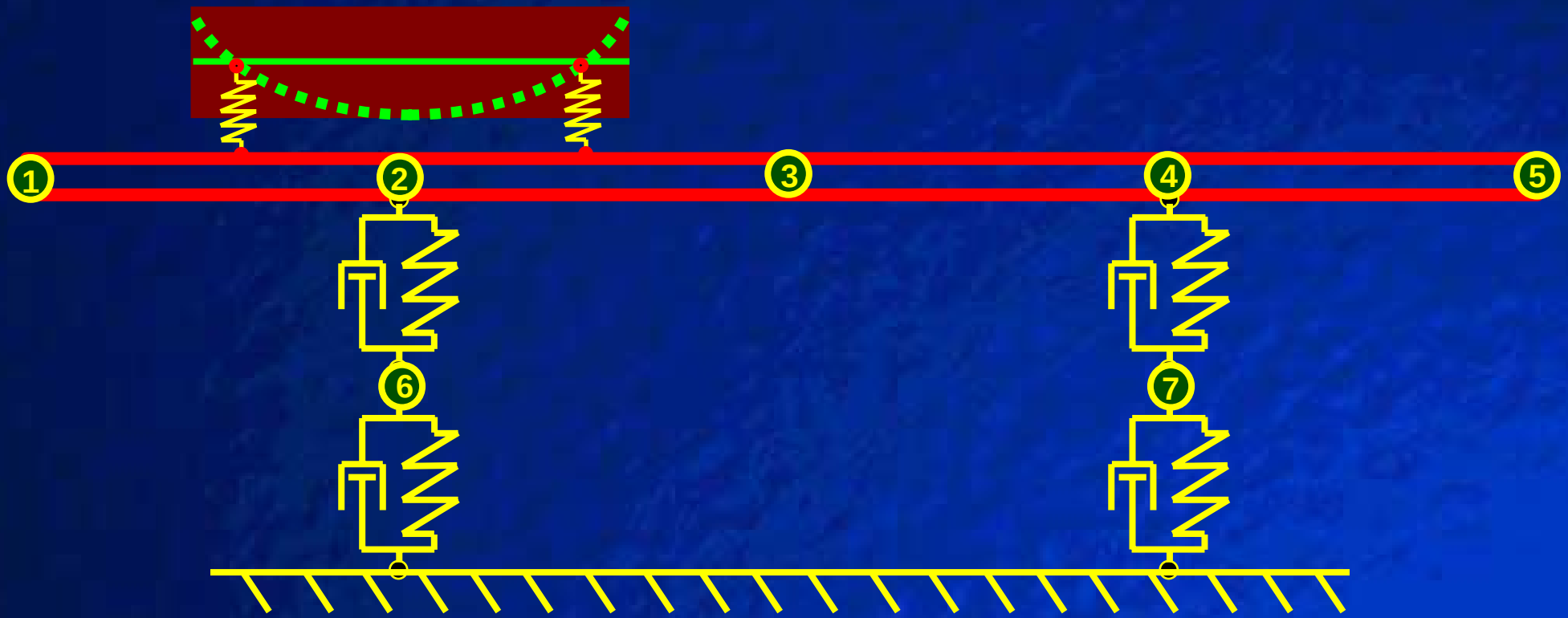
First Torsion: Nodal Point Mounting Examples





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

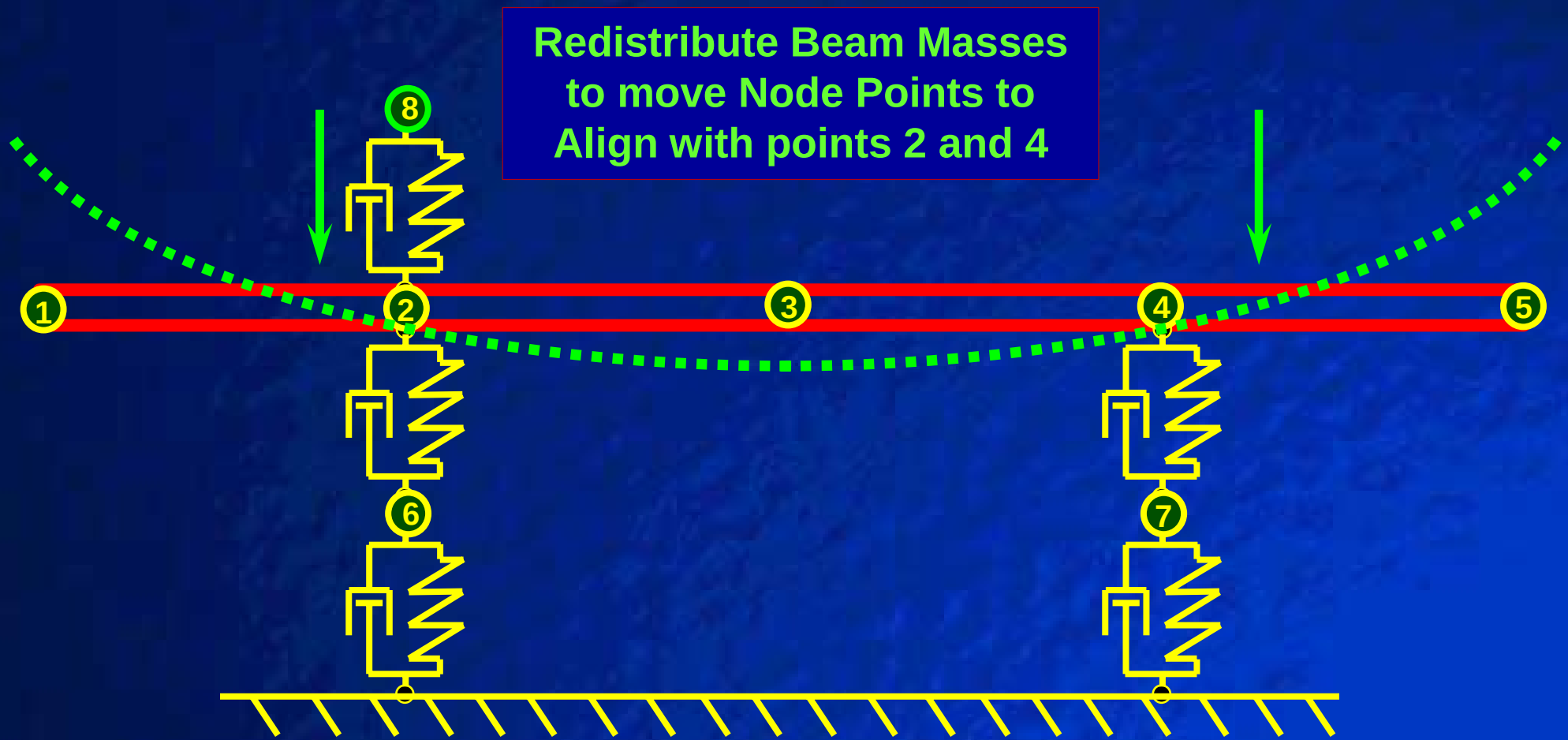
Powertrain Bending Mode Nodal Mounting



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.

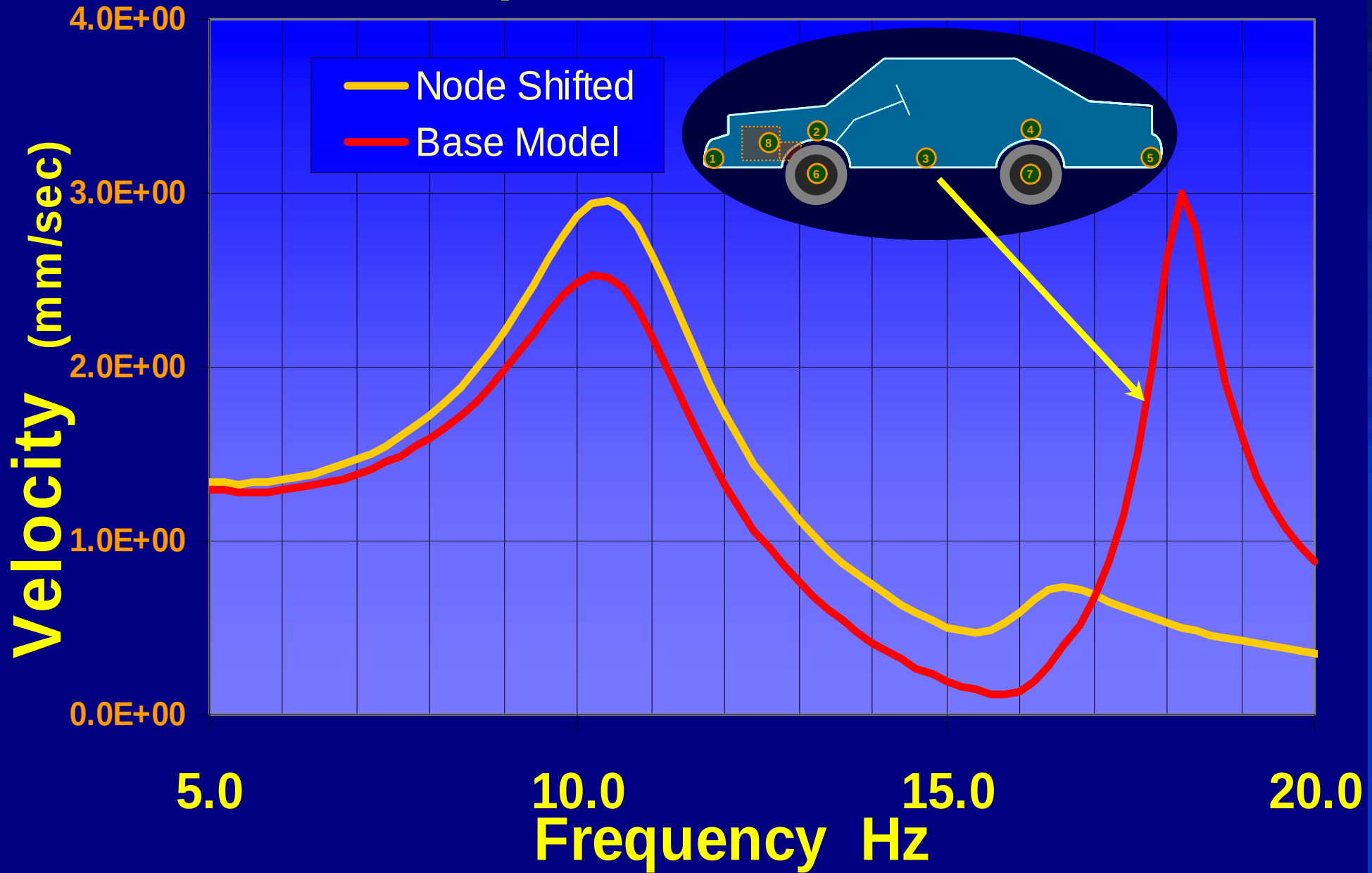
8 Degree of Freedom Vehicle NVH Model

Bending Node Alignment with Wheel Centers



First Bending Nodal Point Alignment

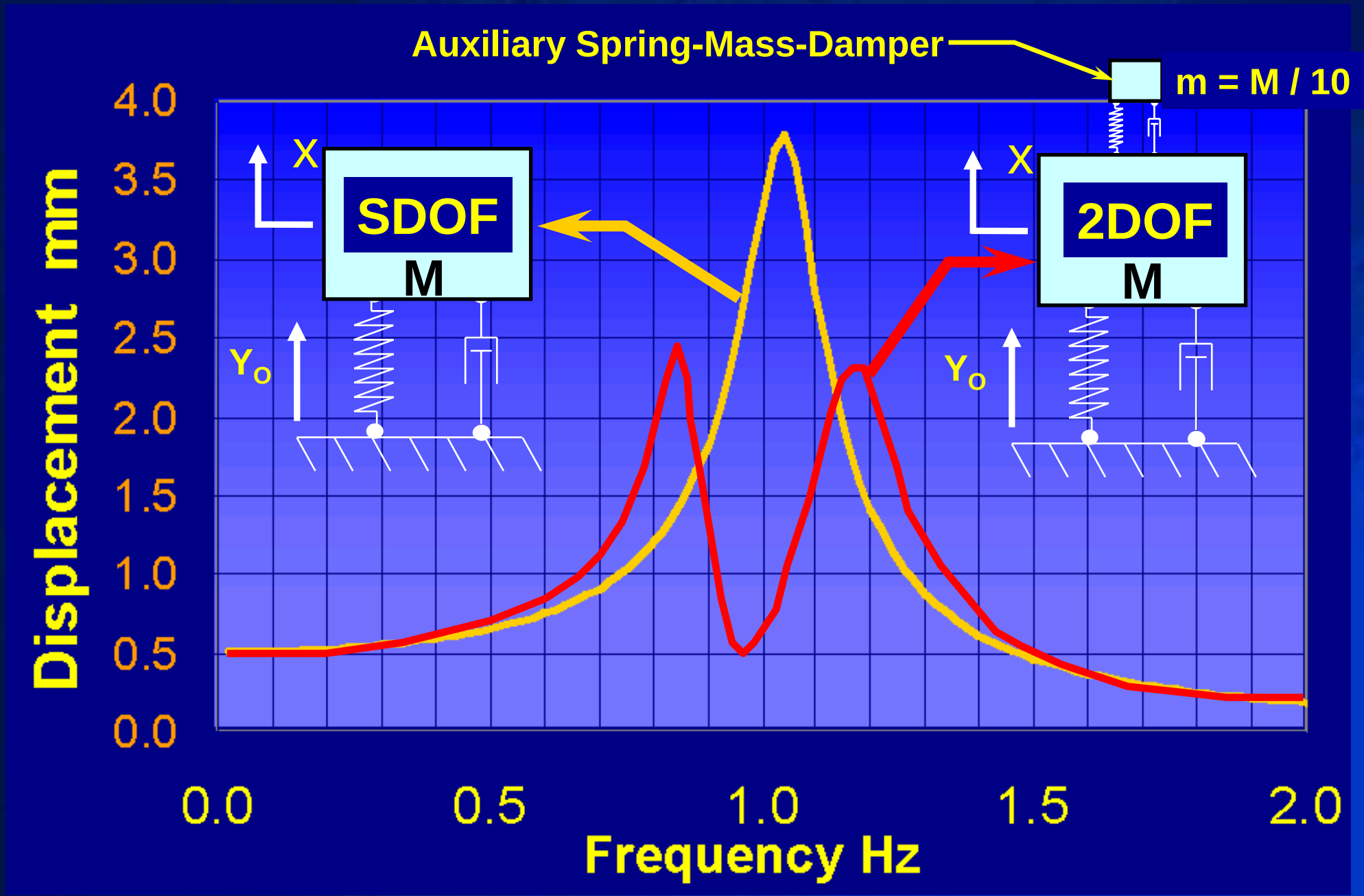
Response at Mid-Car



Low Frequency Basics

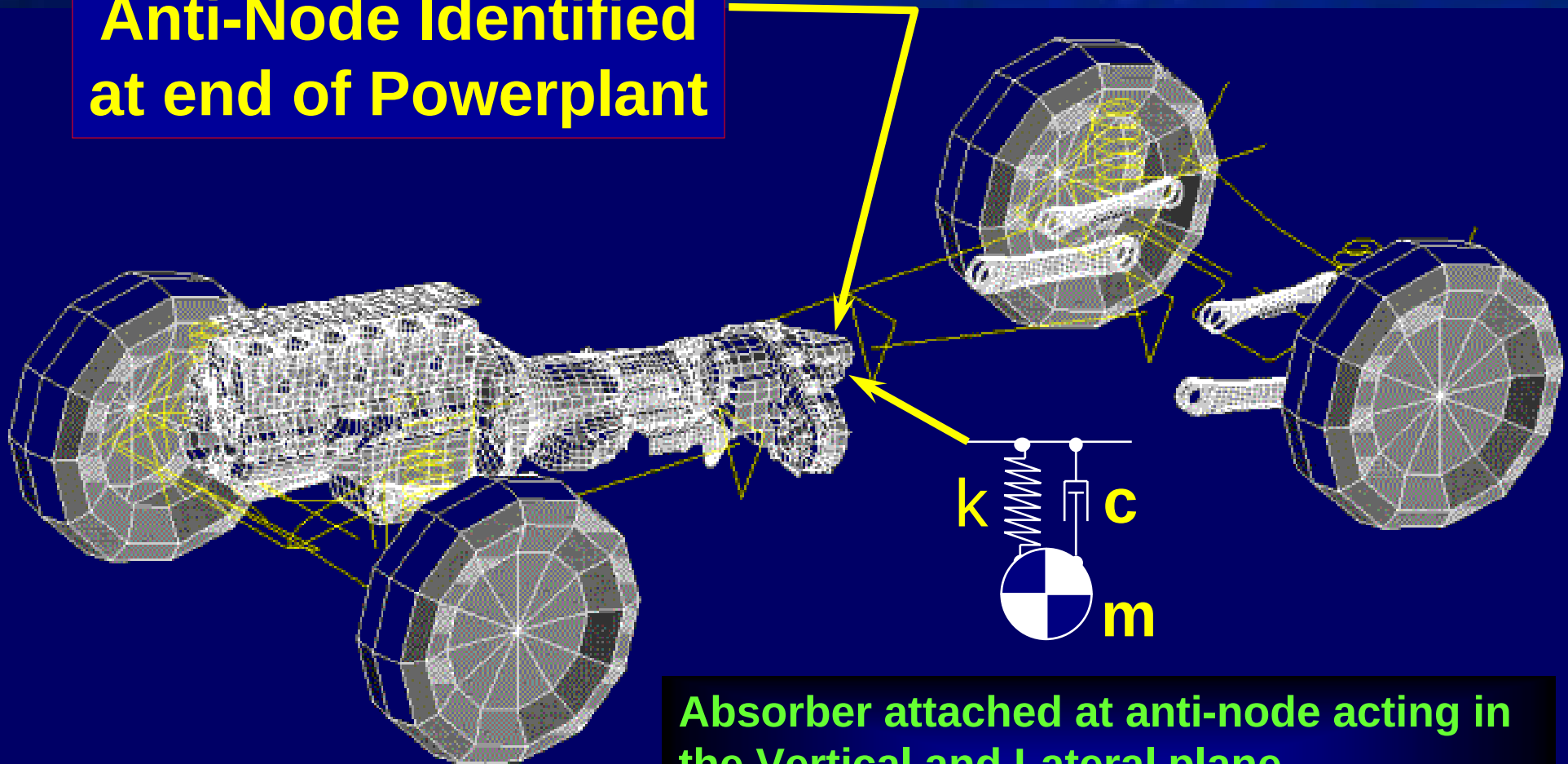
- Source-Path-Receiver Concept
- Single DOF System Vibration
- NVH Source Considerations
- Receiver Considerations
- Vibration Attenuation Strategies
  - Provide Improved Isolation
  - Mode Management
  - Nodal Point Mounting
  - Dynamic Absorbers

Dynamic Absorber Concept



Powertrain Example of Dynamic Absorber

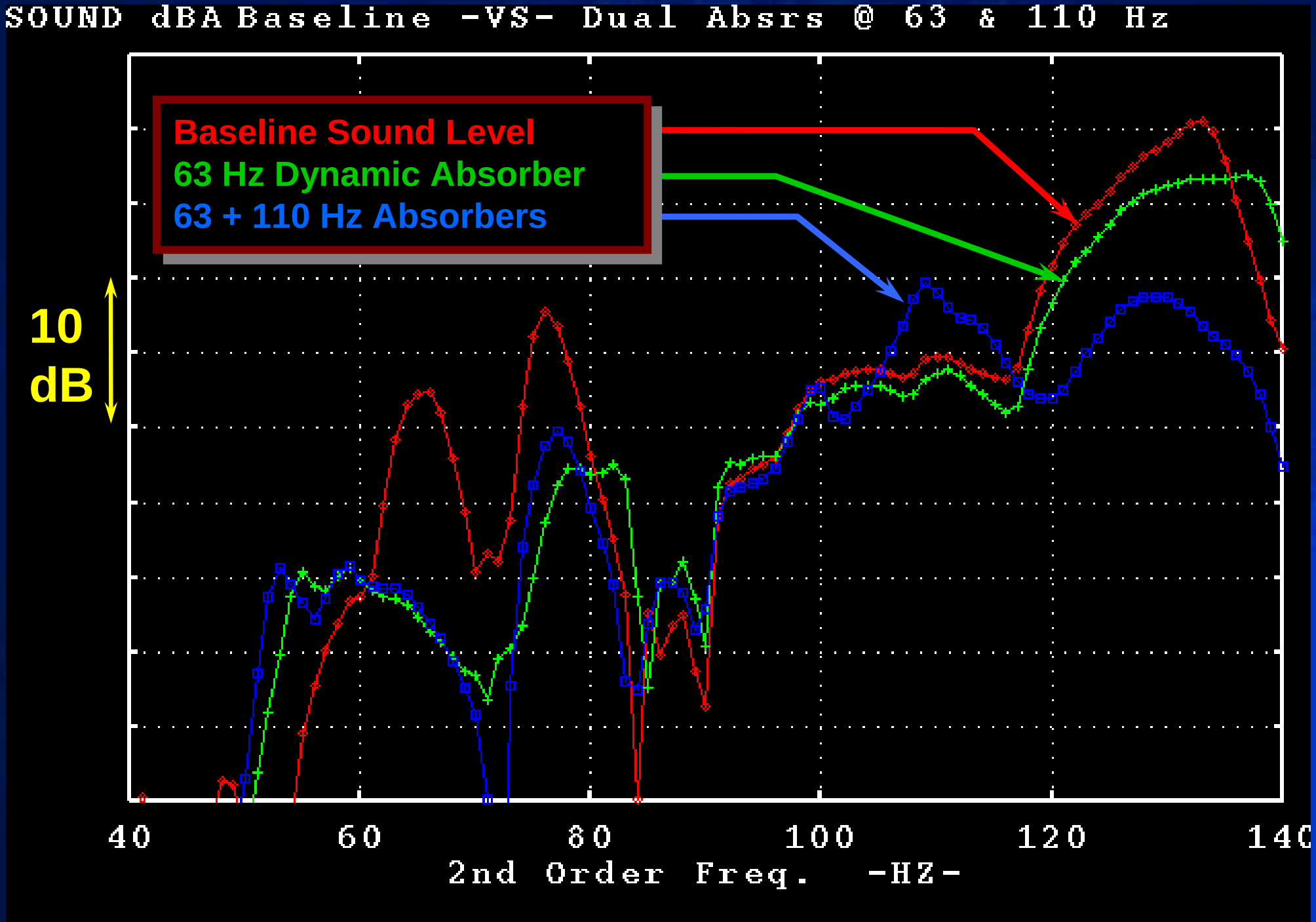
Anti-Node Identified at end of Powerplant



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

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

[Figure Courtesy of DaimlerChrysler Corporation]



[Figure Courtesy of DaimlerChrysler Corporation]

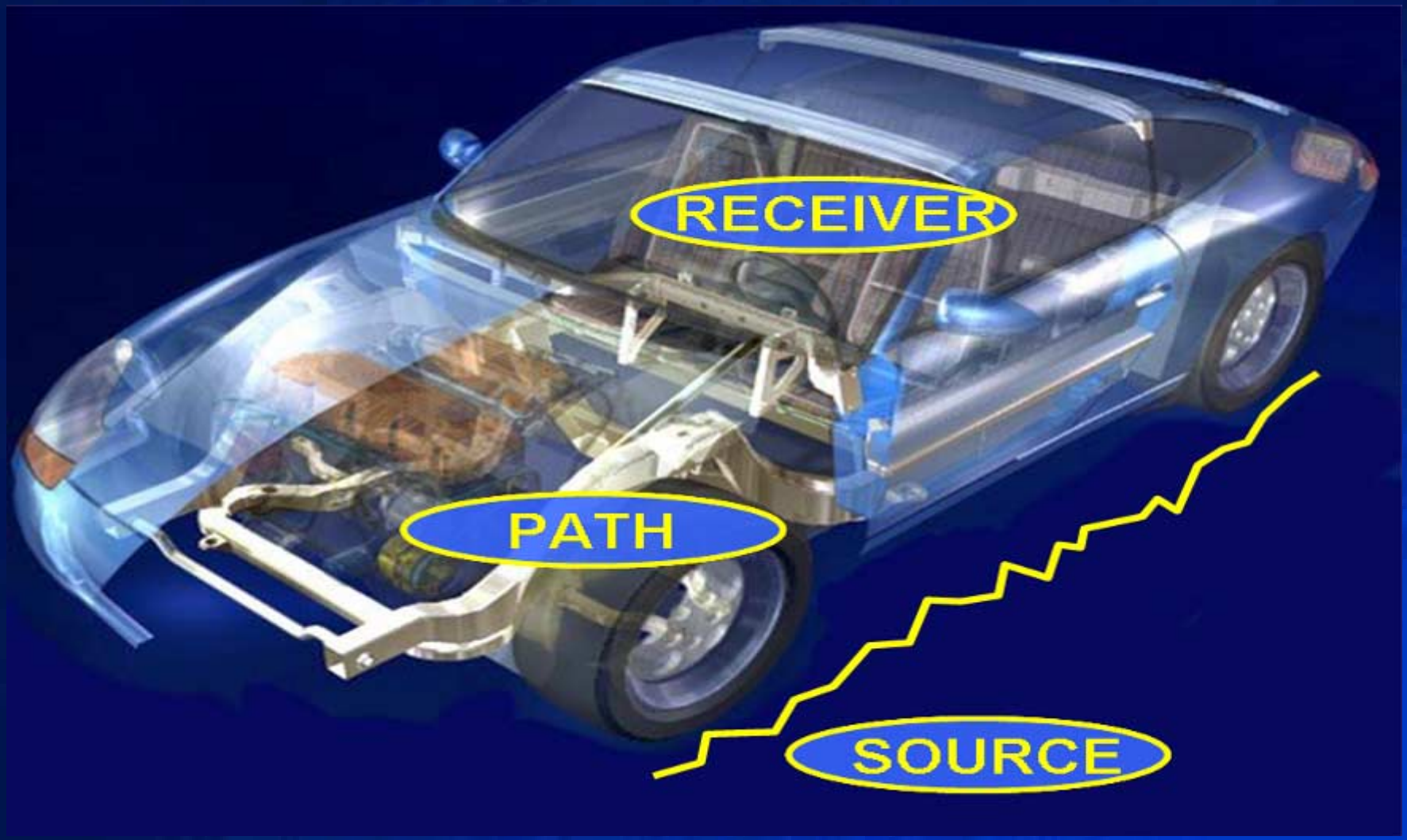
Low Frequency Basics - Review

- 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

Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- **Mid Frequency Basics** **Alan Duncan**
- *Live Noise Attenuation Demo*
- Case Studies
  - Interior Noise Diagnosis using VINS
  - NVH Principles: Applications
- Closing Remarks

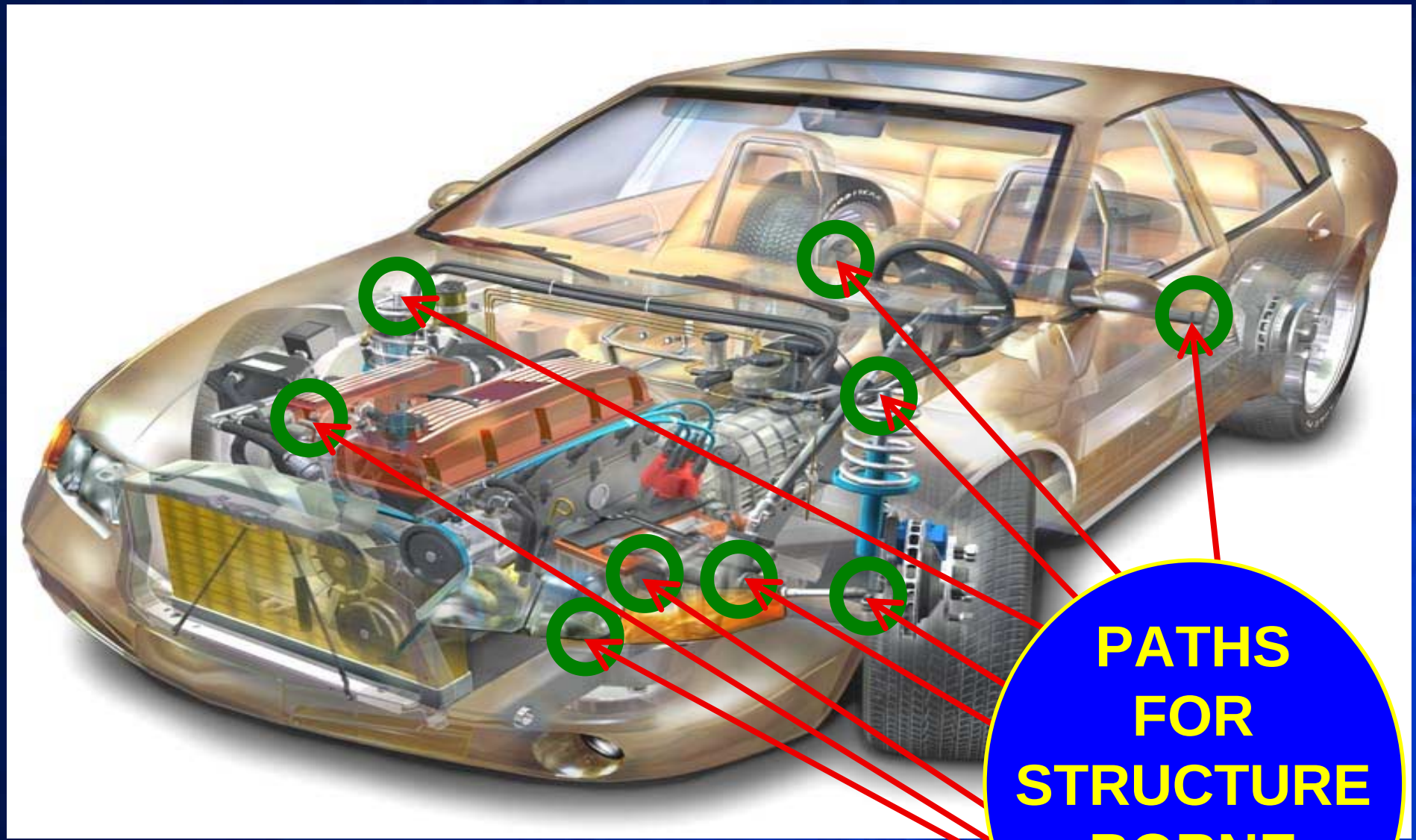
Mid Frequency NVH Fundamentals



This looks familiar!

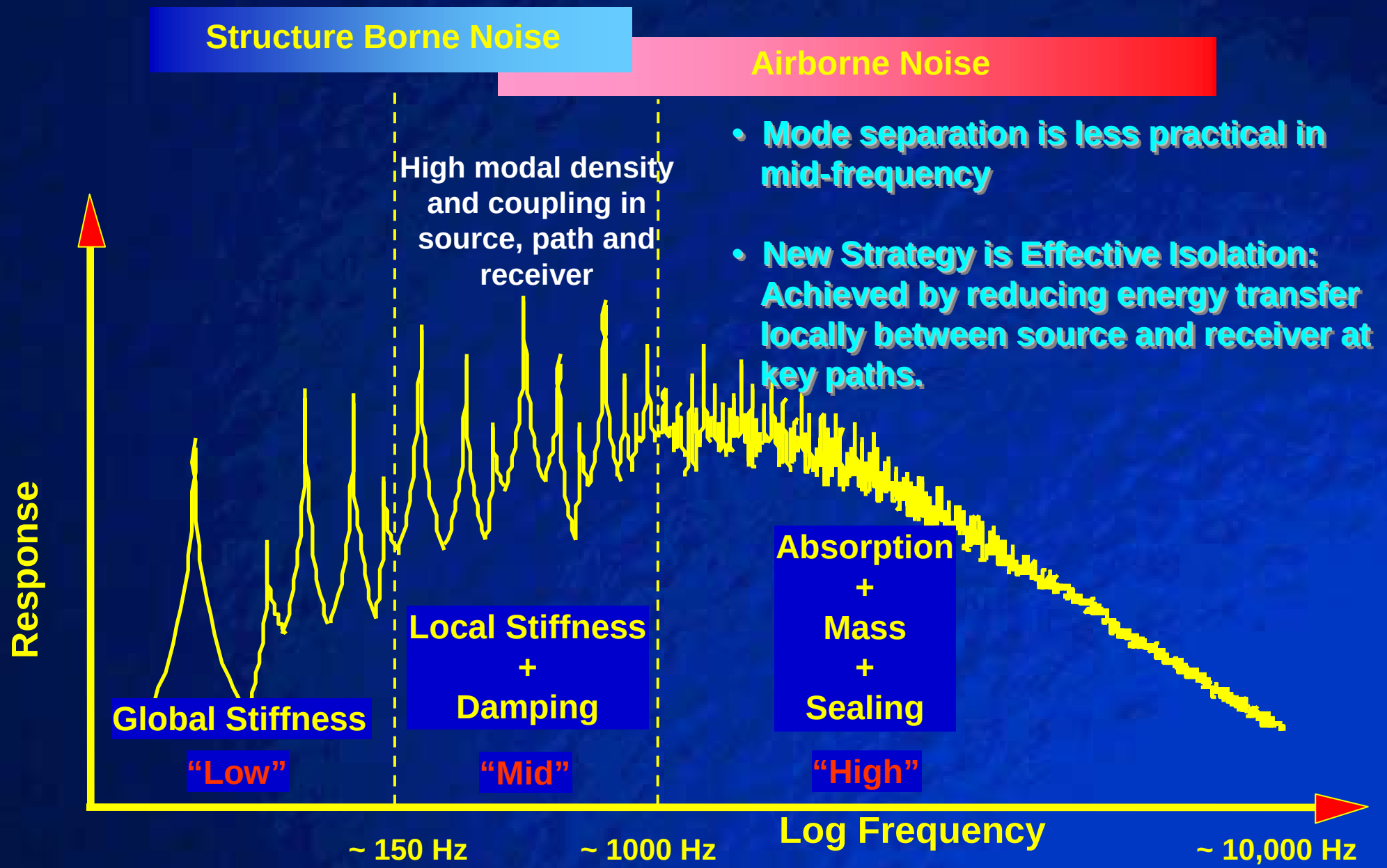
Frequency Range of Interest has changed to 150 Hz to 1000 Hz

Typical NVH Pathways to the Passenger



Noise Paths are the same as Low Frequency Region

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

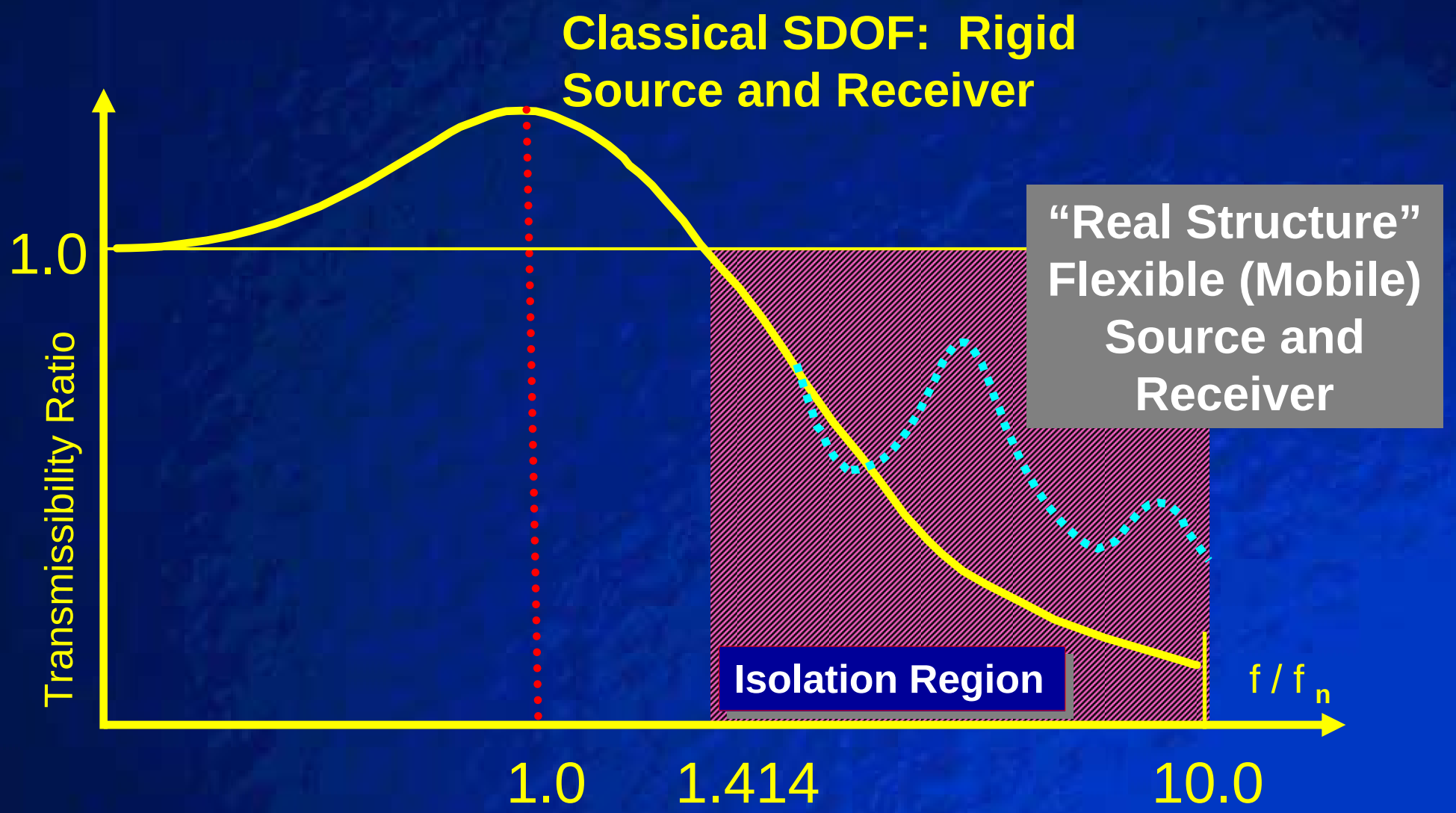
Mid-Frequency Analysis Character

Control Measures for Mid Frequency Concerns

Effective Isolation

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 to the force applied in the direction of the velocity**

$$\text{Mobility} = \frac{\text{Velocity}}{\text{Force}}$$

- **Mobility, also called Admittance, characterizes Dynamic Stiffness of the structure at load application point**

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

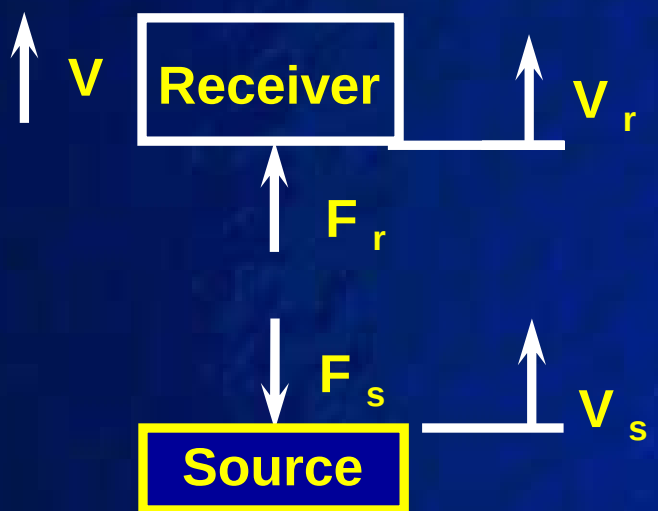
$$= \frac{\text{Frequency}}{\text{Dynamic Stiffness}}$$

Isolation

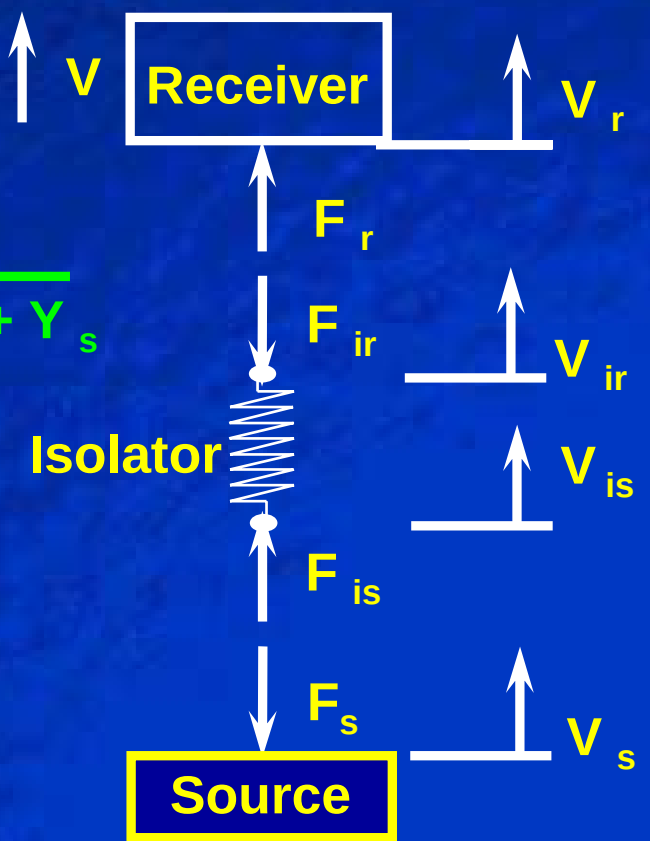
- The isolation effectiveness can be quantified by a theoretical model based on analysis of the SOURCE - PATH - RECEIVER transfer across an isolator

- Transmissibility Ratio (TR) 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

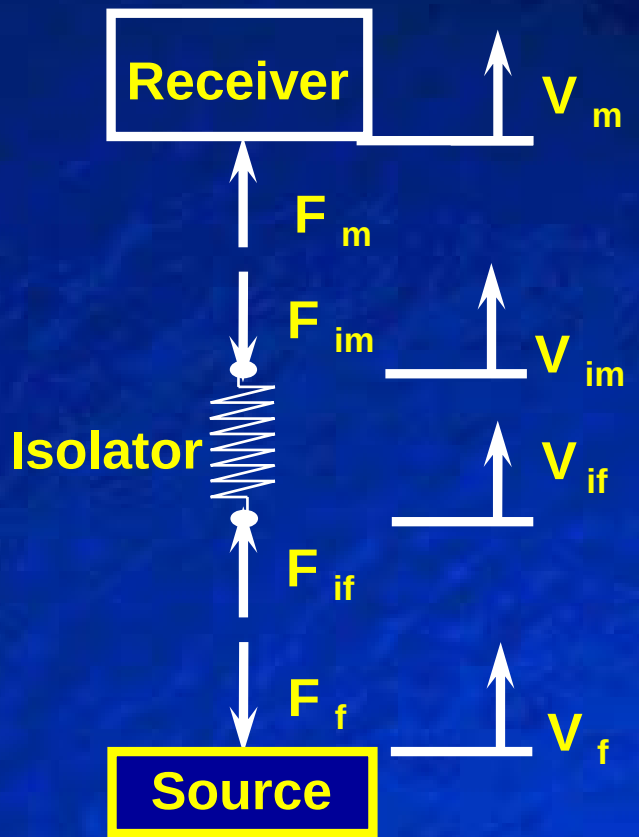
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

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

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 $Y \propto 1/K$

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

Body-to-Bushing Stiffness Ratio Relationship to Transmissibility

Stiffness Ratio; K_{body} / K_{iso}

Mid-Frequency Analysis Character

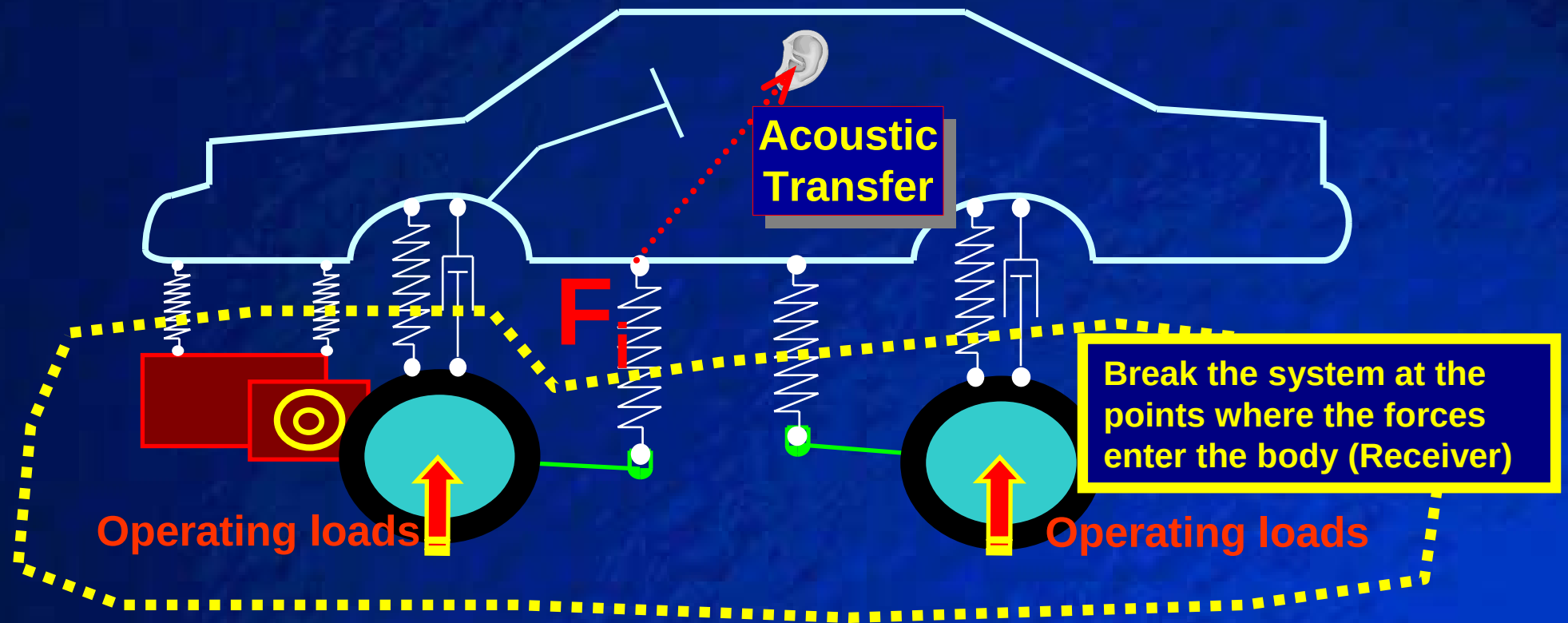
Control Measures for Mid Frequency Concerns

Effective Isolation

Attenuation along Key Noise Paths

Identifying Key NVH Paths

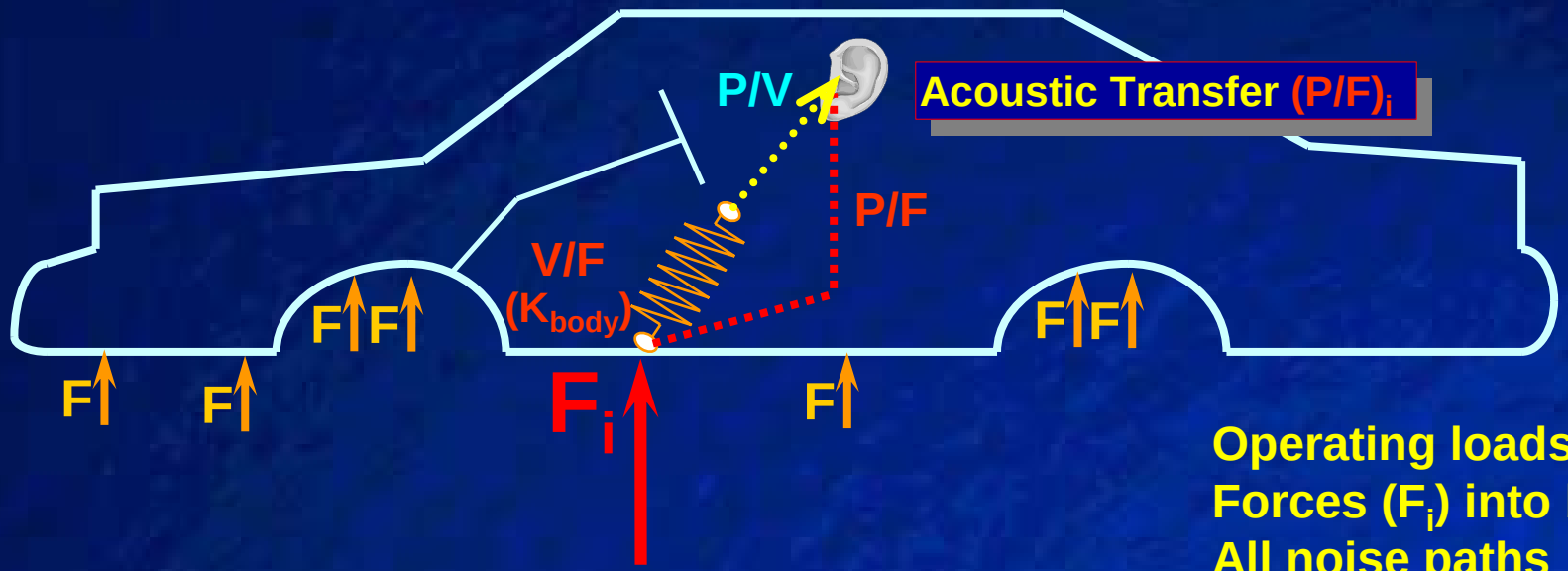
Key NVH paths are identified by Transfer Path Analysis (TPA)



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

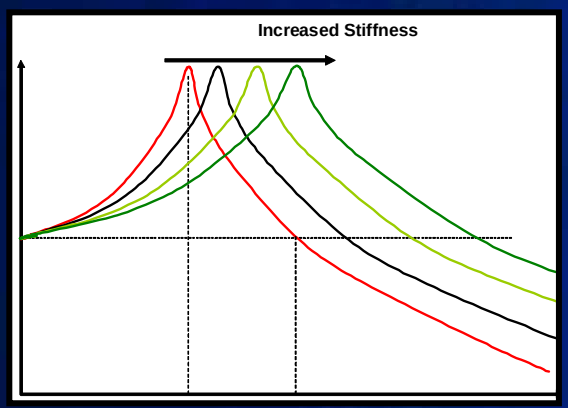
“Downstream” Effects: Body Panels

Recall for Acoustic Response P_t

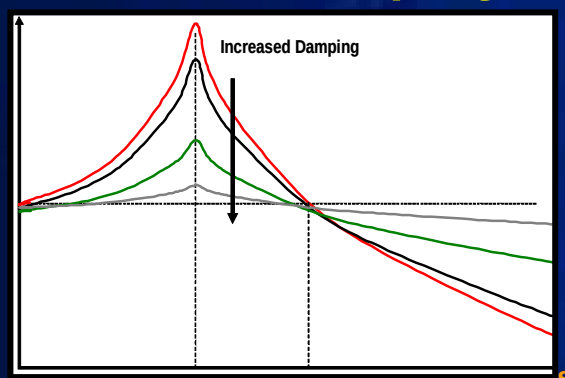
$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [F_i * (P/V)_i * (V/F)_i]$$

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

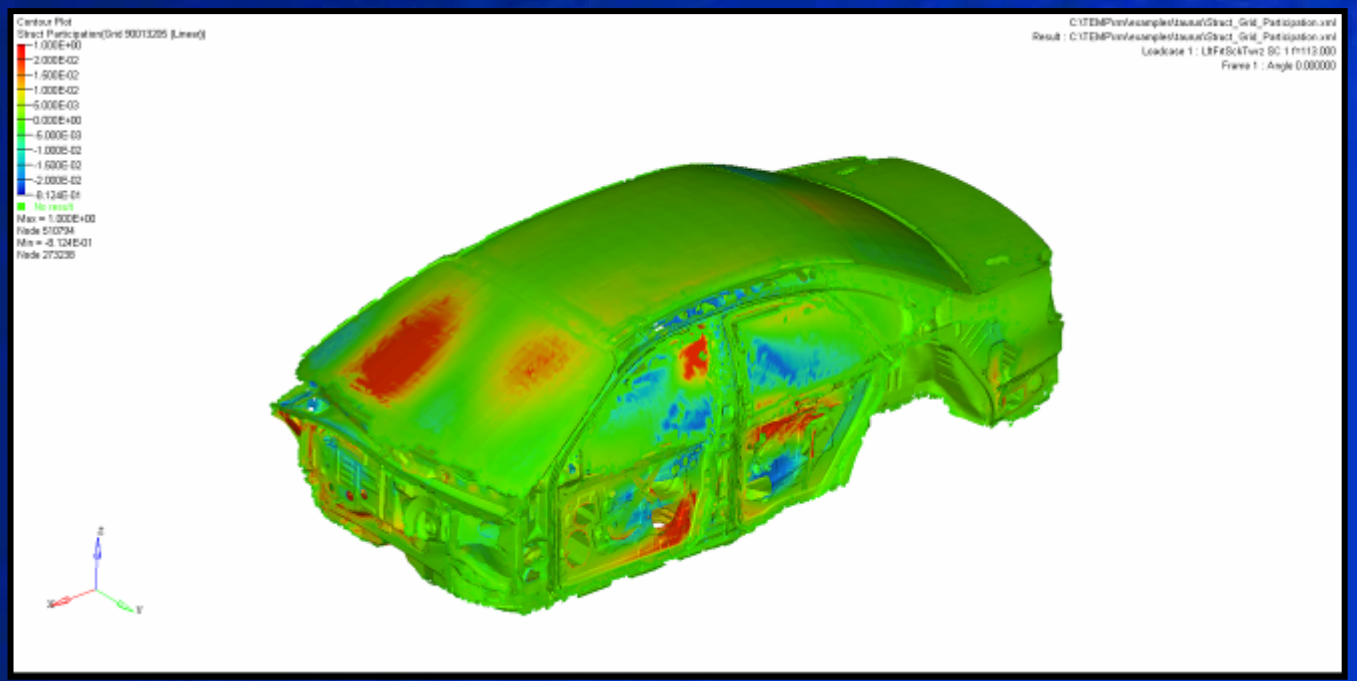
1. Panel Stiffness



2. Panel Damping



3. Panel Acoustic Contribution



Generic Noise Path Targets

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

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

Structural Mobility < 0.2 to 0.3 mm/sec/N

Acoustic Sensitivity < 50 - 60 dBL/N

Primary: Minimize the Source Force

< 1.0 N

Panel Damping Loss Factor

> .10

Final Remarks on Mid Frequency Analysis

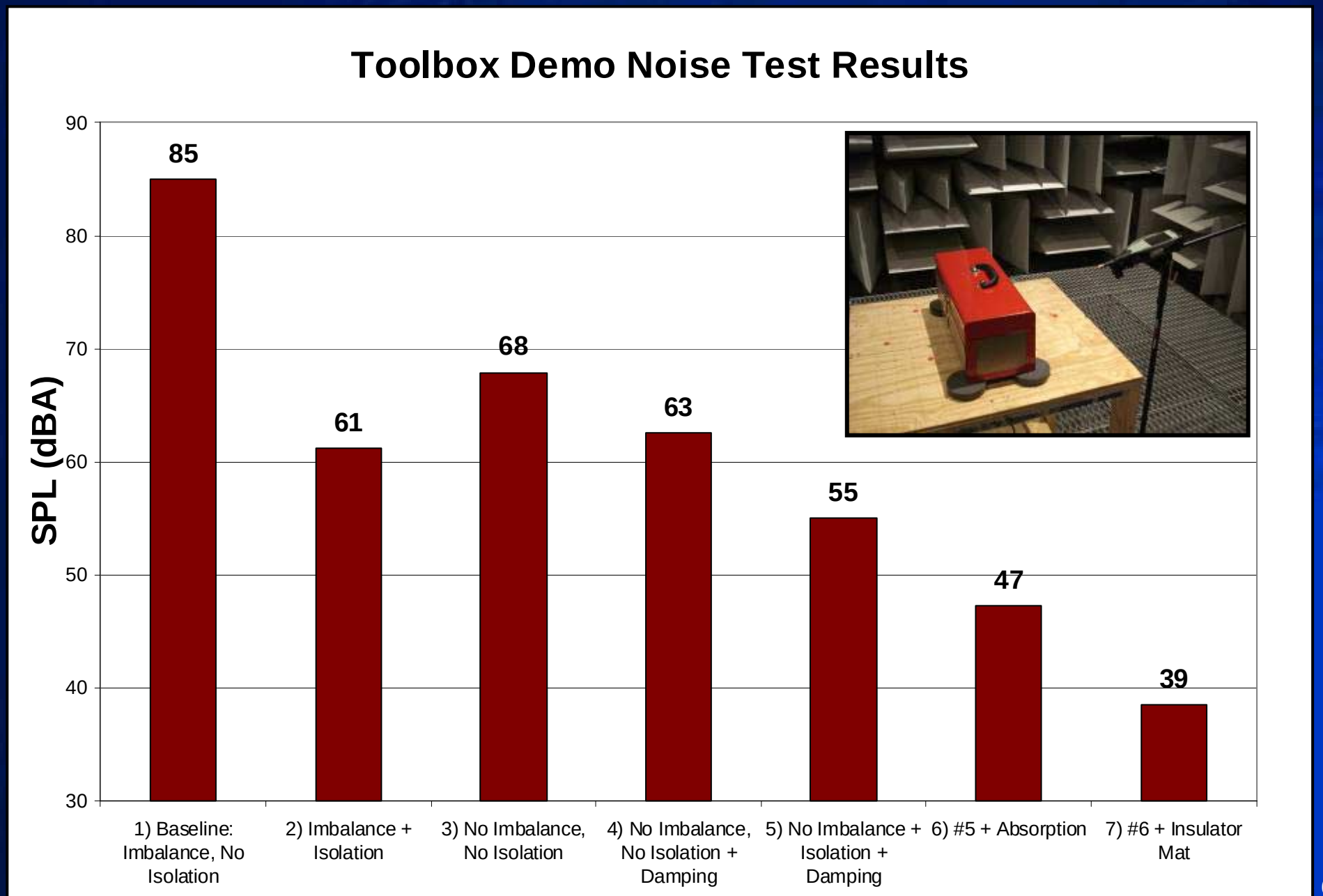
- Effective isolation at dominant noise paths is critical
- Reduced mobilities at body & source and softened bushing are key for effective isolation
- Mode Separation remains a valid strategy as modes in the source structure start to participate
- Other means of dealing with high levels of response (Tuned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

Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- ***Live Noise Attenuation Demo***
- Case Studies
  - Interior Noise Diagnosis using VINS
  - NVH Principles: Applications
- Closing Remarks

Greg Goetchius

Tool Box Demo Test Results



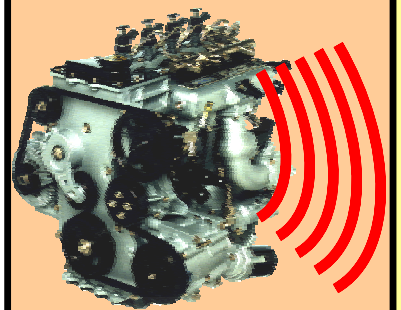
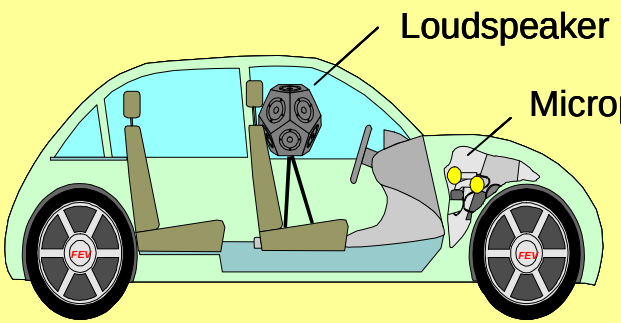
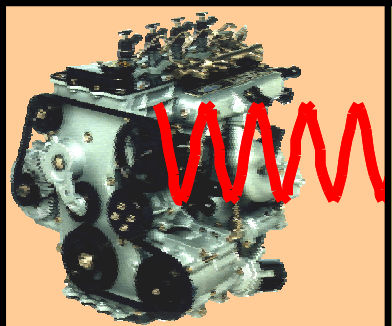
Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- Case Studies
  - Interior Noise Diagnosis using VINS
  - NVH Principles: Applications
- Closing Remarks

Kiran Govindswamy

“Time Domain” Transfer Path Analysis

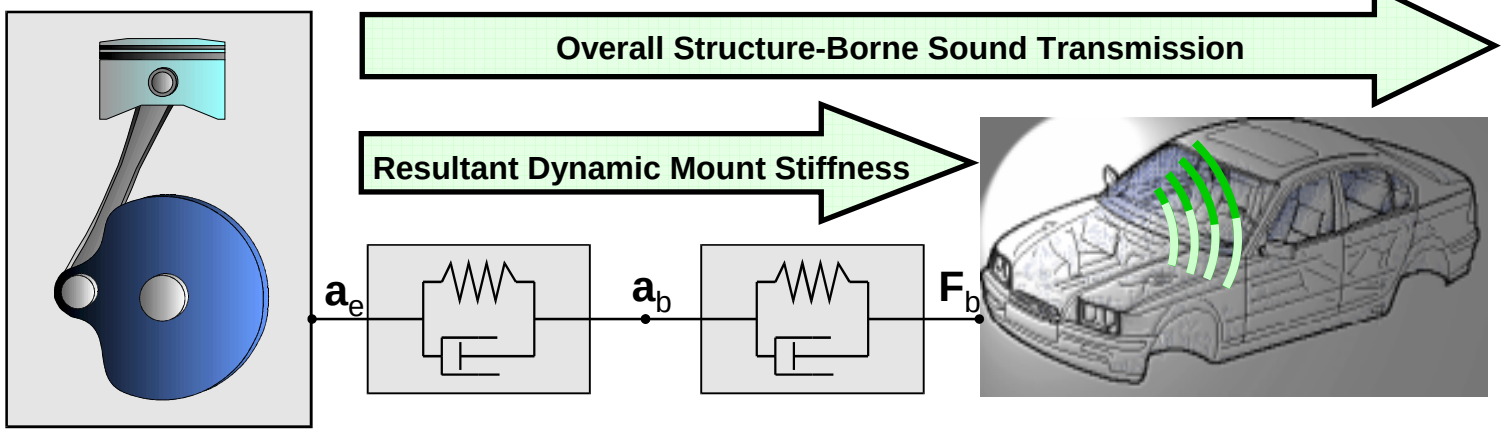
VINS (Vehicle Interior Noise Simulation)

	Excitation	Transfer Behaviour	Interior Noise
Airborne Path	 Power Train Noise Intake&Exhaust Noise . . .		Airborne Noise Share
Structure-Borne Path	 Power Train Vibration Drive Line Vibration . . .	Engine Mount → Vehicle Overall Vehicle Transfer Function	Structureborne Noise Share

Overall Interior Noise

“Time Domain” Transfer Path Analysis

VINS (Vehicle Interior Noise Simulation)

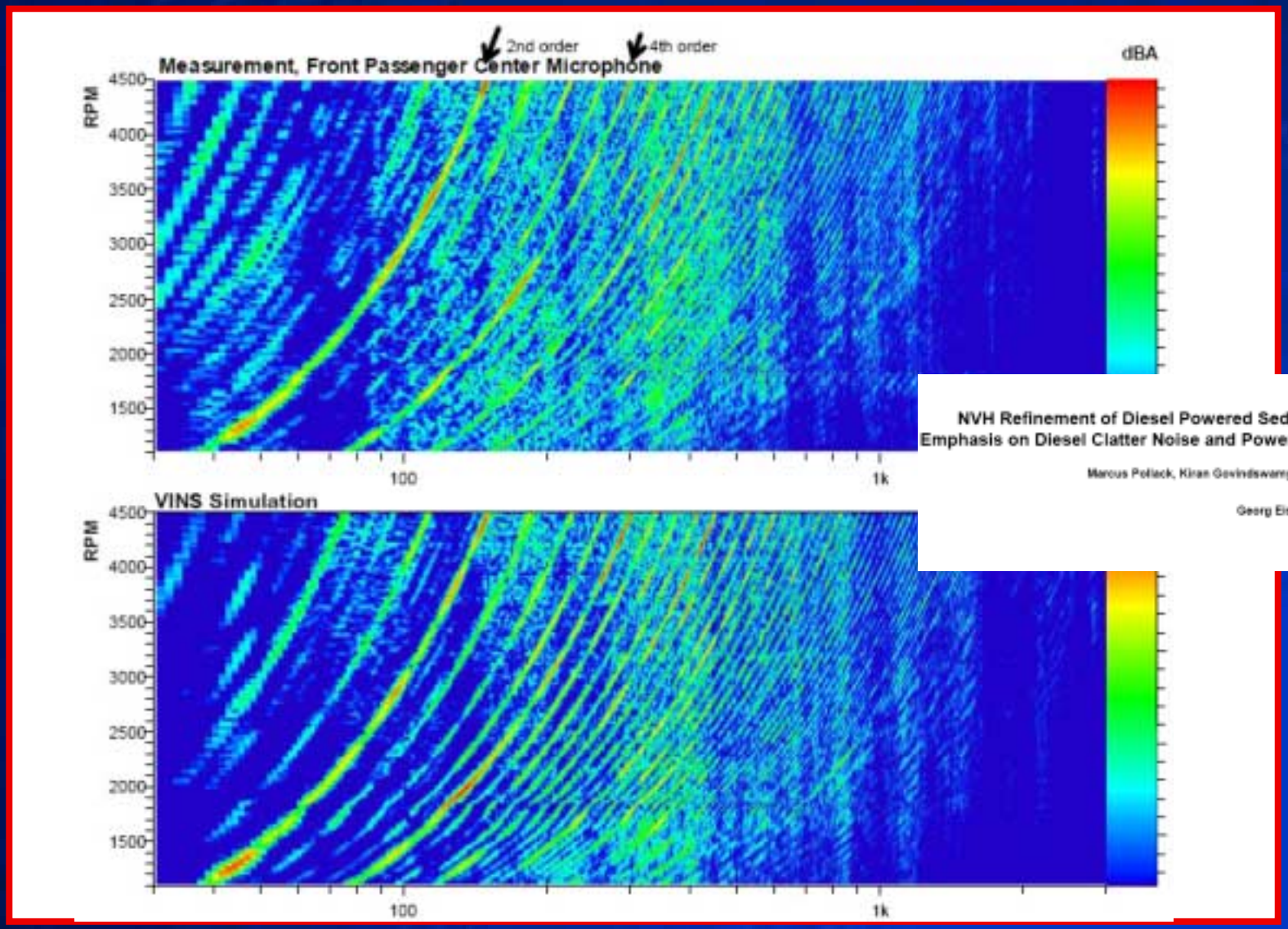


$$a_{engine} \bullet \frac{a_{body}}{a_{engine}} \bullet \frac{F_{body}}{a_{body}} \bullet \frac{P_{interior}}{F_{body}} = P_{interior}$$

Resultant Dynamic Mount Stiffness

“Time Domain” Transfer Path Analysis

VINS: Correlation (Full Load Speed Sweep)



2007-01-2378

NVH Refinement of Diesel Powered Sedans with Special Emphasis on Diesel Clatter Noise and Powertrain Harshness

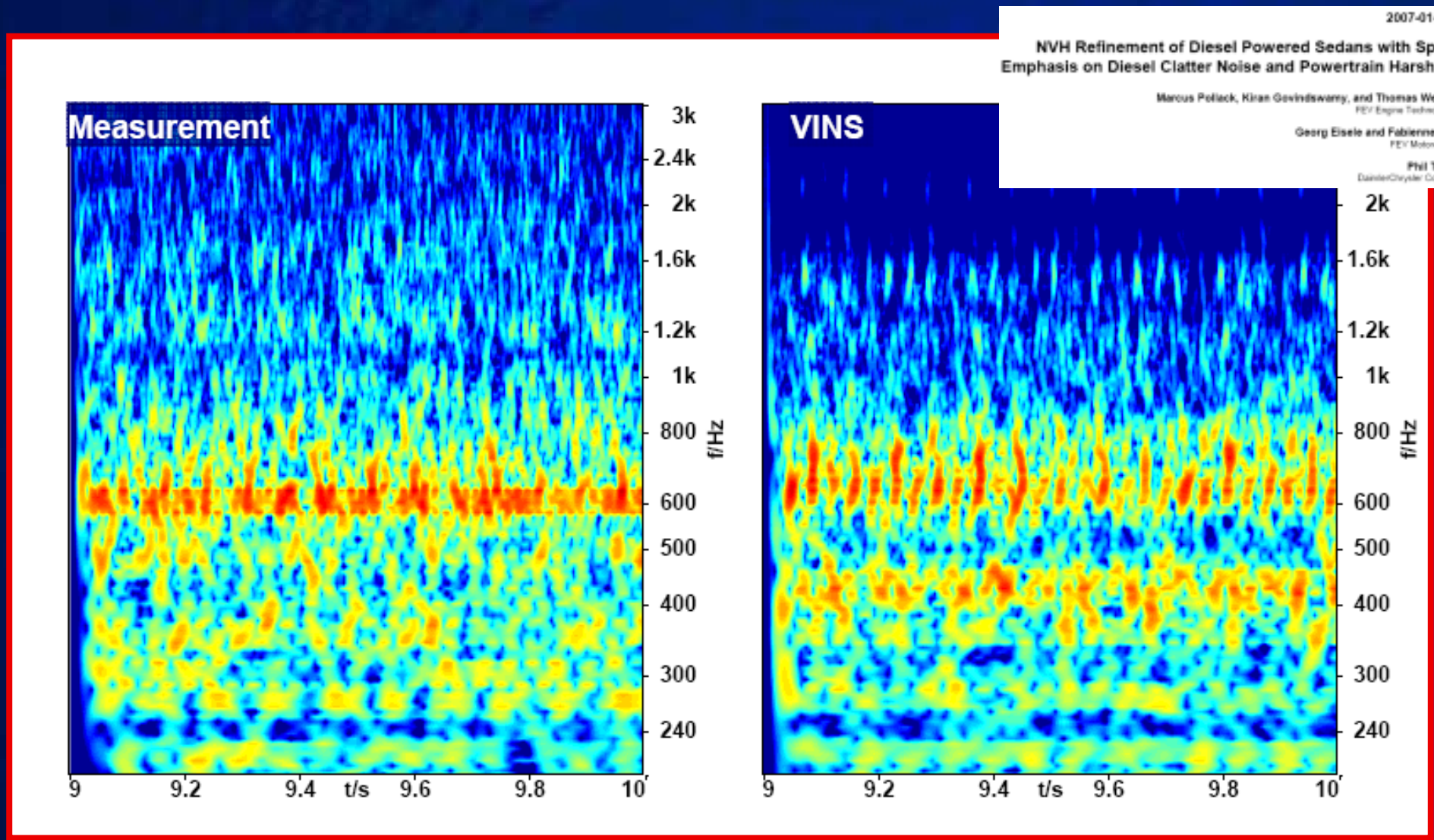
Marcus Pollack, Kiran Govindswamy, and Thomas Wellmann
FEV Engine Technology, Inc.

Georg Eisele and Fabienne Pichot
FEV Motorentechnik

Phil Thomas
DaimlerChrysler Corporation

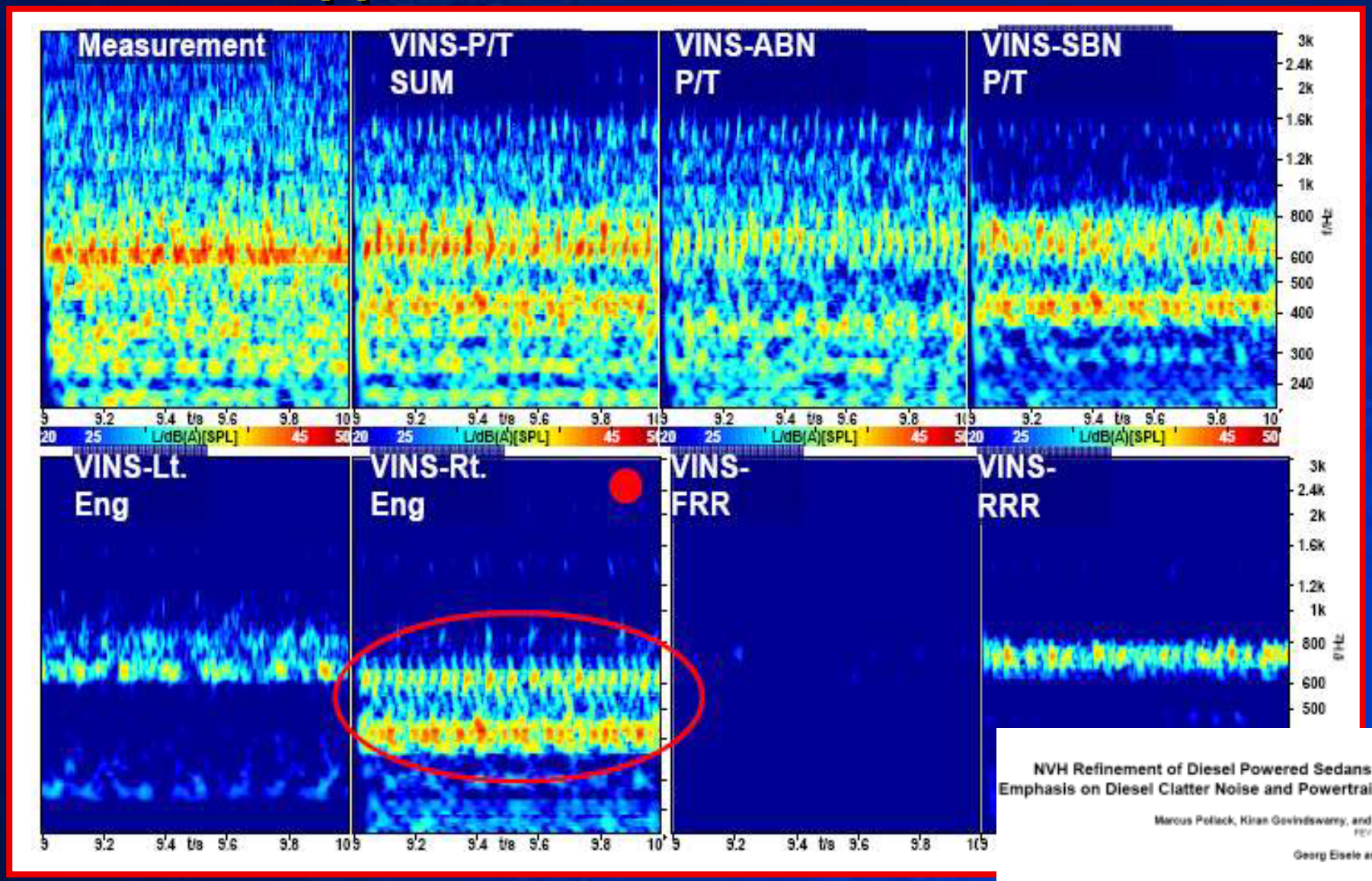
“Time Domain” Transfer Path Analysis

VINS: Application to “Idle Modulation Noise”



“Time Domain” Transfer Path Analysis

VINS: Application to “Idle Modulation Noise”



2007-01-2378

NVH Refinement of Diesel Powered Sedans with Special Emphasis on Diesel Clatter Noise and Powertrain Harshness

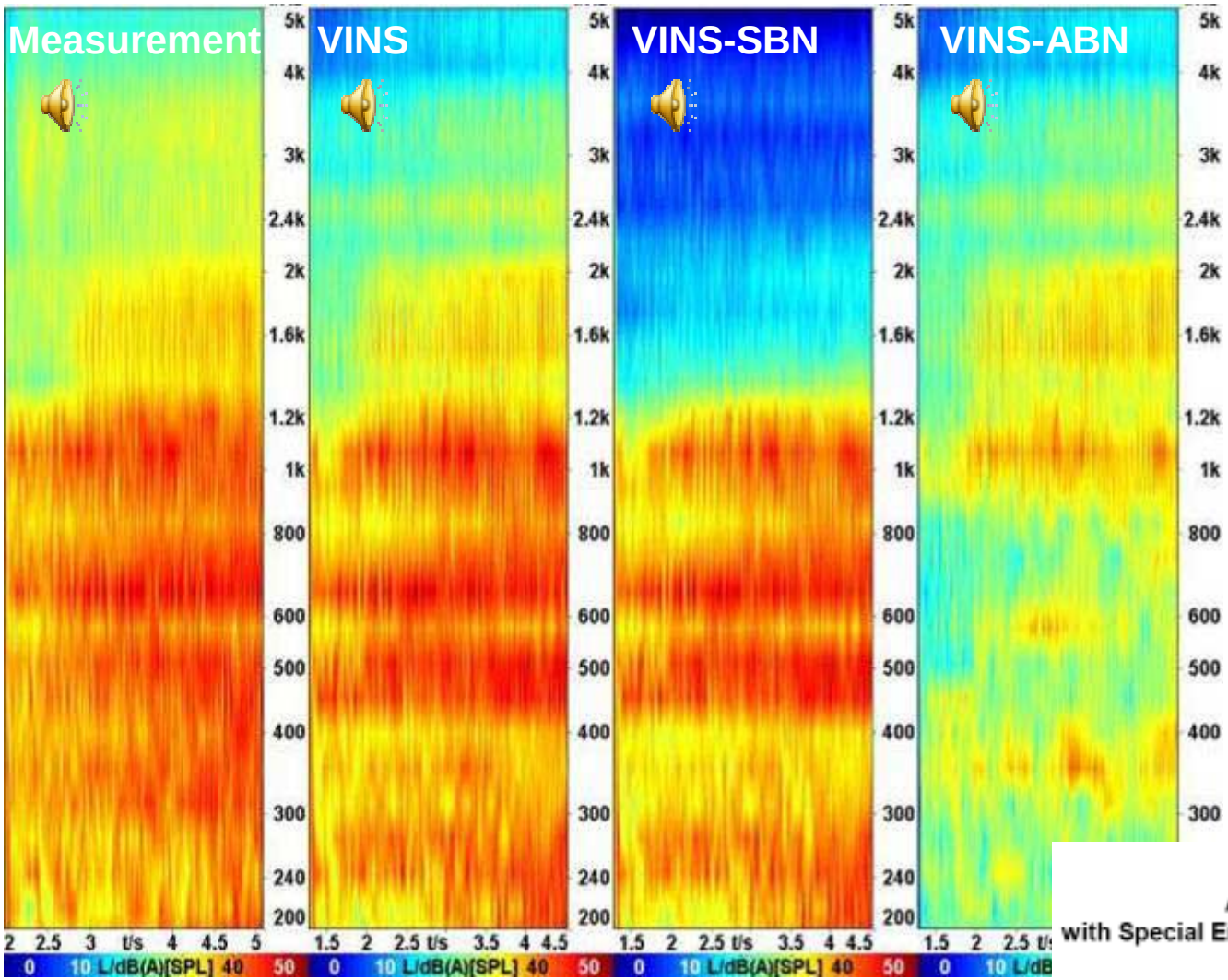
Marcus Pollack, Kiran Govindswamy, and Thomas Wellmann
FEV Engine Technology, Inc.

Georg Eisele and Fabienne Pichot
FEV Motorentechnik

Phil Thomas
DaimlerChrysler Corporation

“Time Domain” Transfer Path Analysis

VINS: Application to “Transient Diesel Clatter”



2007-01-2411

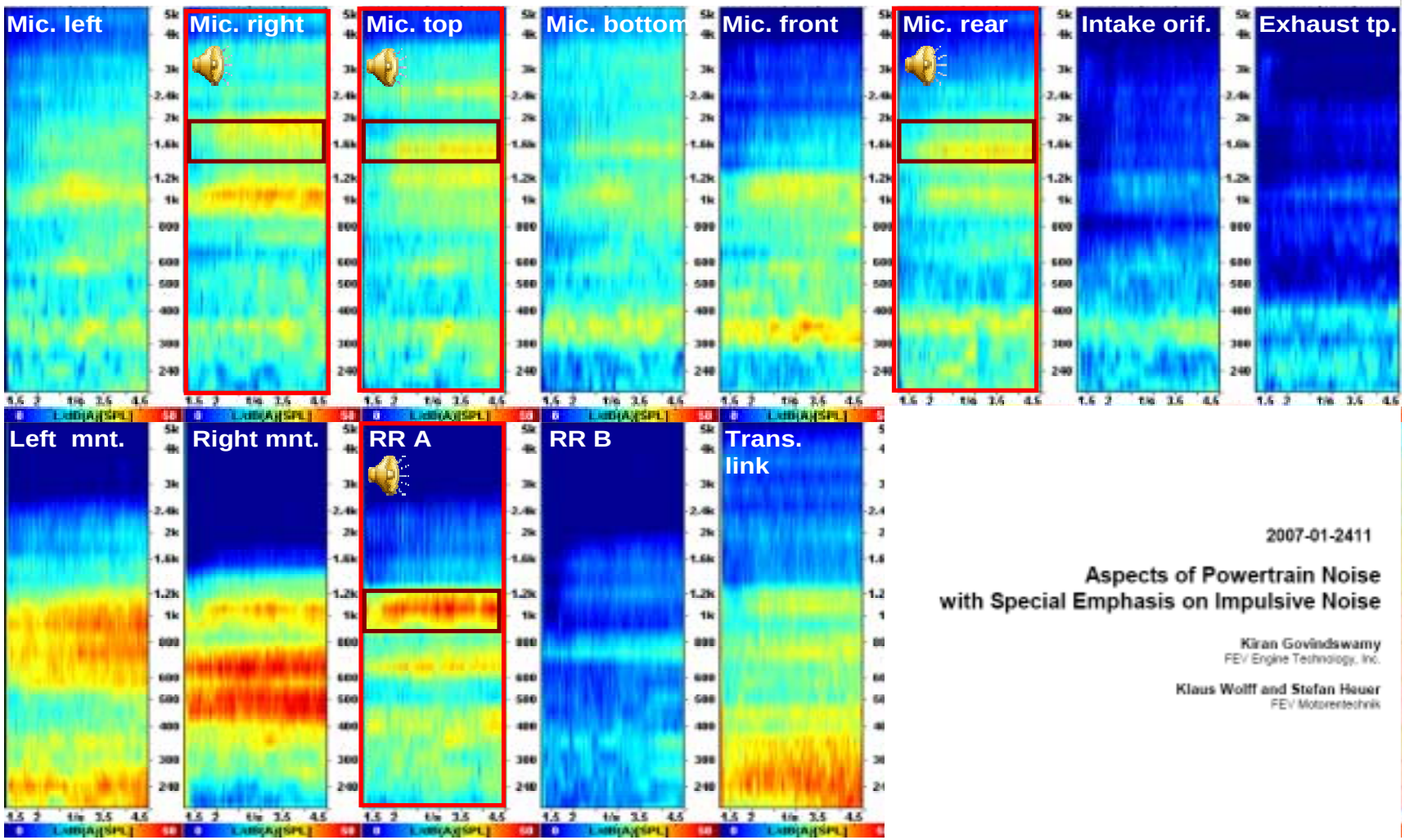
Aspects of Powertrain Noise with Special Emphasis on Impulsive Noise

Kiran Govindswamy
FEV Engine Technology, Inc.

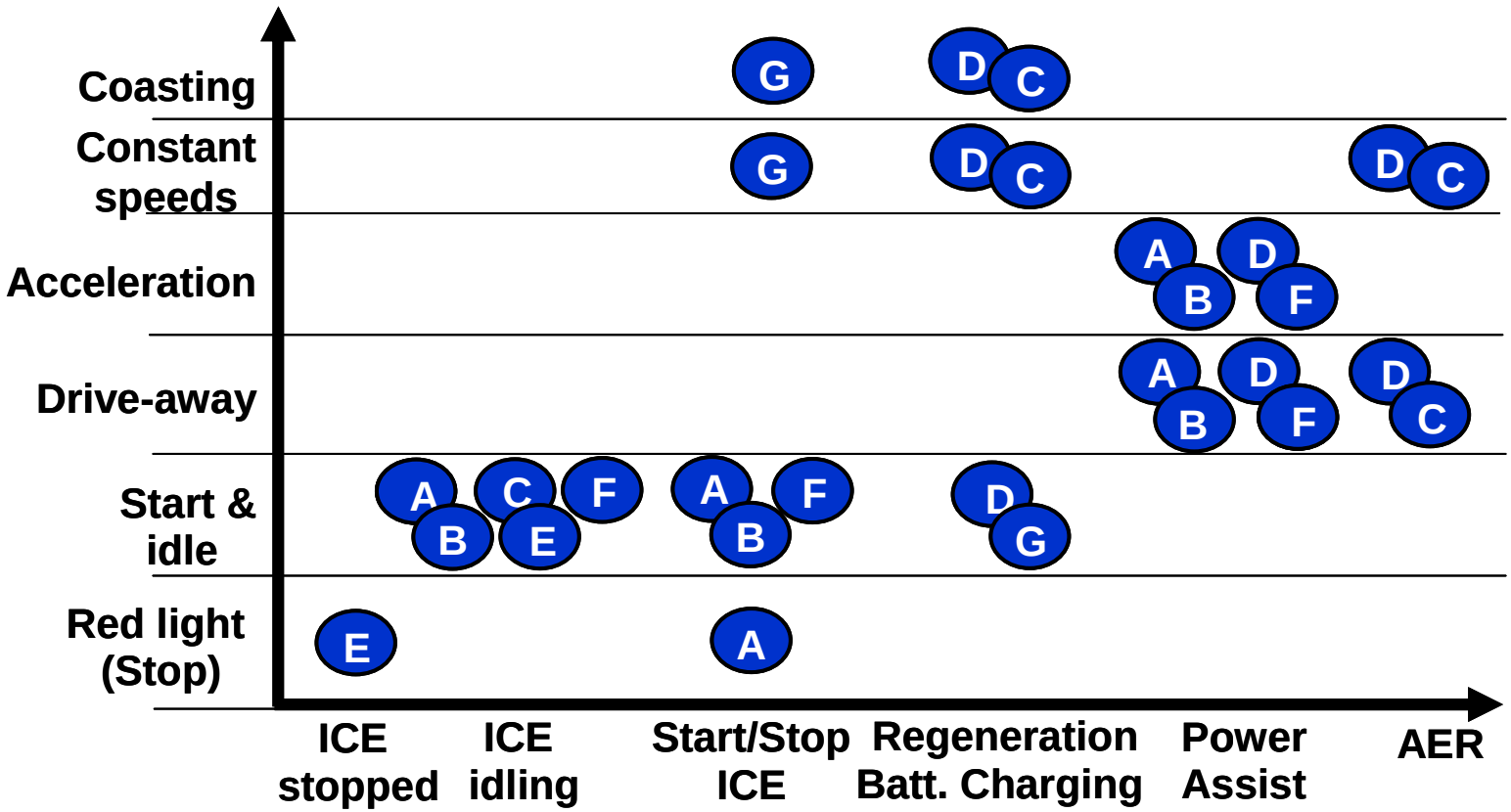
Klaus Wolff and Stefan Heuer
FEV Motorentechnik

“Time Domain” Transfer Path Analysis

VINS: Application to “Transient Diesel Clatter”



HEV, PHEV, ReEV, and EV NVH



- A: Global powertrain vibration
- B: Driveline vibration
- C: Noise of HEV specific components
- D: Magnetic noise E-Motor/(Generator)
- E: Noise of accessories
- F: Gearbox rattle noise due to ICE (non-uniformity) and E-Motor
- G: Noise pattern changeover

2009-01-2085

Aspects of NVH Integration in Hybrid Vehicles

Kiran Govindswamy and Thomas Wellmann
FEV, Inc.

Georg Eisele
FEV Motorentechnik

Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- **Case Studies**
  - Interior Noise Diagnosis using VINS
  - **NVH Principles: Applications**
- Closing Remarks

Jianmin Guan

NVH Principles: 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

Copyright © 2007 SAE International

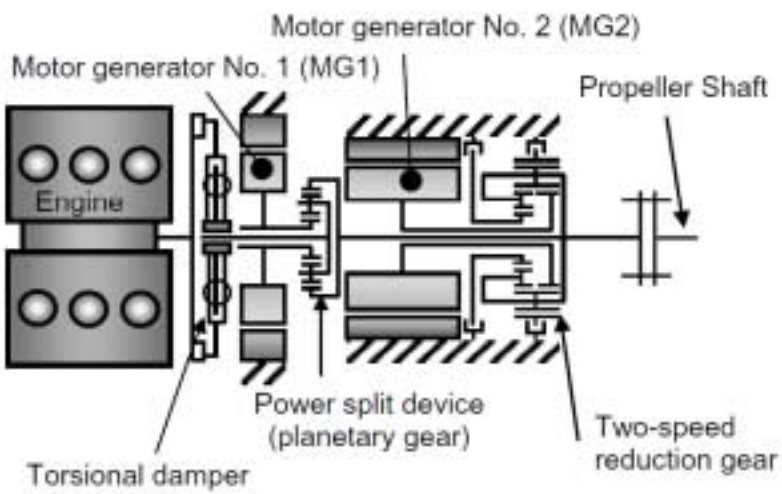
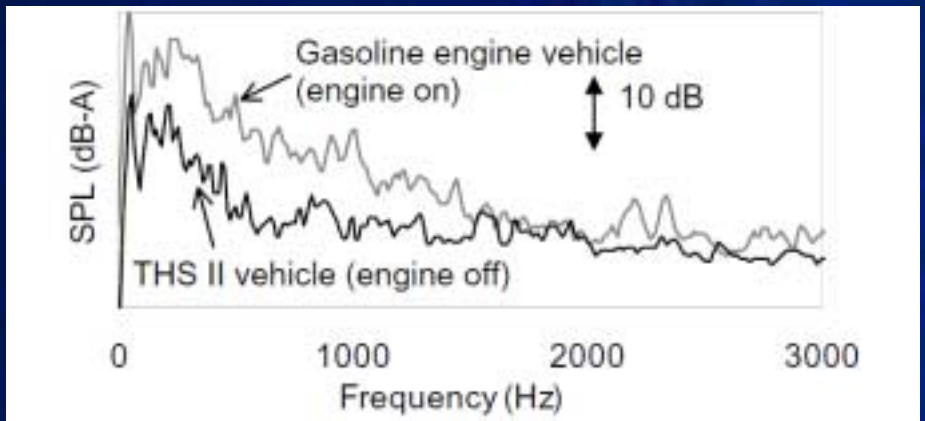


Fig.1 Longitudinal Power Train Configuration

Copyright © 2009 SAE International

Quietness during Idle and Electric-Vehicle Operation

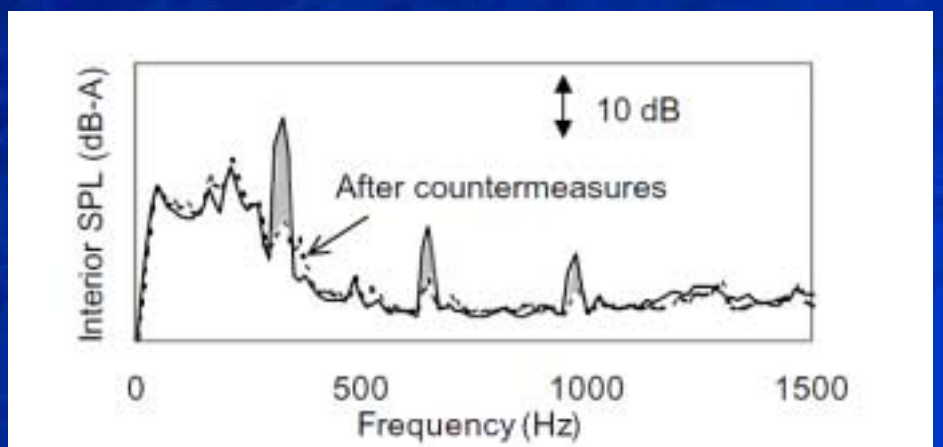


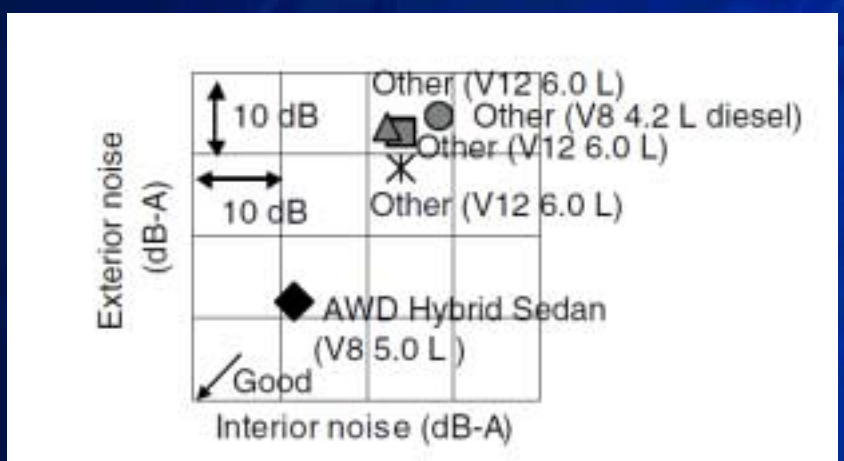
Root Cause Diagnostics:

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



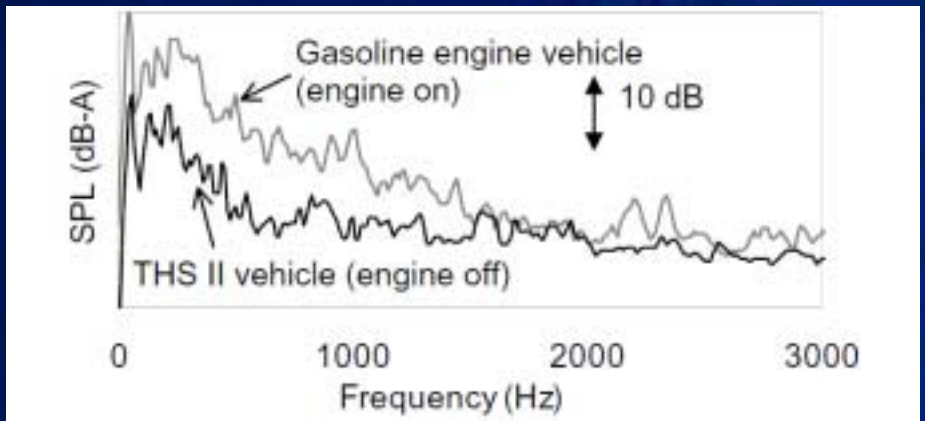


Improve Isolation from Body:

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

Copyright © 2009 SAE International

Quietness during Idle and Electric-Vehicle Operation



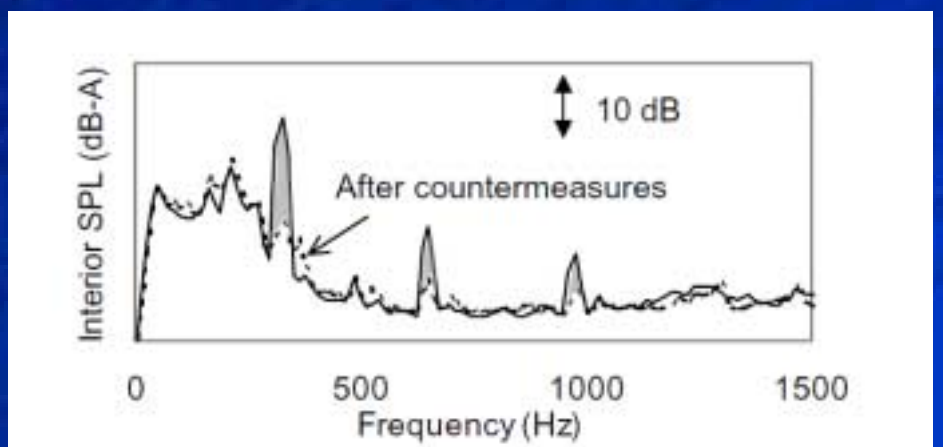
Root Cause Diagnostics:

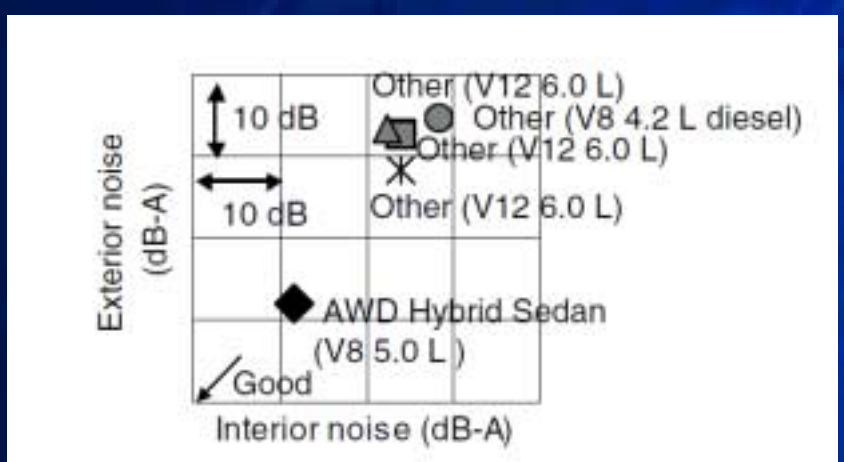
1. Water pump in the inverter cooling system
2. Electromagnetic noise of the motor, the inverter, and other units

Reduce Inverter Water Pump Loads:

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

Reduce Source





Improve Isolation from Body:

Effective Isolation

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

Attach. Stiffness

Downstream

Copyright © 2009 SAE International

Engine Start Vibration

Root Cause Diagnostics:

1. Engine torque fluctuations
2. Engine torque reaction forces

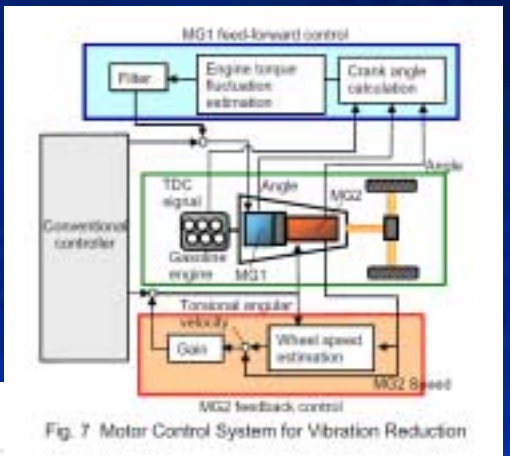
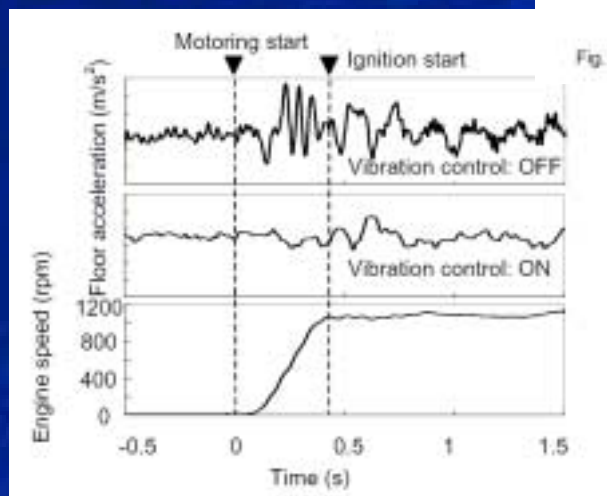


Fig. 7 Motor Control System for Vibration Reduction

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

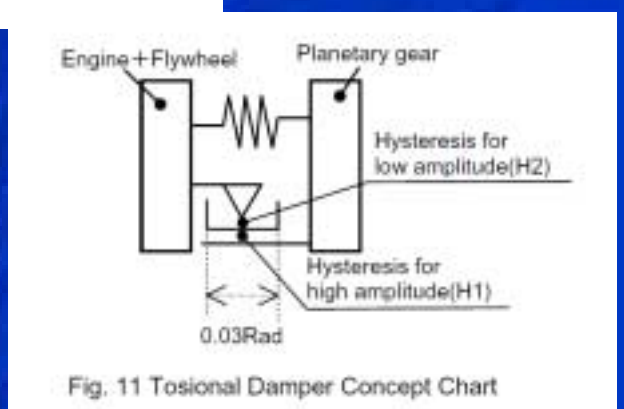


Fig. 11 Tosional Damper Concept Chart

Copyright © 2009 SAE International

Engine Start Vibration

Root Cause Diagnostics:

1. Engine torque fluctuations
2. Engine torque reaction forces

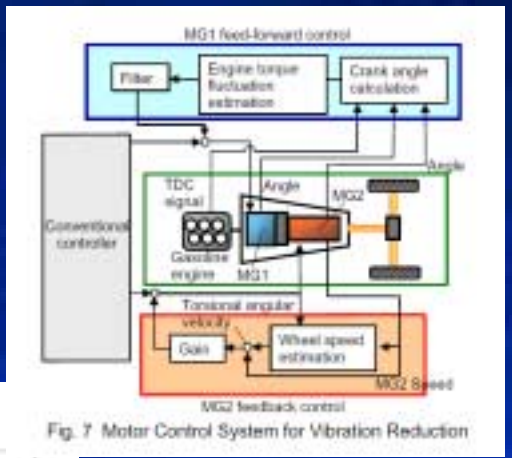
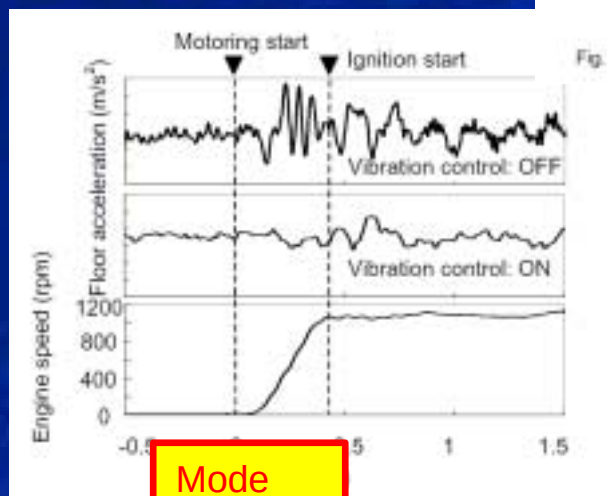


Fig. 7 Motor Control System for Vibration Reduction

Reduce Torque Fluctuation:

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

Reduce Source



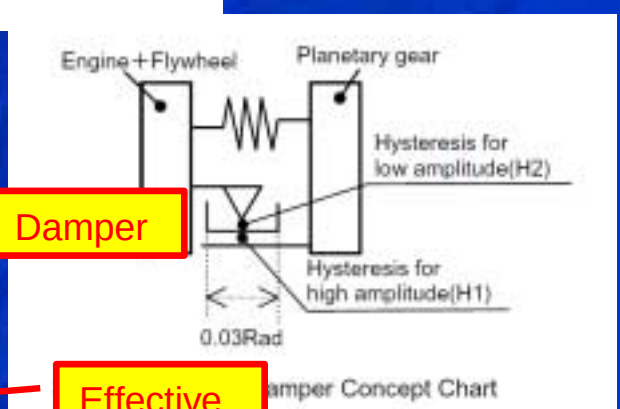
Mode Manage.

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

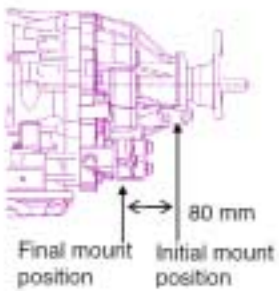


Copyright © 2009 SAE International

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



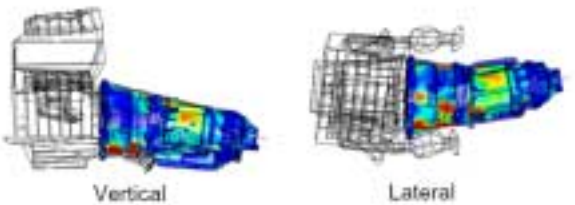
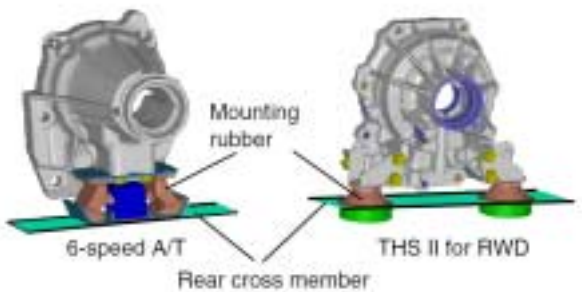


Fig. 14 Vibration Mode of Power Plant Bending Resonance



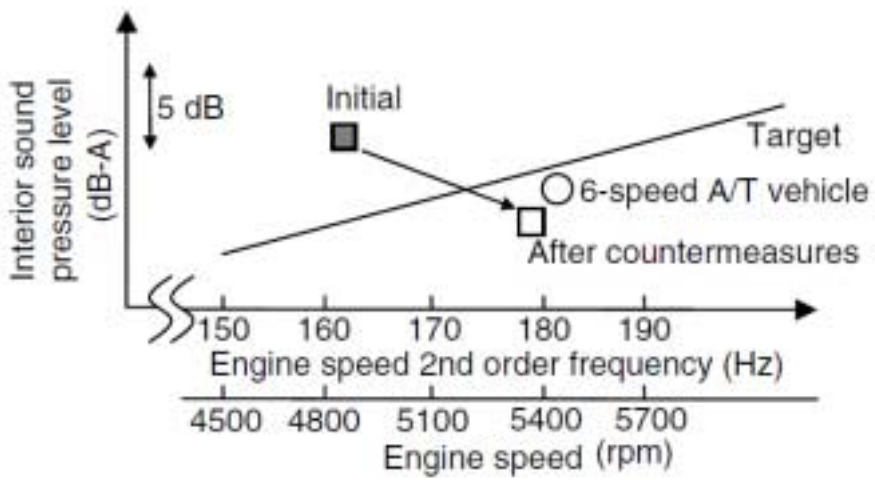


Fig. 17 Interior Booming Noise Level Caused by the Second Order Component of Engine Speed and Power Plant Bending Resonance Frequency

Reduce Effect of 2nd order Couple:

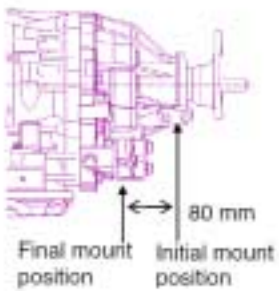
1. Increase power plant bending mode
2. Move mount to a nodal point
3. Embed mounts inside cross member
4. Reduces distance from principle elastic axis to CG
5. Optimized vertical to lateral rate ratio

Copyright © 2009 SAE International

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



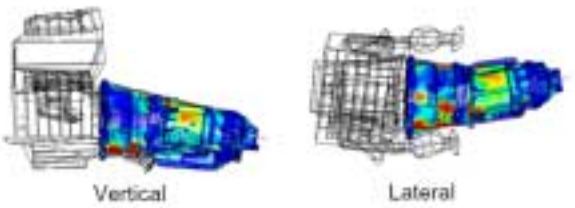
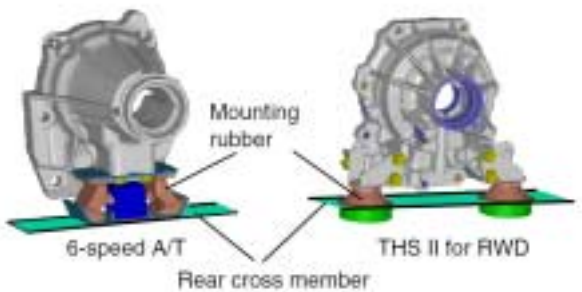


Fig. 14 Vibration Mode of Power Plant Bending Resonance



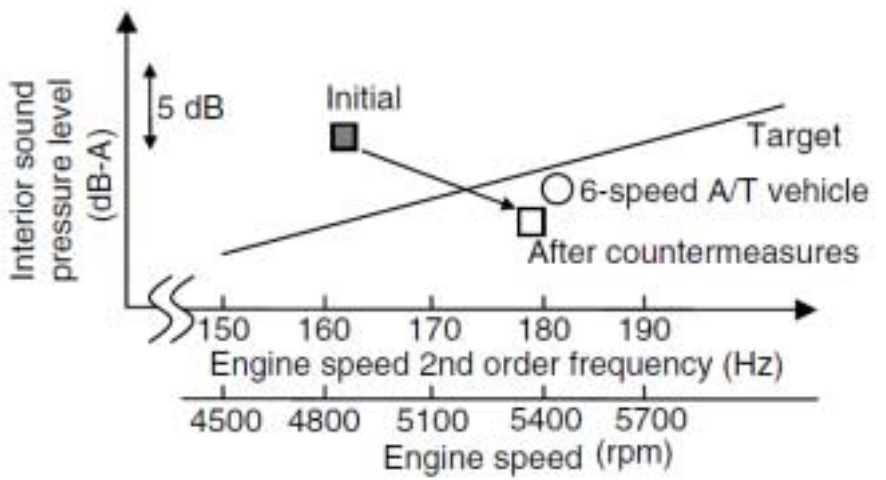


Fig. 17 Interior Booming Noise Level Caused by the Second Order Component of Engine Speed and Power Plant Bending Resonance Frequency

Reduce Effect of 2nd order Couple:

Mode Manage.

1. Increase power plant bending mo
2. Move mount to a nodal point

Nodal Mounting

3. Embed mounts inside cross mem
4. Reduces distance from principle elastic axis to CG

Effective Isolation

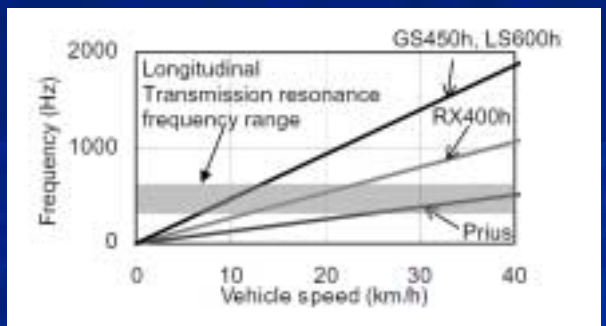
5. Optimized vertical to lateral rate ratio

Copyright © 2009 SAE International

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

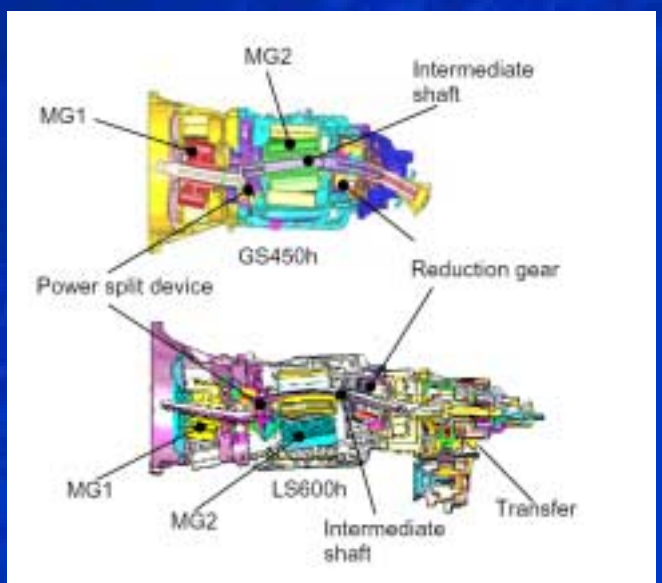


Reduce 24th order Loads:

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

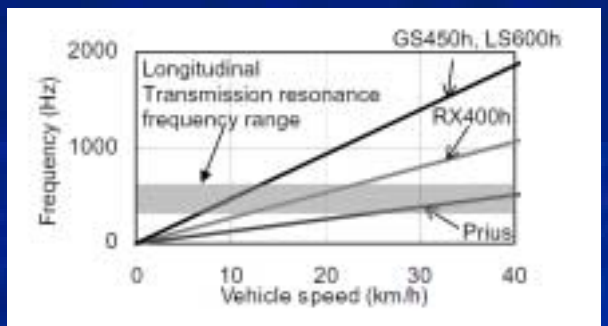


Copyright © 2009 SAE International

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

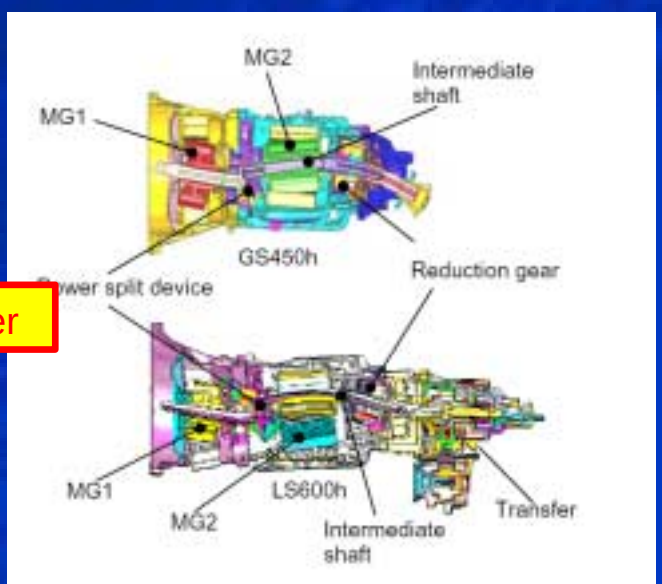
Reduce Effect of 24th order Loads:

Mode Manage.

Damper

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

Downstream



Copyright © 2009 SAE International

Structure Borne NVH Workshop

- Introduction
- Low Frequency Basics
- Mid Frequency Basics
- *Live Noise Attenuation Demo*
- Case Studies
  - Interior Noise Diagnosis using VINS
  - NVH Principles: Applications
- Closing Remarks

Jianmin Guan

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 2011 NVH Conference
Structure Borne NVH Workshop

Thank You for Your Time!

Q & A

Structure Borne NVH References

Primary References (Workshop Basis: 4 Papers)

1. A. E. Duncan, et. al., “Understanding NVH Basics”, IBEC, 1996

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.

4. S. D. Gogate, et. al., “’Digital Prototype’ Simulations to Achieve Vehicle Level NVH Targets in the Presence of Uncertainties’”, SAE 2001-01-1529, NVH Conference Proceedings, May 2001

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

Supplemental Reference Recommendations

5. T.D. Gillespie, Fundamentals of Vehicle Dynamics, SAE 1992
(Also see SAE Video Lectures Series, same topic and author)

6. D. E. Cole, Elementary Vehicle Dynamics, Dept. of Mechanical Engineering, University of Michigan, Ann Arbor, Michigan, Sept. 1972

7. J. Y. Wong, Theory of Ground Vehicles, John Wiley & Sons, New York, 1978

8. N. Takata, et.al. (1986), "An Analysis of Ride Harshness" Int. Journal of Vehicle Design, Special Issue on Vehicle Safety, pp. 291-303.

9. T. Ushijima, et.al. "Objective Harshness Evaluation" SAE Paper No. 951374, (1995).