SAE 2005 NVH Conference
Structure Borne NVH Workshop

Presenters:

Alan Duncan	**Material Sciences Corp.**
Greg Goetchius	**Material Sciences Corp.**
Sachin Gogate	**DaimlerChrysler Corp.**
Greg Hopton	**LMS North America**

Structure Borne NVH Outline

- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics
- *Live Noise Attenuation Demo*
- Mid Frequency Basics
- Utilization of the NVH Toolbox
- Closing Remarks

The Fundamental Secret of Structure Borne NVH Performance

1st Corollary

Revealed here today !

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

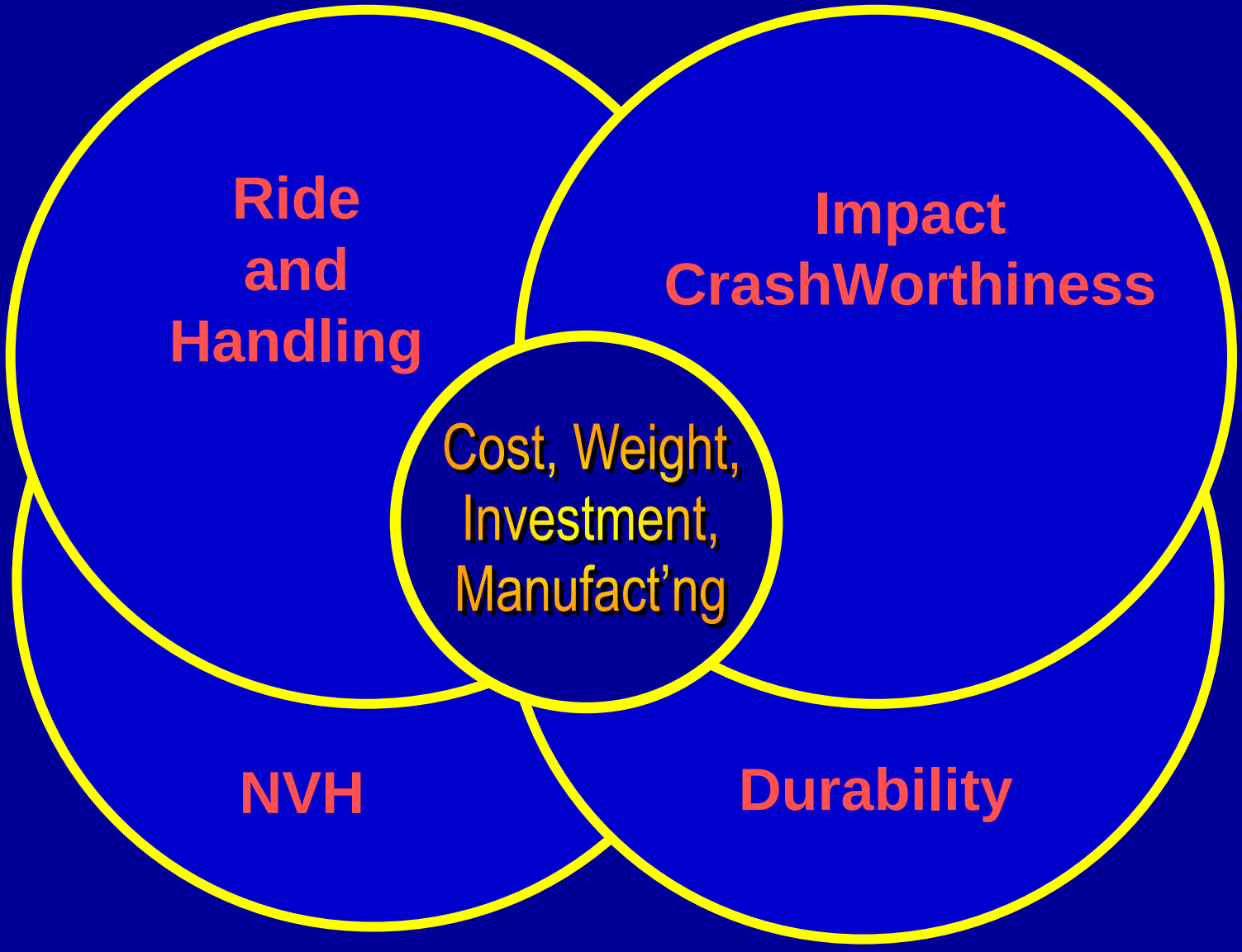
Supplemental Reference Recommendations

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).

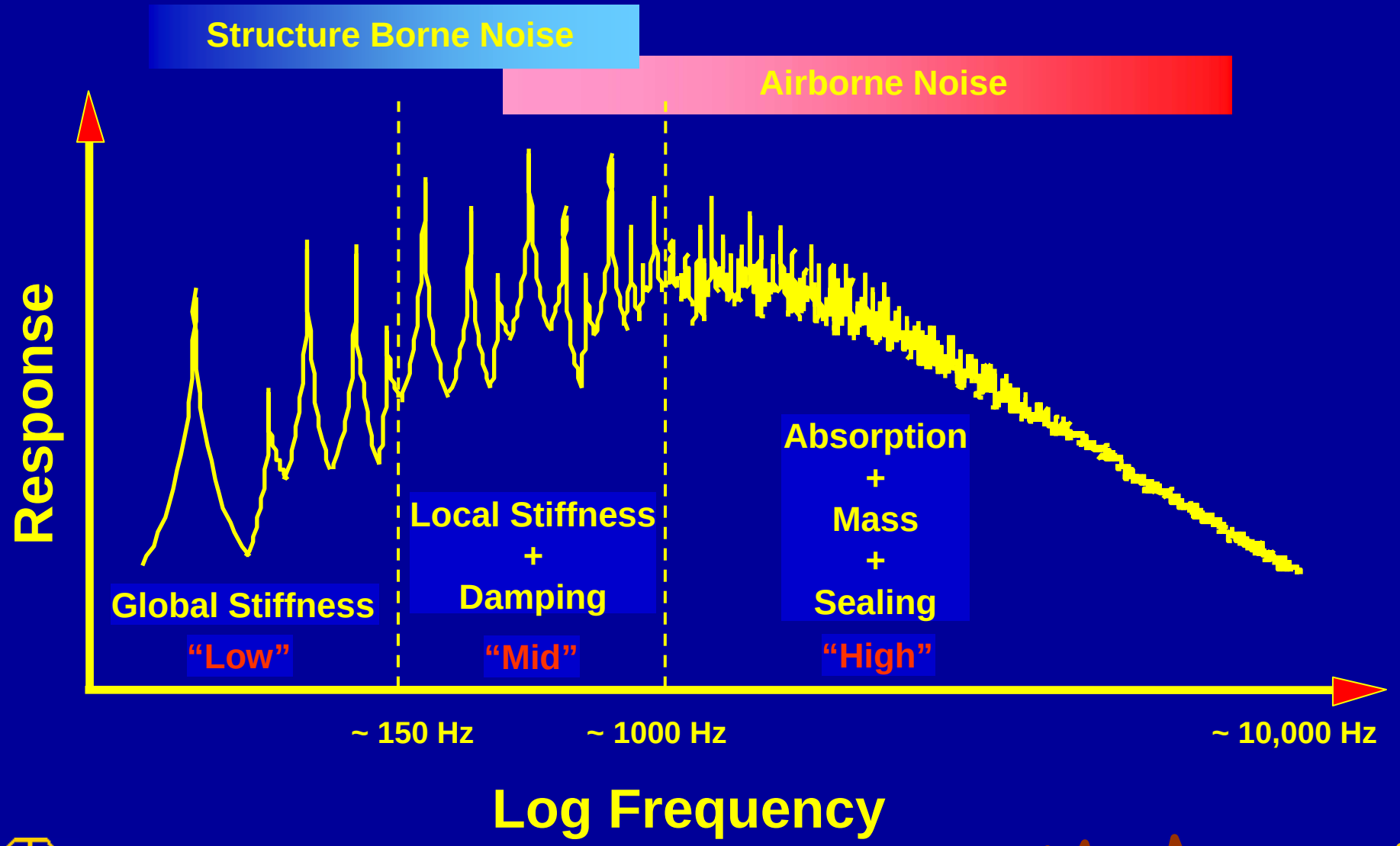
Structure Borne NVH Outline

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

Competing Vehicle Design Disciplines



Automotive NVH Frequency Range



Structure Borne NVH Outline

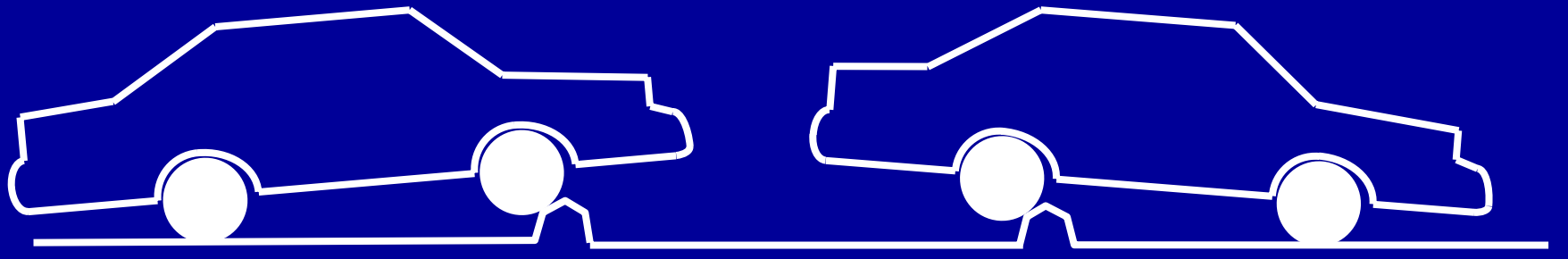
- Introduction
- **Ride Balance in the Ride Range**
- NVH Load Conditions
- Low Frequency Basics

Study of Ride Balance

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

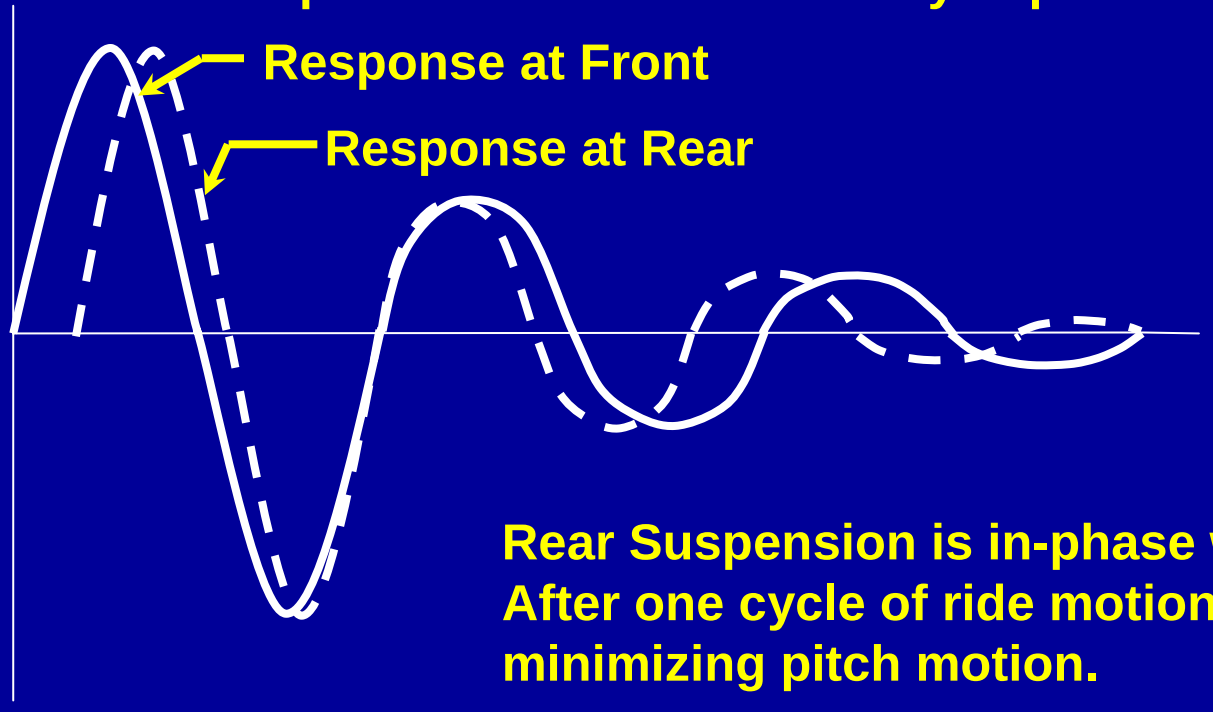
Ride Balance Study

Vehicle Traversing a Bump



Impact at Front Suspension

Followed by Impact at Rear Suspension

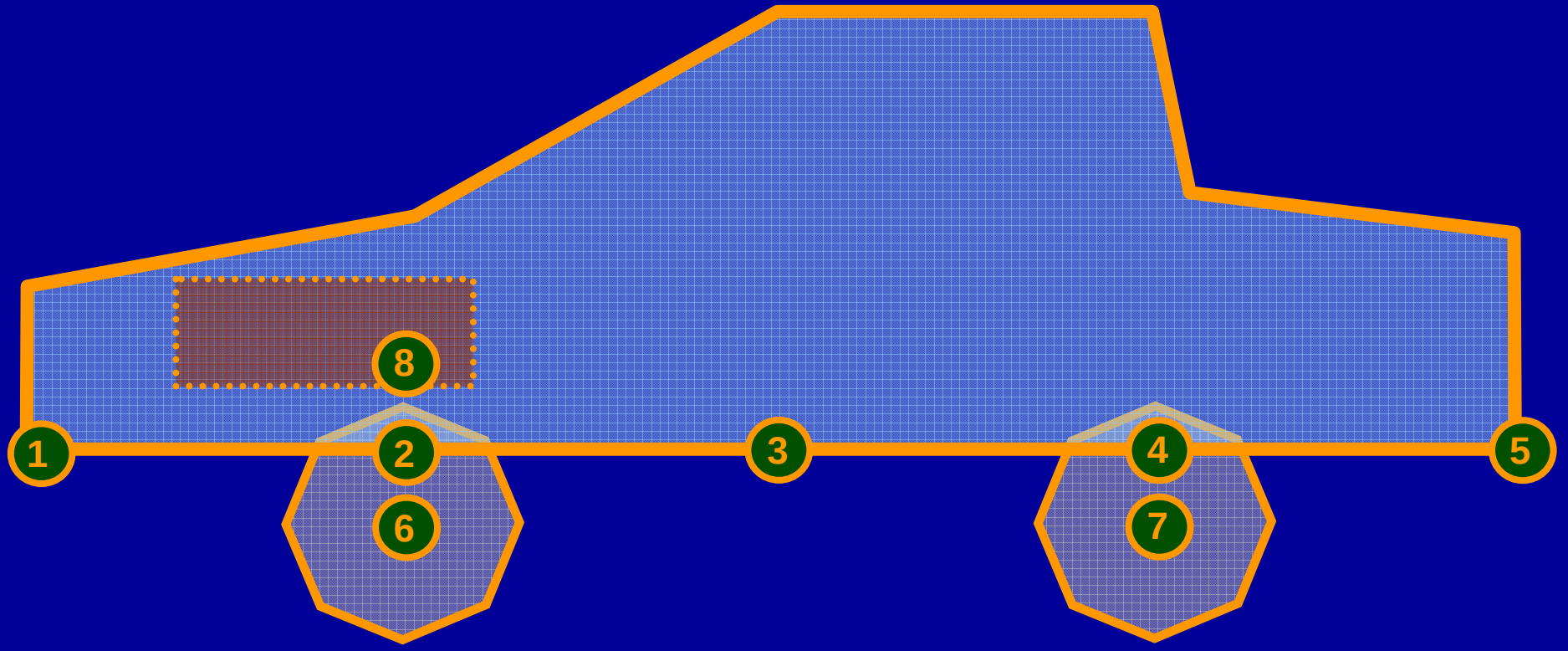


Rear Suspension is in-phase with Front After one cycle of ride motion, thus minimizing pitch motion.

See Ref. 5, Gillespie

NVH Model of Unibody Passenger Car

Symbolic Outline



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

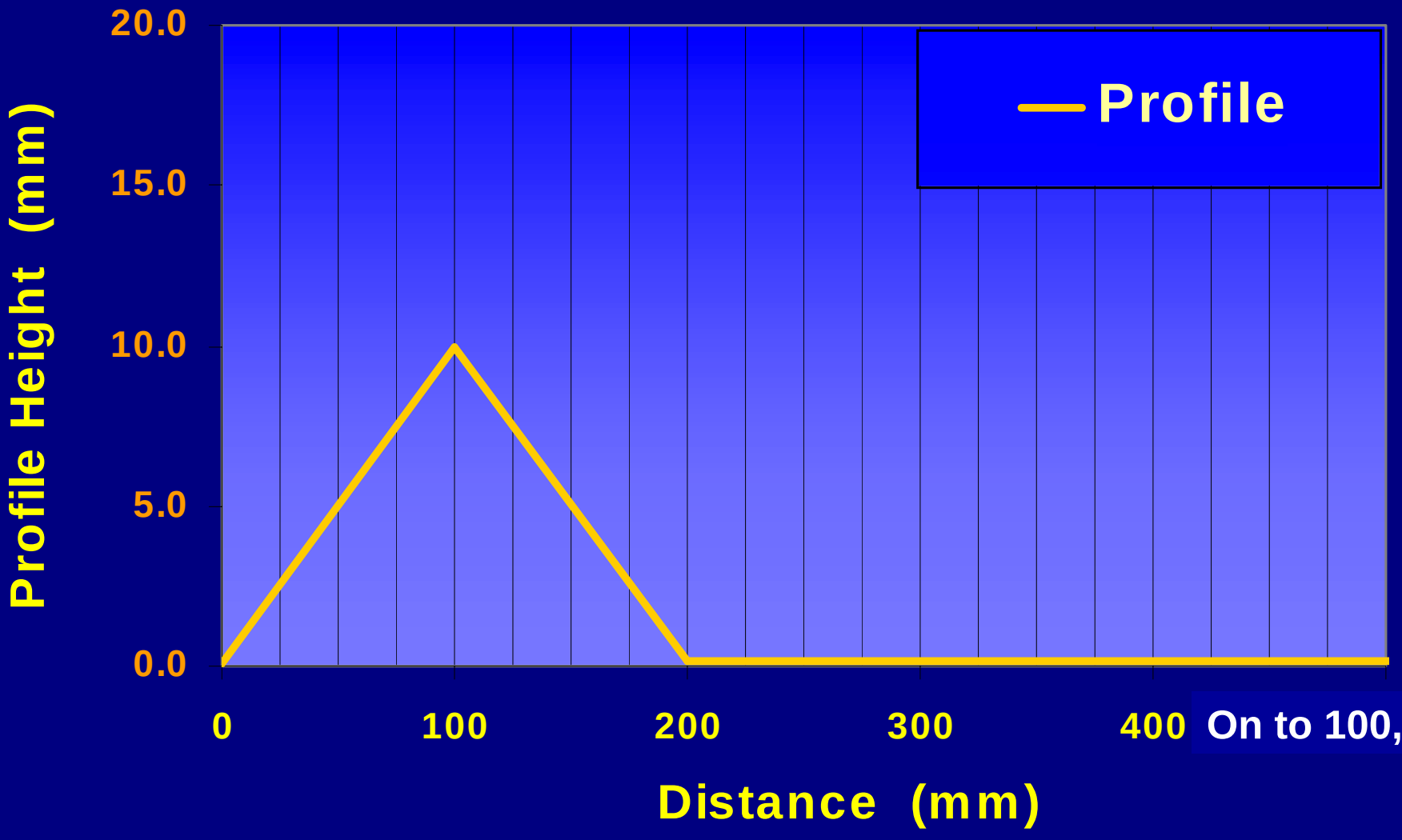
Beam mass lumped on grids like a beam

M2,3,4 =2 * M1,5

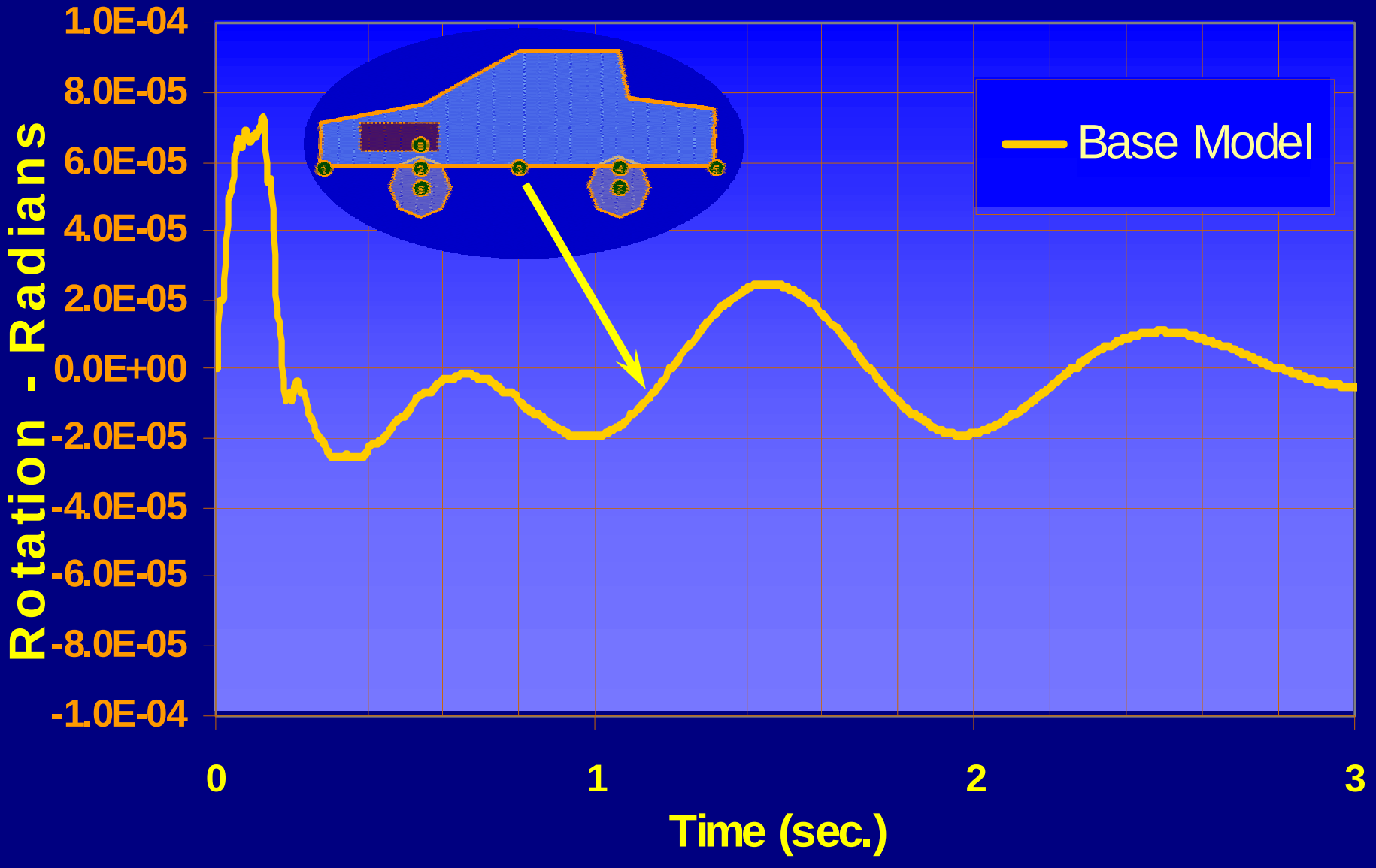
From Reference 6

www.AutoAnalyst.com

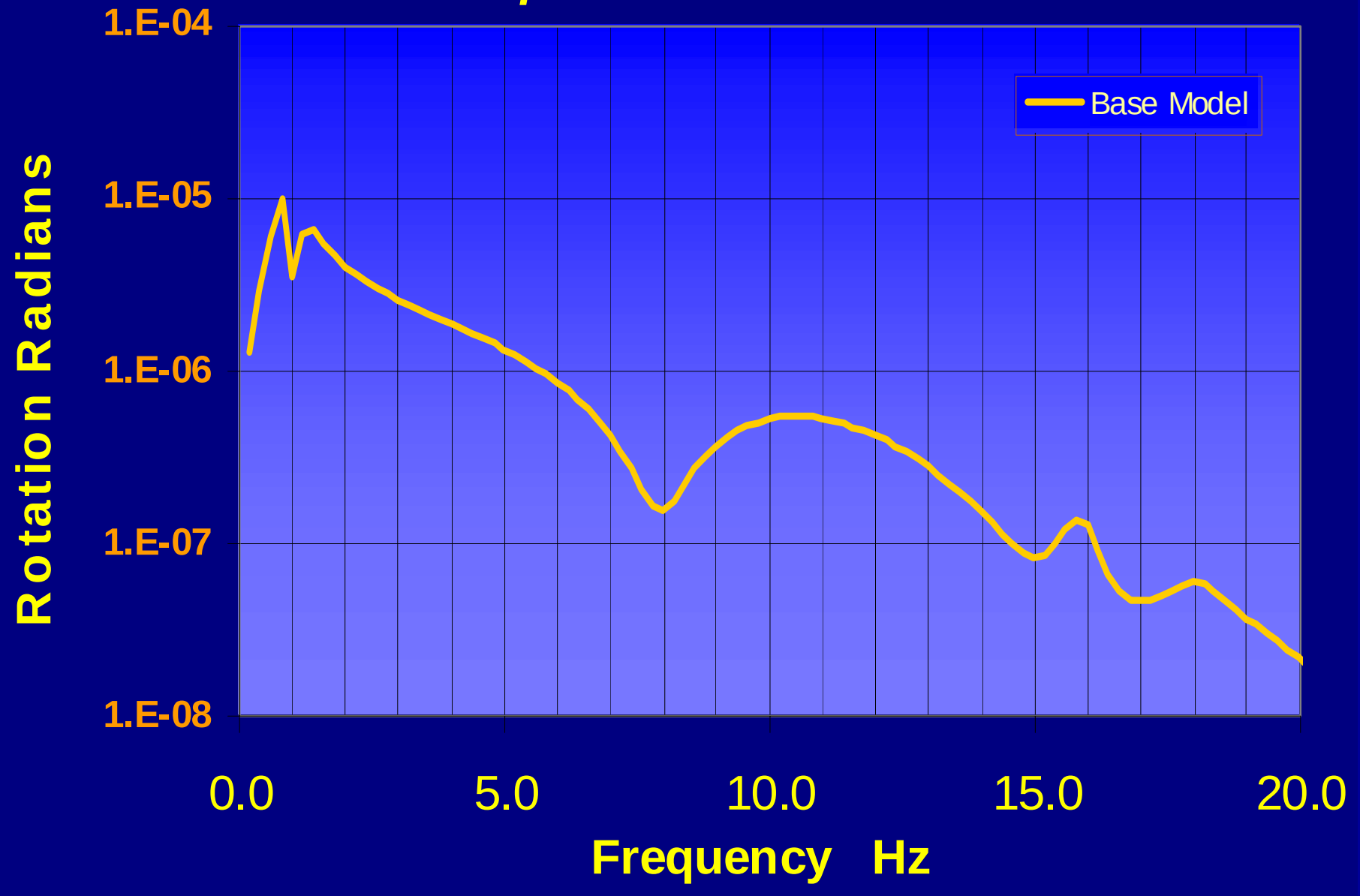
Excitation Bump Profile



Pitch at Mid-Car DOF3



Pitch Response - Baseline Model



X (f) = Fourier Transform of X (t)

$$\{X(f)\} = [H(f)] * \{F(f)\}$$

Fourier transform of the output for unit input forces system **OUT / IN** FRF (Known Load = Unity)

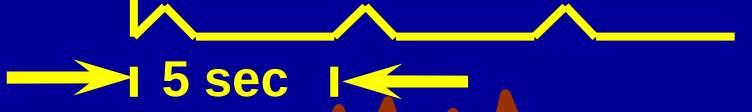
Fourier transform of the input forces, vector of all input possibilities

Spring Analogy (C = 1 / K) X = C * F

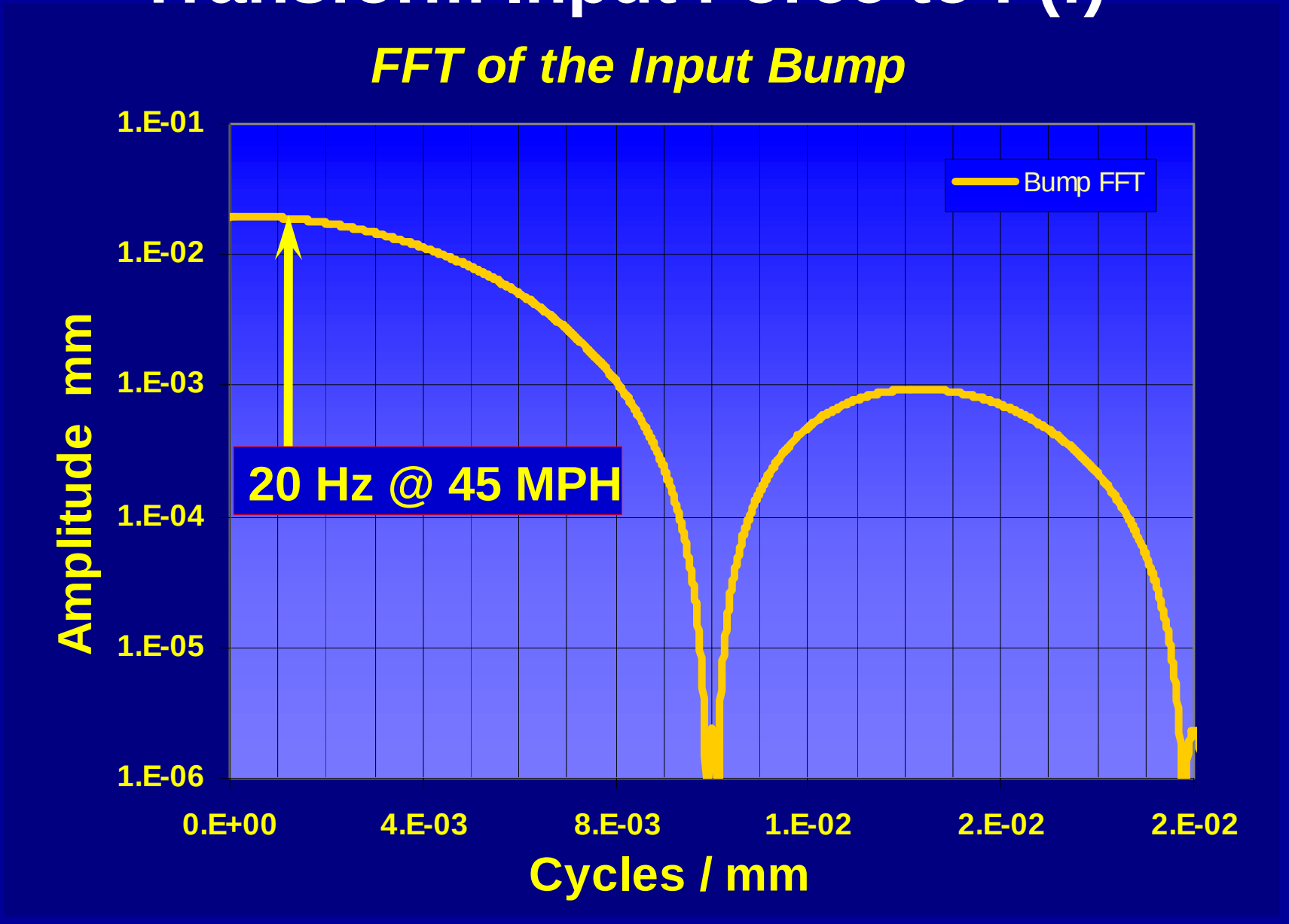
Caveat

X (t) must be Periodic
Linear System

F(t) is Periodic

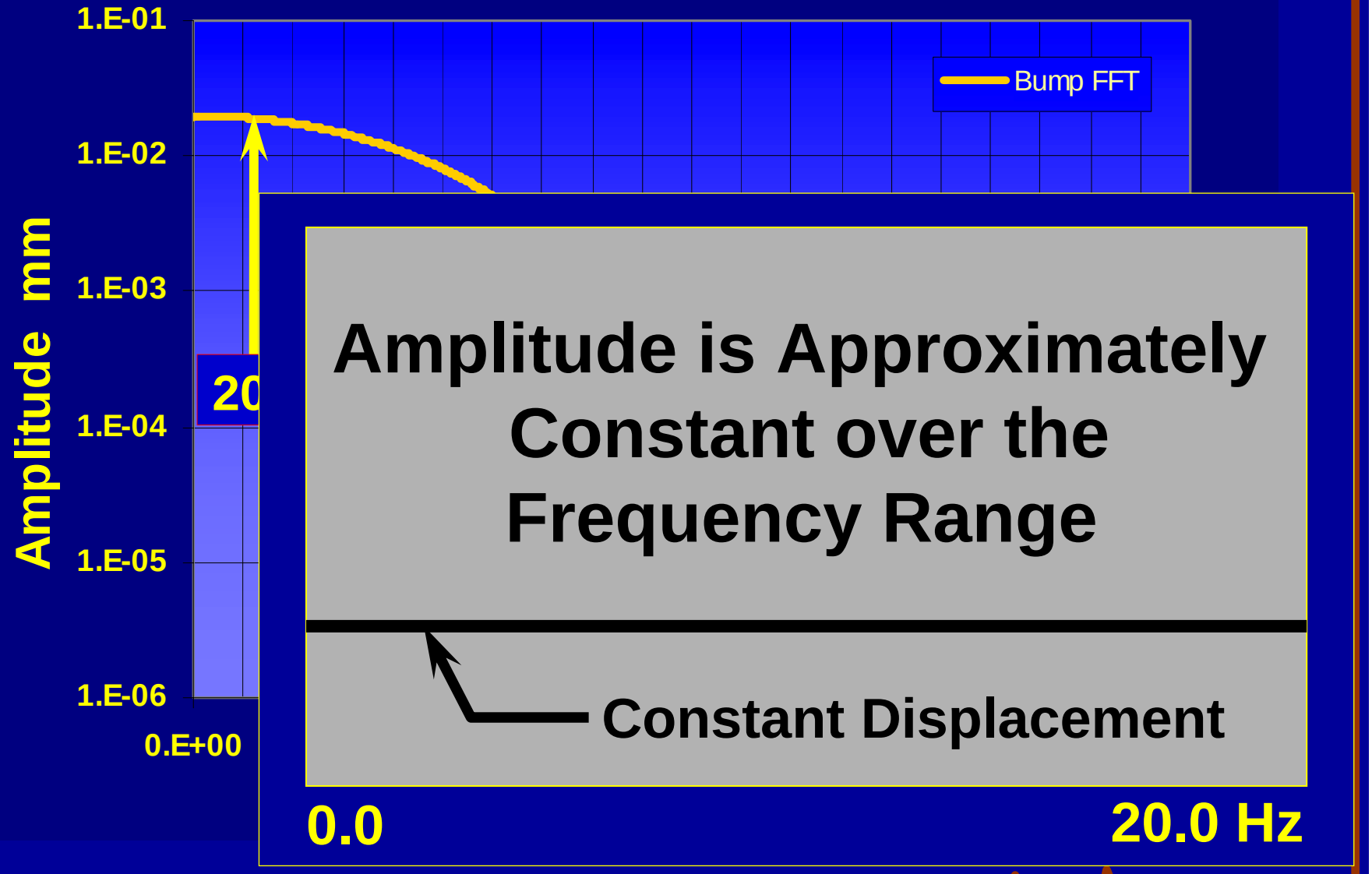


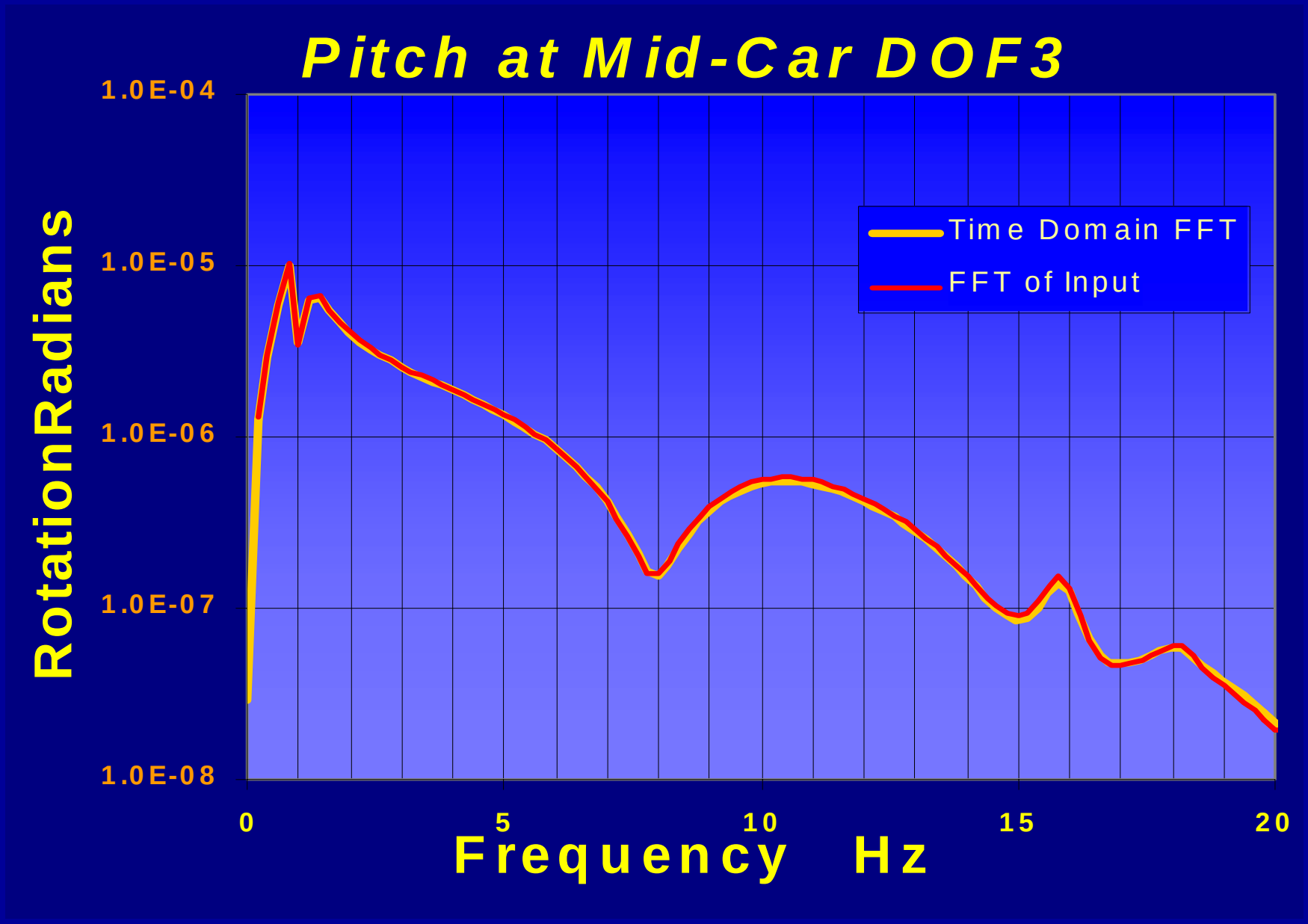
Transform Input Force to F(f)



Transform Input Force to F(f)

FFT of the Input Bump




Pitch at Mid-Car DOF3
RotationRadians
1.0E-04
1.0E-05
1.0E-06
1.0E-07
1.0E-08
Time Domain FFT
FFT of Input
0
5
10
15
20
Frequency Hz

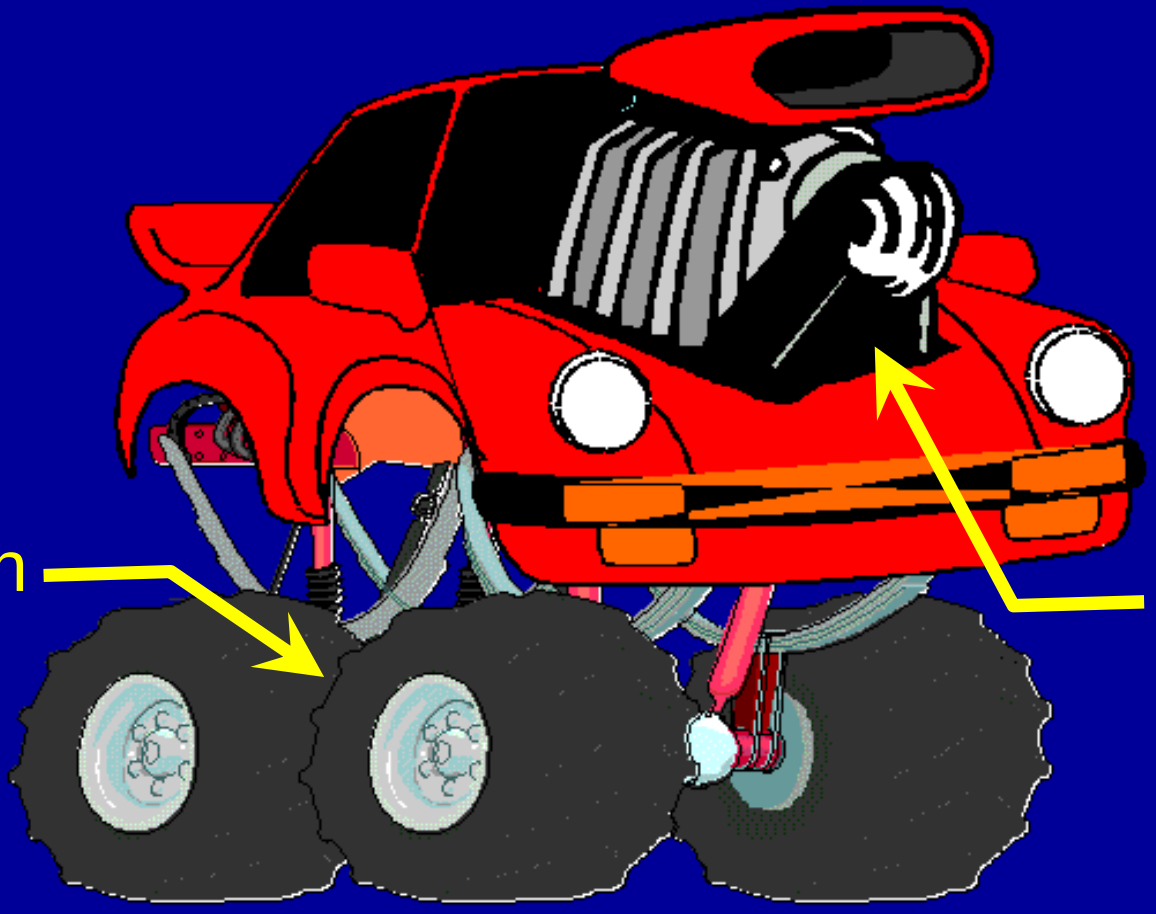
Ride Balance Study Summary

- Demonstrated the Fundamental Bounce and Pitch Modes in the Ride Range
- Demonstrated Transient Response in Time Domain then Obtained the FRF
- Derived Direct Computation of FRF in the Frequency Domain

Structure Borne NVH Outline

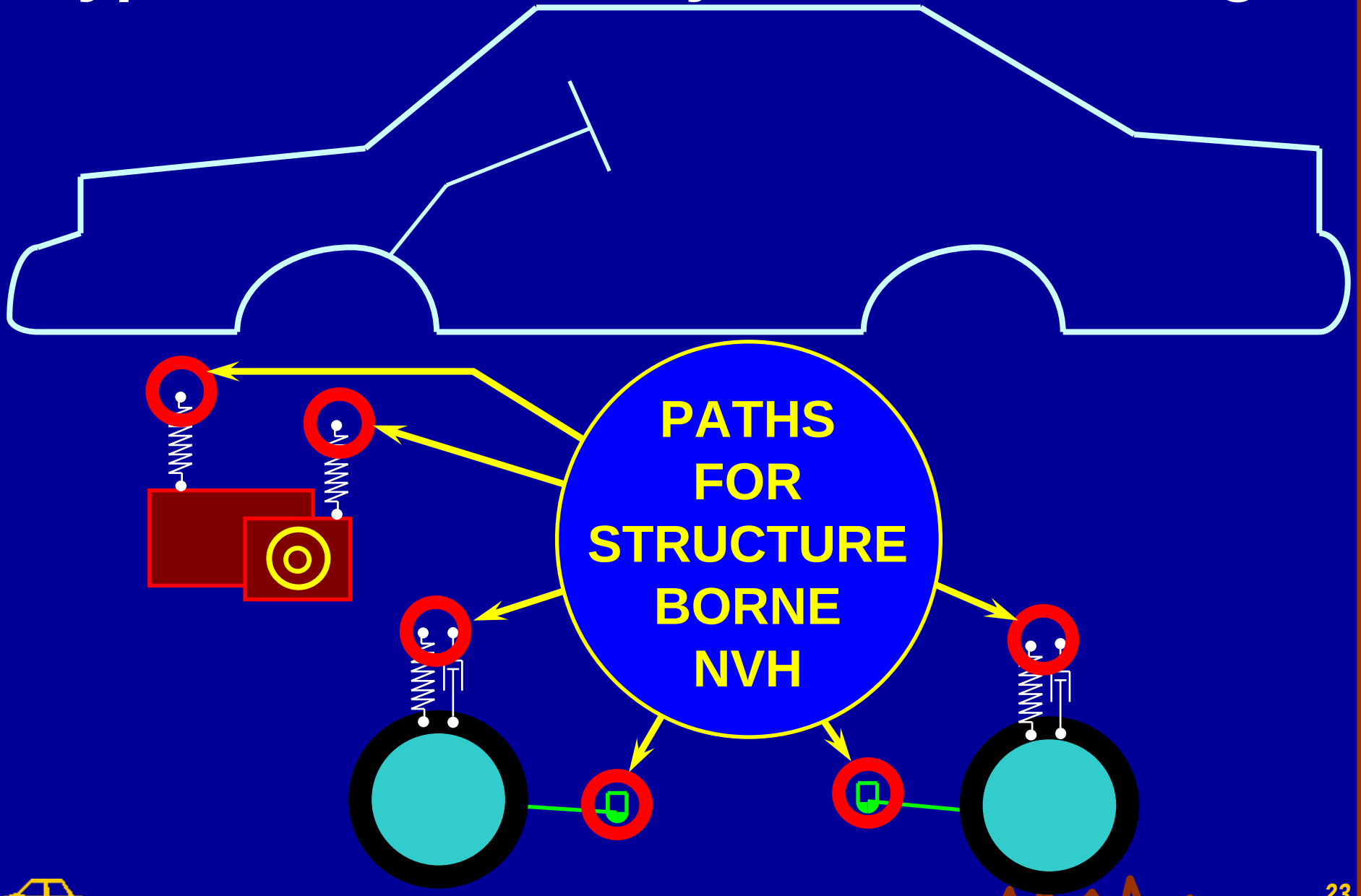
- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics

Noise and Vibration Sources



Two Main Sources

Typical NVH Pathways to the Passenger



NVH Load Conditions

	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 Outline

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New: Harshness Discussion

Structure

Borne

N (Noise)

V (Vibration)

H (Harshness)

Harshness Definition

Excitation:

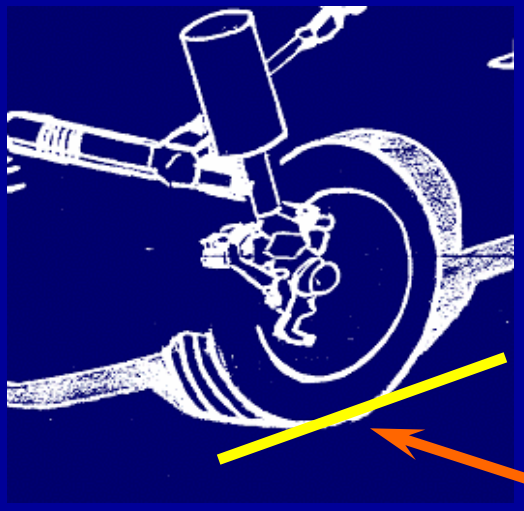
Transient event transmitted through the tires due to road irregularities such as tar strip impacts or similar discrete inputs.

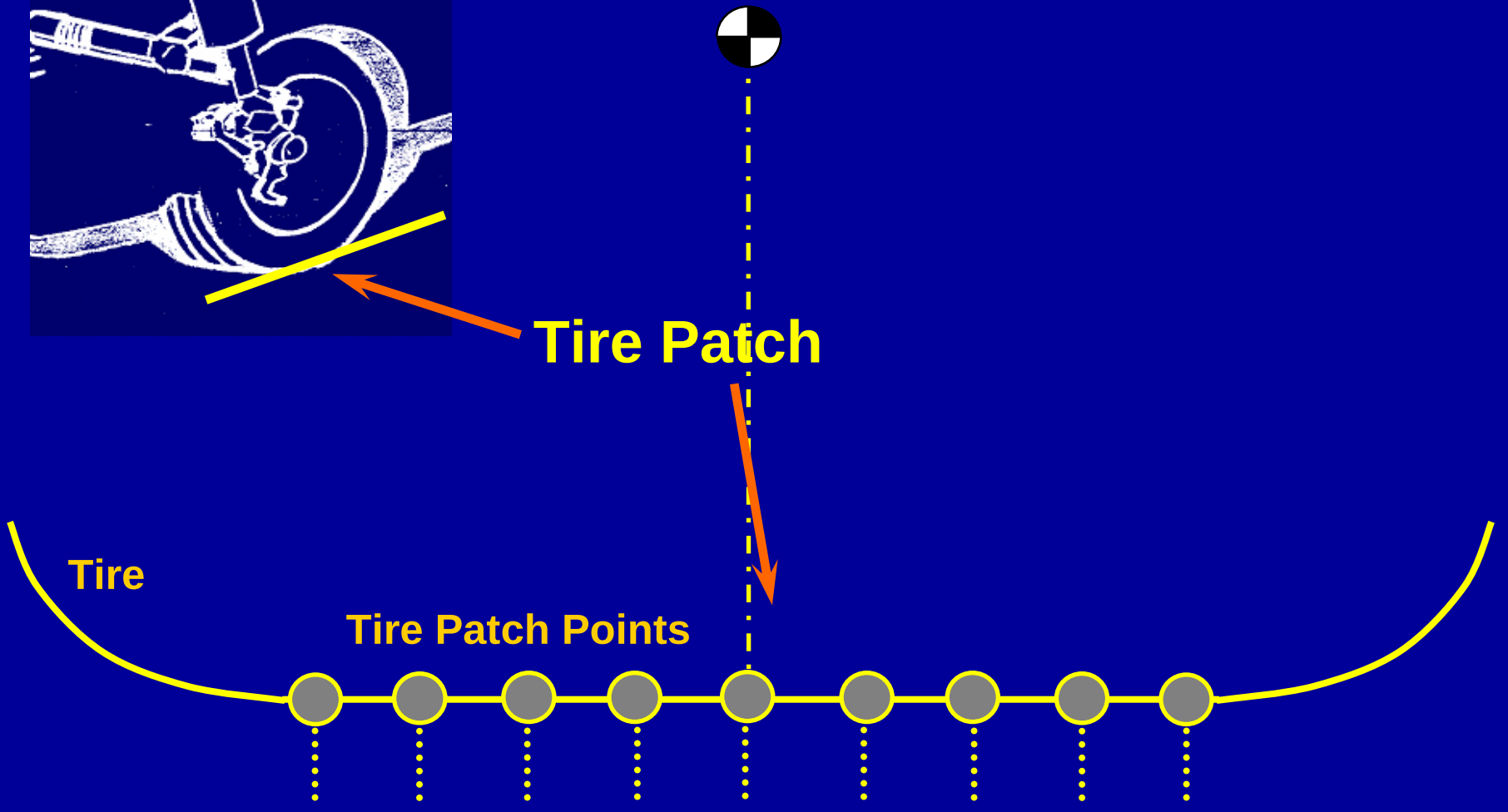
Response:

An unpleasant ride experience characterized by:

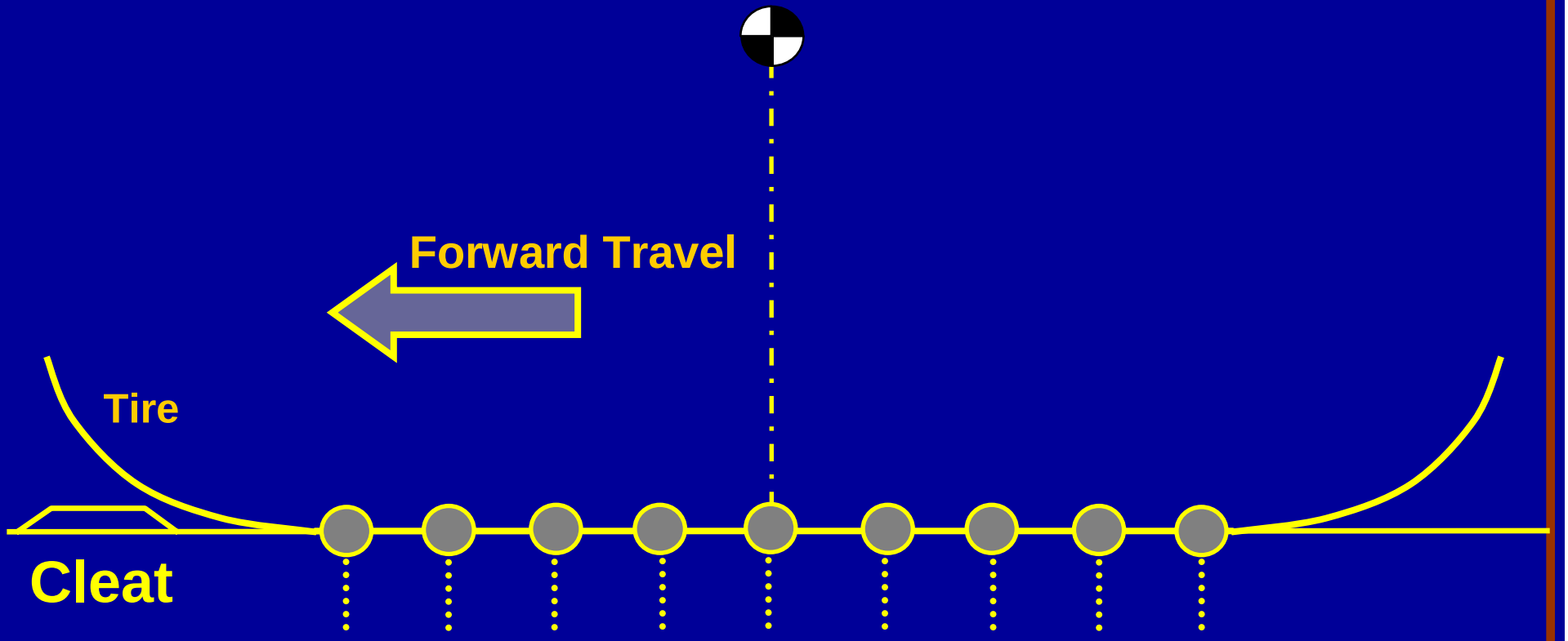
1) **A short duration For/Aft tactile spike (Shock)**
2) **A short duration Noise spike (Impact Boom)**
3) **Extended post-event oscillatory vibration (After Shake)**
4) **A combination of the above ! ! ! ! ! ! !**

Tire Patch Loading Schematic



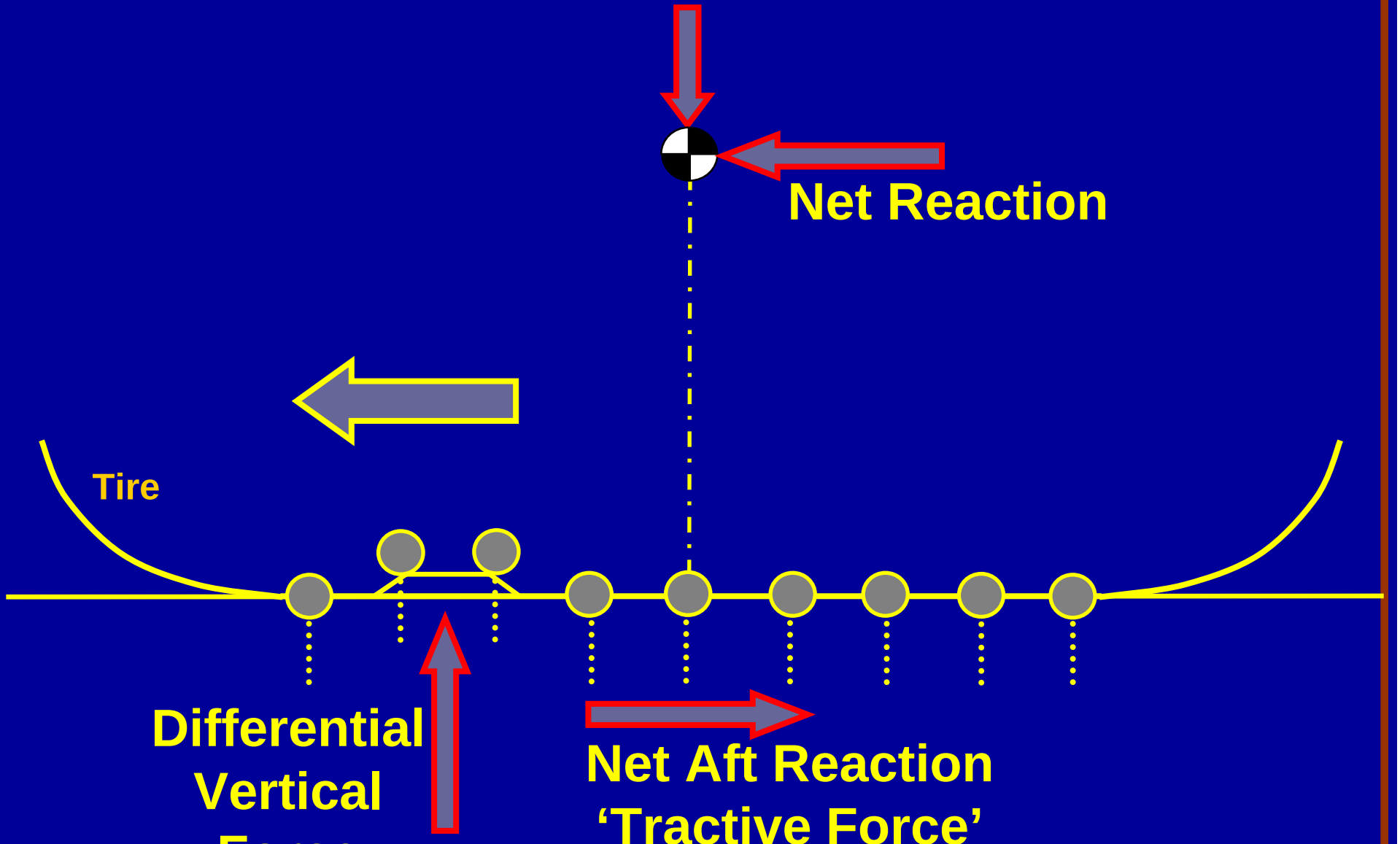


Tire Patch Loading Schematic

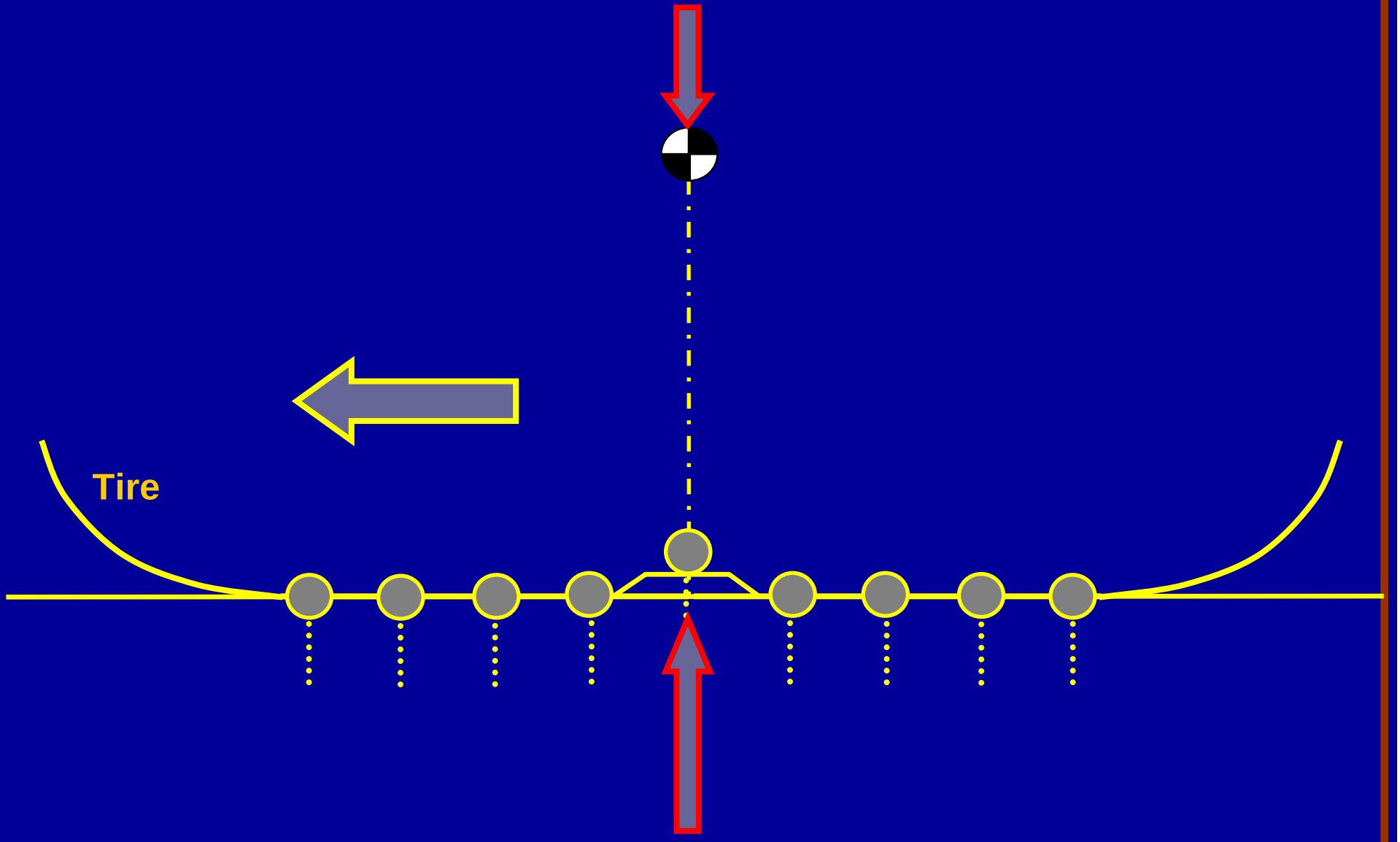


Start of the Impact Event

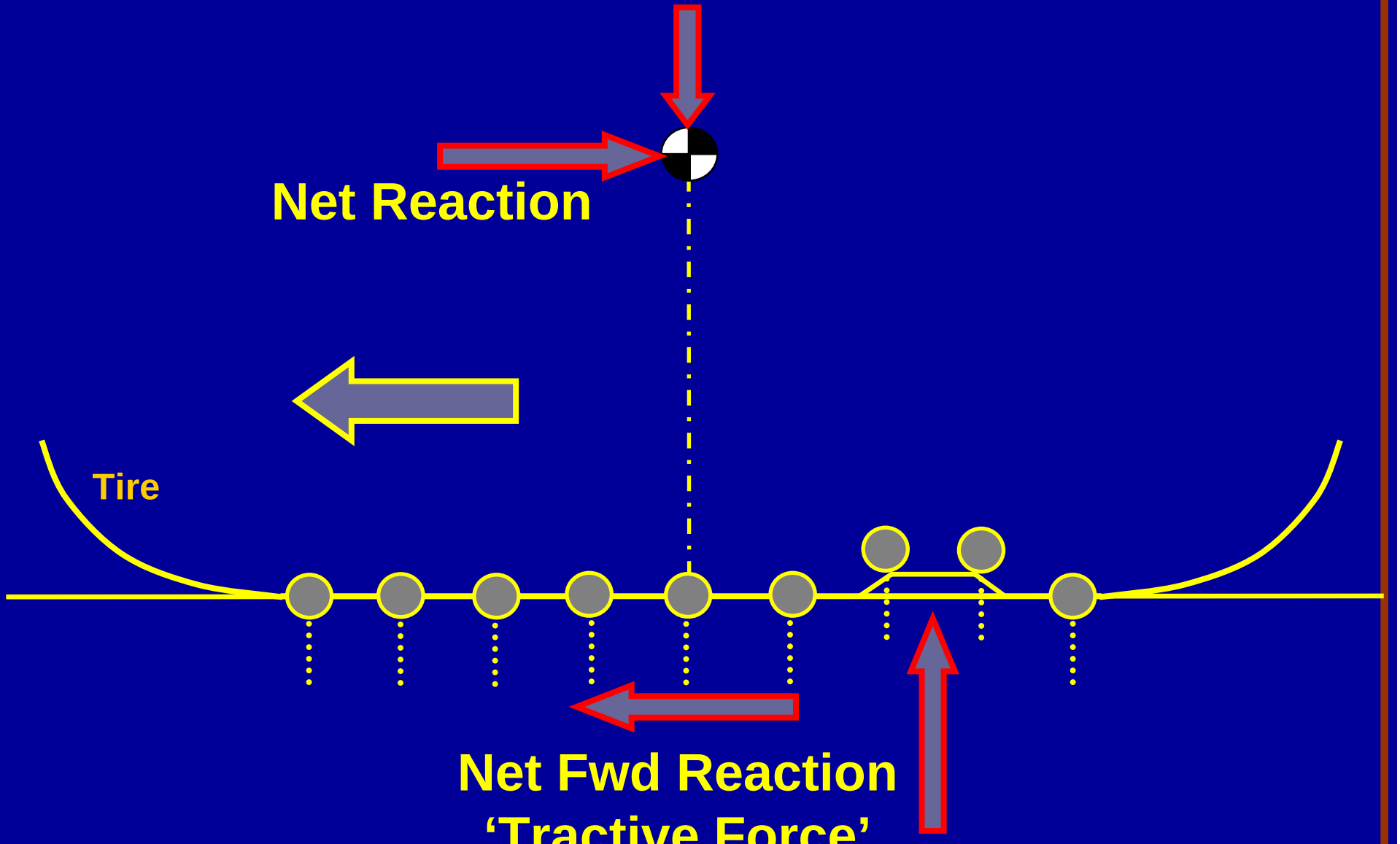
Tire Patch Loading Schematic



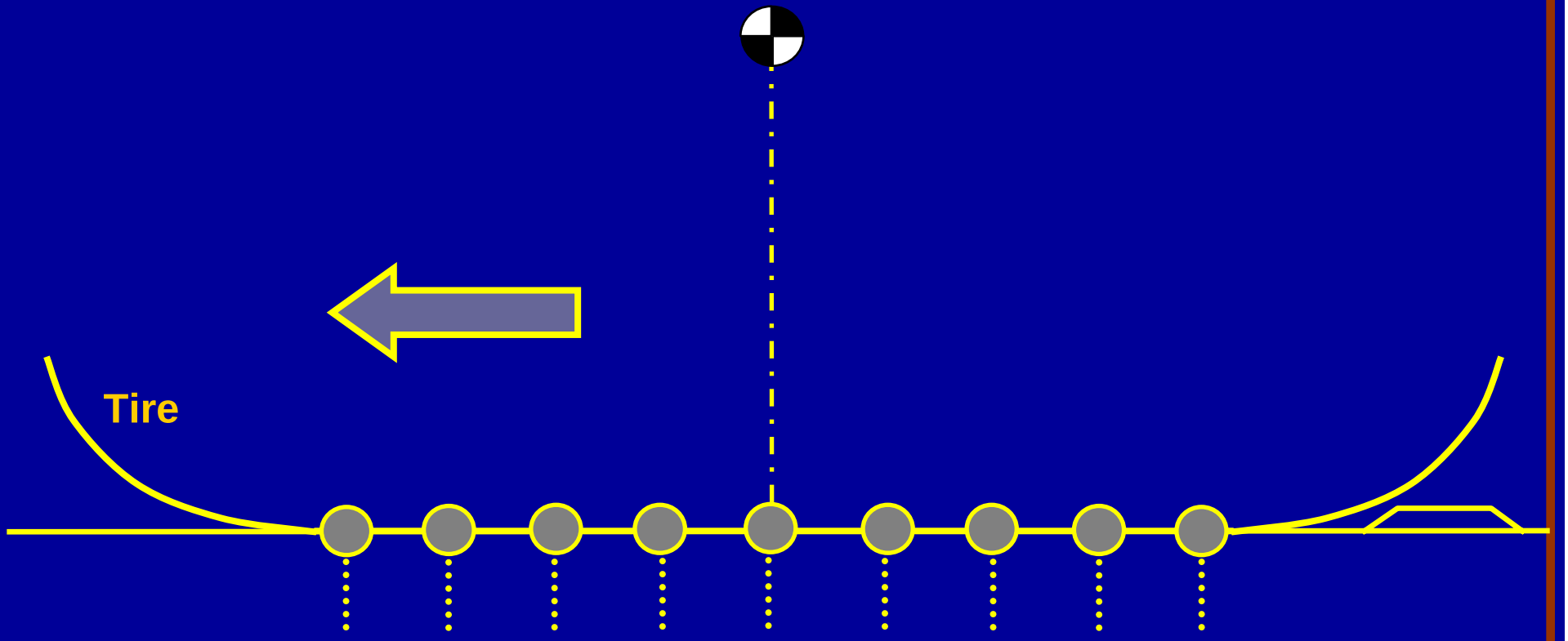
Tire Patch Loading Schematic



Tire Patch Loading Schematic



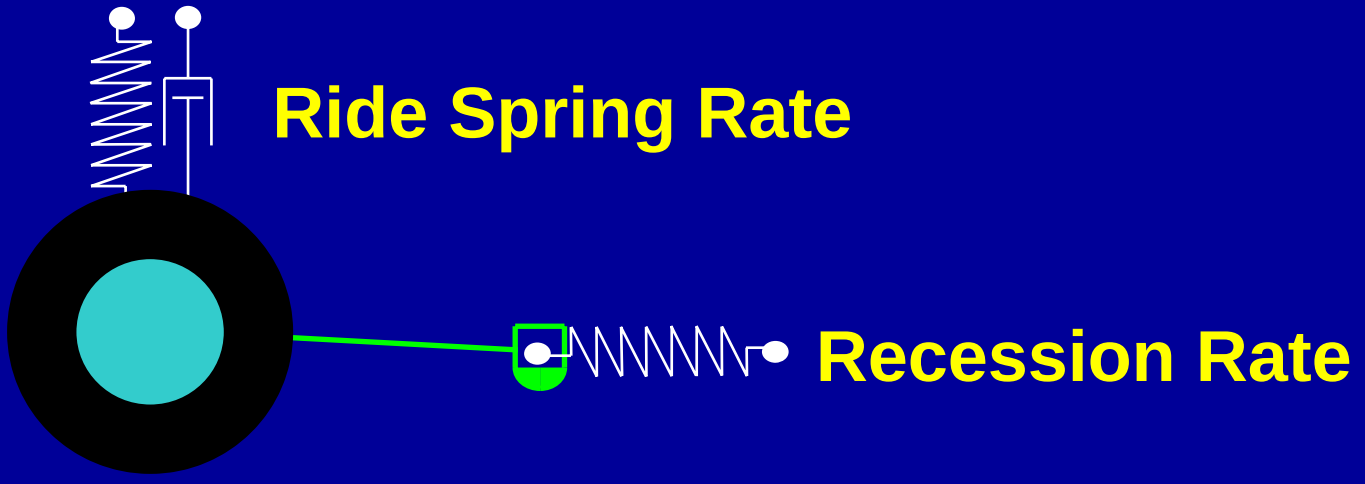
Tire Patch Loading Schematic



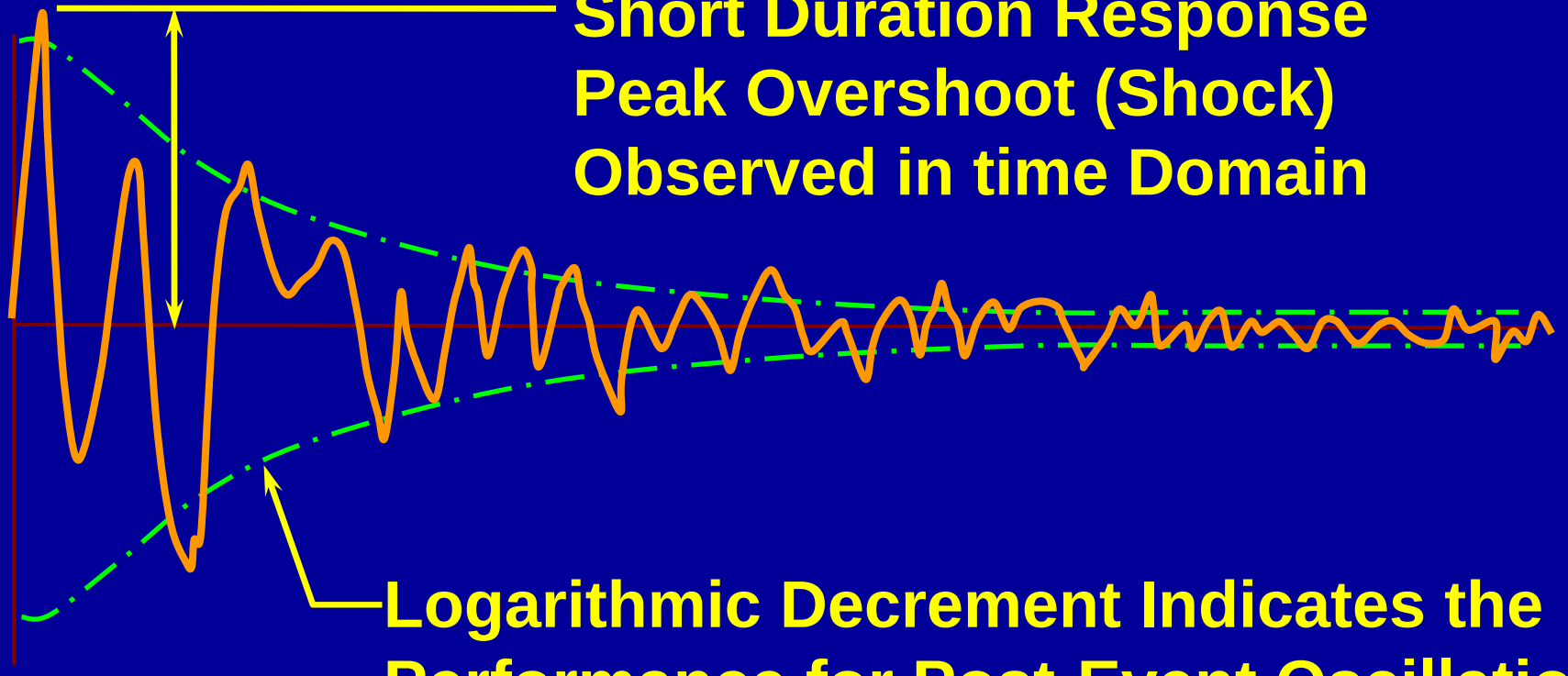
End of the Impact Event

Factors Leading to Increased For/Aft Response

- Sign Reversal in For/Aft Direction
- Recession Rate is Higher than Ride Rate



Vehicle Response to a Cleat Impact Event

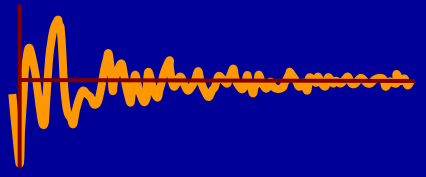
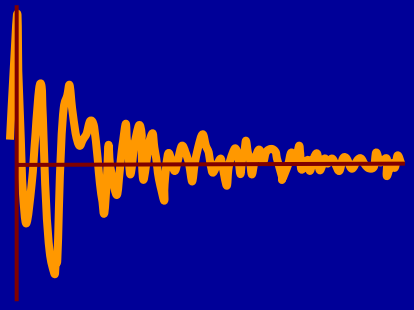
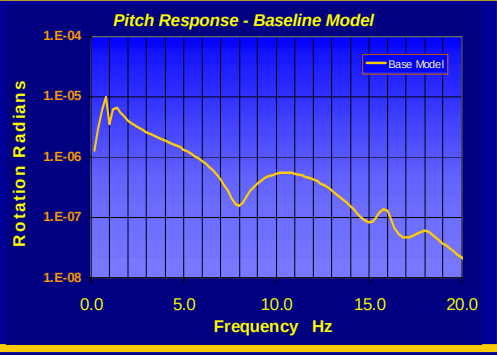
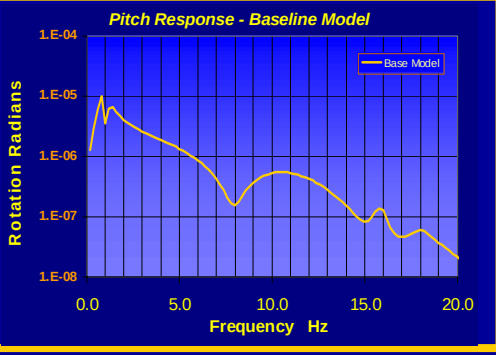


Short Duration Response
Peak Overshoot (Shock)
Observed in time Domain

Logarithmic Decrement Indicates the Performance for Post-Event Oscillation (After Shake)

Ref. 9

Harshness Criteria : Tactile and Noise Response

Harshness Parameters	Tactile Response	Noise Response
Time Domain (Peak Overshoot) Ref. 9		
Frequency Domain (Resonance Content) Ref. 8		

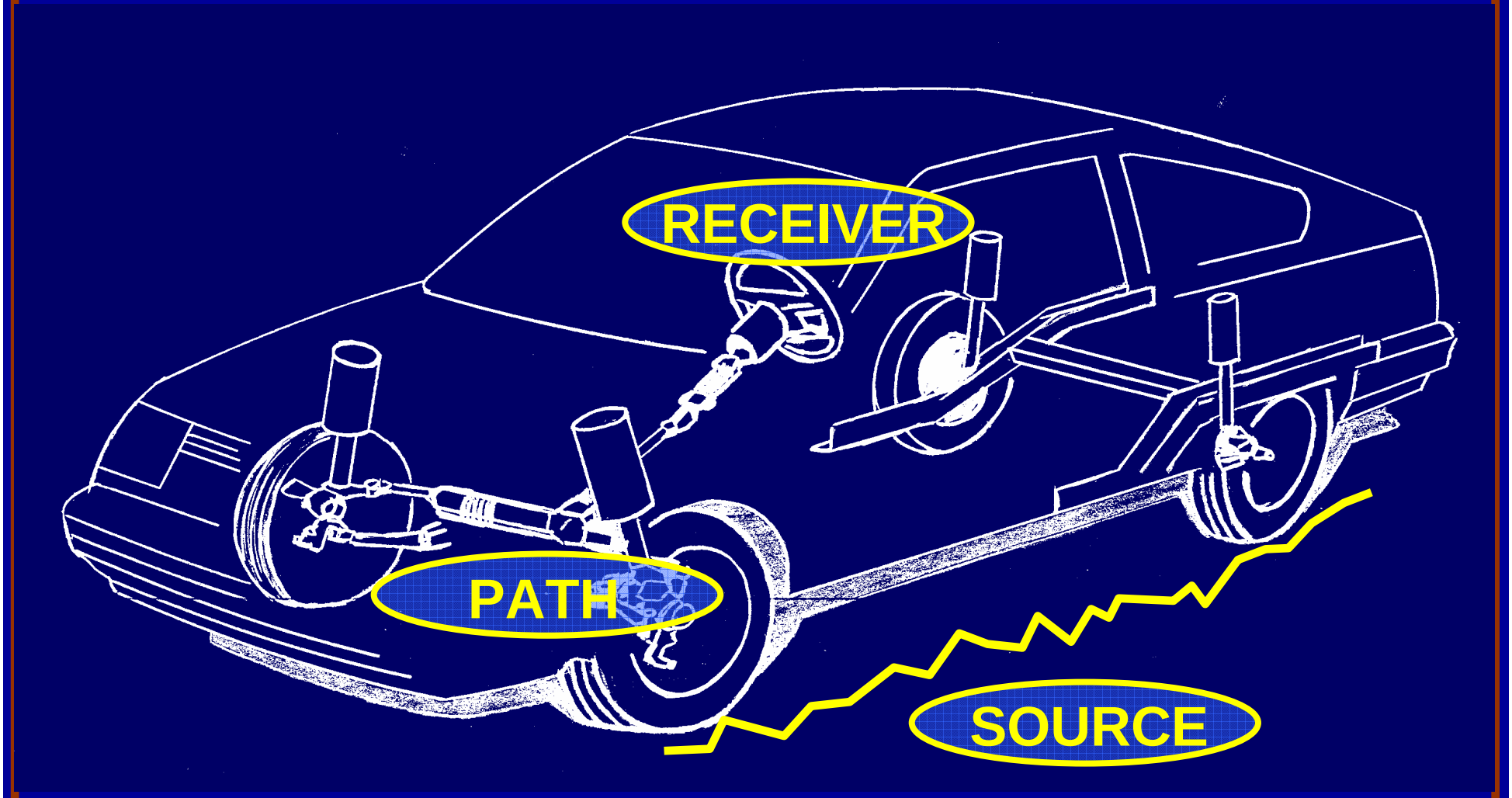
NVH Workshop Topic Outline

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

Low Frequency NVH Basics

- Subjective to Objective Relationships
- Single Degree of Freedom Vibration
- Vibration and Noise Attenuation Strategies

Low Frequency NVH Fundamentals



Subjective to Objective Conversions

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

Receiver Sensitivity is a Key Consideration

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

Represents 1.0 Rating Change

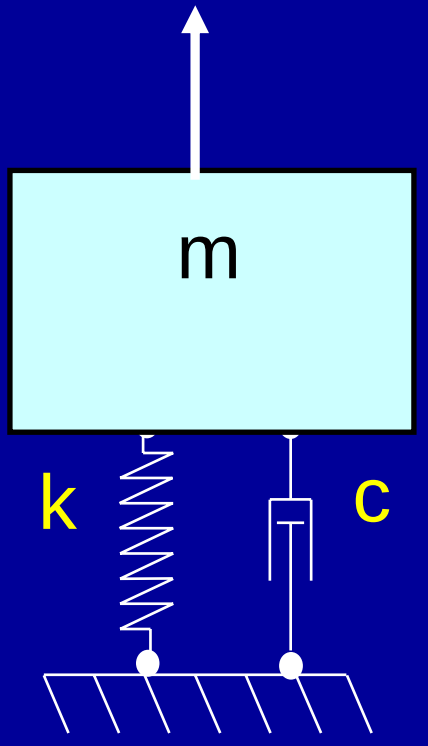
TACTILE: 50% reduction in motion

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

Single Degree of Freedom Vibration

APPLIED FORCE

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



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

Transmitted Force

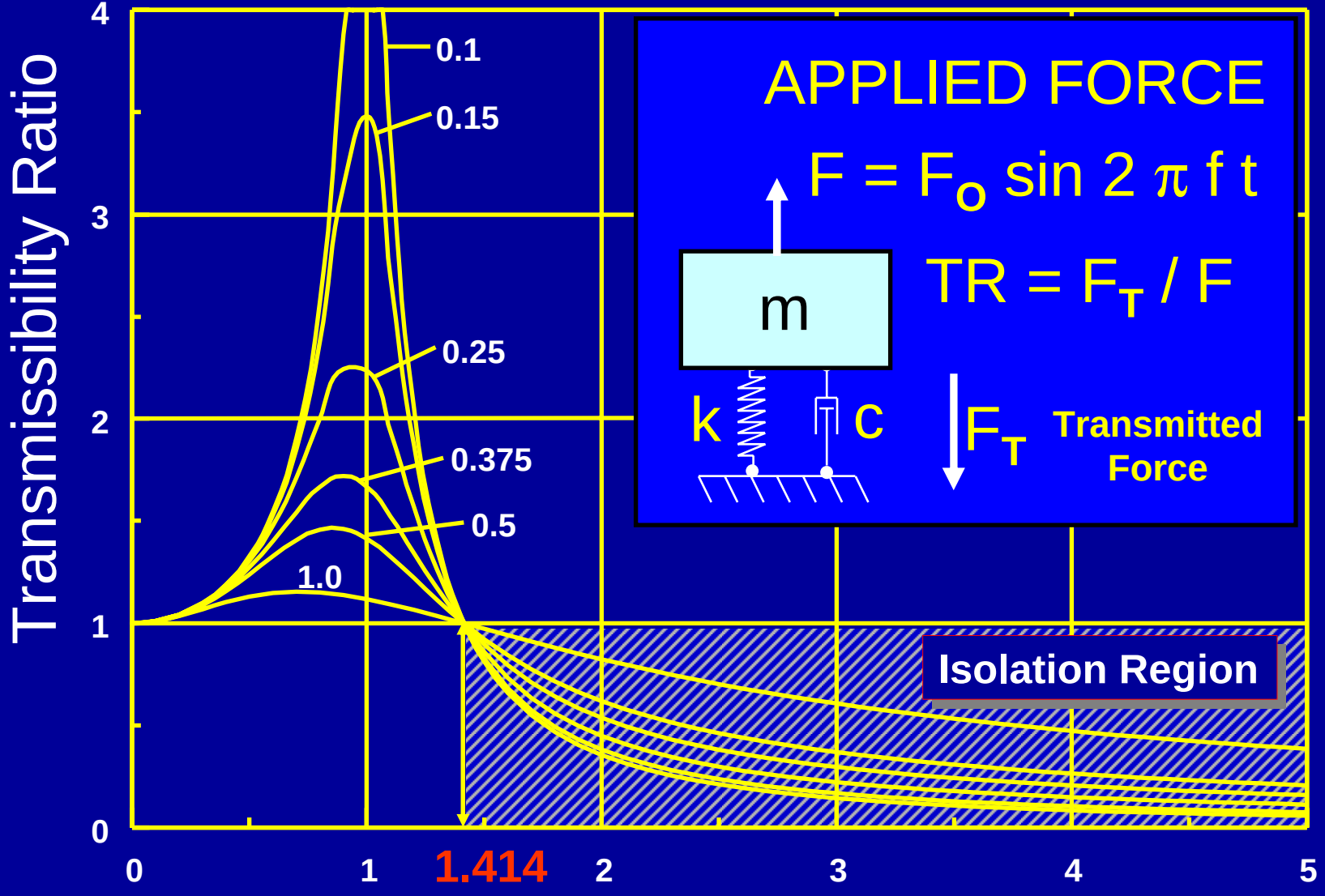
F_T

d = fraction of critical damping

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

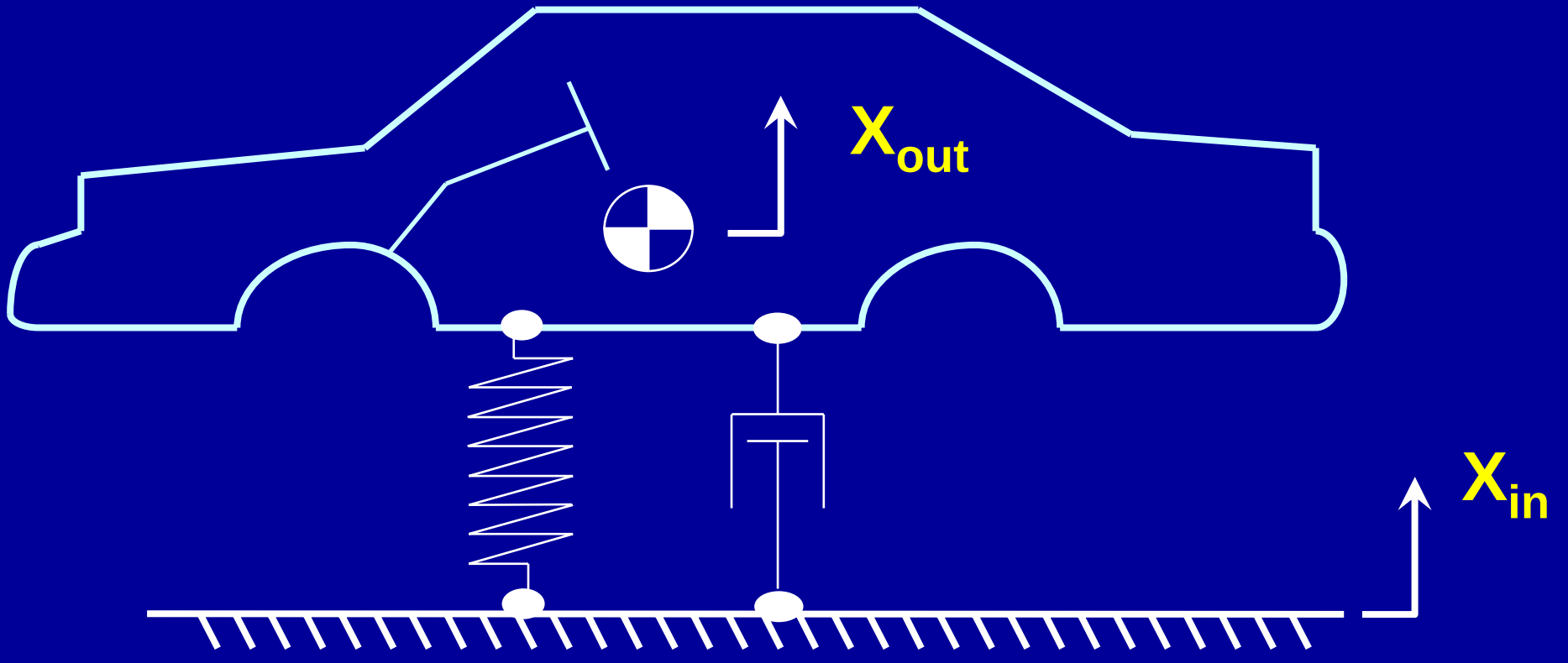
f = operating frequency

Vibration Isolation Principle



Body on Suspension Single DOF Model

Isolation from Base Excitation

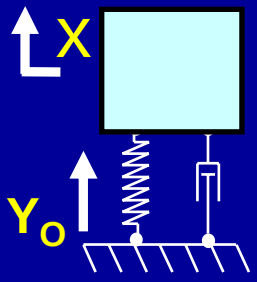


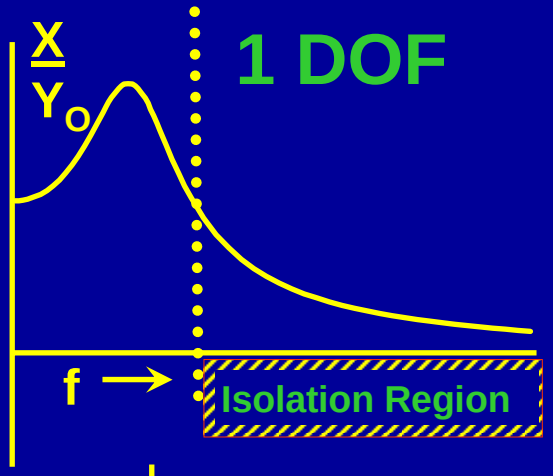
Transmissibility Ratio is X_{out} / X_{in}

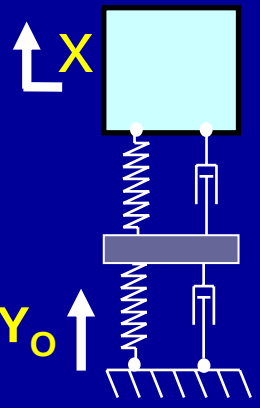
Example: Vertical Ride Mode at 1.3 Hz provides isolation starting at 1.8 Hz. This provides isolation for the first order Hop and Tramp modes.

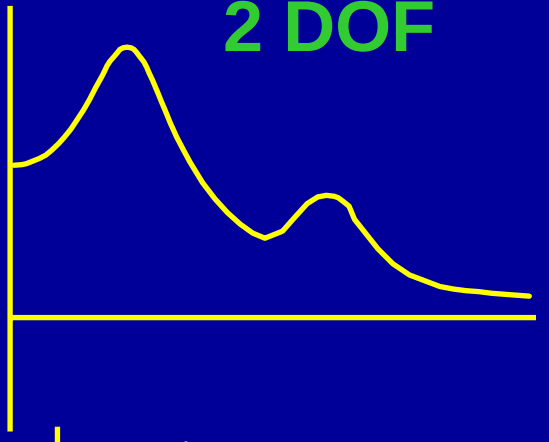
Simplified Models from 1 to 8 DOF's

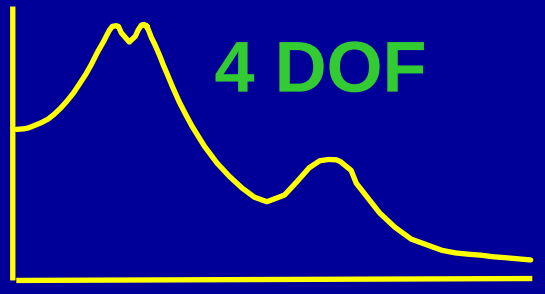
Enforced Base Motion

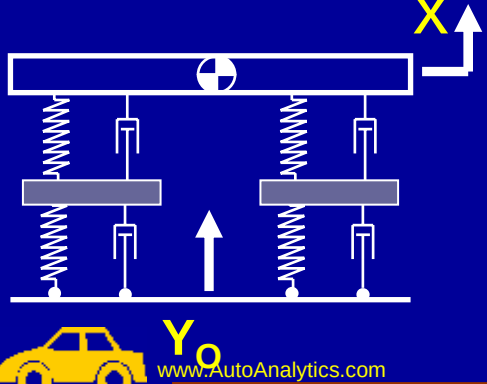


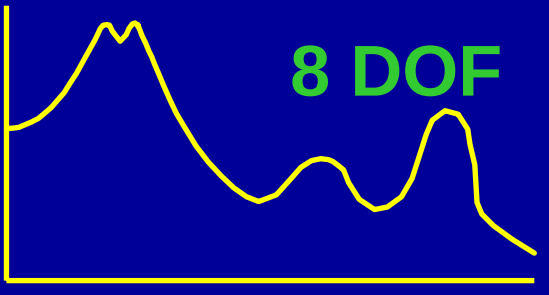


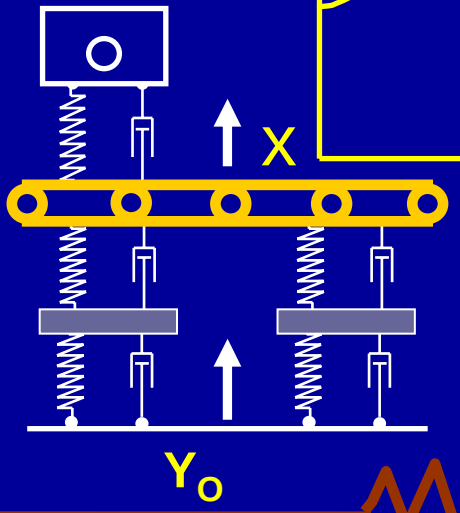




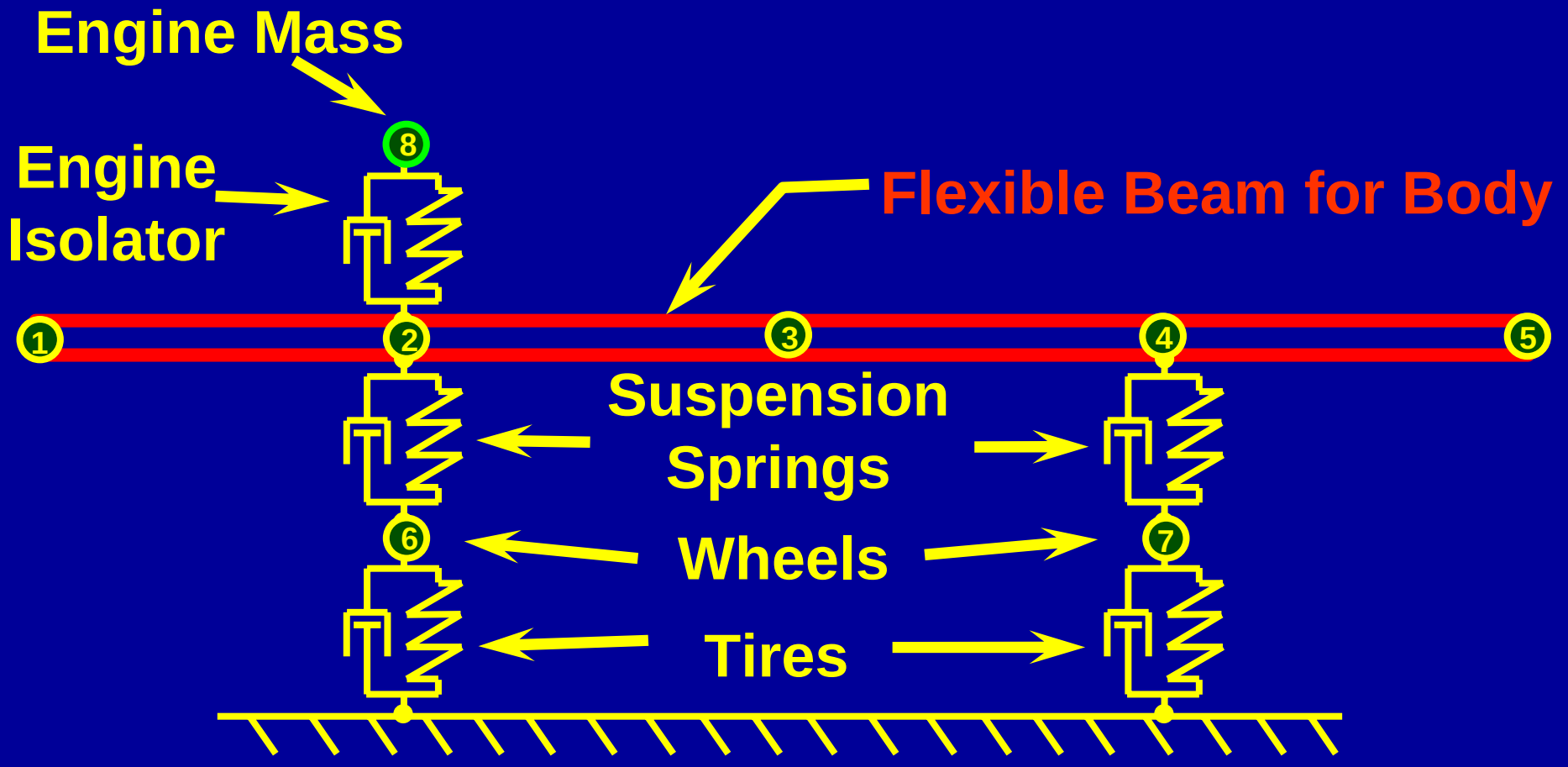








8 Degree of Freedom Vehicle NVH Model



Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

Reduction of Input Forces from the Source

Road Load Excitation

- Use Bigger / Softer Tires
- Reduce Tire Force Variation
- Drive on Smoother Roads

Powertrain Excitation

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

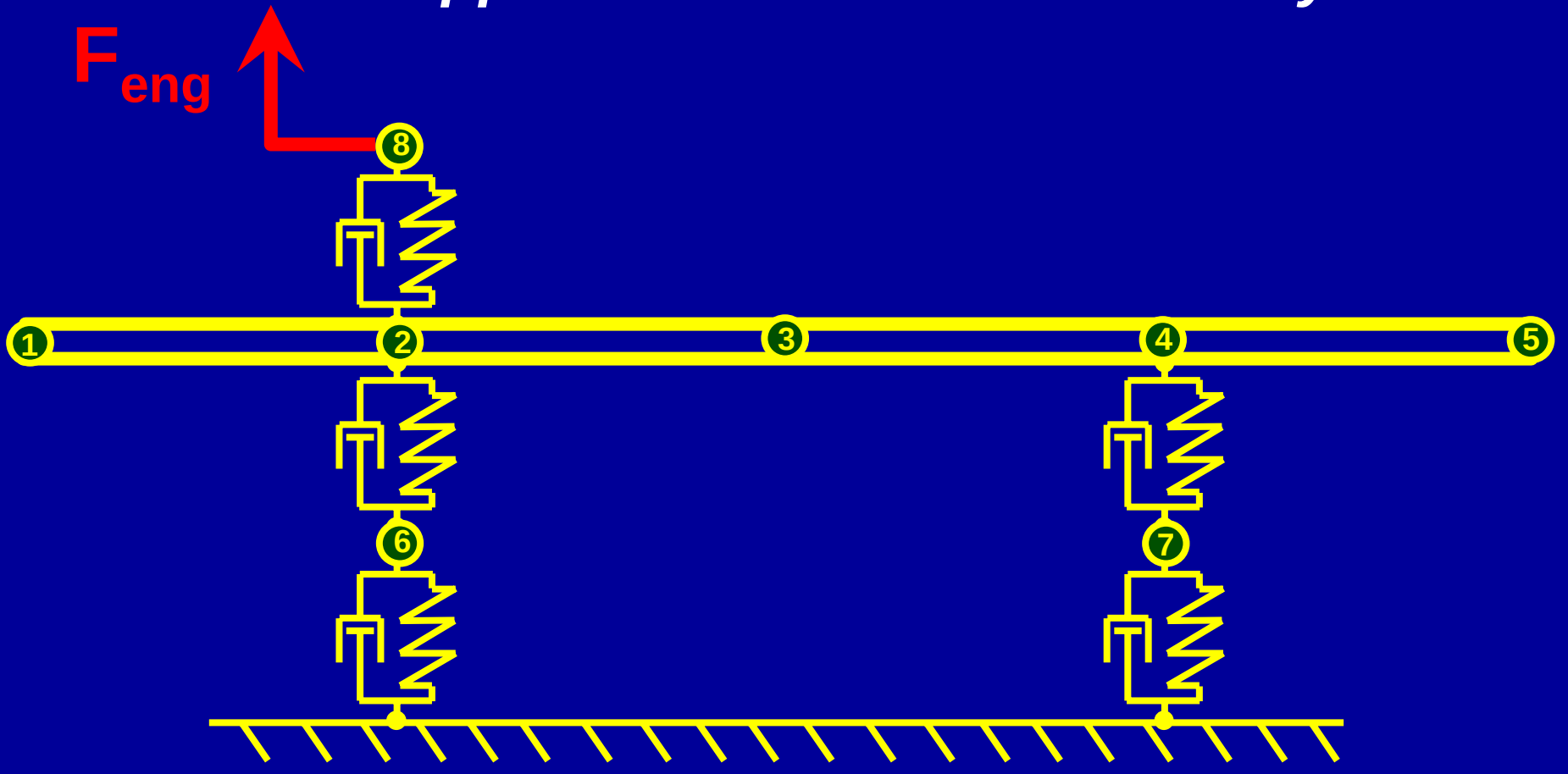
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

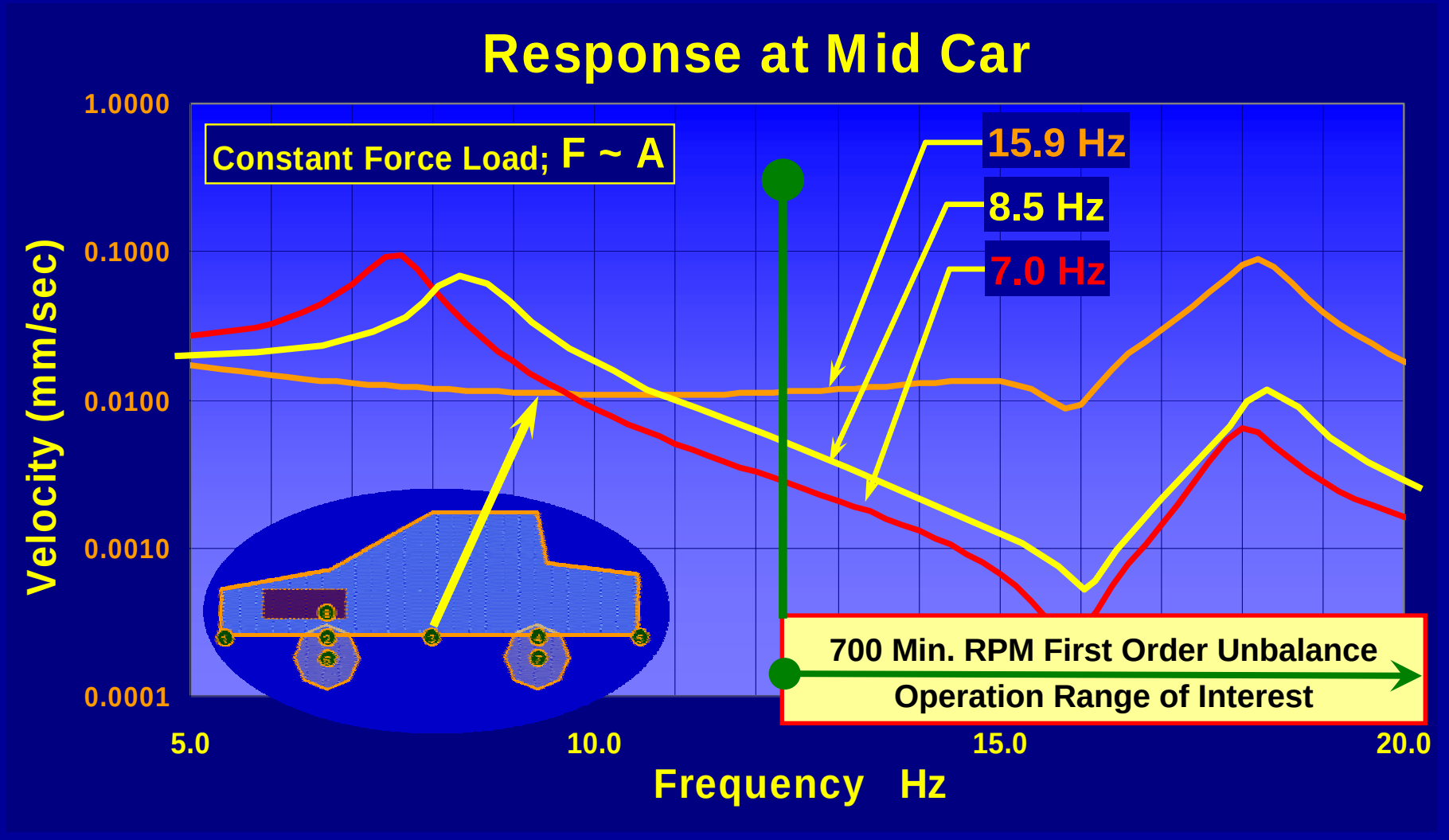
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



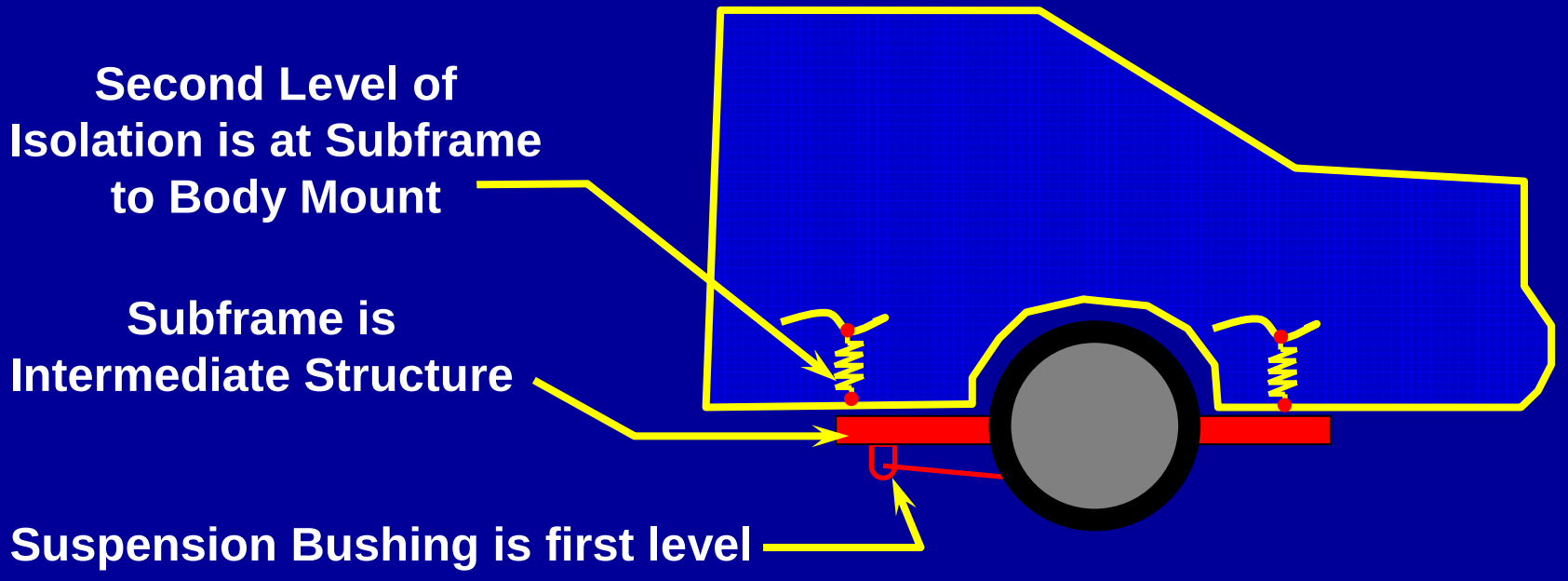
Forces at Powertrain could represent a First Order Rotating Imbalance

Engine Isolation Example



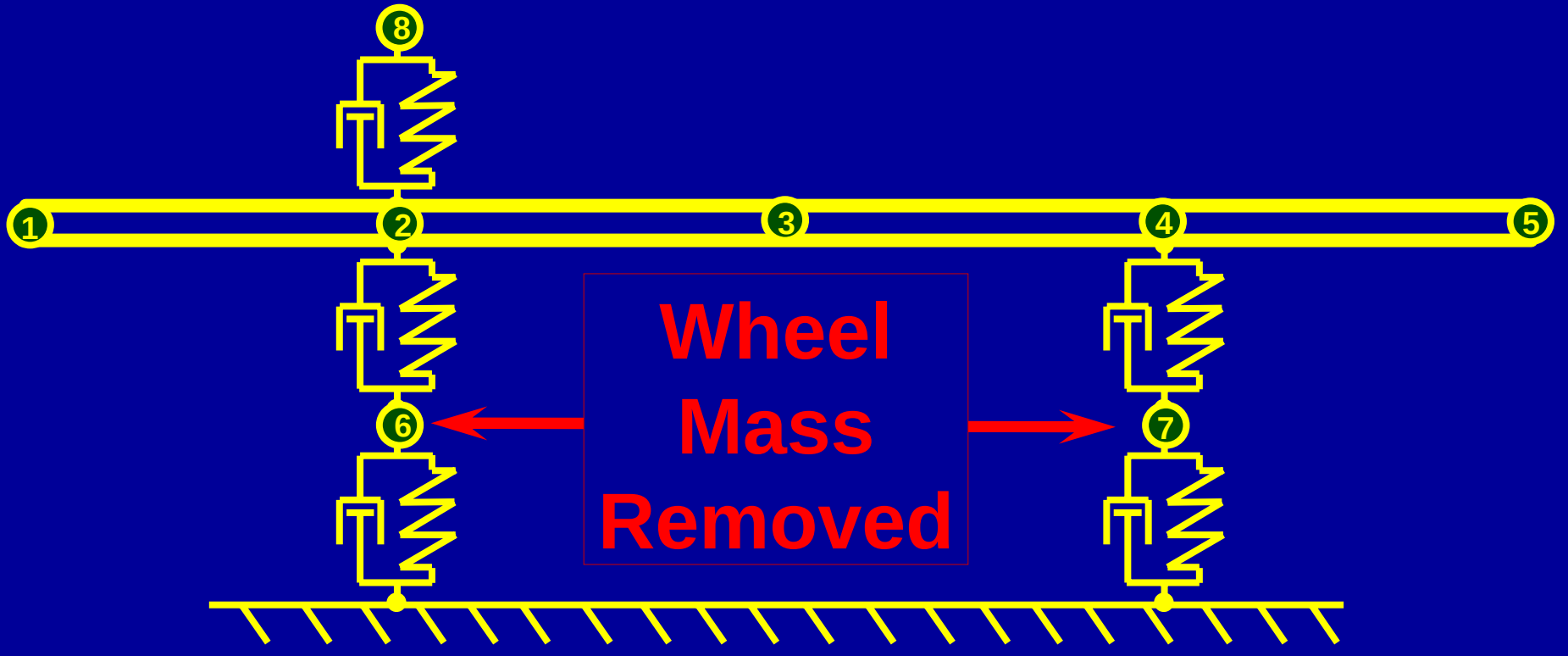
Concepts for Increased Isolation

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

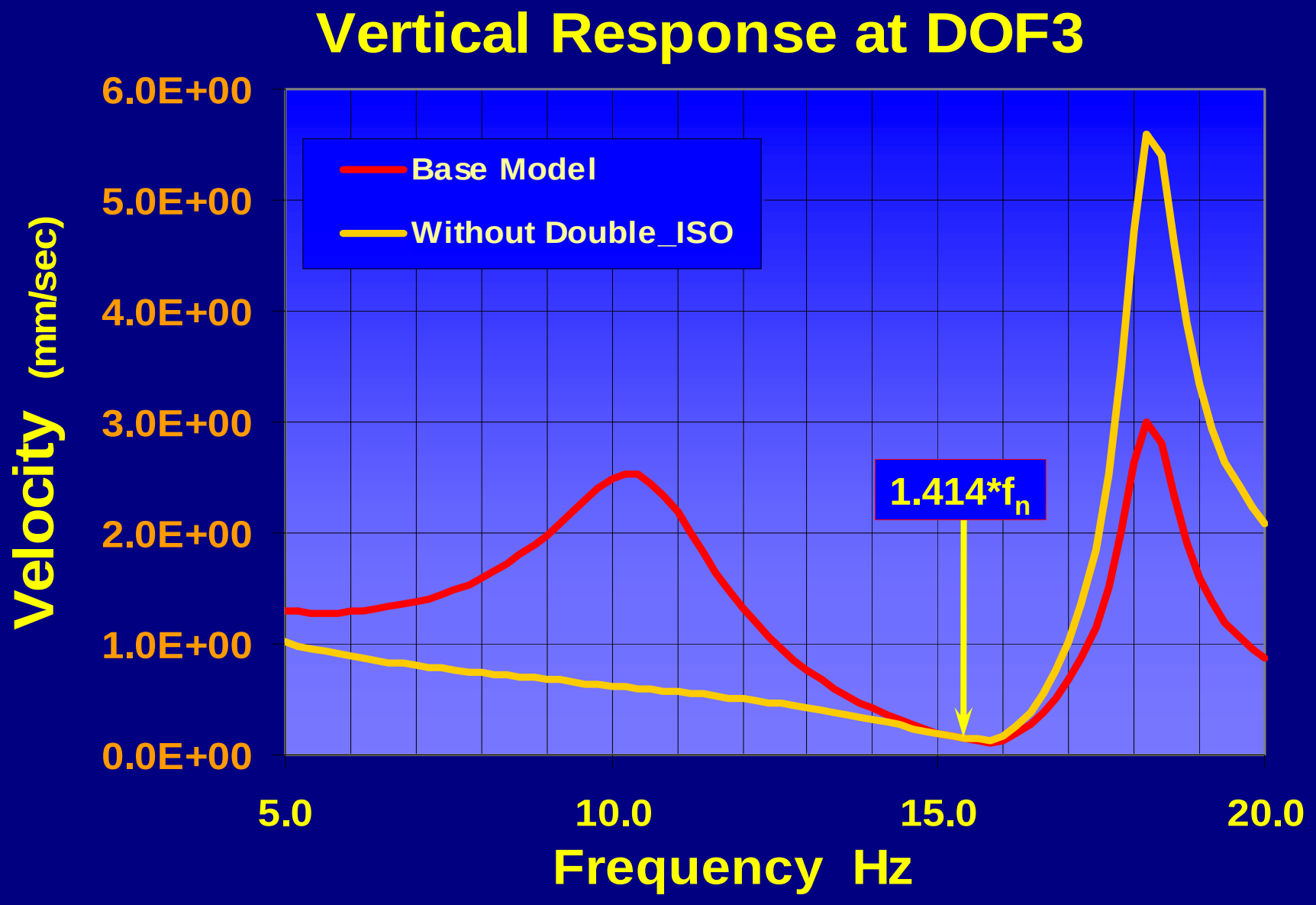


8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



Double Isolation Example

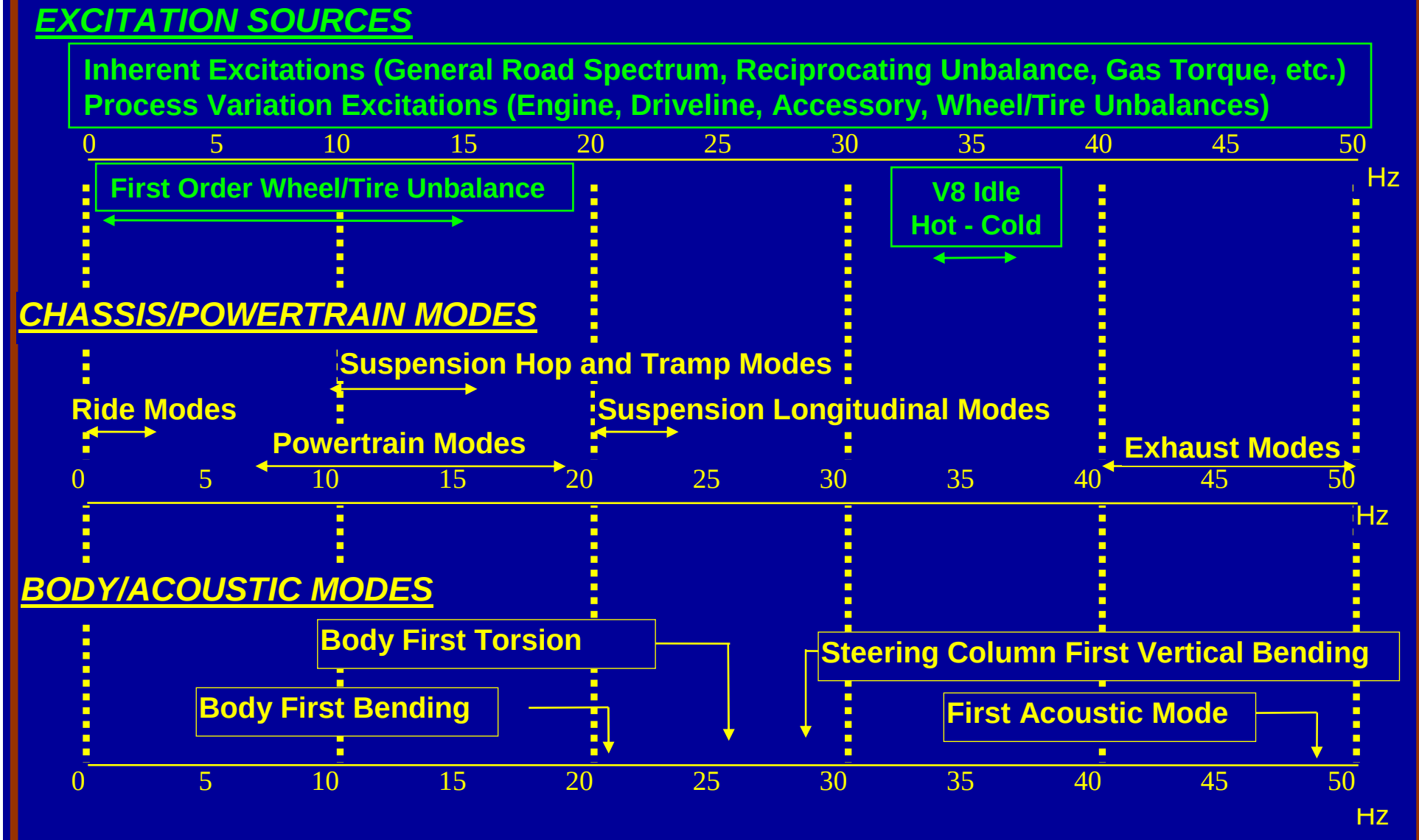


Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

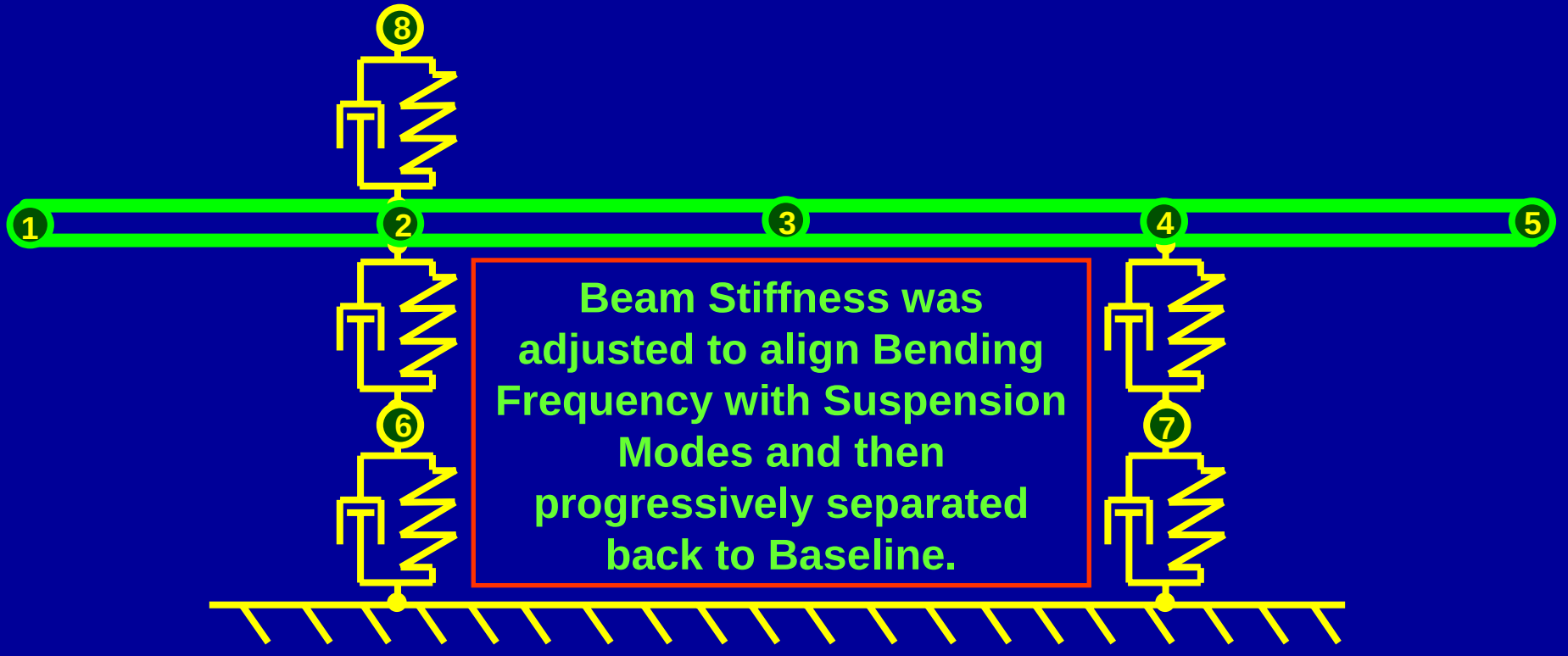
Mode Management Chart



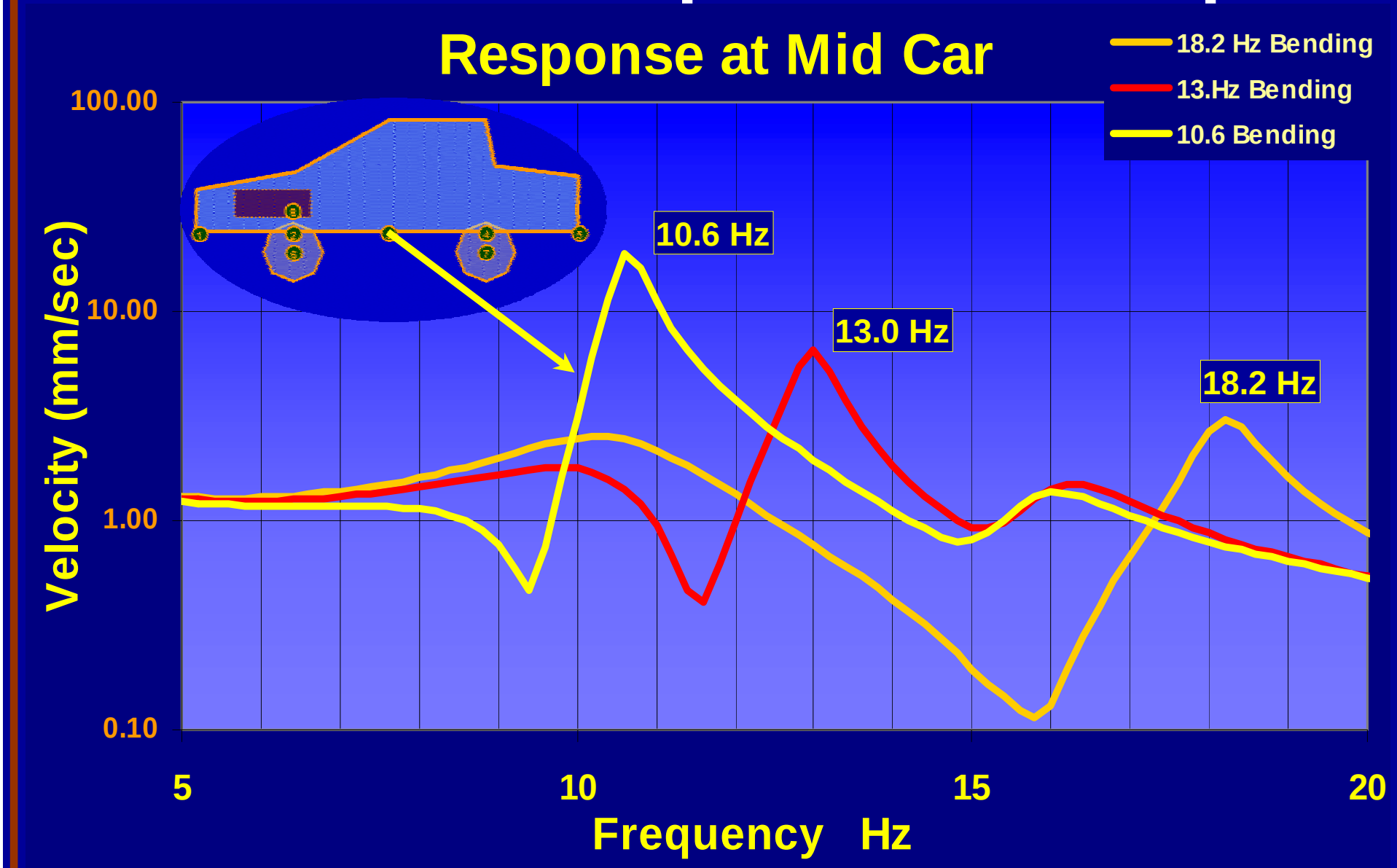
(See Ref. 1)

8 Degree of Freedom Vehicle NVH Model

Bending Mode Frequency Separation



8 DOF Mode Separation Example



Vibration and Noise Attenuation Methods

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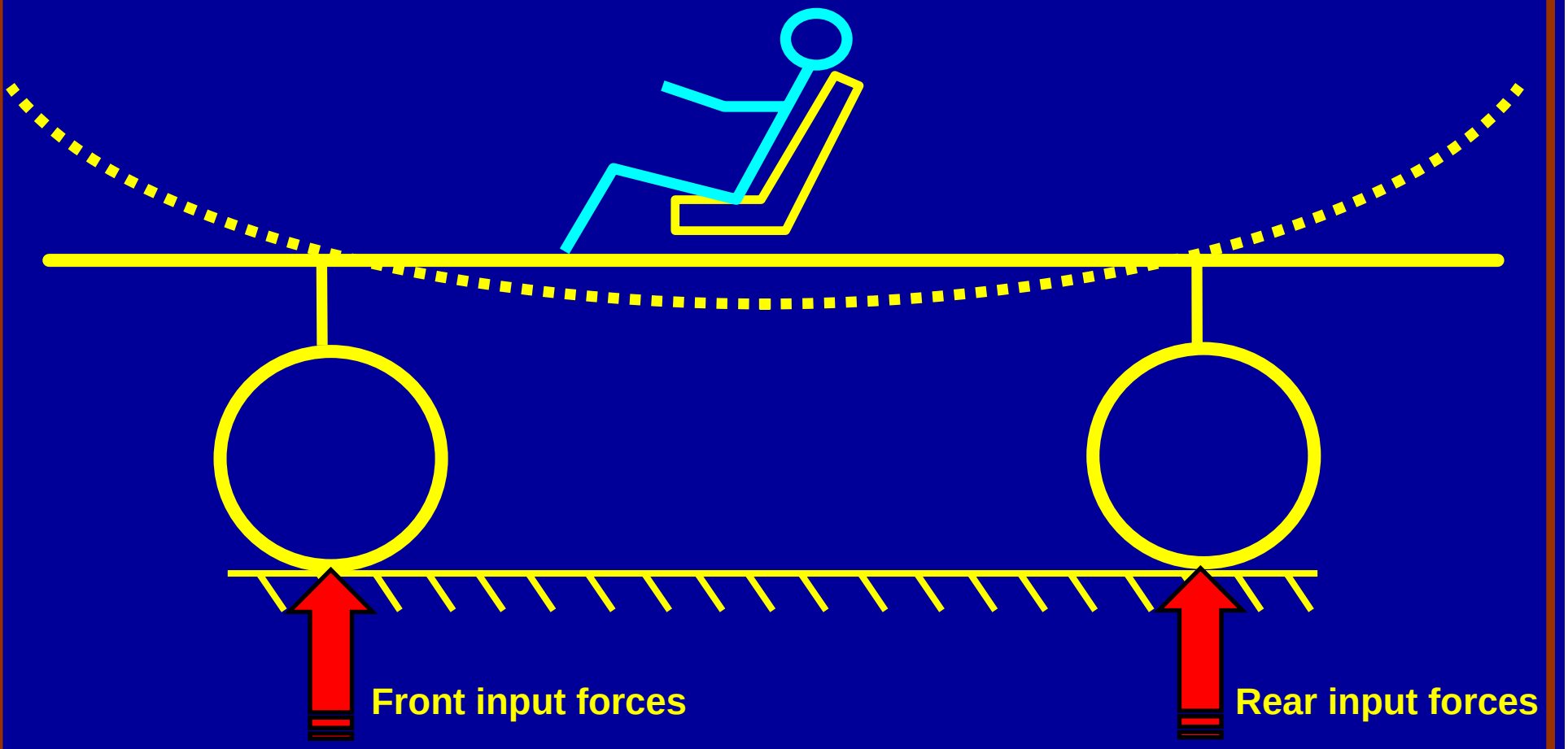
- Dynamic Absorbers

Mount at Nodal Point

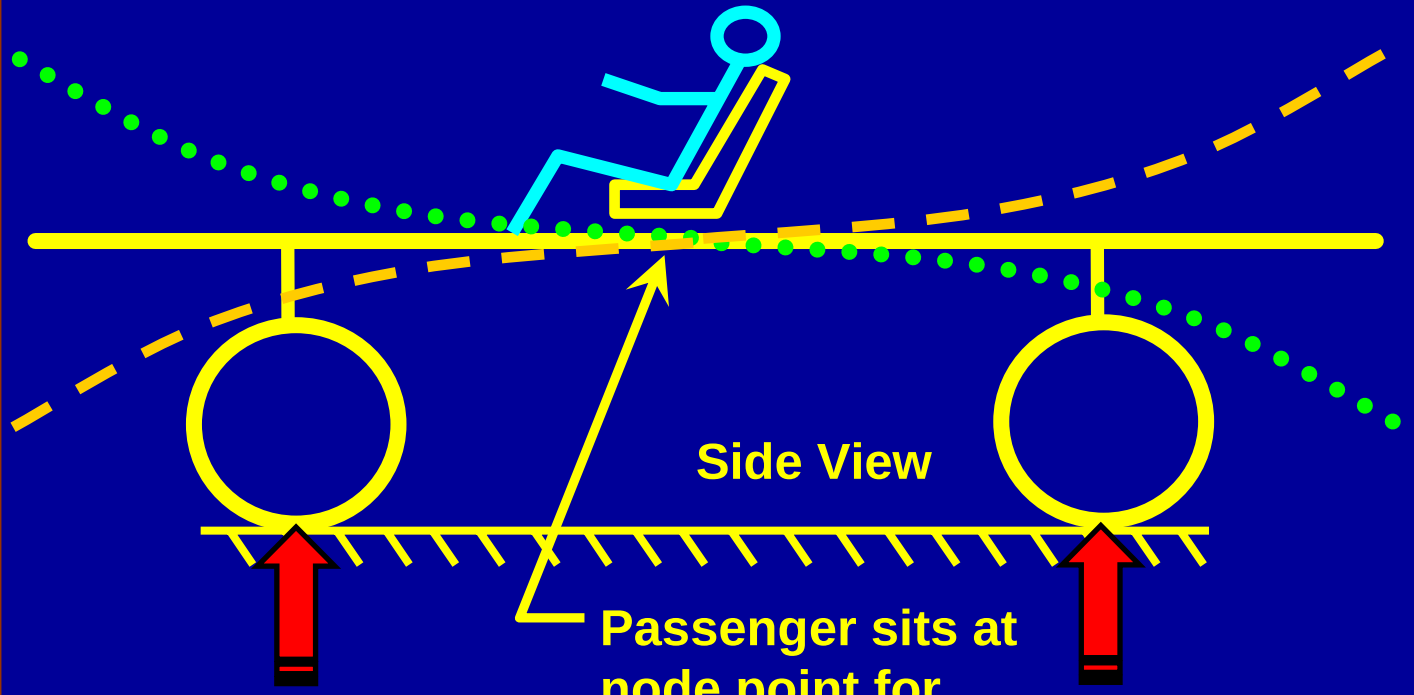
First Bending: Nodal Point Mounting Example

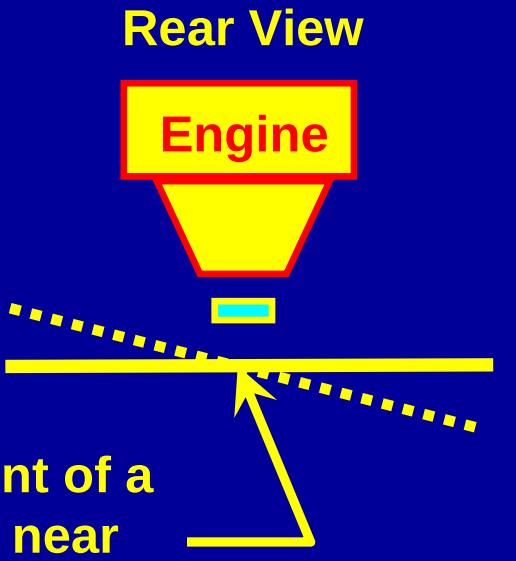


Locate wheel centers at node points of the first bending modeshape to prevent excitation coming from suspension input motion.

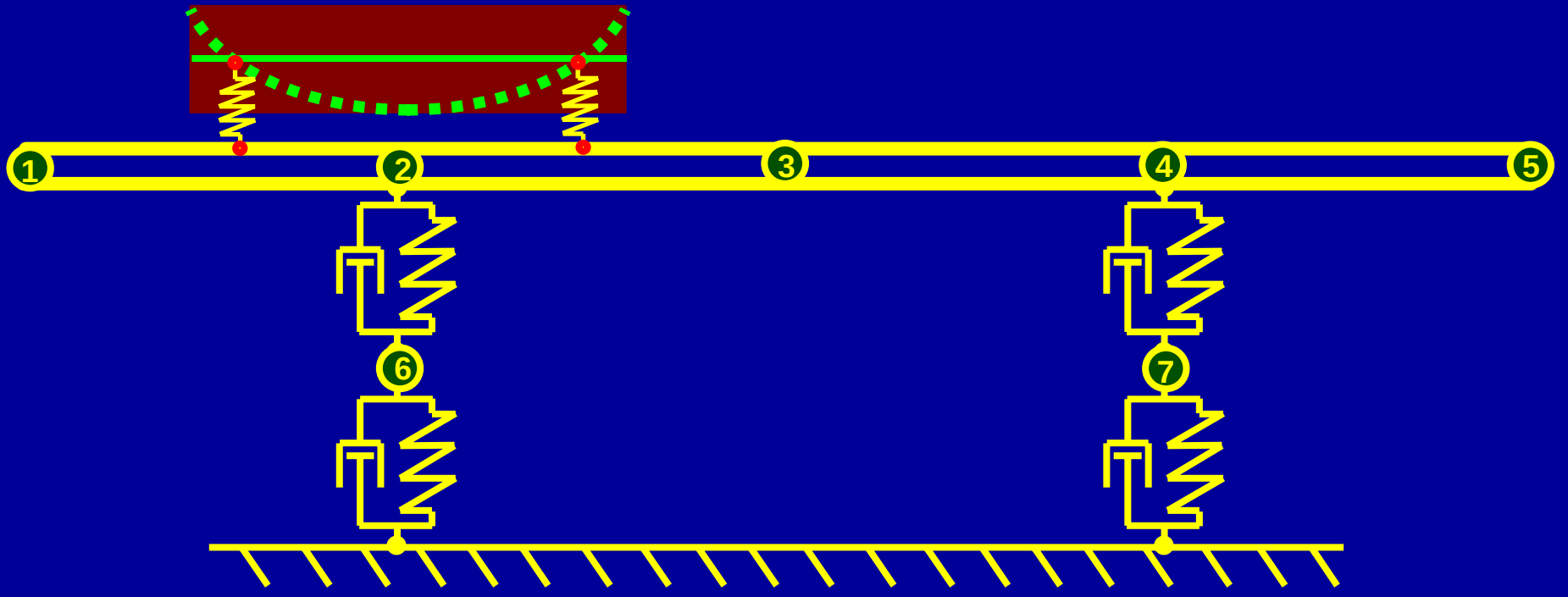
Mount at Nodal Point

First Torsion: Nodal Point Mounting Examples





Powertrain Bending Mode Nodal Mounting

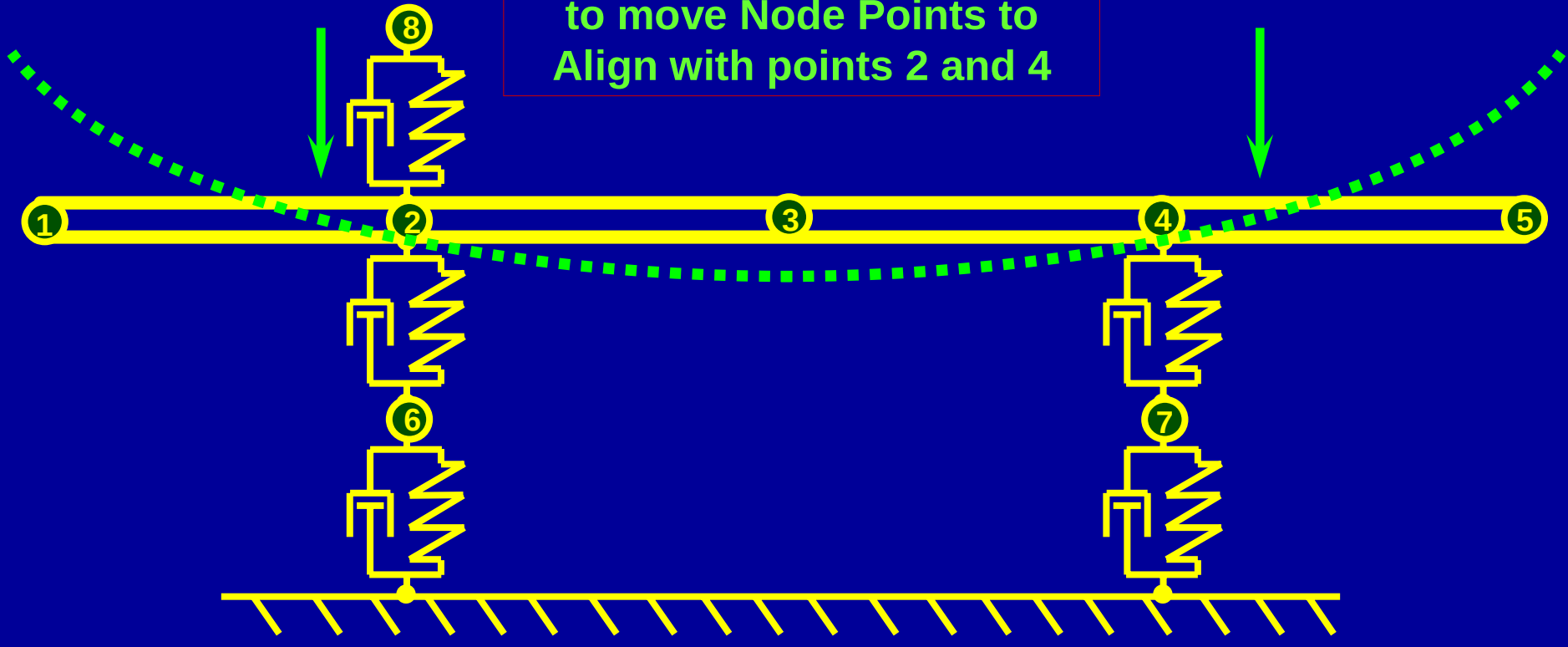


Mount system is placed to support Powertrain at the Nodal Locations of the First order Bending Mode.
Best compromise with Plan View nodes should also be considered.

8 Degree of Freedom Vehicle NVH Model

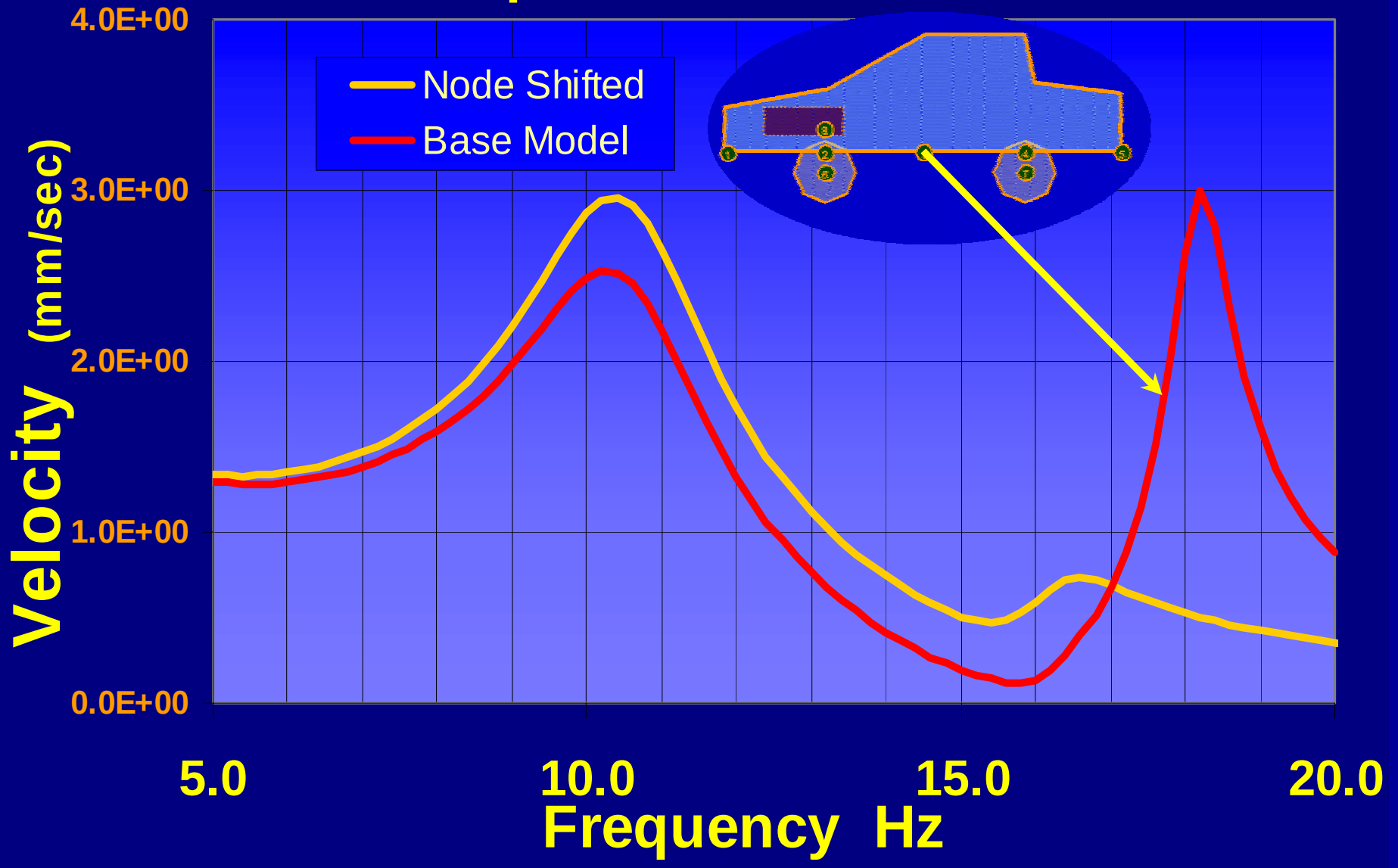
Bending Node Alignment with Wheel Centers

Redistribute Beam Masses to move Node Points to Align with points 2 and 4



First Bending Nodal Point Alignment

Response at Mid-Car



Vibration and Noise Attenuation Methods

Main Attenuation Strategies

- Reduce the Input Forces from the Source
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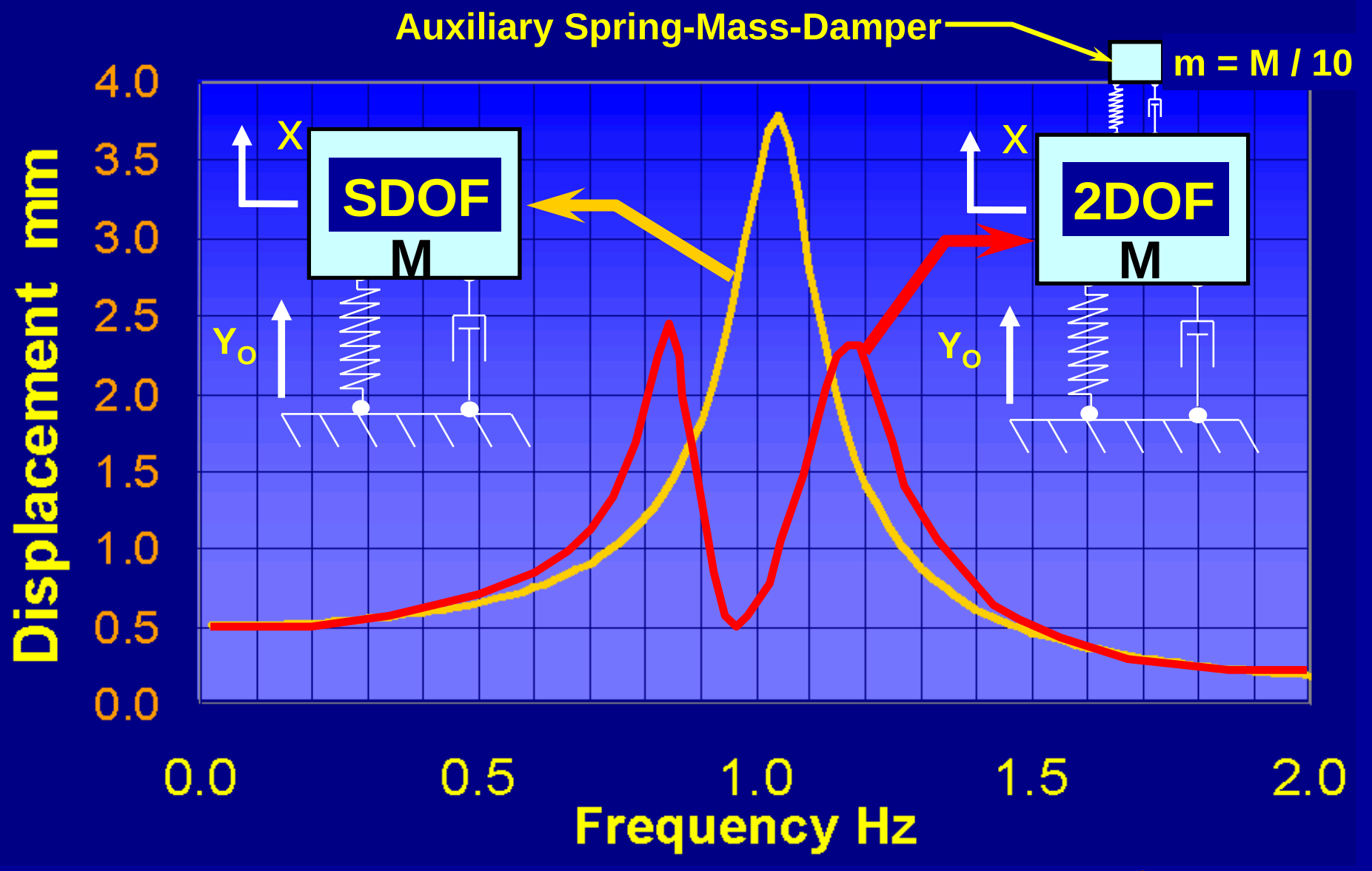
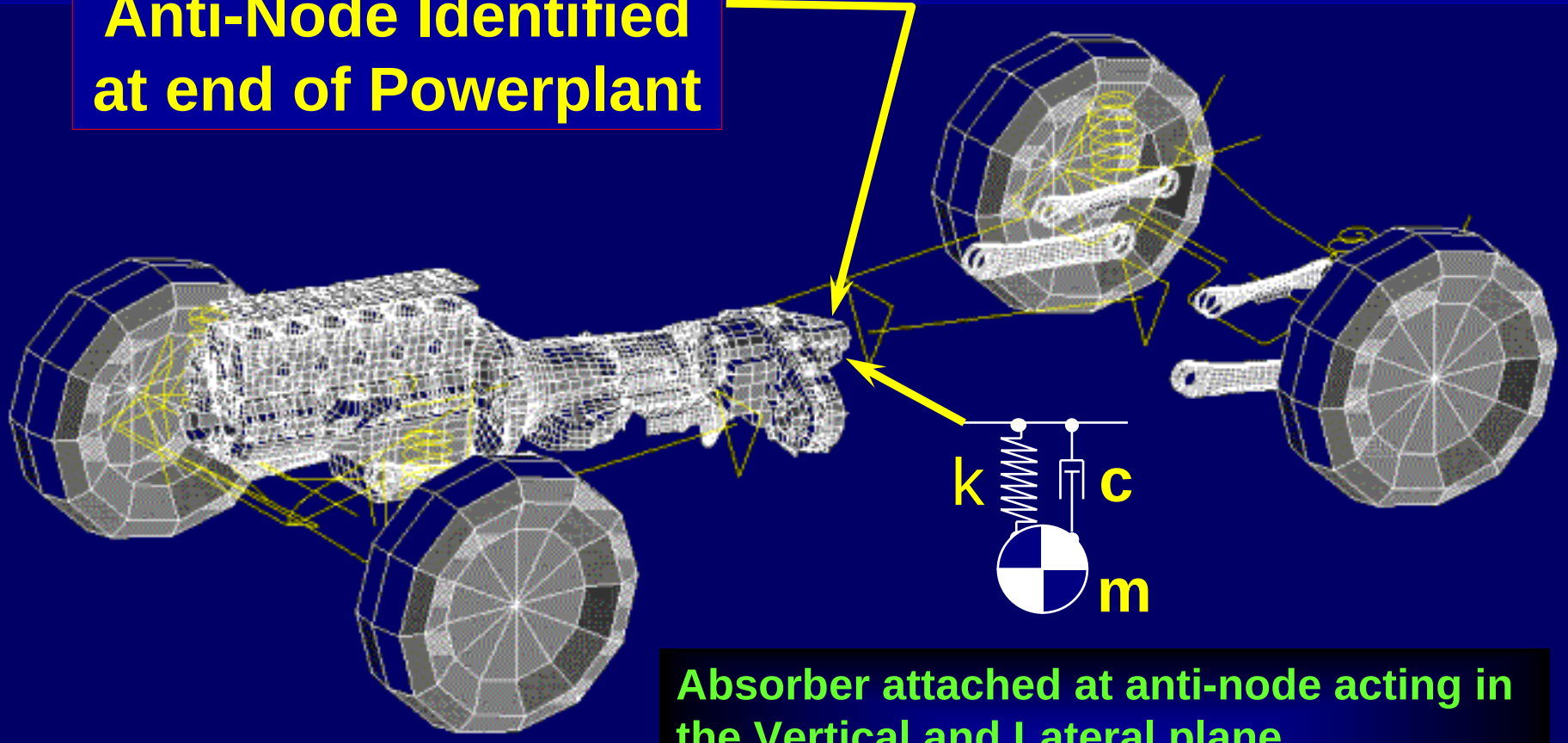
Dynamic Absorber Concept



Powertrain Example of Dynamic Absorber

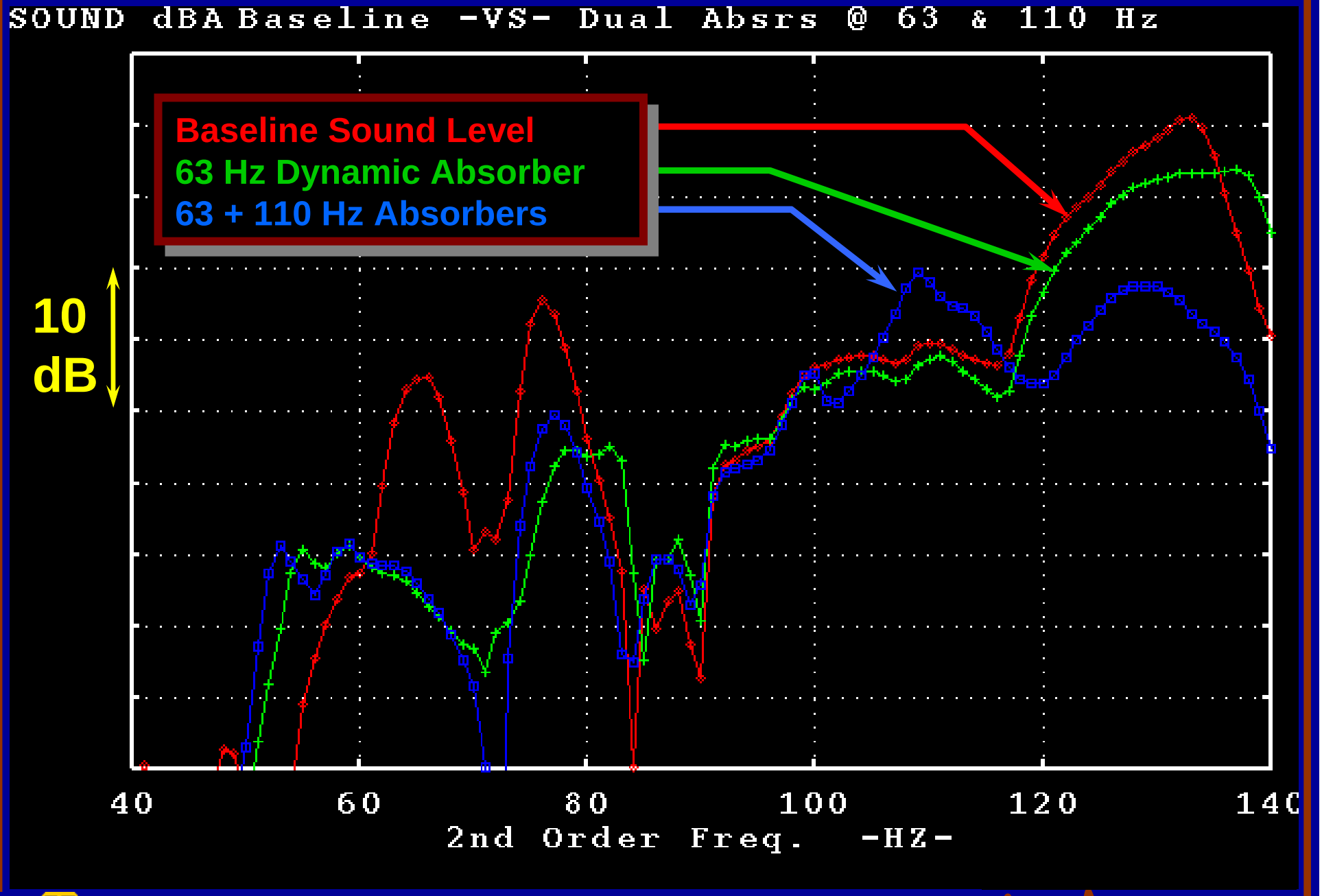
Anti-Node Identified at end of Powerplant



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]

Vibration and Noise Attenuation Methods

Main Attenuation Strategies

- Reduce the Input Forces from the Source
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Structure Borne NVH Outline

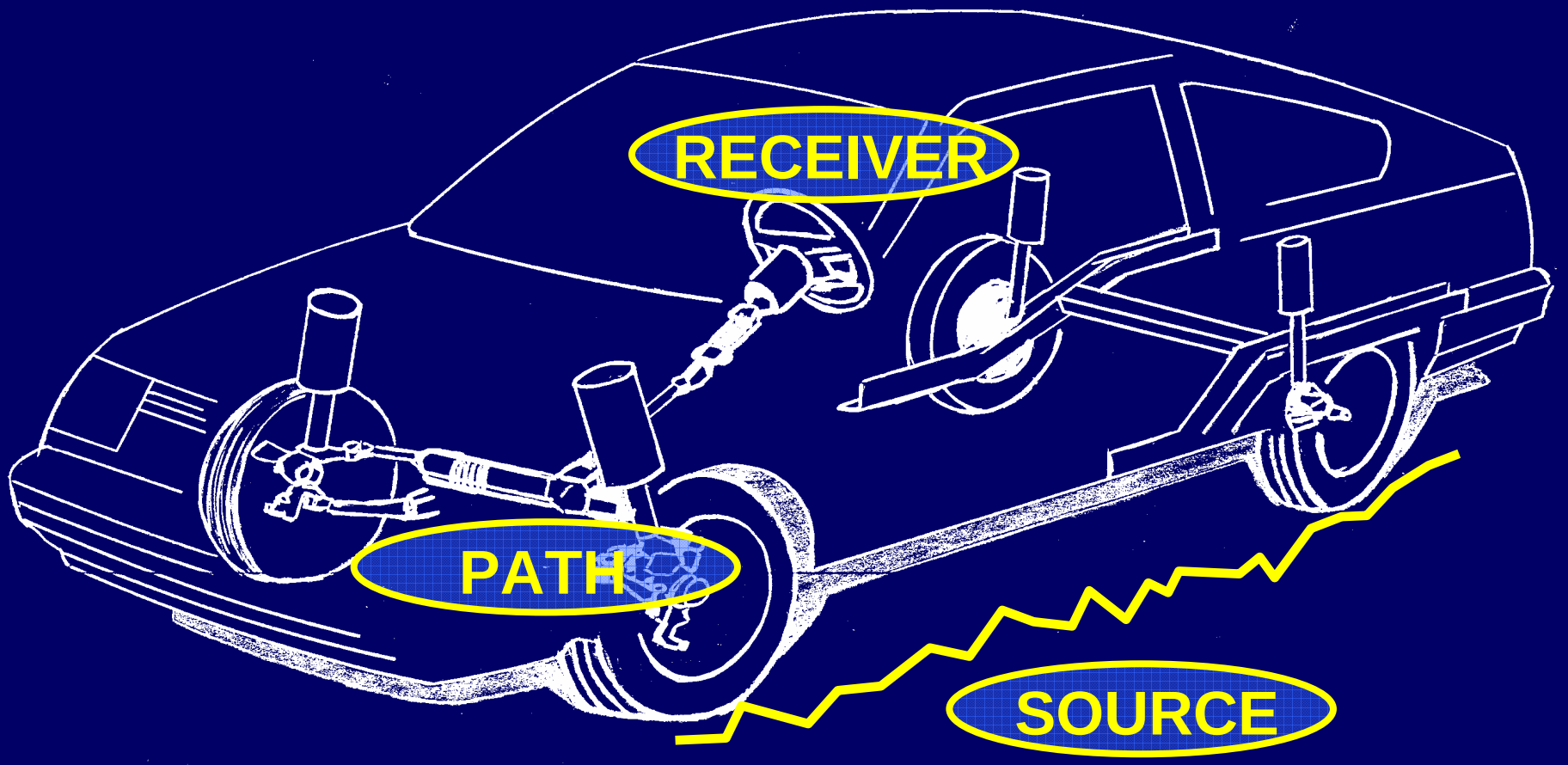
- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics
- ***Live Noise Attenuation Demo***
- Mid Frequency Basics
- Utilization of the NVH Toolbox
- Closing Remarks

Greg Goetchius

Structure Borne NVH Outline

- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics
- *Live Noise Attenuation Demo*
- **Mid Frequency Basics** ***Sachin Gogate***
- Utilization of the NVH Toolbox
- Closing Remarks

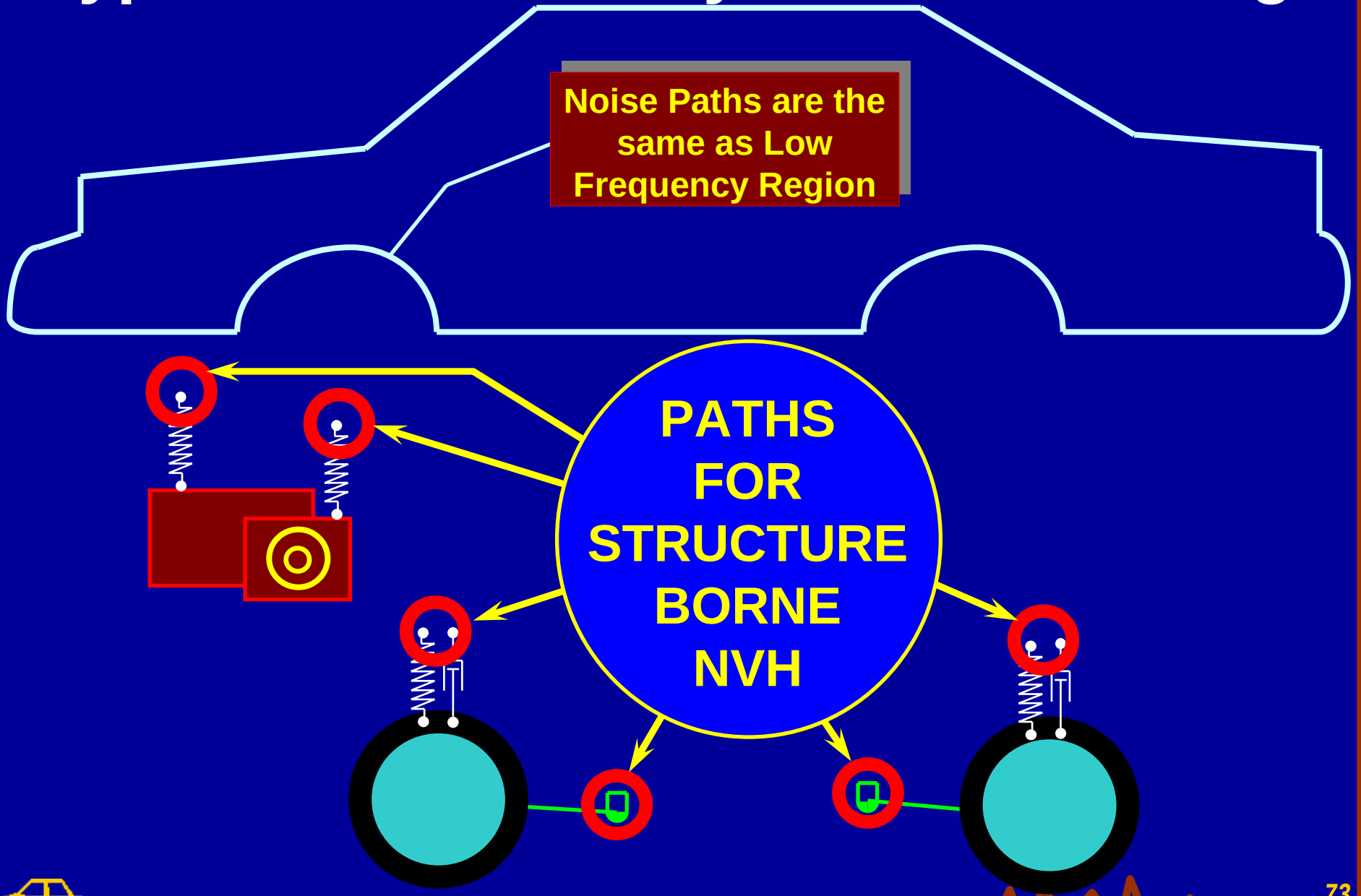
Mid Frequency NVH Fundamentals



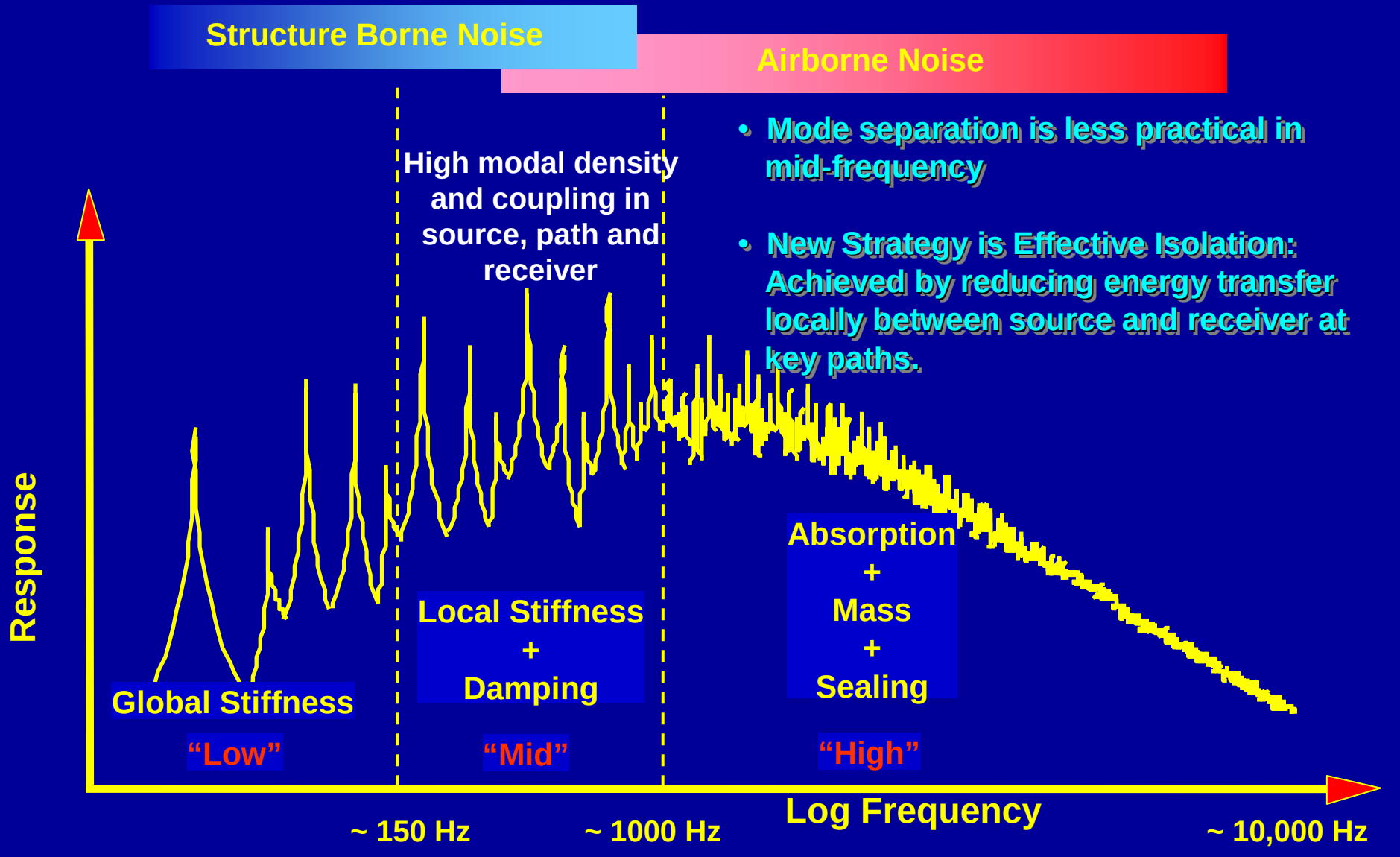
This looks familiar!

Frequency Range of Interest has changed to 150 Hz to 500 Hz

Typical NVH Pathways to the Passenger



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

- **Important characteristics of mid frequency analysis**

Effective Isolation

&

Identifying Key Noise Paths

Mid-Frequency Analysis Character

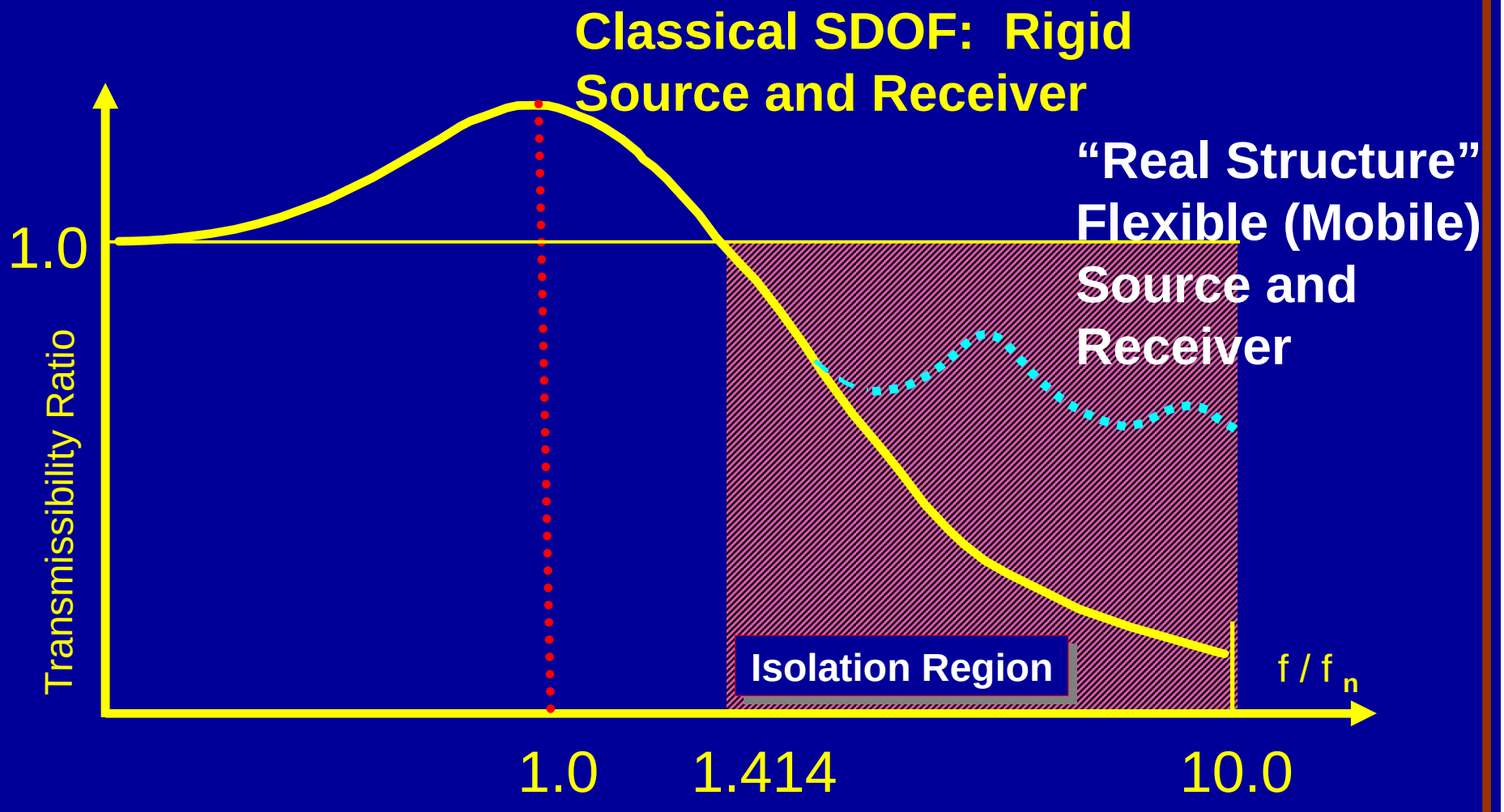
- **Important characteristics of mid frequency analysis**

Effective Isolation

&

Identifying Key Noise Paths

Isolation Effectiveness



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

Mobility

- 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

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

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

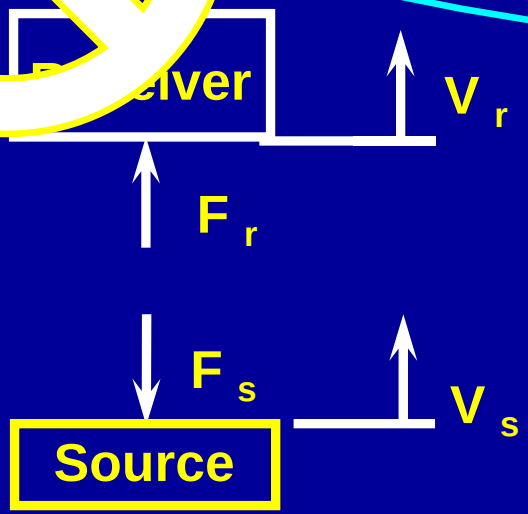
Isolation

- a theoretical model based on
rce

2003, 2005 ERRATA

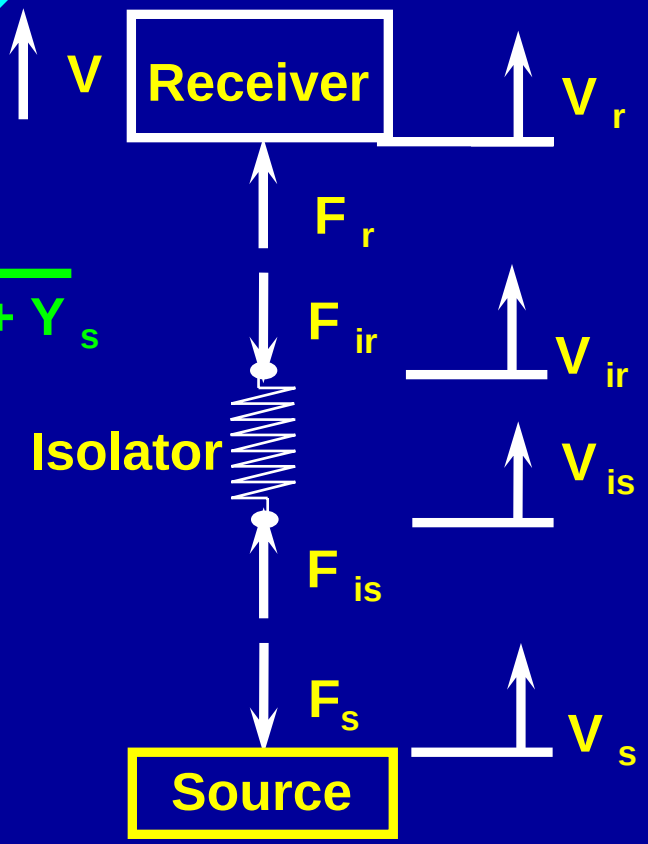
- Tr ility ratio is used to objectively define measure of isolation

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



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

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

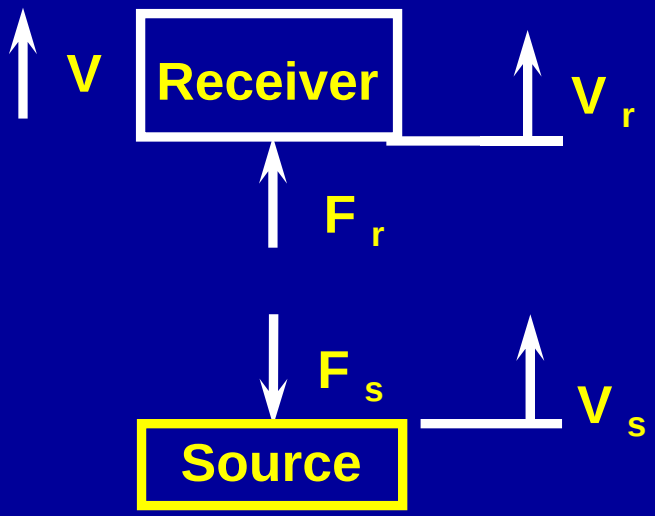


Isolation

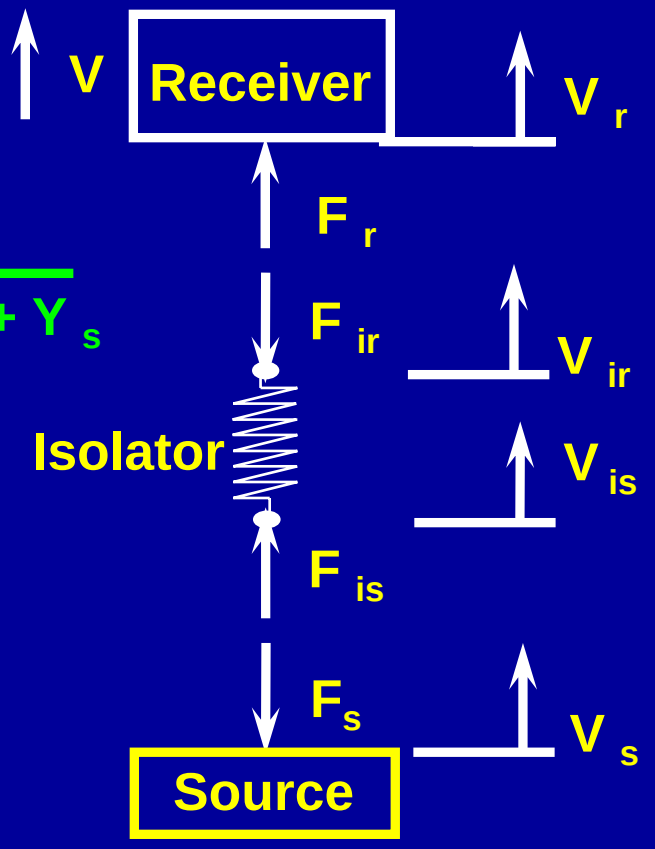
- The isolation effectiveness can be quantified by a theoretical model based on analysis of mobilities of receiver, isolator and source

- 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

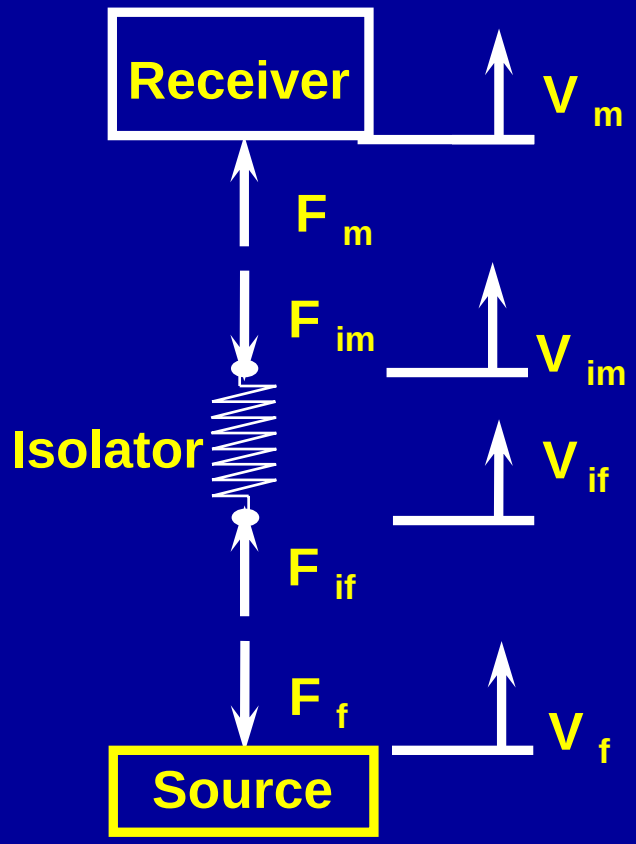
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

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

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.***

Mid-Frequency Analysis Character

- **Important characteristics of mid frequency analysis**

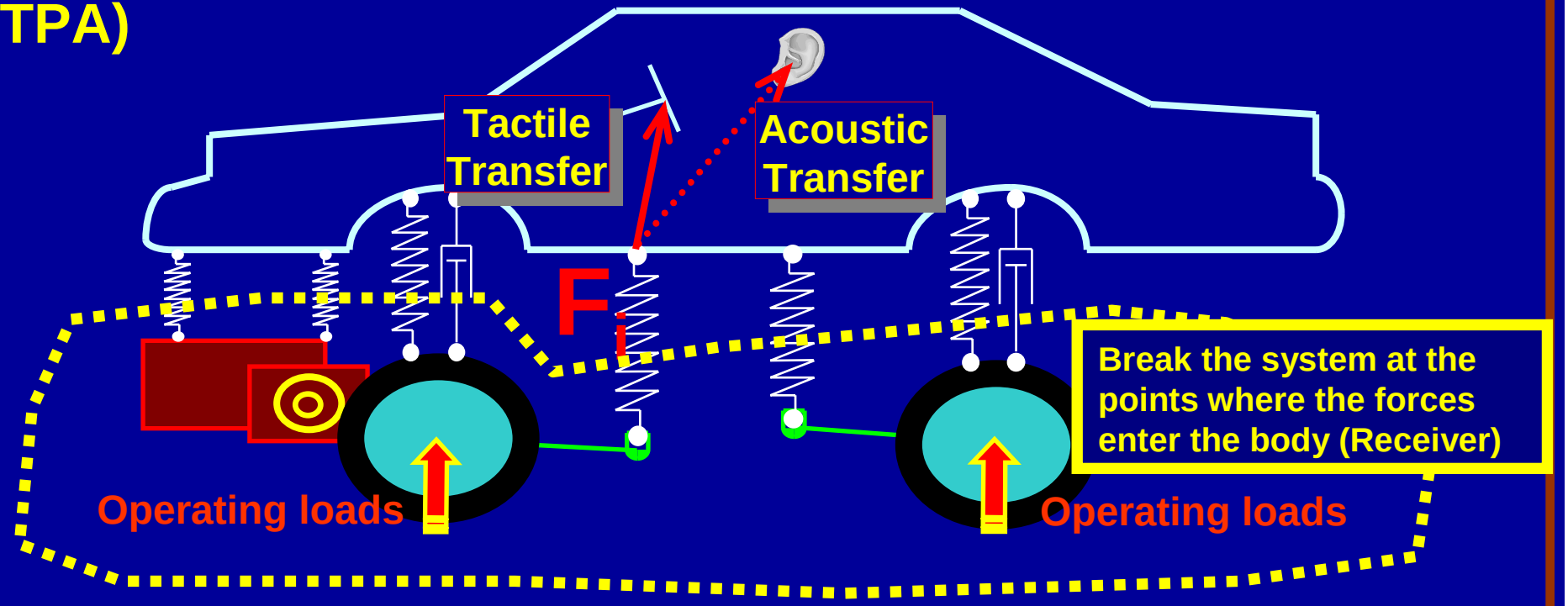
Effective Isolation

&

Identifying Key Noise Paths

Identifying Key NVH Paths

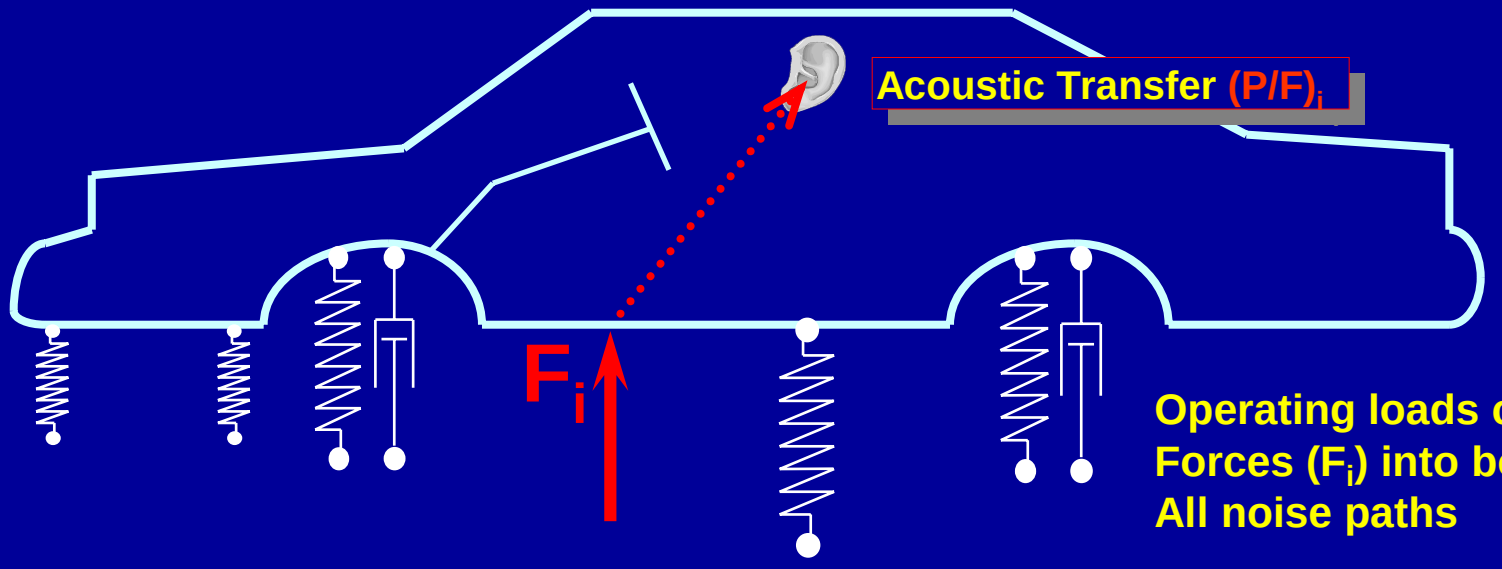
Key NVH paths are identified by Transfer Path Analysis (TPA)



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

- TPA is applicable in both testing and simulation scenarios to identify key paths

Noise Path Analysis



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

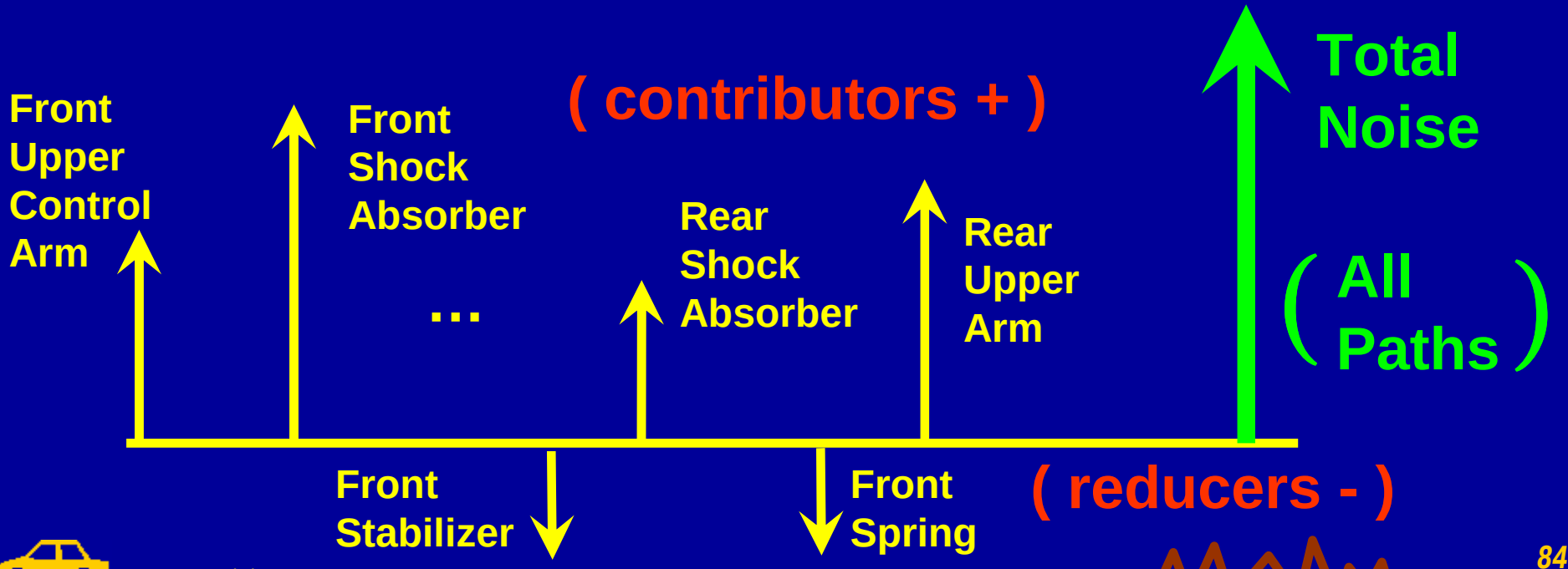
$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [(P/F)_i * F_i]$$

P_i : Partial contribution of path i due to operating force

$(P/F)_i$: Acoustic Transfer Function of the i^{th} Path

Transfer Path Analysis

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



Designing for Mid Frequency

- **Important characteristics of mid frequency analysis**

Effective Isolation

&

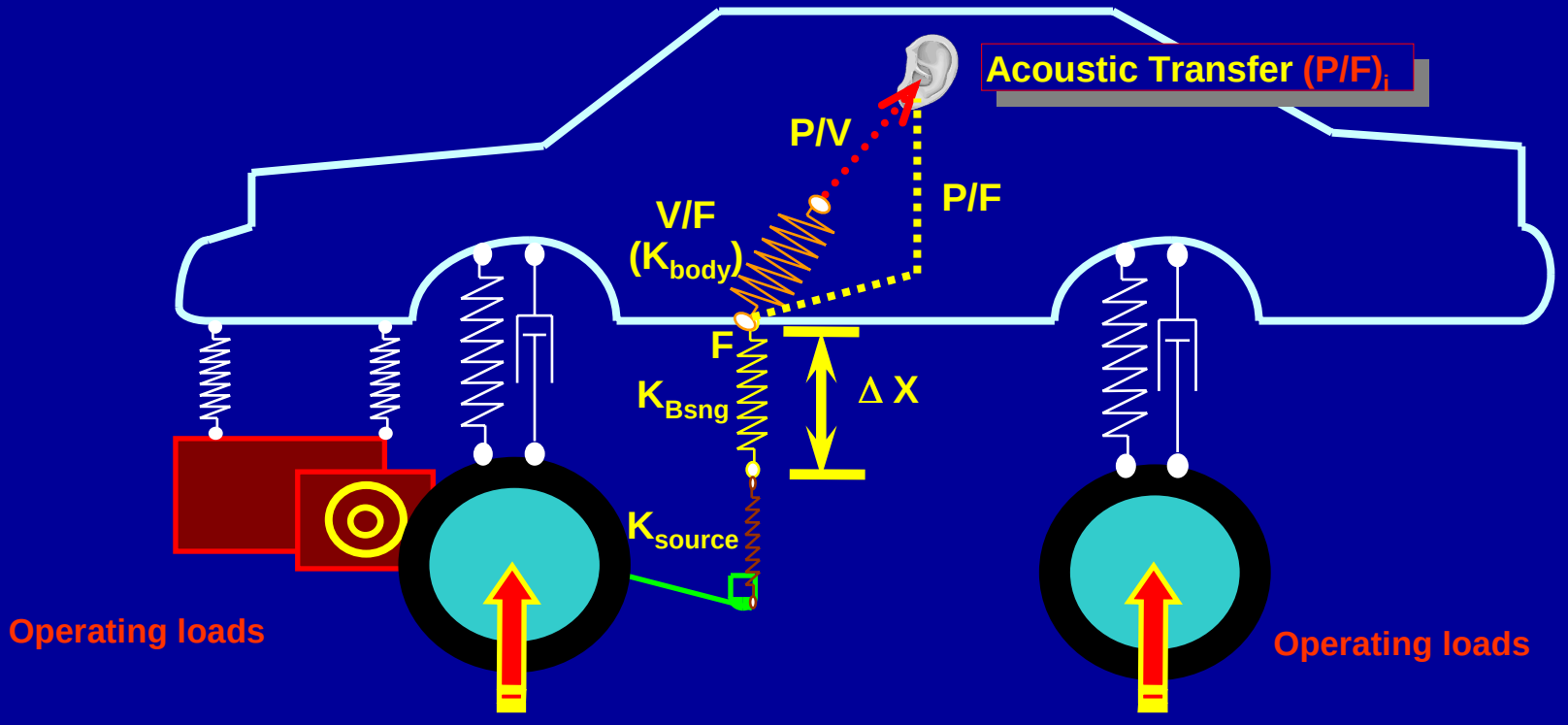
Identifying Key Noise Paths

Designing for Mid Frequency

When designing a new vehicle, the first phase is to satisfy generic targets for key parameters along *all noise paths* in order to achieve effective isolation.

What are these generic targets and key parameters ?

Generic Noise Path Targets



Transmissibility along a given noise path (TR_i)

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

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

Generic Noise Path Targets

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

$\frac{K_{body}}{K_{iso}}$ \ $\frac{K_{source}}{K_{iso}}$ →	1.0	5.0	Infinite
1.0	0.67	0.54	0.50
5.0	0.54	0.28	0.17
Infinite	0.50	0.17	0.00

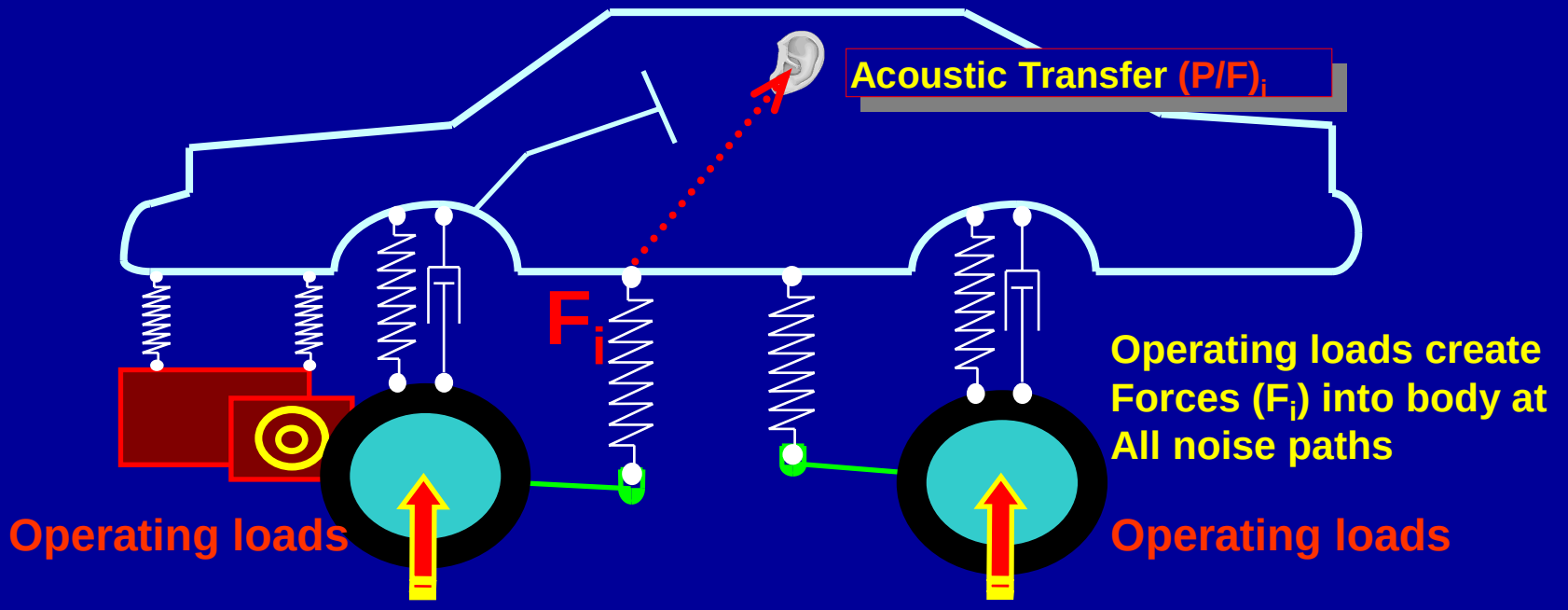
As a generic target, body to bushing stiffness ratio of at least 5.0 and very high source to bushing stiffness ratio (~ infinite) is desired to achieve "good" TR of 0.17

Relationship of Body-to-Bushing Stiffness Ratio to Transmissibility

For a very stiff source

$$TR = \left| \left(K_{bsg} \right) / \left(K_{bsg} + K_{body} \right) \right|$$

Generic Noise Path Targets



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

(P/F) Acoustic Sensitivity
(V/F) Structural Point Mobility (Receiver)

Generic Noise Path Targets

For example, for Acoustic Response P_t

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

- For a given force generated at a source attachment to body, lowering sensitivities (P/F) or (V/F) along a path would reduce total response
- Typical generic targets,
  - Acoustic Sensitivity (P/F) in 50 - 60 dBL/N range
  - Structural Mobility (V/F) less than 0.312 mm/sec/N

Generic Noise Path Targets

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

$$\frac{K_{source}}{K_{iso}} \sim \text{very high}$$

Acoustic Sensitivity < 50 - 60 dBL/N

Structural Mobility < 0.312 mm/sec/N

How does one achieve these generic targets ?

Generic Noise Path Targets

How does one achieve

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

$$\text{Structural Mobility} < 0.312$$

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

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

Generic Noise Path Targets

How does one achieve

$$\frac{K_{source}}{K_{iso}} \sim \text{very high}$$

- **Increase source side attachment stiffness (K_{source})**
- **Reduce attachment isolator stiffness (K_{iso})**

In Automotive Structures, it is realistic to expect that the Source to isolator stiffness ratio is high since Source usually corresponds to a stiff structure (such as powertrain or axle).

Generic Noise Path Targets

How does one achieve

Acoustic Sensitivity	<	50 - 60 dBL/N

- **At a given frequency, Acoustic Sensitivity (P/F) is**

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

- **Based on the above equation, increasing body stiffness usually reduces Acoustic Sensitivity.**

Generic Noise Path Targets

How does one achieve:

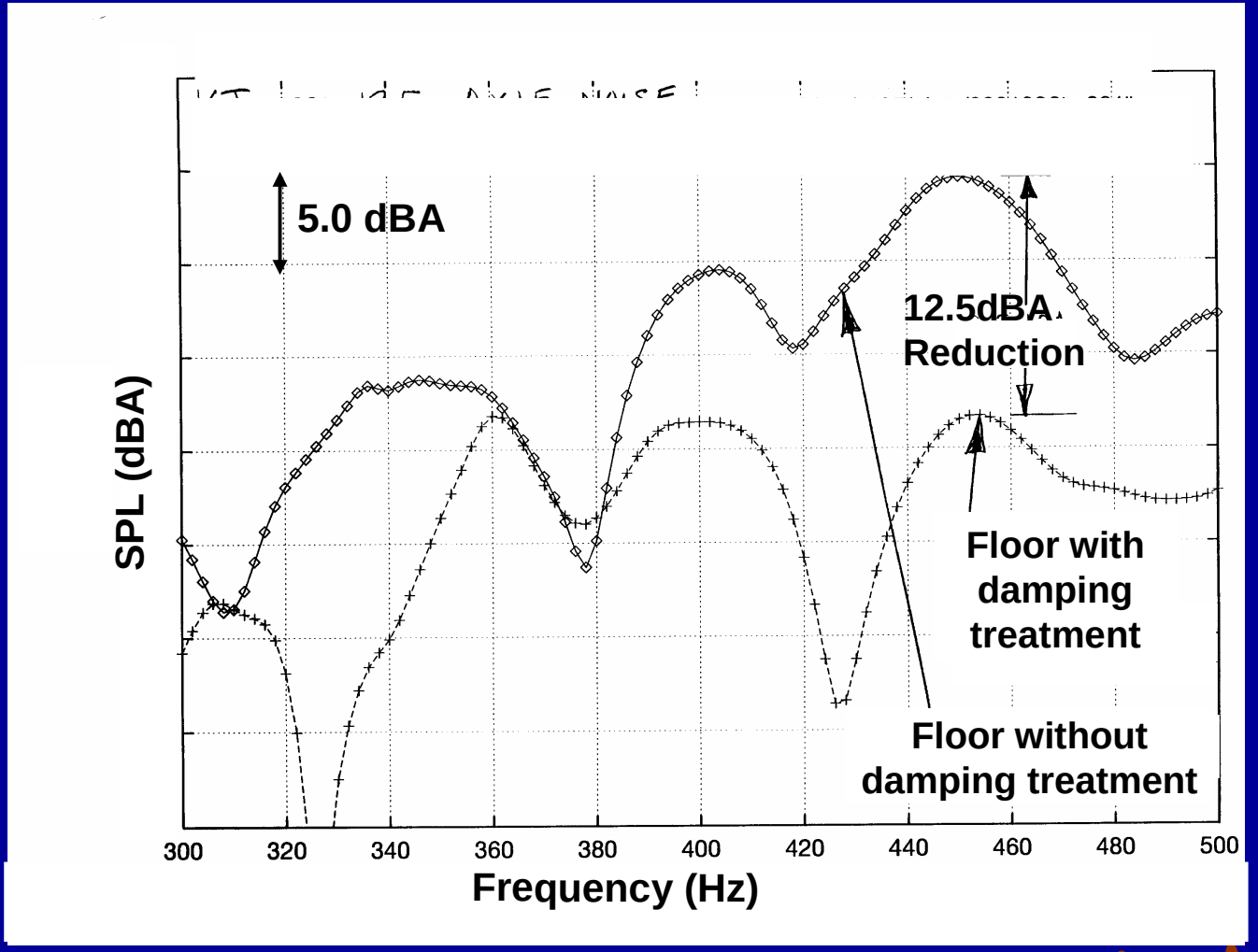
Acoustic Sensitivity	<	50 - 60 dBL/N

- **There are situations when increasing body stiffness does not reduce Acoustic Sensitivity**

- **In such cases, the Acoustic Sensitivity can be reduced by reducing the overall body panel velocity through application of damping treatments**

Application of Damping Treatment

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



[Figure Courtesy of DaimlerChrysler Corporation]

Designing for Mid Frequency

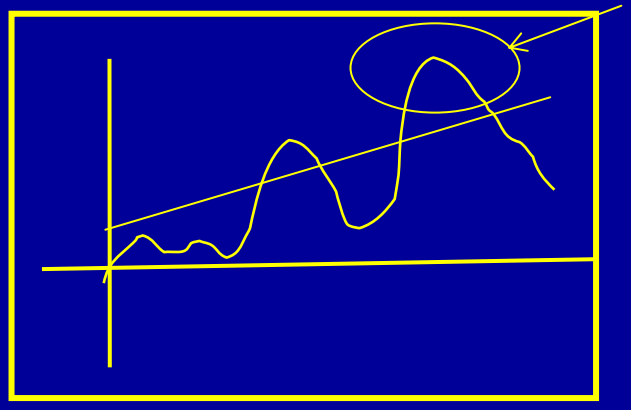
While designing a new vehicle, generic targets are set for key parameters along *all noise paths* in order to achieve effective isolation.

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

Probably Not !!

Designing for Mid Frequency

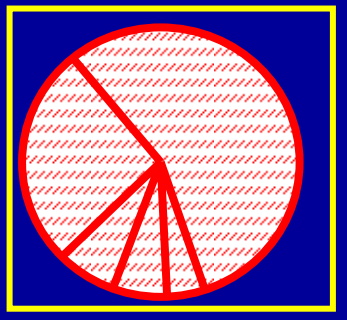
Driver's Ear Noise



Vehicle Level Response

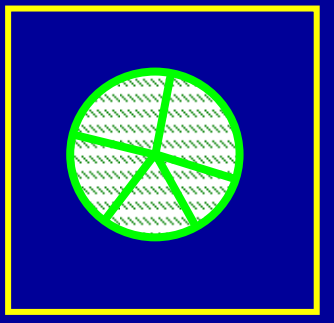
Transfer Path Analysis

Original Noise Path Contributions



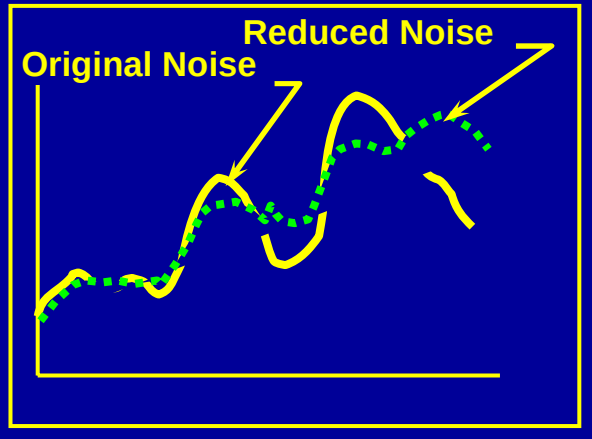
Some noise paths are more dominant than others

Improved Noise Path Contributions



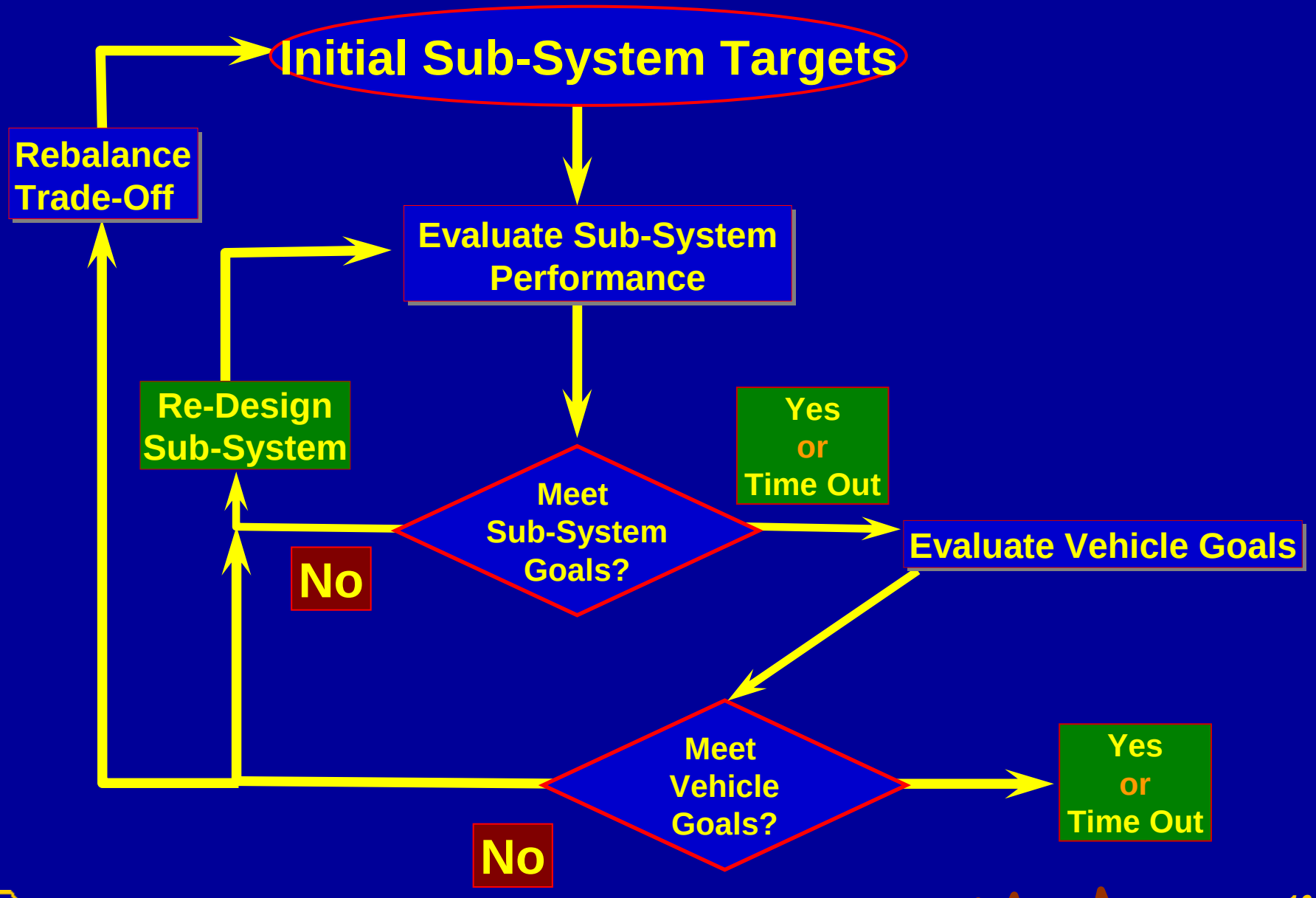
Impose more strict requirements for these dominant paths and relax requirements for other paths to achieve more "rebalanced" noise

Driver's Ear Noise



Vehicle Response to Meet NVH Targets

Mid Frequency NVH Goal Achievement Process



Designing for Mid Frequency

Principles to follow

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

- As the design is firmed out, shift focus to key contributing noise paths using Transfer Path Analysis in order to meet target for all NVH operating load conditions.
- Perform path “rebalancing” to arrive at revised path targets if Sub-System goal achievement is not possible due to architectural constraints.

Final Remarks on Mid Frequency Analysis

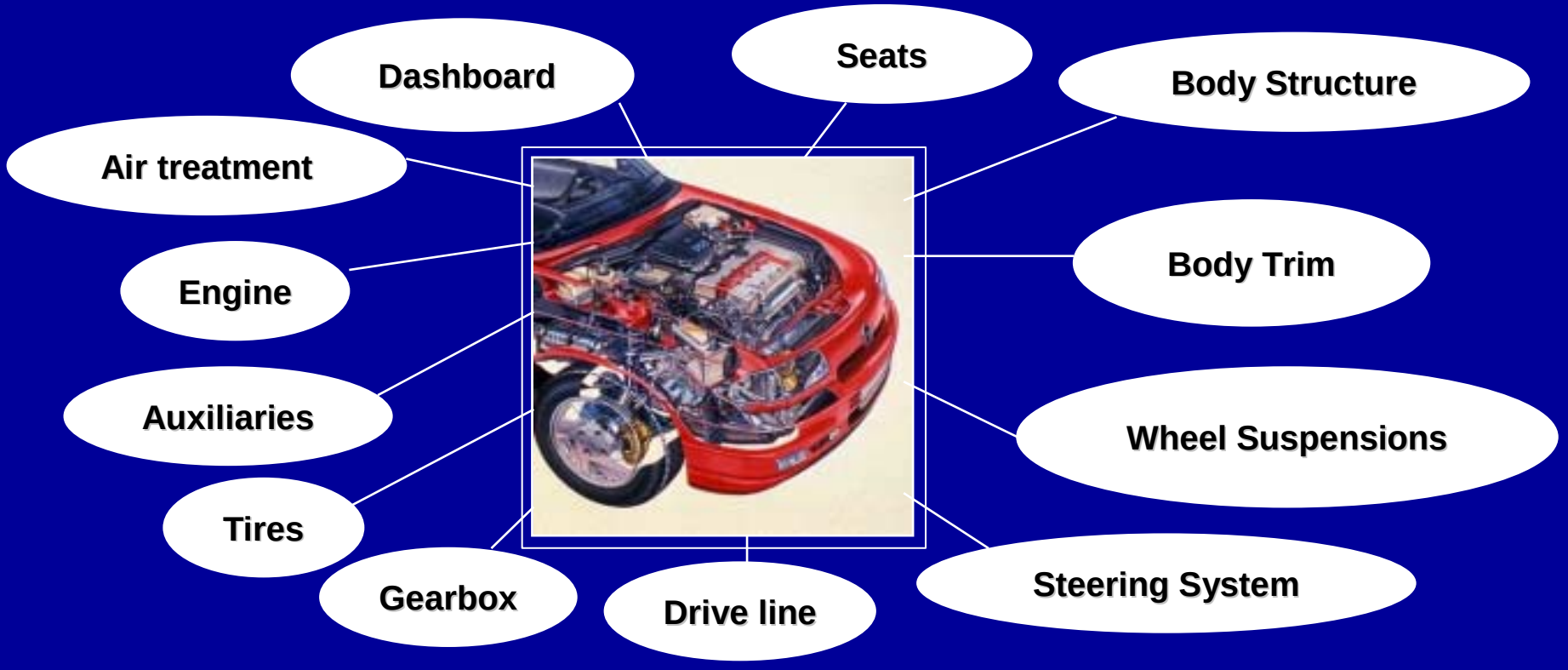
- Effective isolation at dominant noise paths is critical
- Reduced mobilities at body & source and softened bushing are key for effective isolation
- Other means of dealing high levels of source input such as damping treatments are also effective
- It is important to balance NVH requirements against other functionalities (Ride and Handling, Impact)
- It is important to understand the robustness of design recommendations

Structure Borne NVH Outline

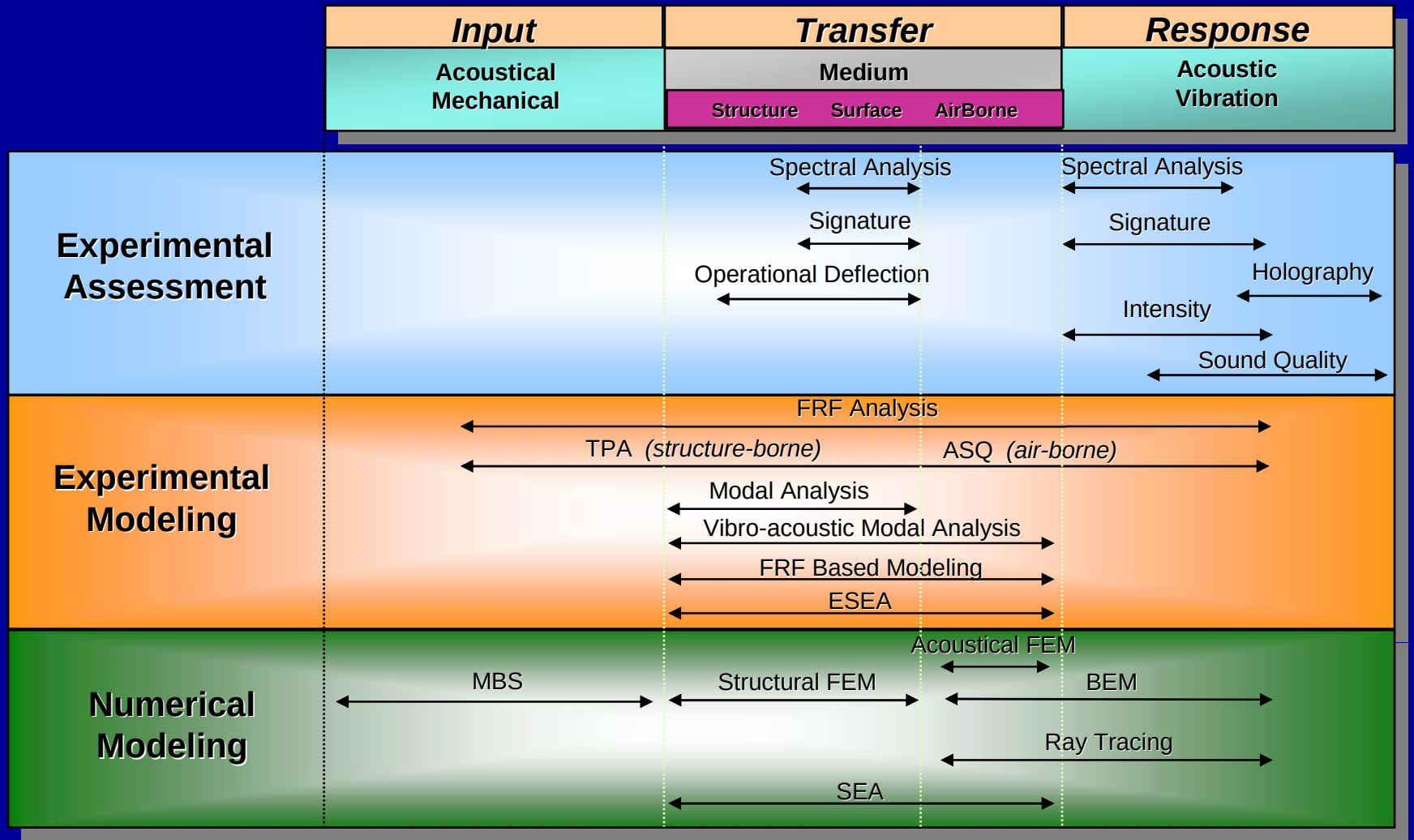
- Introduction
- Ride Balance in the Ride Range
- NVH Load Conditions
- Low Frequency Basics
- *Live Noise Attenuation Demo*
- Mid Frequency Basics
- **Utilization of the NVH Toolbox**
- Closing Remarks

Greg Hopton

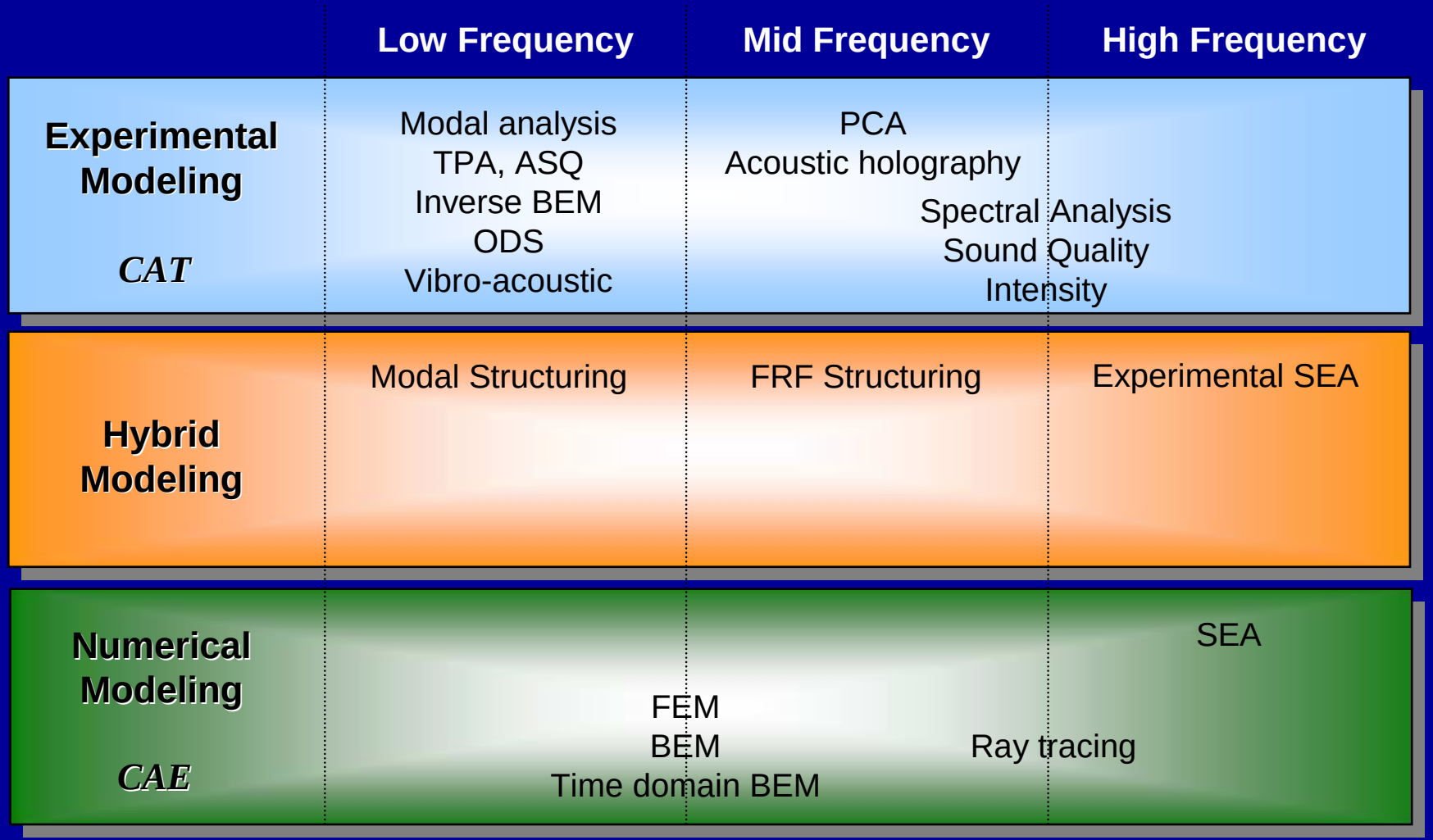
Utilization of the NVH Toolbox



Technologies for NVH

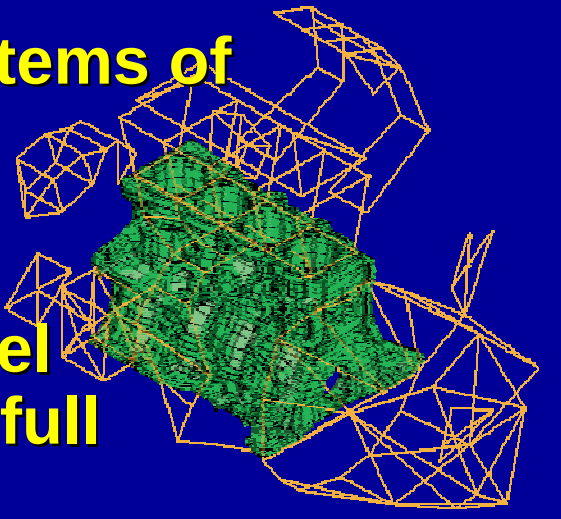


Various Techniques



Hybrid Modeling

- A “Hybrid” model is a full system performance model composed of both numerical and experimental based models
- Test-based models are used for existing / carryover sub-systems
  - Faster than building numerical models
  - More accurately match hardware
- Numerical models are used for sub-systems of design interest
  - Automatic optimization
  - Design Space Studies
- Performance is evaluated at system level
  - More accurate, faster run-times than full numerical models
  - More flexibility than full test-based models



When to Use These Methods

Experimental Modeling

- Utilize previous generation hardware
- Utilize prototype builds
- Numerical models are not available

Numerical Modeling

- Physical hardware is not available
- Evaluate a variety of analysis cases quickly
- Understand variability

Hybrid Modeling

- Both numerical and experimental models are available
- Fast run times
- Evaluate variety of sub-assemblies or components quickly

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- Introduction
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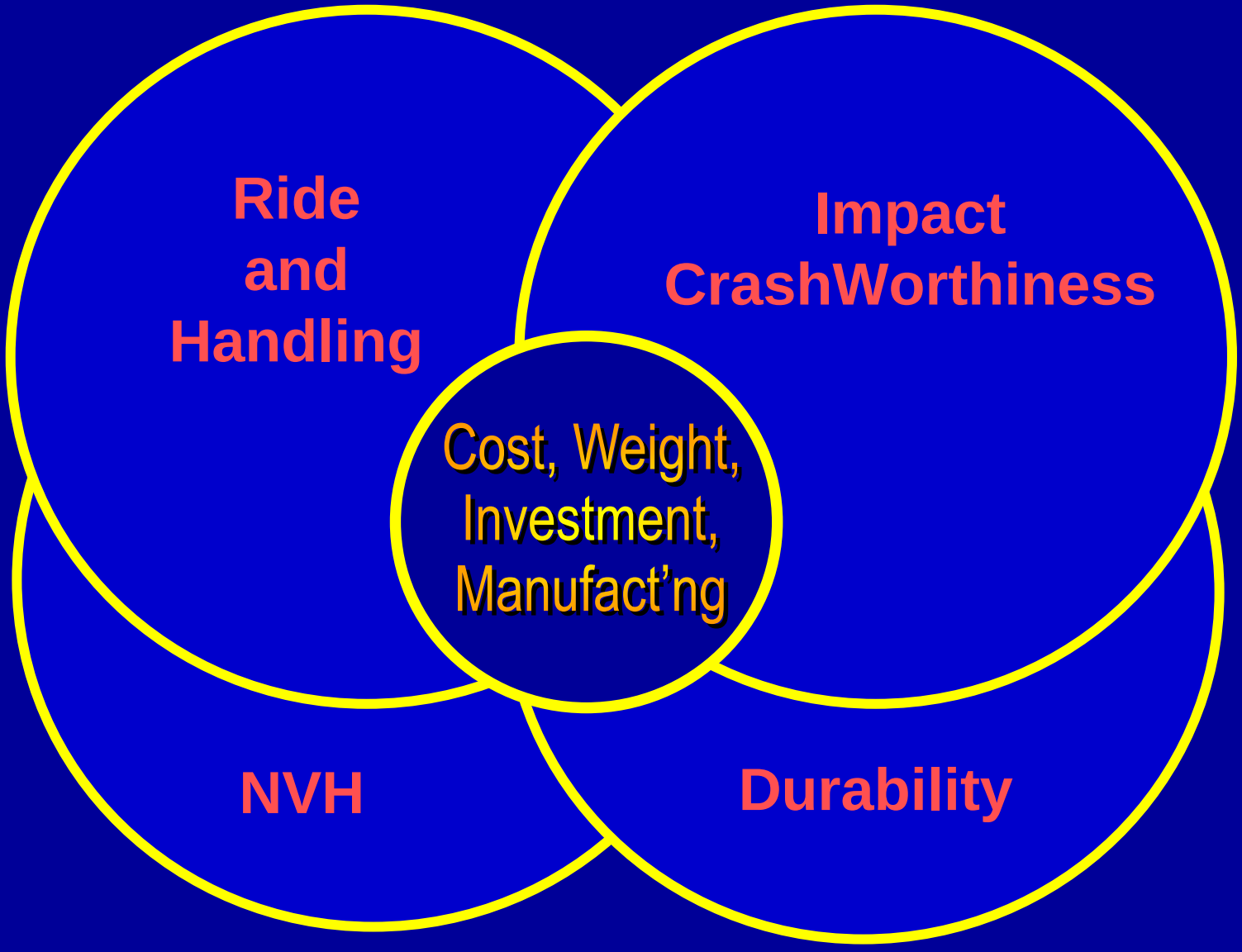
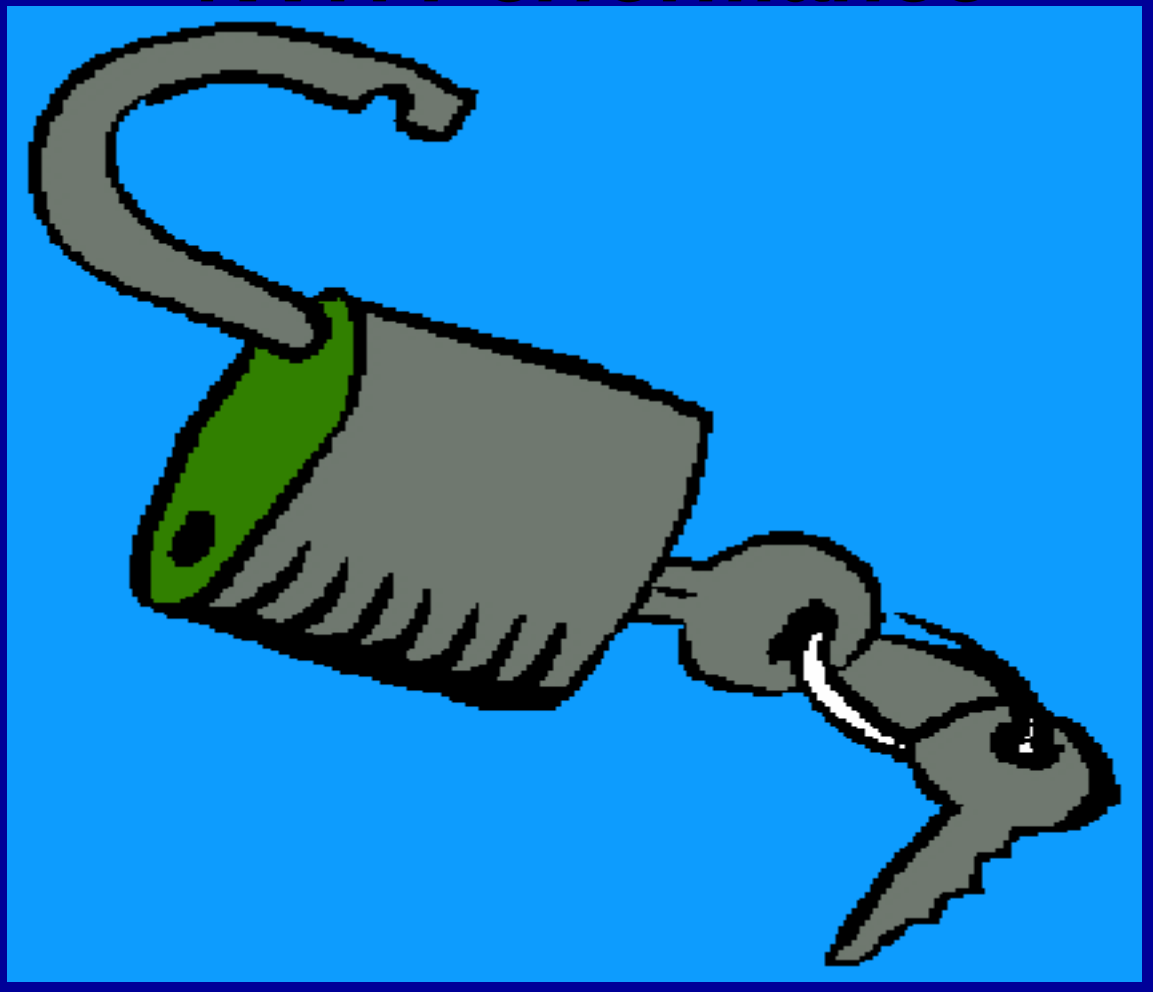
- *Live Noise Attenuation Demo*
- Mid Frequency Basics
- Utilization of the NVH Toolbox
- **Closing Remarks** *Alan Duncan*

Competing Vehicle Design Disciplines



The Fundamental Secret of Structure Borne NVH Performance



Revealed here today !

The Fundamental Secret of Structure Borne NVH Performance

To Minimize Structure Borne NVH, connect between Source and Receiver Sub-Systems at Locations where the Motion is at a Minimum.

Ist Corollary

First Best Principle for NVH Improvement is Minimization / Understanding of the Source Excitation.

The Fundamental Secret of Structure Borne NVH Performance

To Minimize Structure Borne NVH, connect between Source and Receiver Sub-Systems at Locations where the Motion is at a Minimum.

Thank You for Attending the Structure Borne NVH Workshop

Your Presenters today were:

Alan Duncan, Material Sciences Corp.
Contact: Alan.Duncan@matsci.com or AEDuncan@AutoAnalytics.com

Greg Goetchius, Material Sciences Corp.
Contact: Greg.Goetchius@matsci.com

Sachin Gogate, DaimlerChrysler Corp.
Contact: sg57@dcx.com

Greg Hopton, LMS North America
Contact: Greg.Hopton@lmsna.com

Visit: SAE at www.sae.org/events/nvc/specialevents.htm

Also with Ref.s at: www.AutoAnalytics.com/papers.html

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