Structure Borne NVH Basics

SAE 2003 NVH Conference; Traverse City, Michigan
Wednesday Evening, May 7, 2003

Presenters:

A. E. Duncan — **Automotive Analytics, Inc.**

G. Goetchius — **Material Sciences Corp.**

S. Gogate — **Altair Engineering, Inc.**

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NVH Workshop Topic Outline

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

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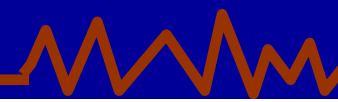
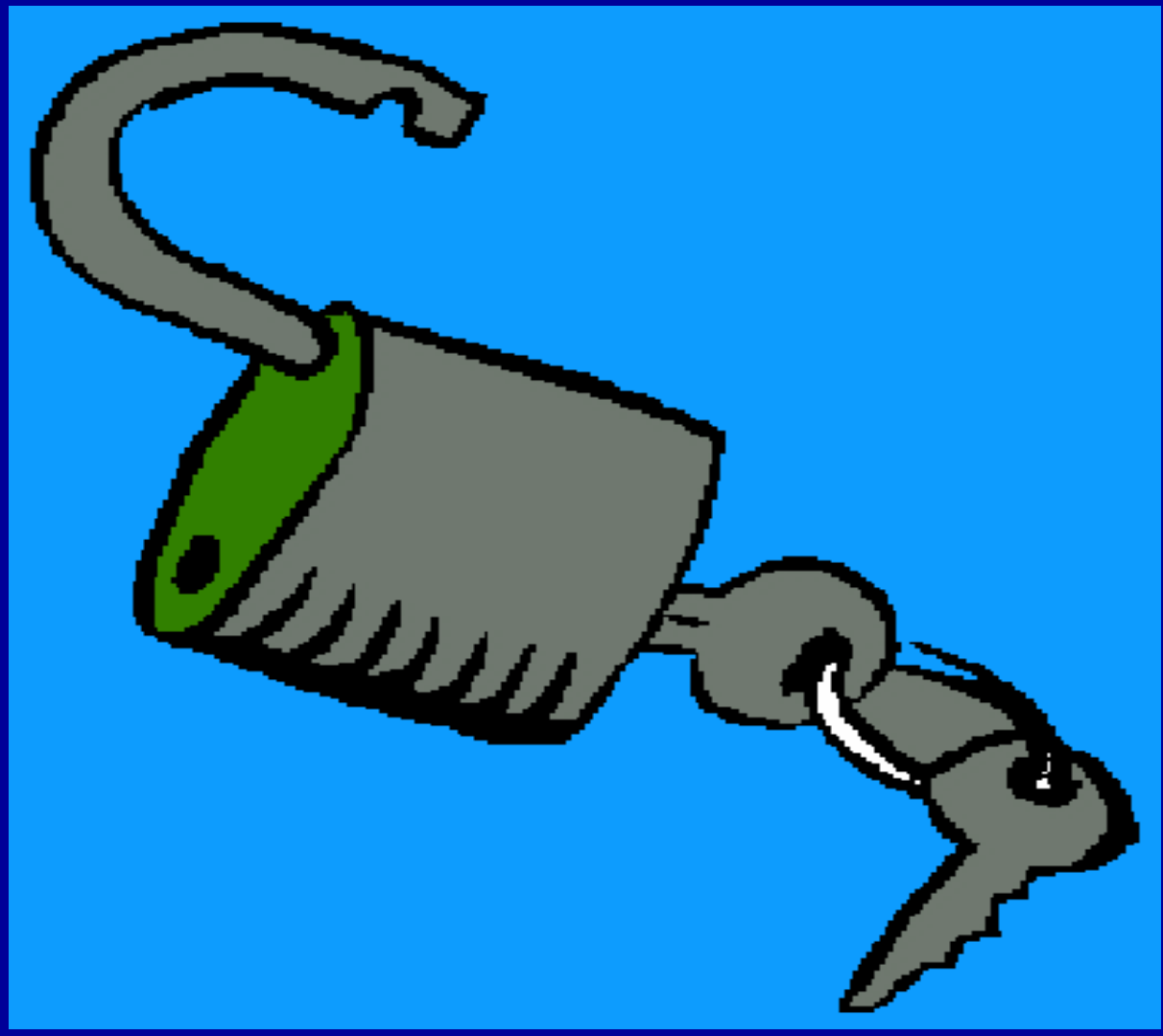


The Fundamental Secret of Structure Borne NVH Performance



Revealed here tonight !

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Structure Borne NVH Basics 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

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Structure Borne NVH Basics References

Supplemental References

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. Kompella, M. S., and Bernhard, J., "Measurement of the Statistical Variation of Structural-Acoustic Characteristics of Automotive Vehicles", SAE No. 931272, 1993

9. Freymann, R., and Stryczek, R., "A New Optimization Approach in the Field of Structural-Acoustics", SAE No. 2000-01-0729, 2000

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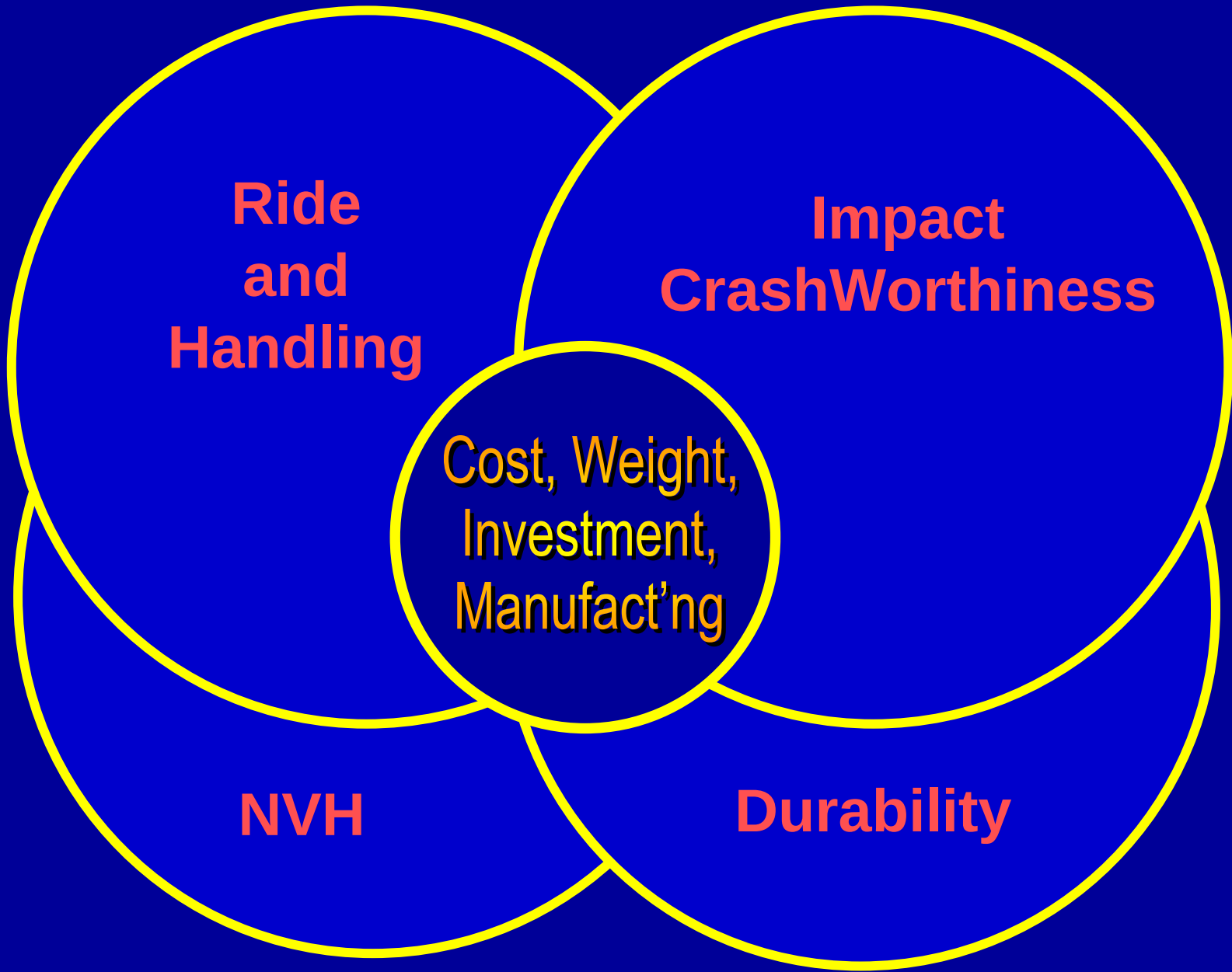
- NVH Load Conditions
- Low Frequency Basics

Competing Vehicle Design Disciplines



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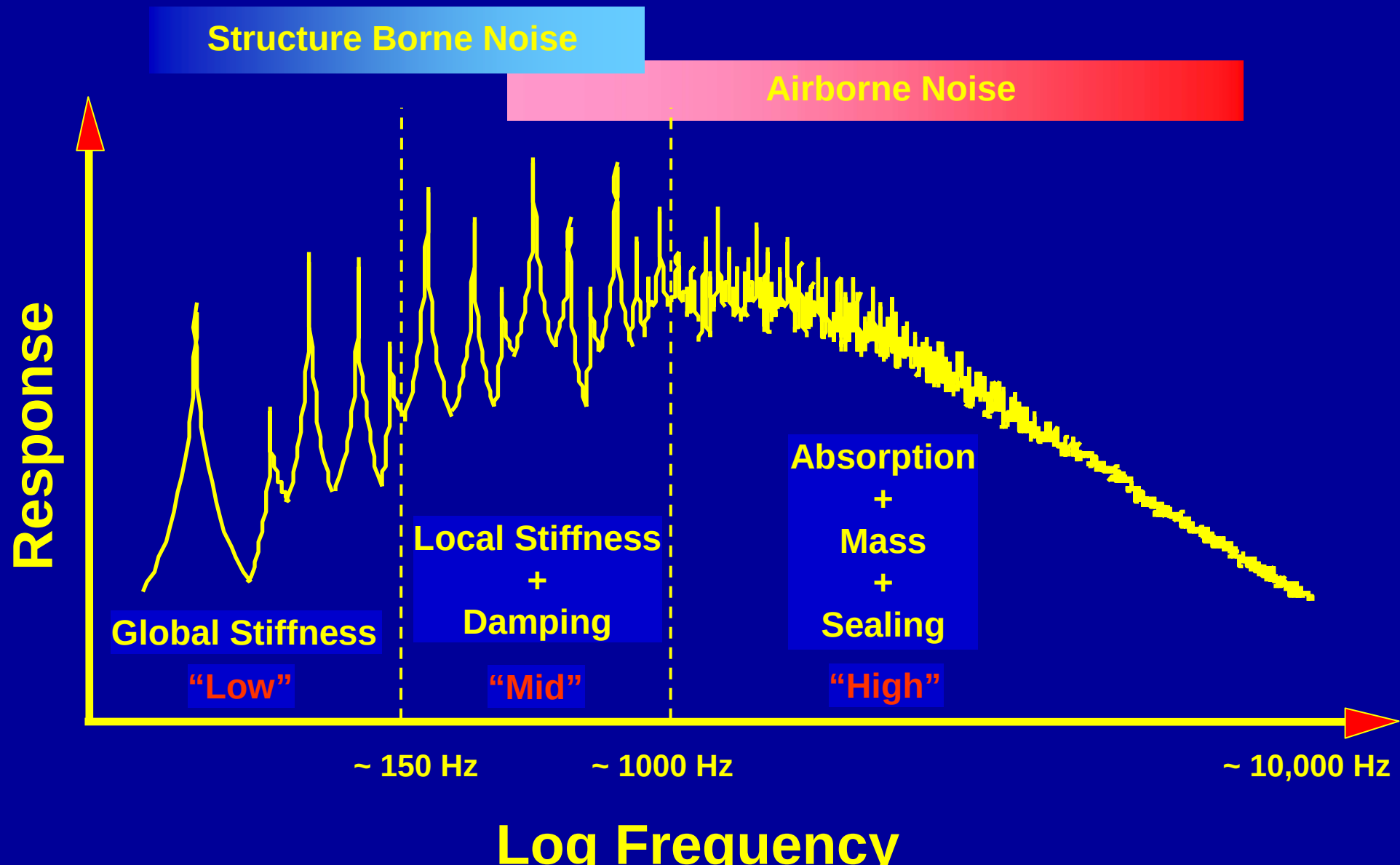
Automotive NVH Frequency Range



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

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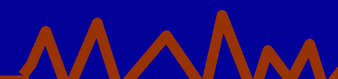

Study of Ride Balance

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

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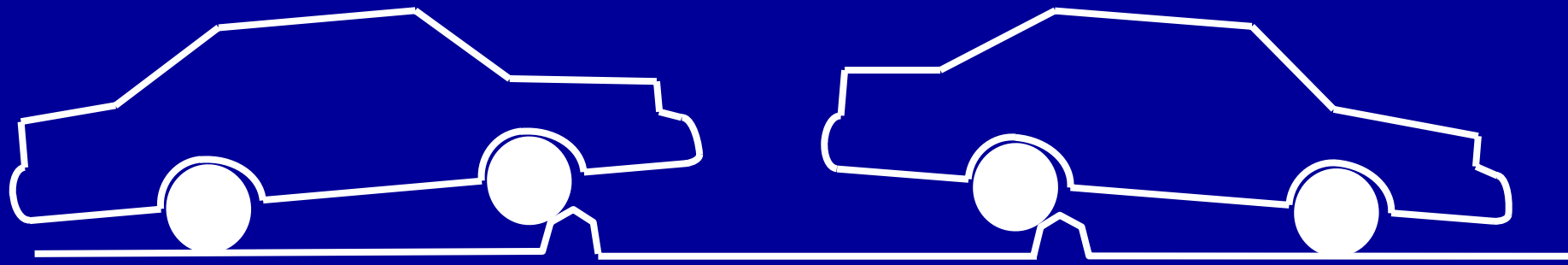
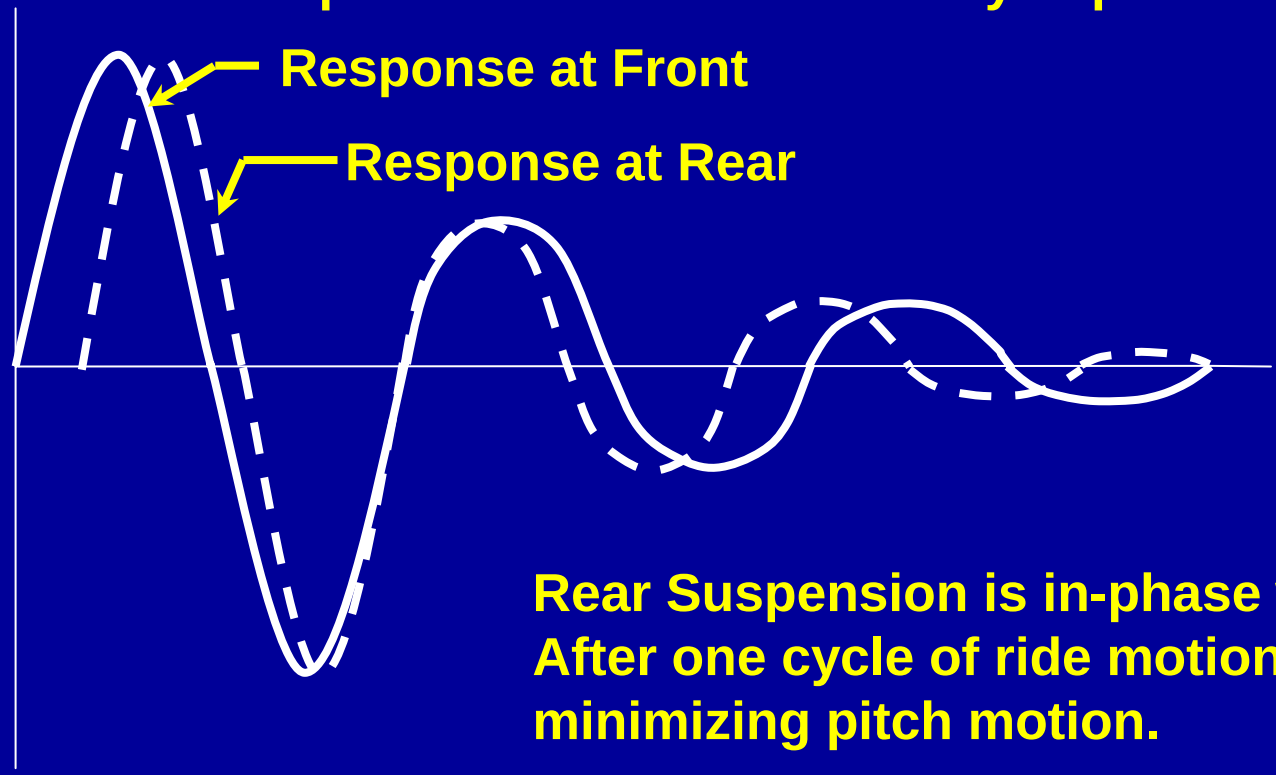
Ride Balance Study

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

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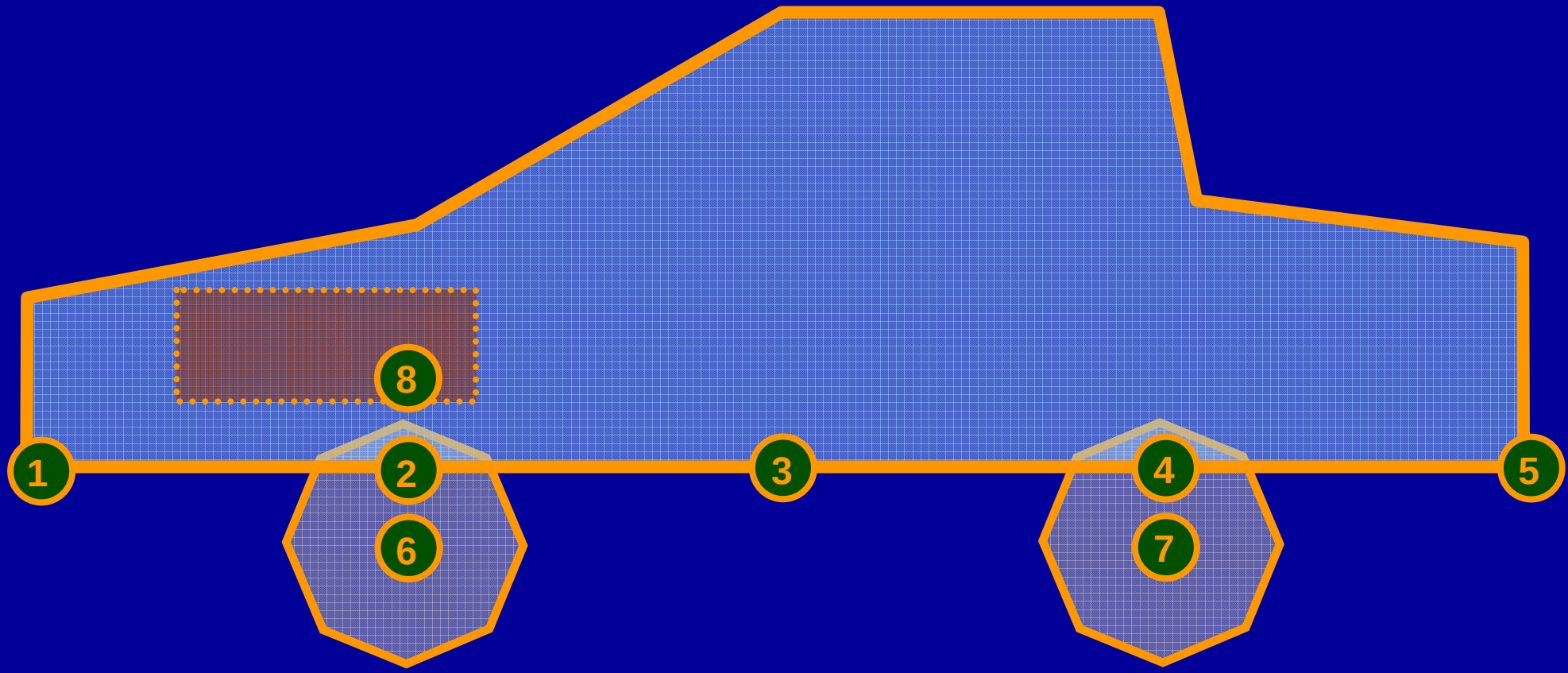
NVH Model of Unibody Passenger Car

Symbolic Outline



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

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

Beam mass lumped on grids like a beam

M2,3,4 =2 * M1,5

From Reference 6

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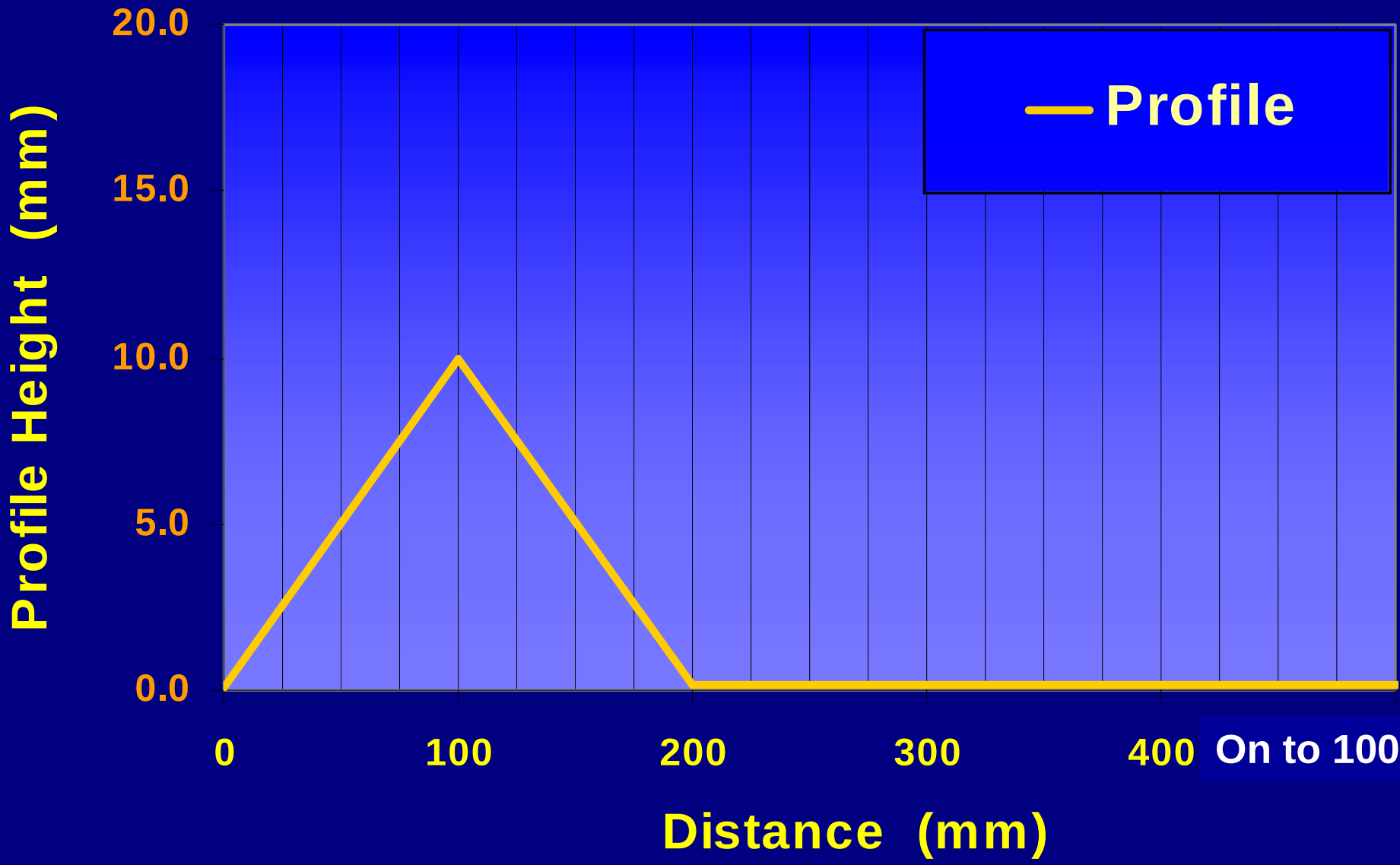
Excitation Bump Profile



On to 100,380

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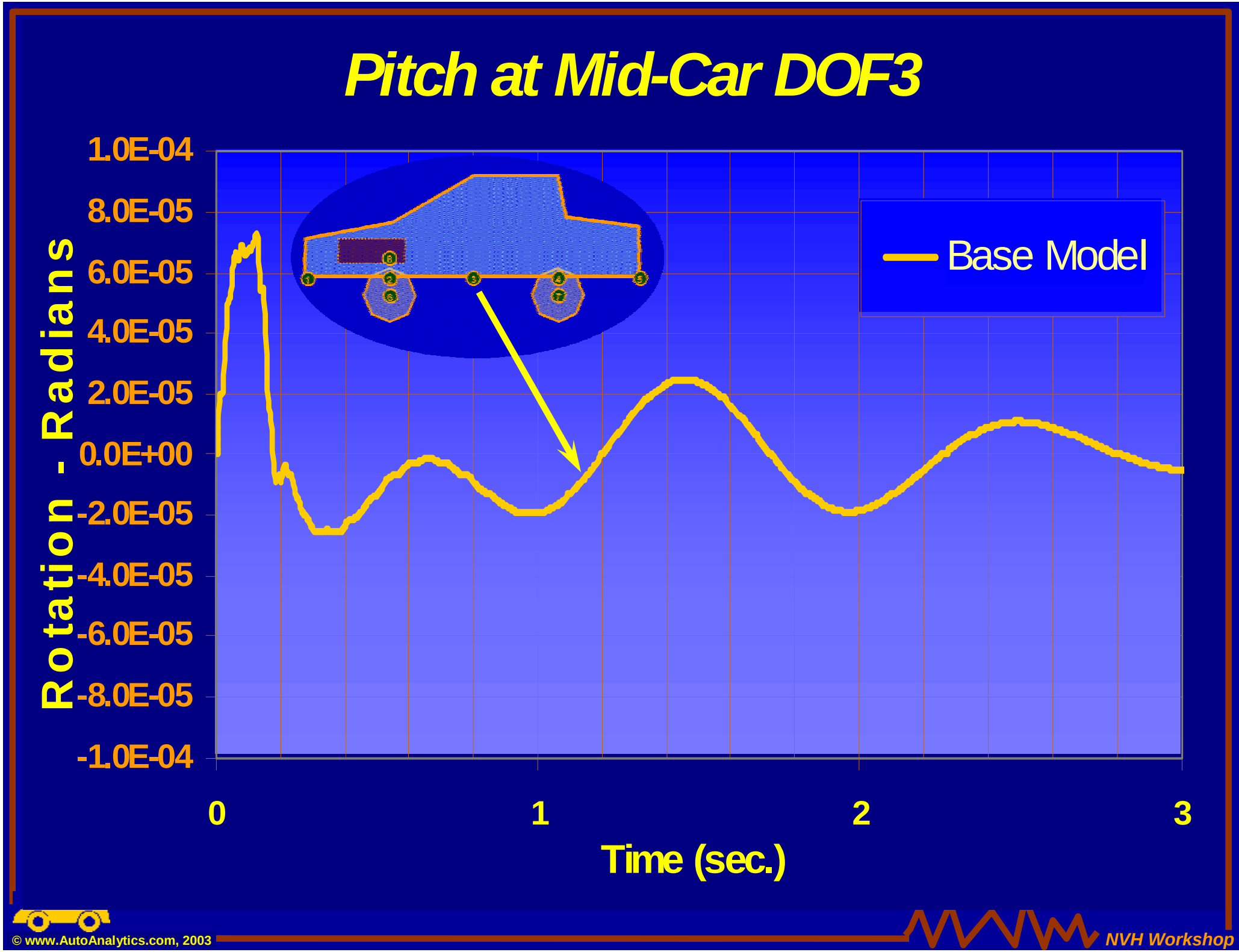
Pitch at Mid-Car DOF3



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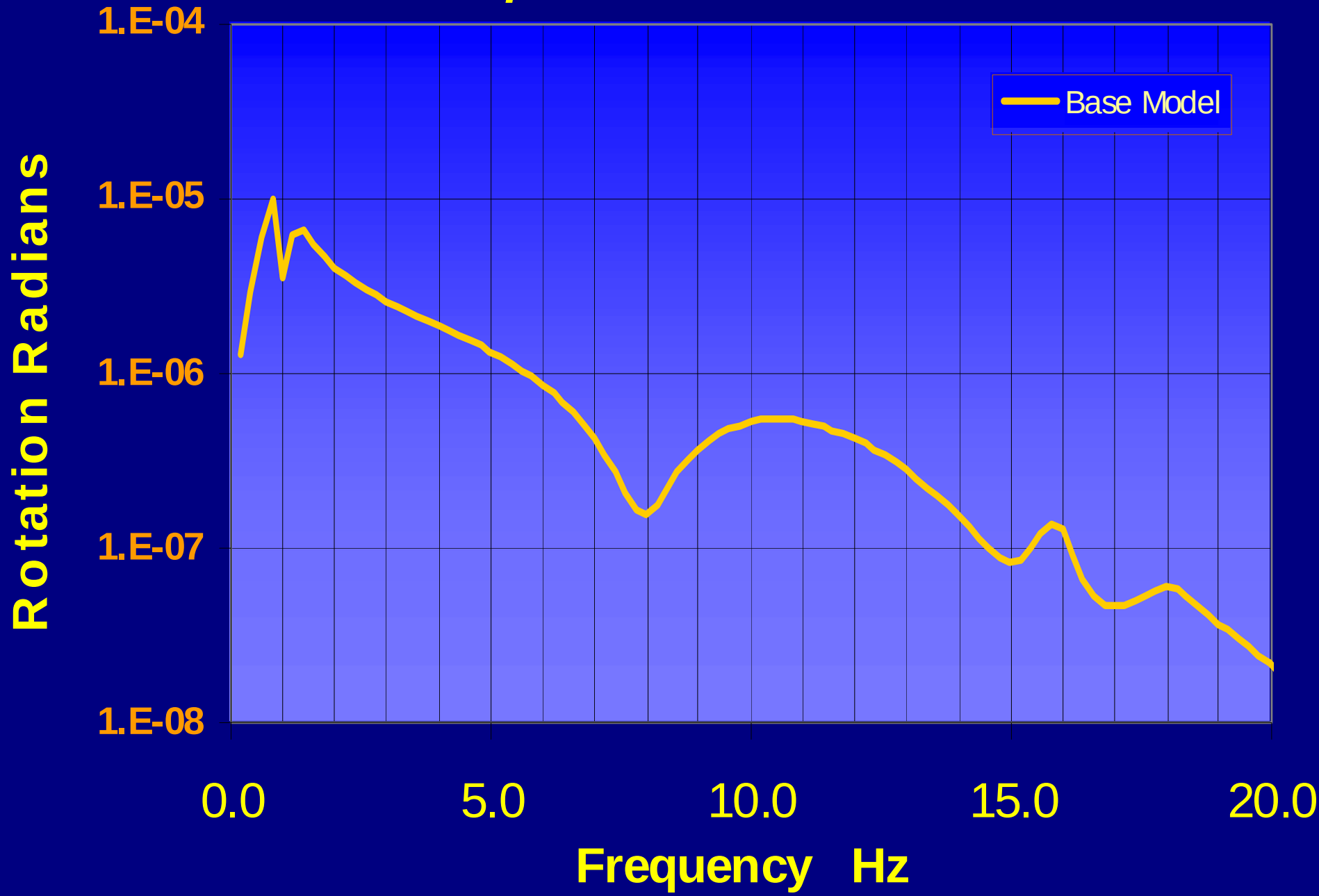
Pitch Response - Baseline Model



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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)

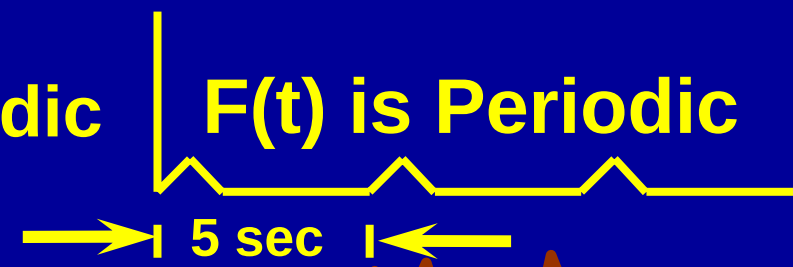
Fourier transform of the input forces, vector of all input possibilities

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

Caveat

X (t) must be Periodic
Linear System

F(t) is Periodic



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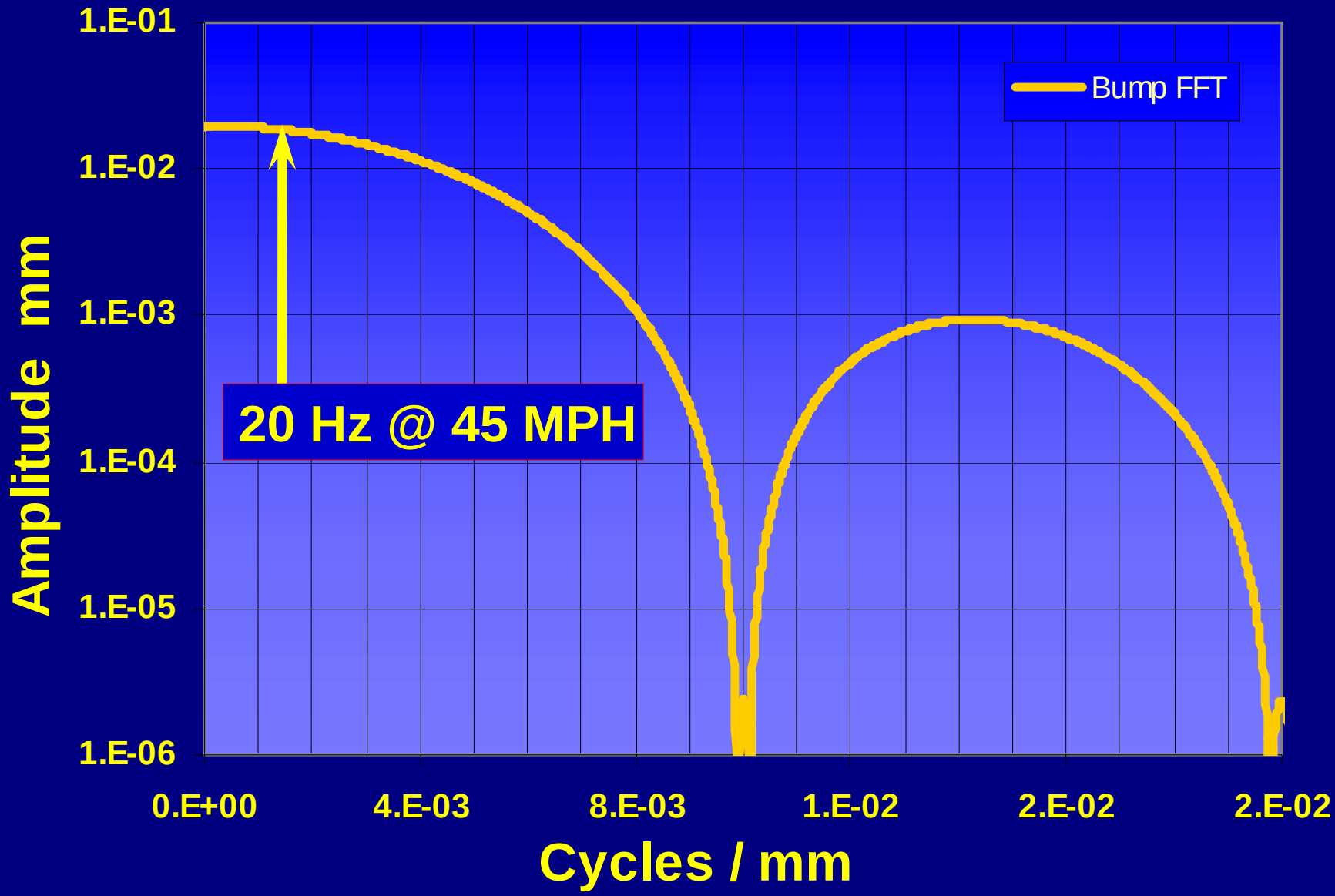
Transform Input Force to F(f)

FFT of the Input Bump



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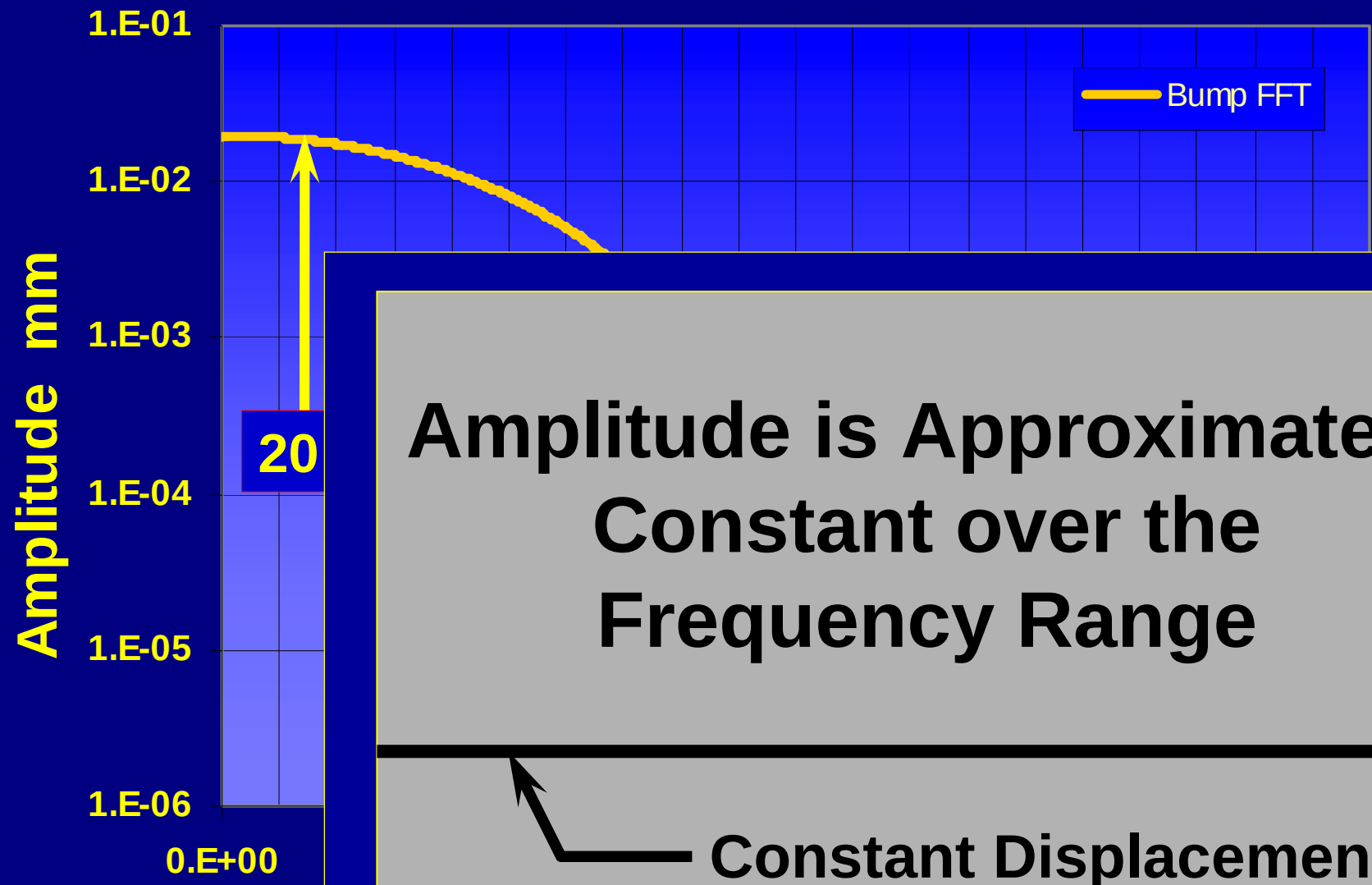

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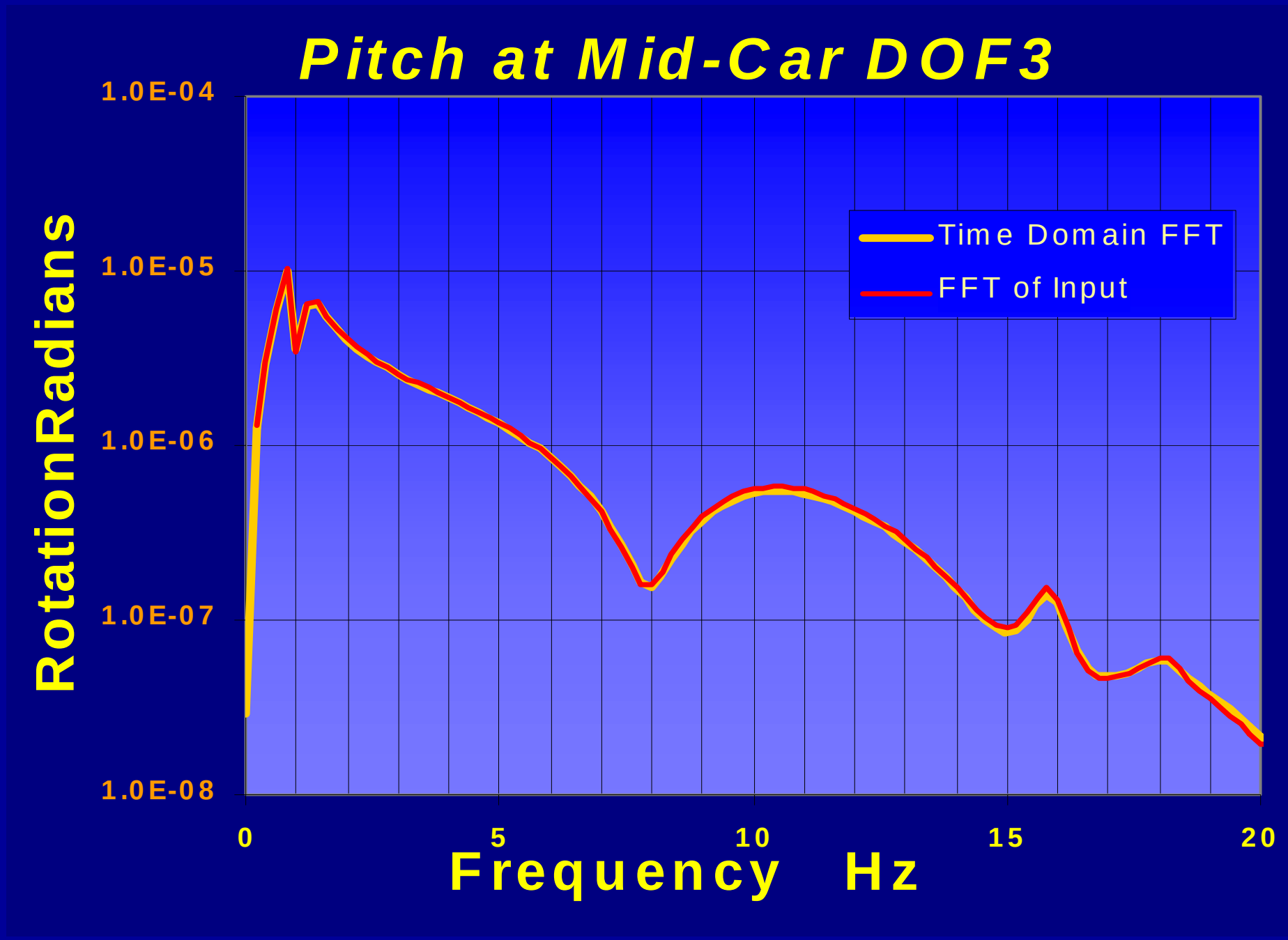

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Ride Balance Study Summary

- 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

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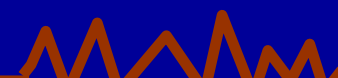

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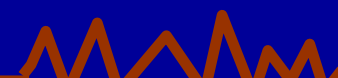
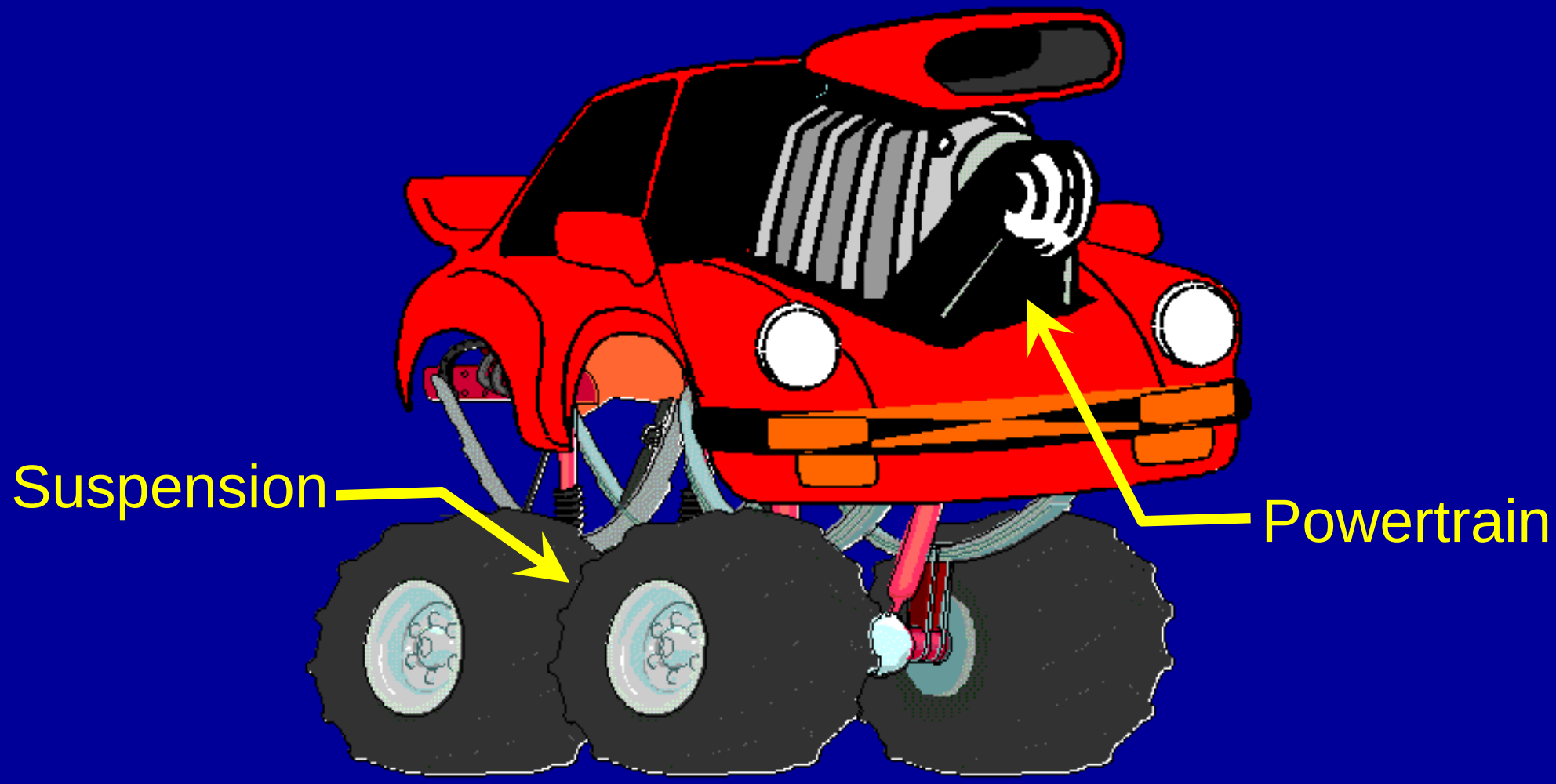


Noise and Vibration Sources



Two Main Sources

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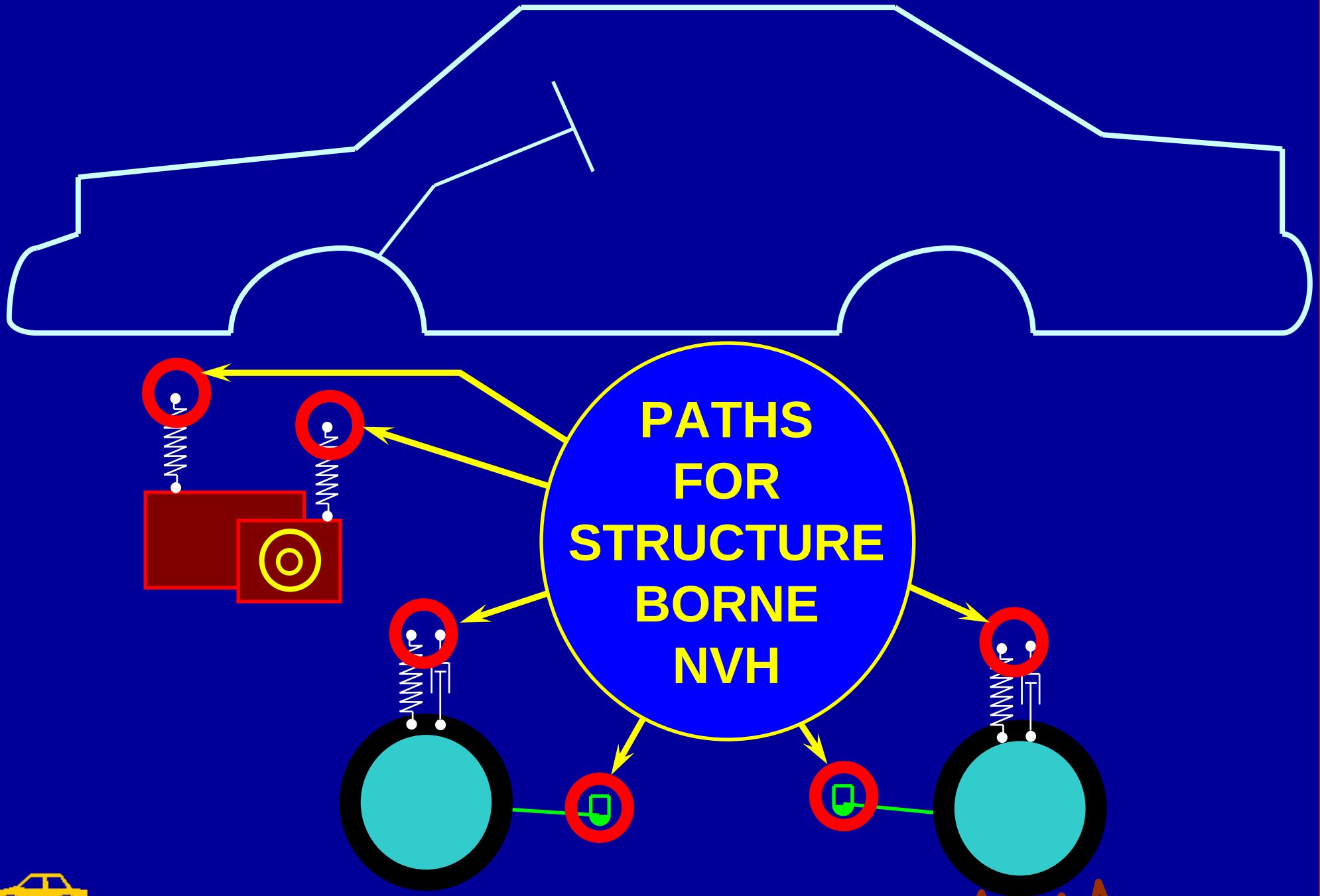
Typical NVH Pathways to the Passenger



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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

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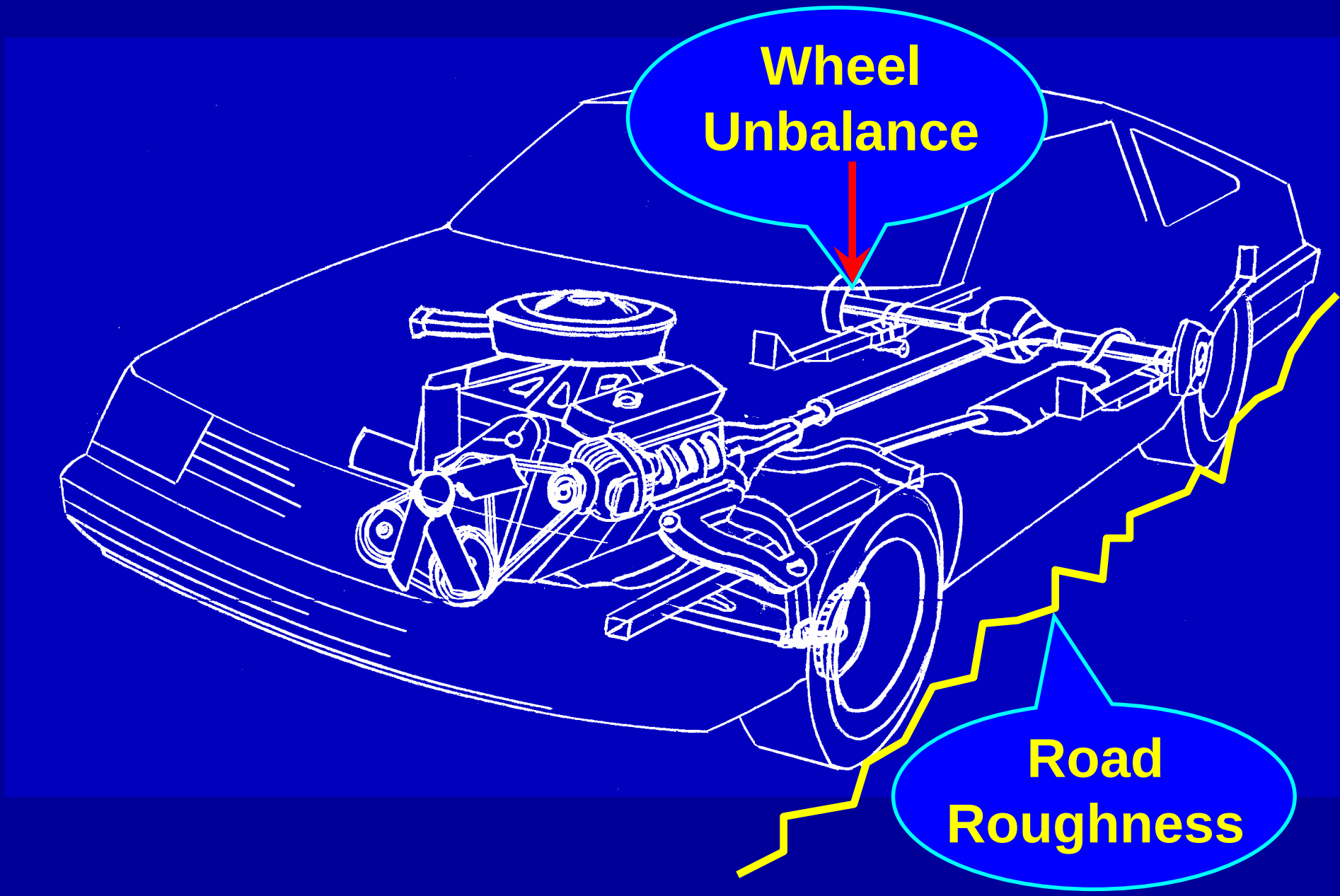
Suspension Excitation Load Conditions



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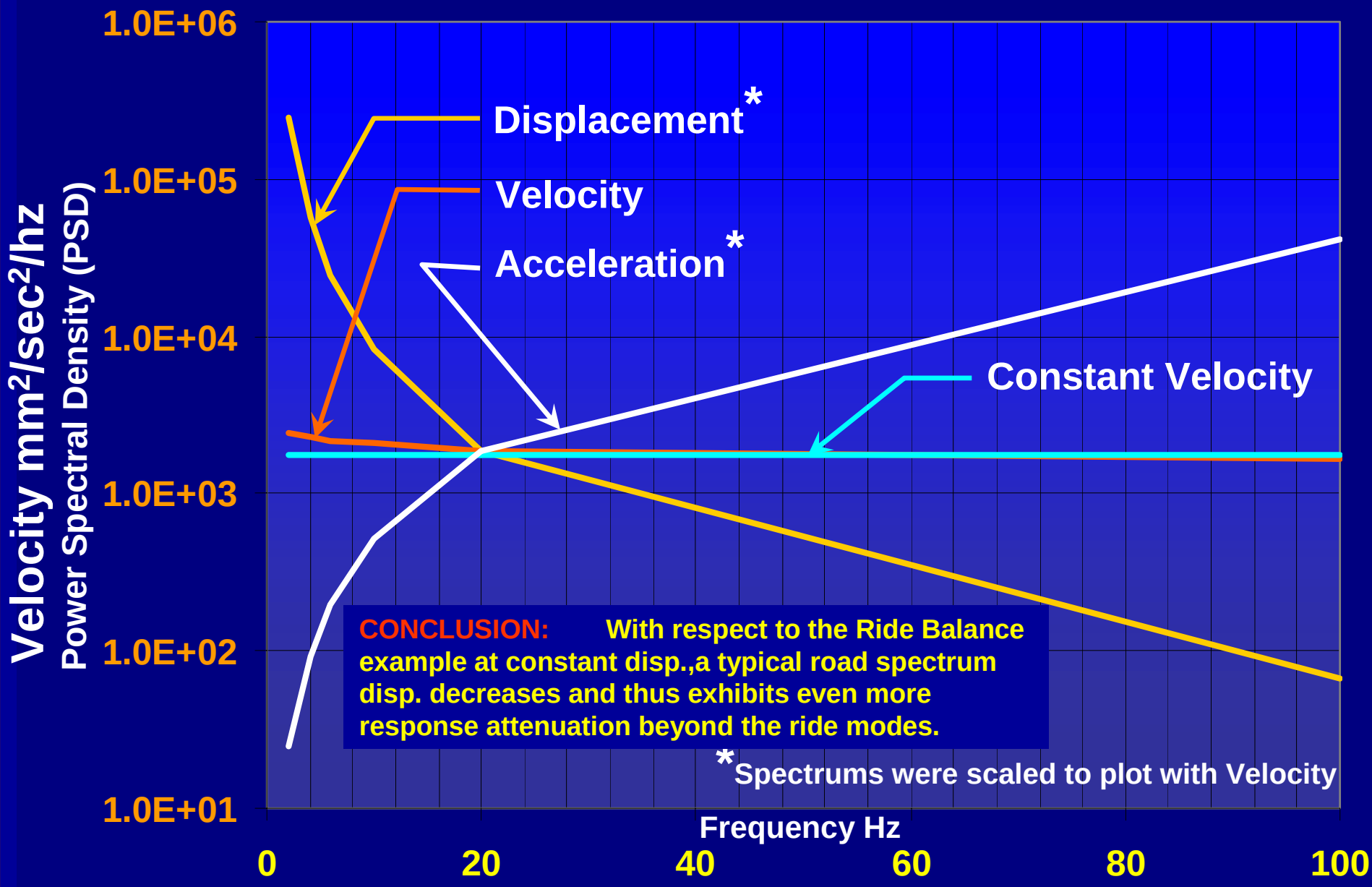
Medium Rough Road Spectrums from Wong, Ref. 7



Road Spectrum at Constant Speed of 45 MPH

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Low Frequency NVH Basics

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

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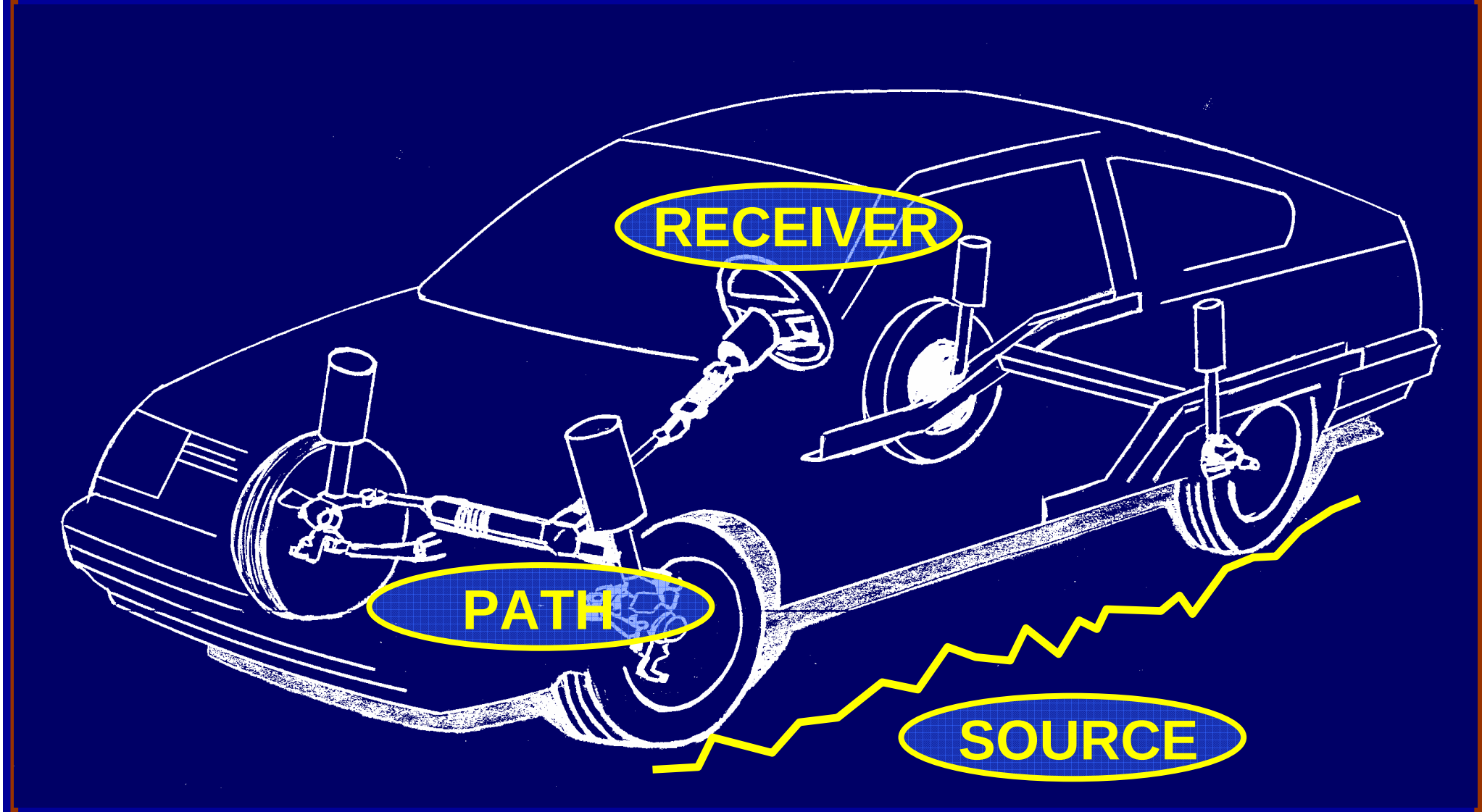
Low Frequency NVH Fundamentals



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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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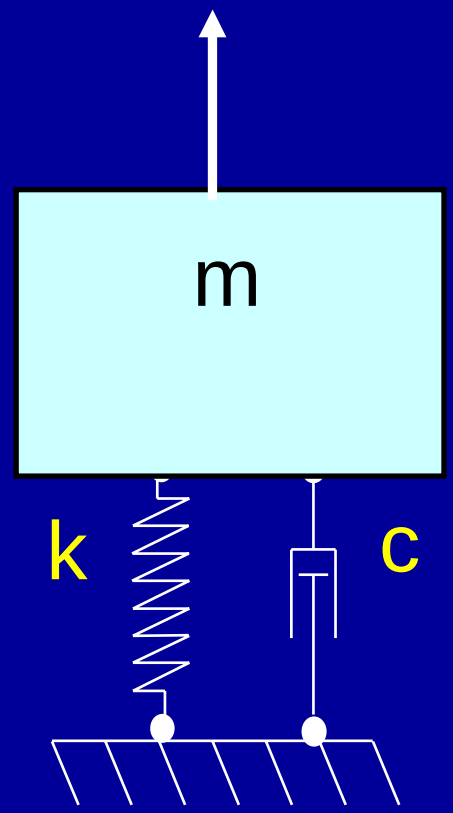
Single Degree of Freedom Vibration

APPLIED FORCE

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



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

Transmitted Force

F_T

d = fraction of critical damping

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

f = operating frequency

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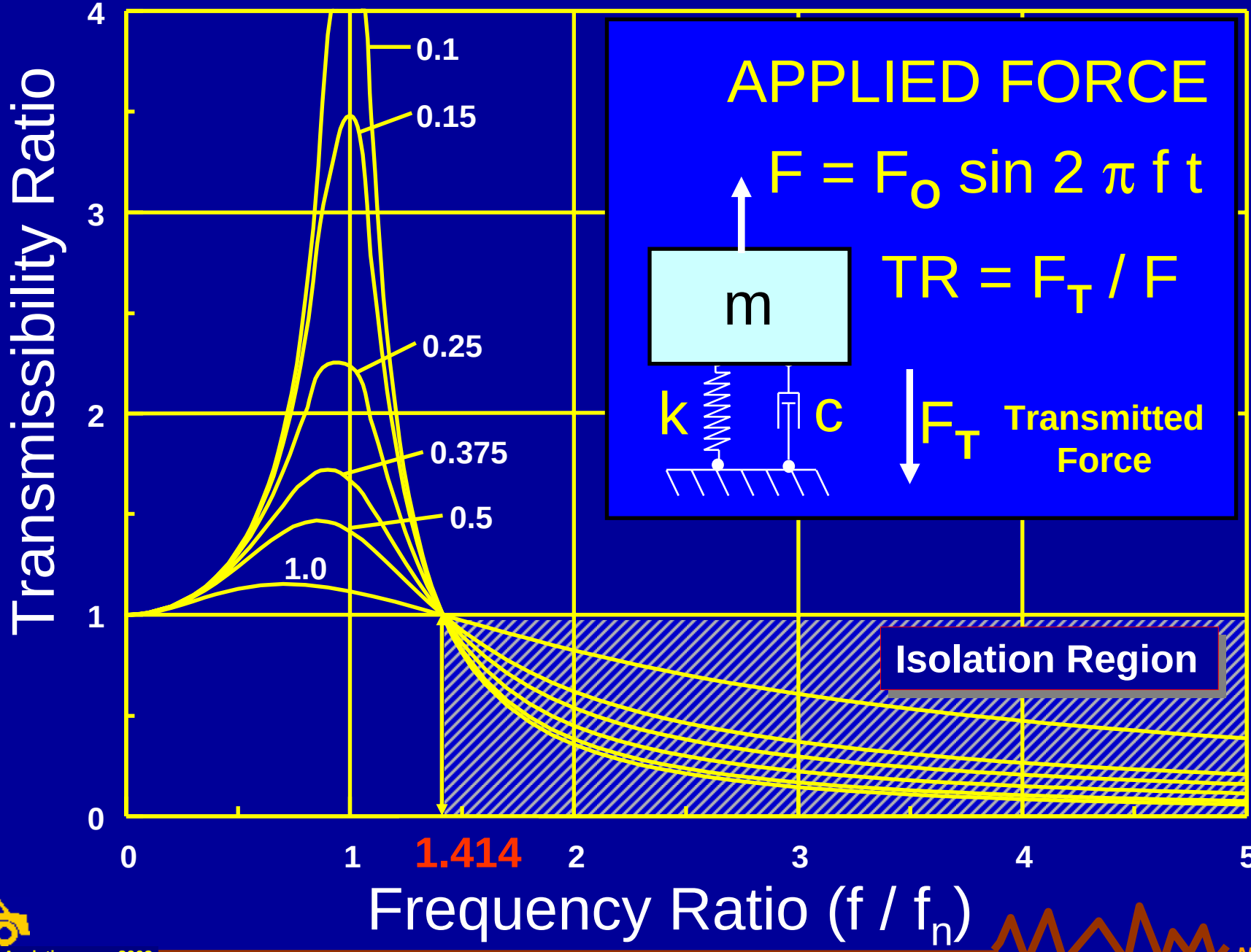
Vibration Isolation Principle



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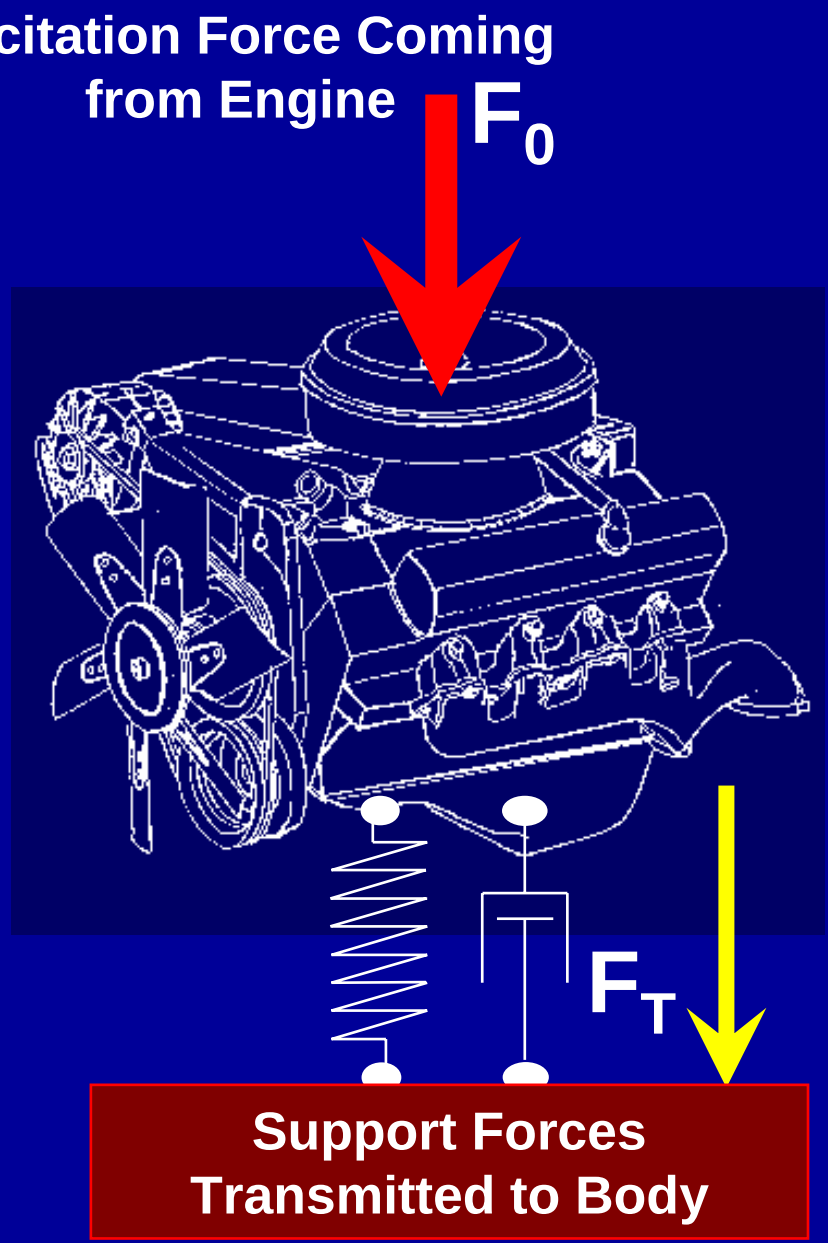
Isolation from an Applied Force



Transmissibility Force Ratio is F_T/F_0

Example:

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

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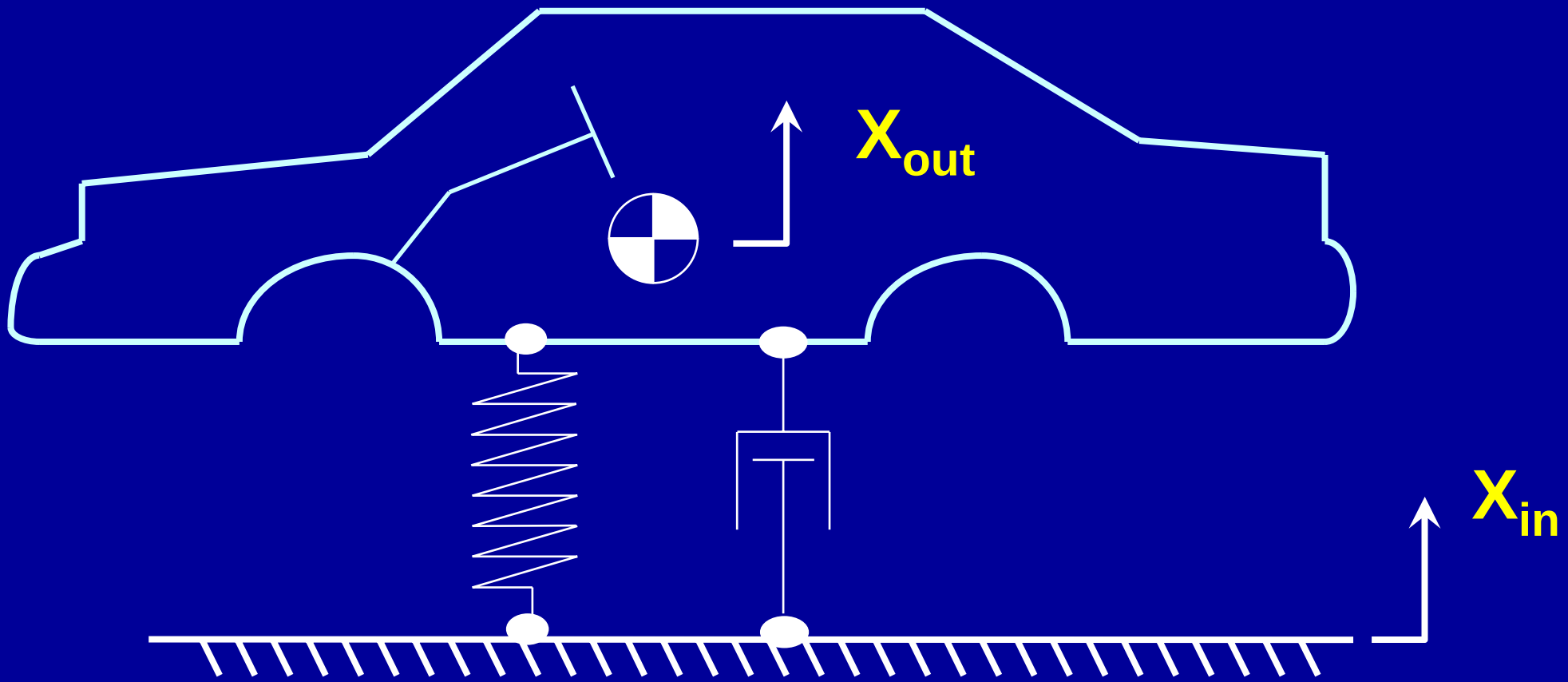
Body on Suspension Single DOF Model

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.

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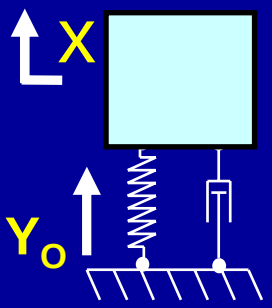
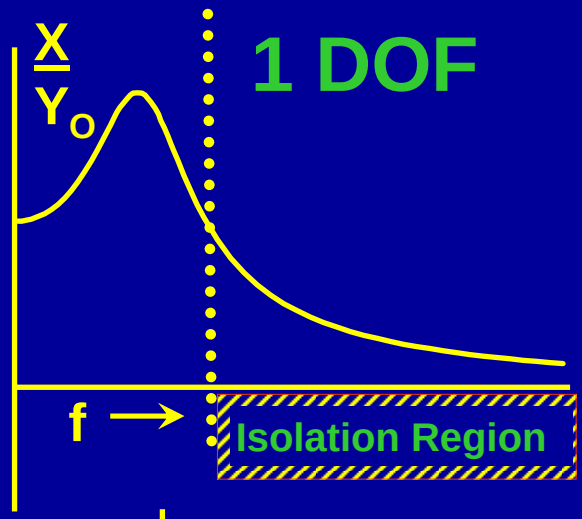
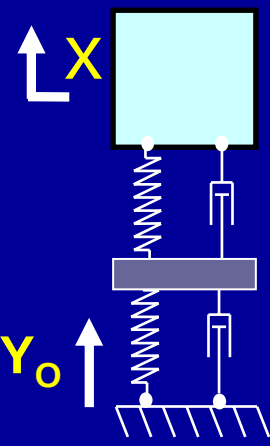
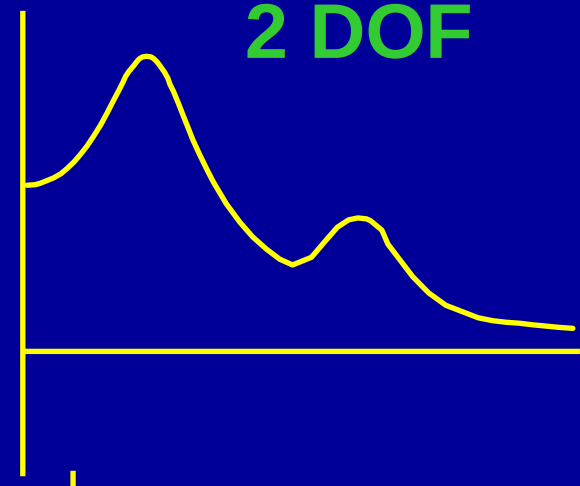
Simplified Models from 1 to 8 DOF's

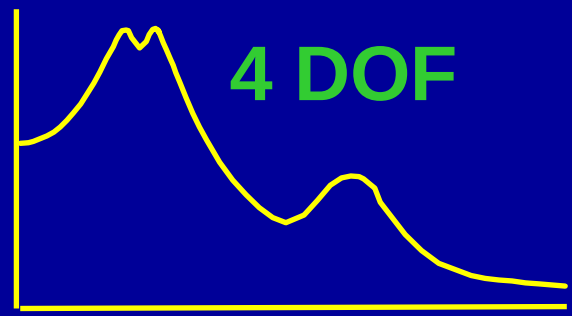
Enforced Base Motion

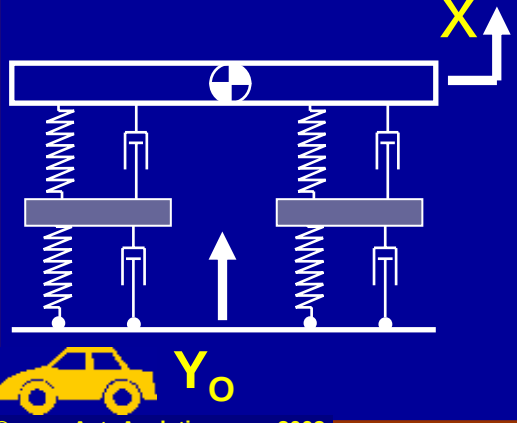


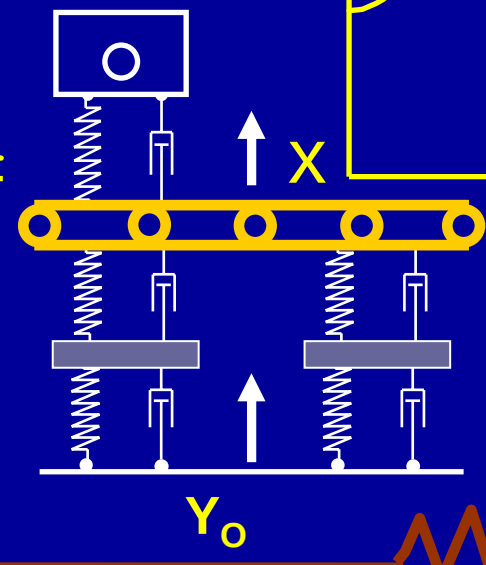


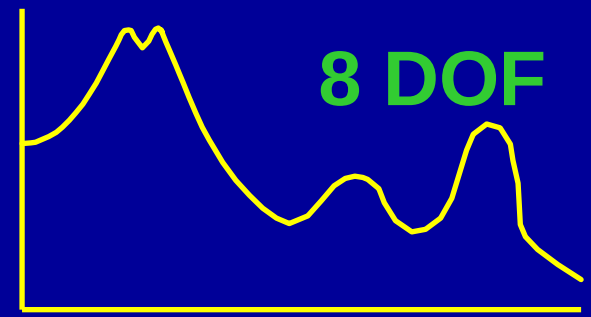












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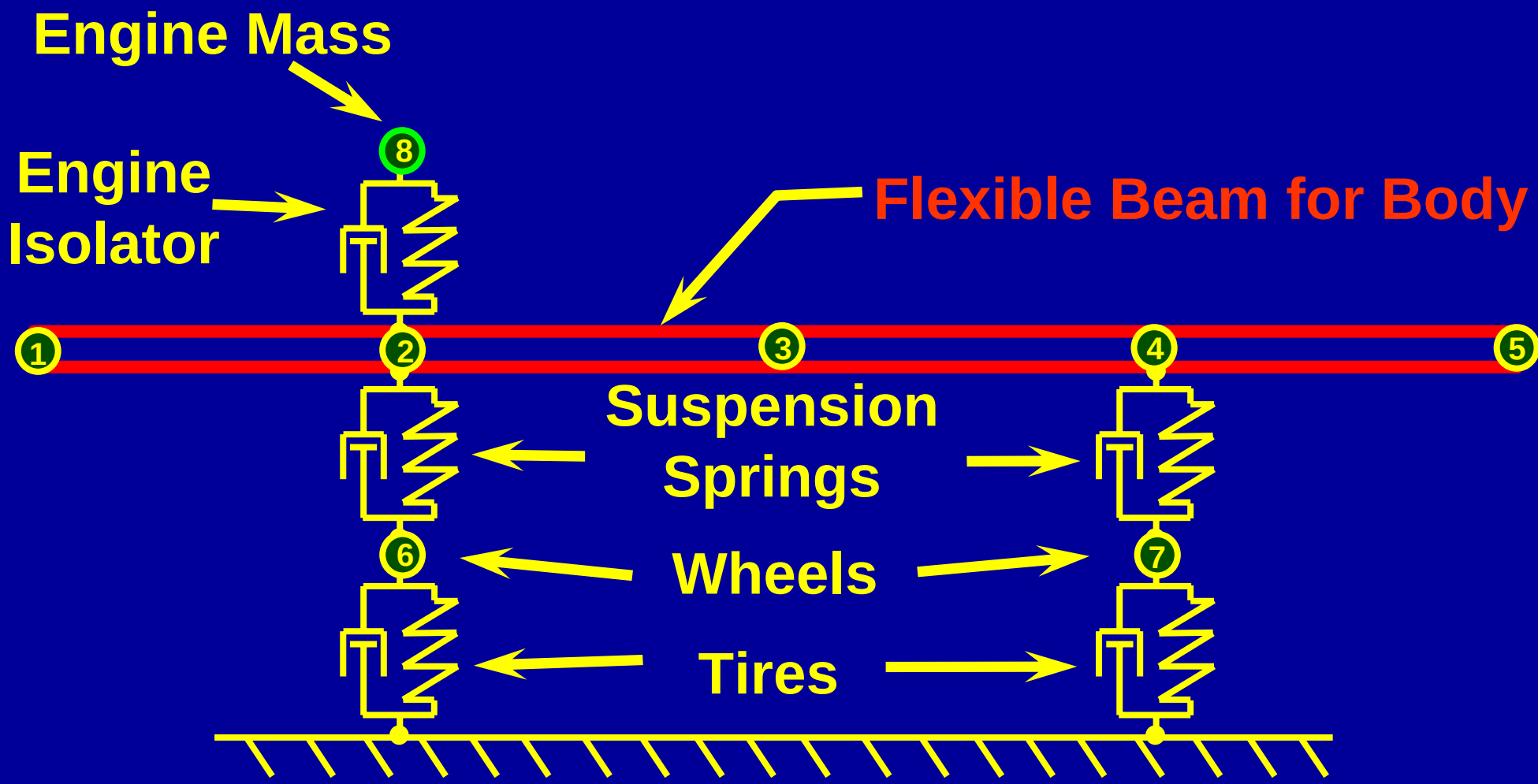
8 Degree of Freedom Vehicle NVH Model



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Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

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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.

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Vibration and Noise Attenuation Methods

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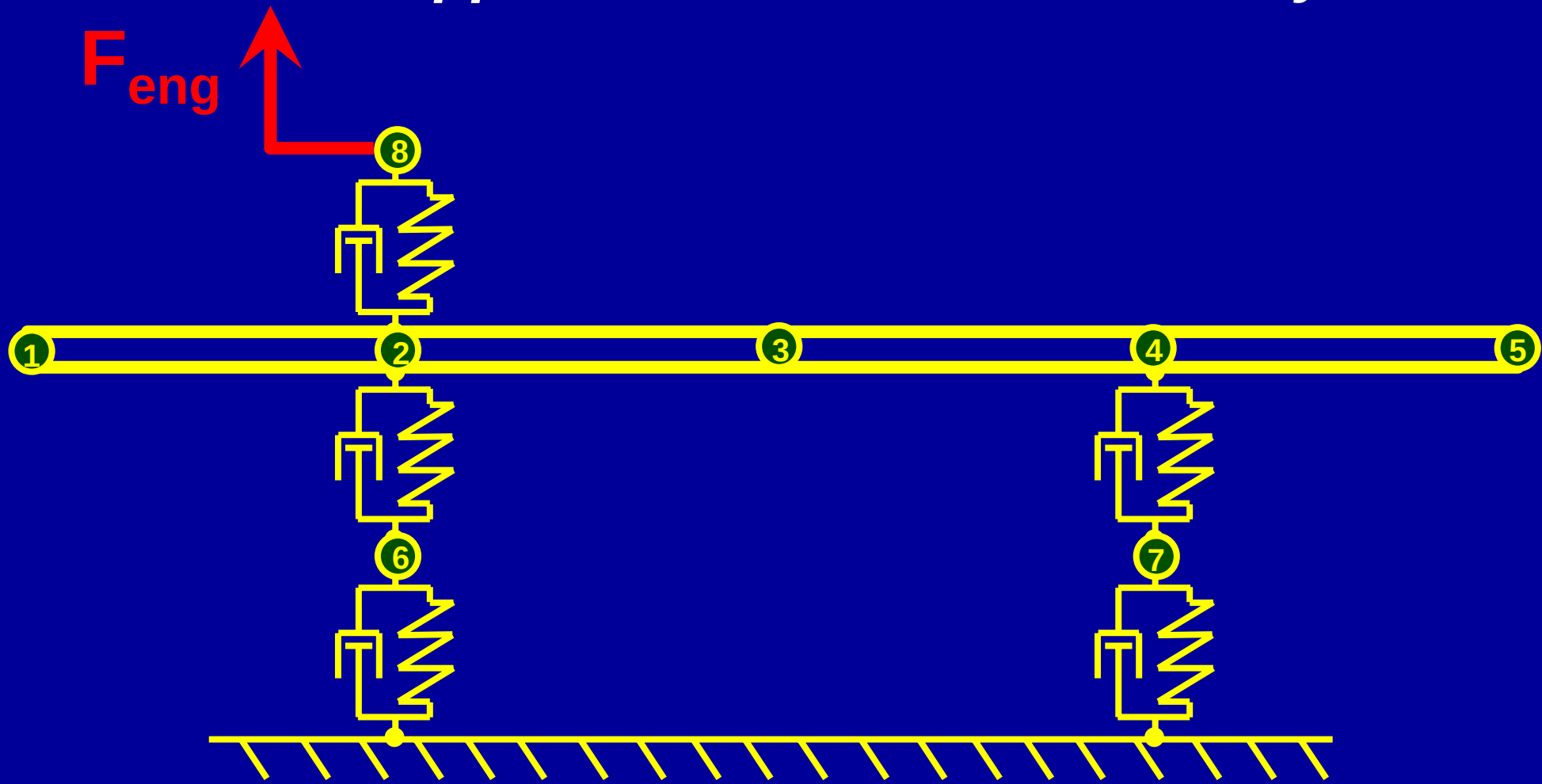
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



Forces at Powertrain could represent a First Order Rotating Imbalance

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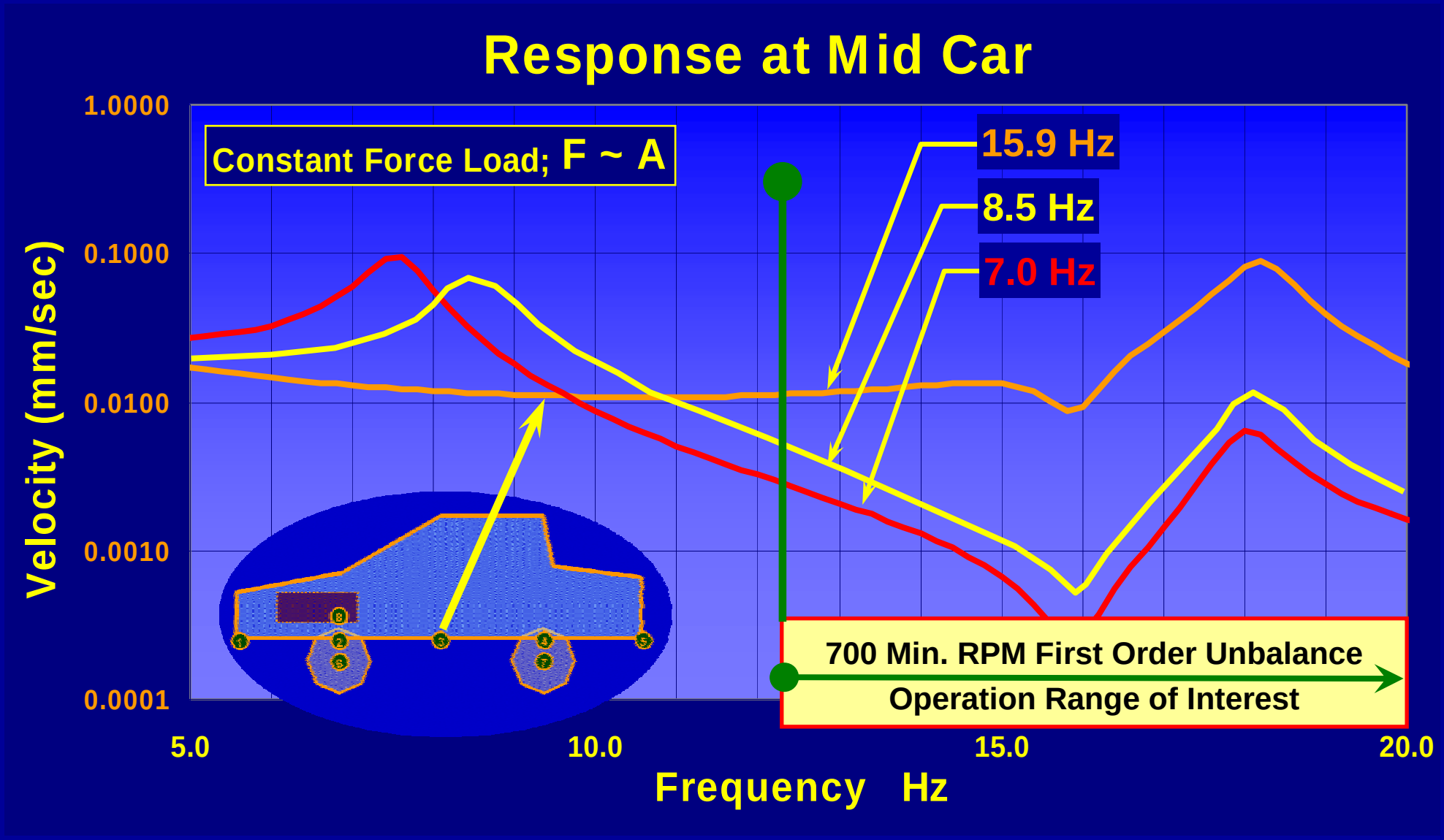
Engine Isolation Example



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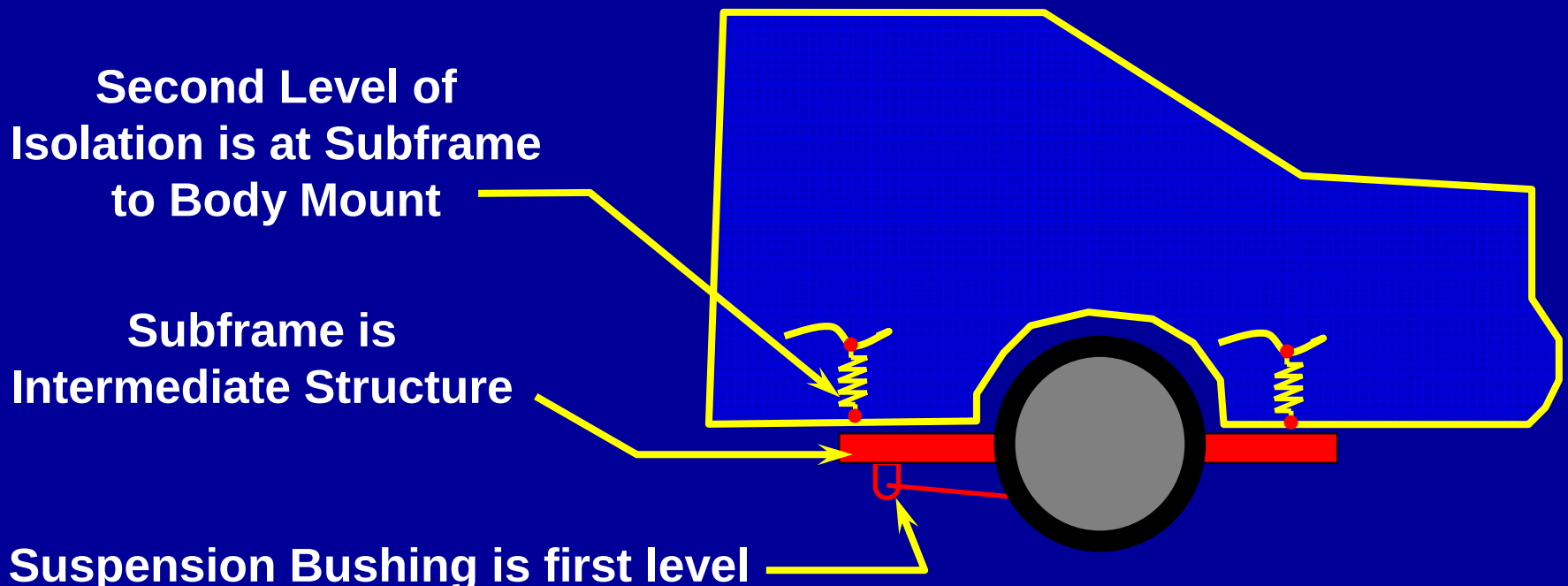
Concepts for Increased Isolation

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



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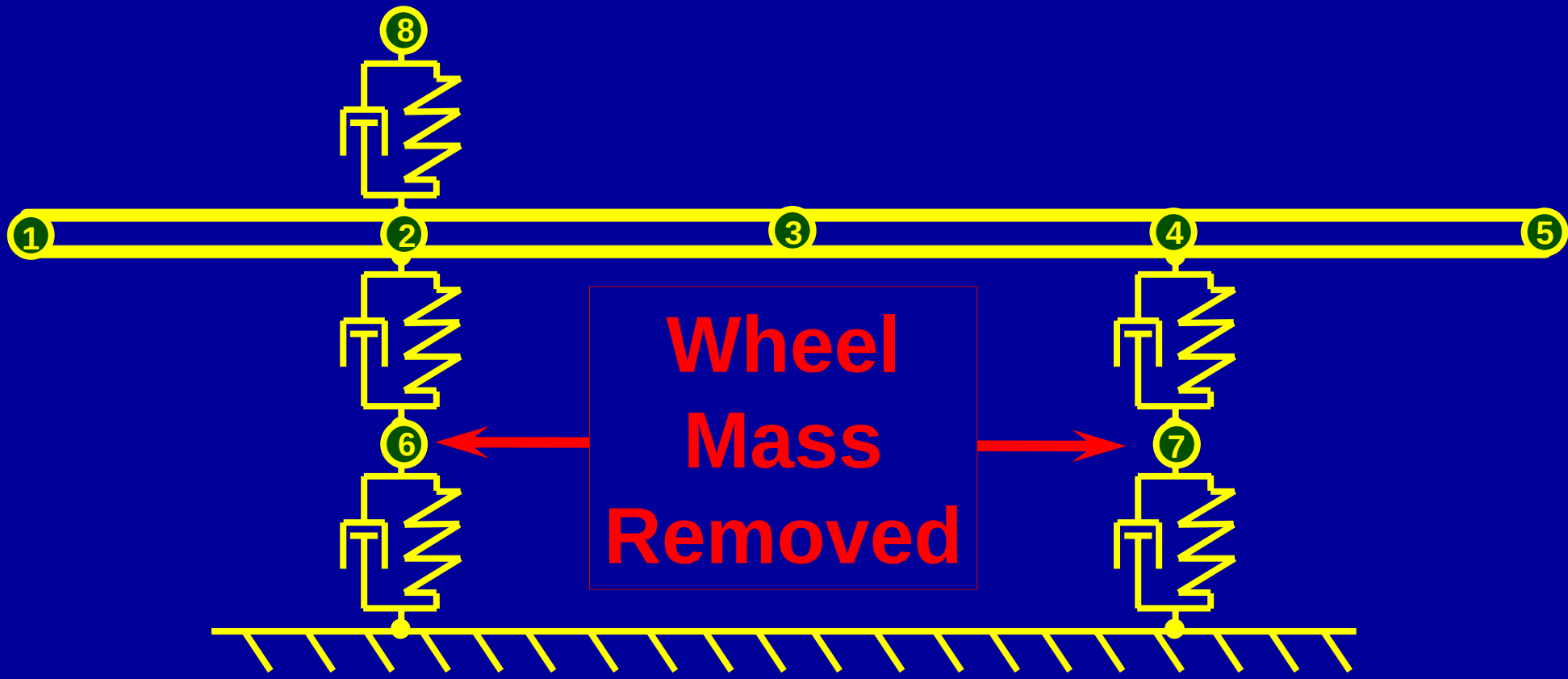
8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



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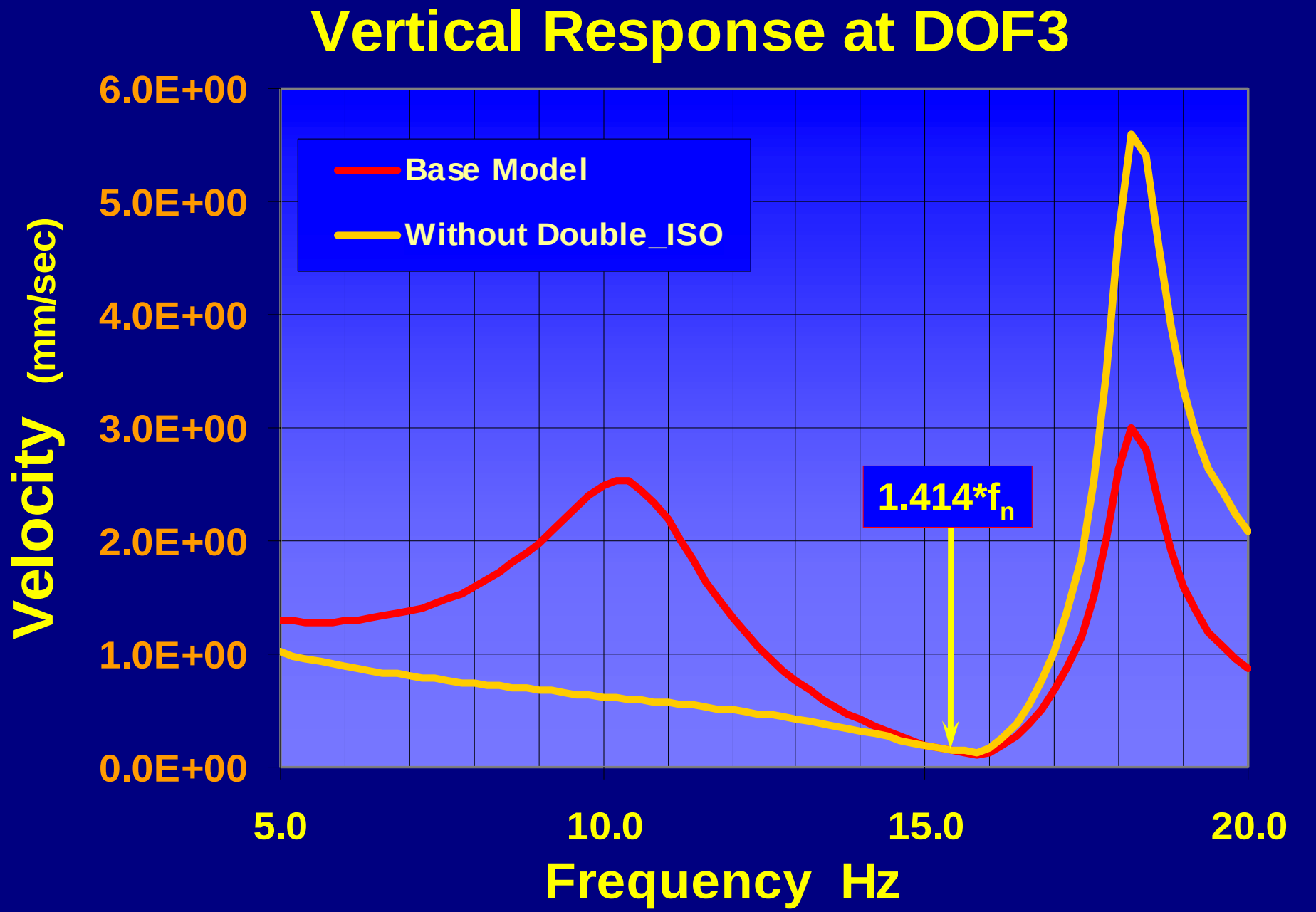
Double Isolation Example



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Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

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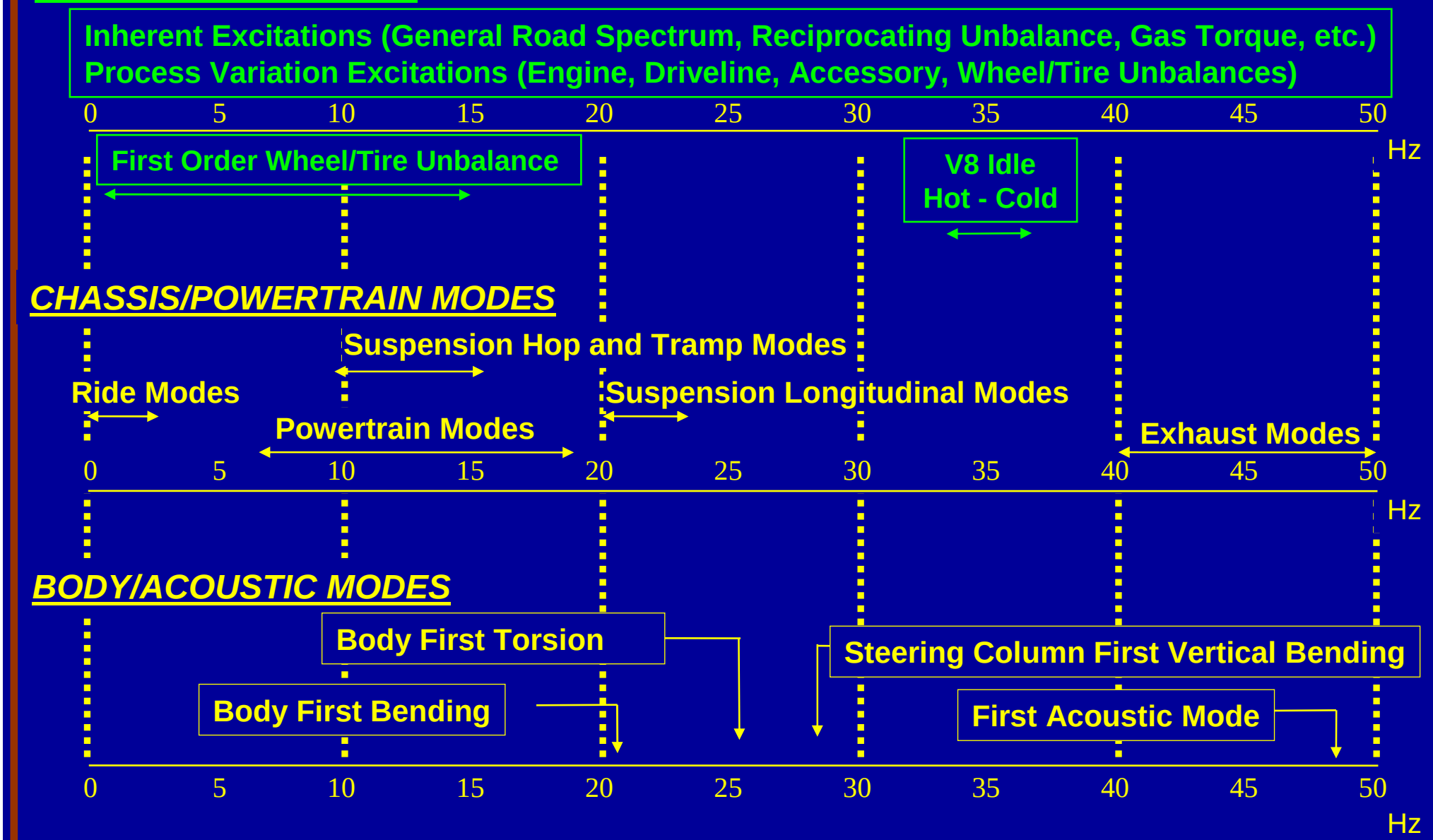
Mode Management Chart



(See Ref. 1)

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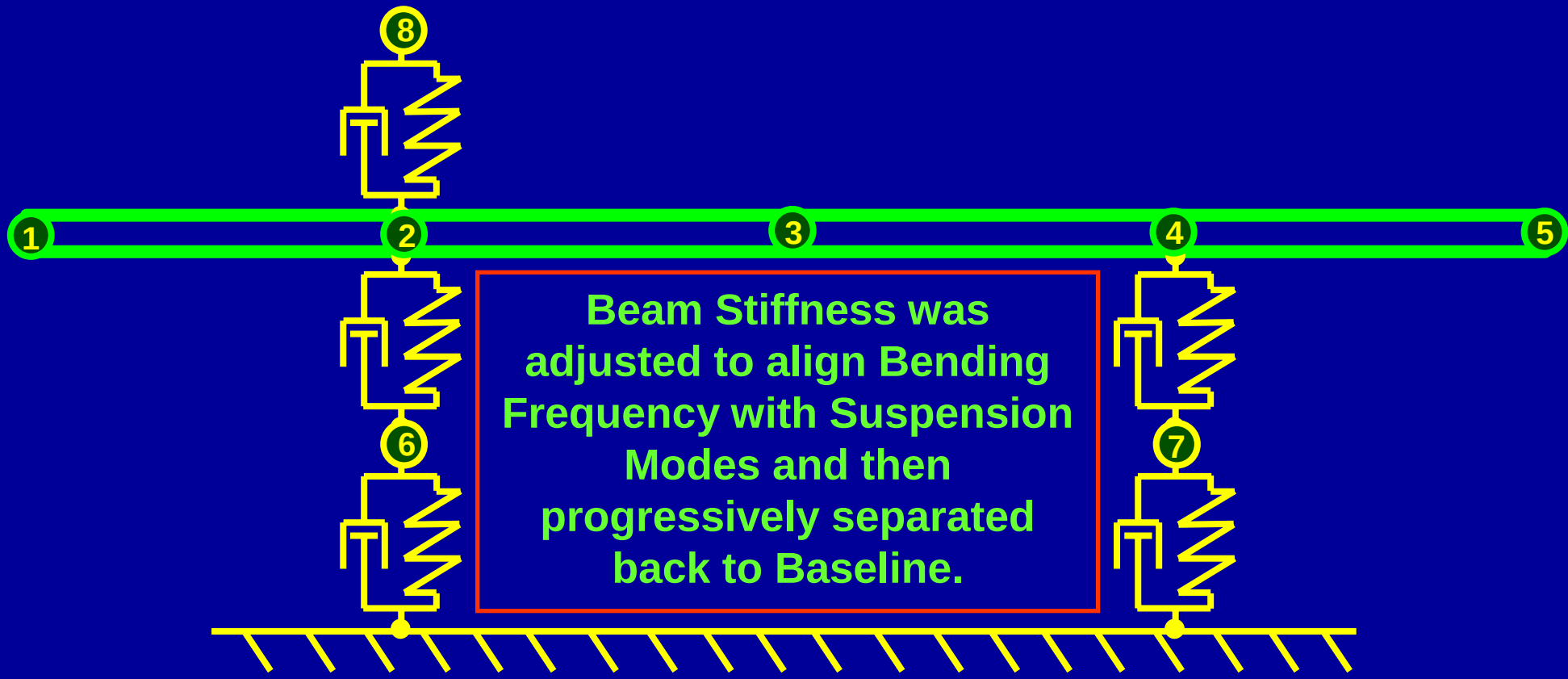
8 Degree of Freedom Vehicle NVH Model

Bending Mode Frequency Separation



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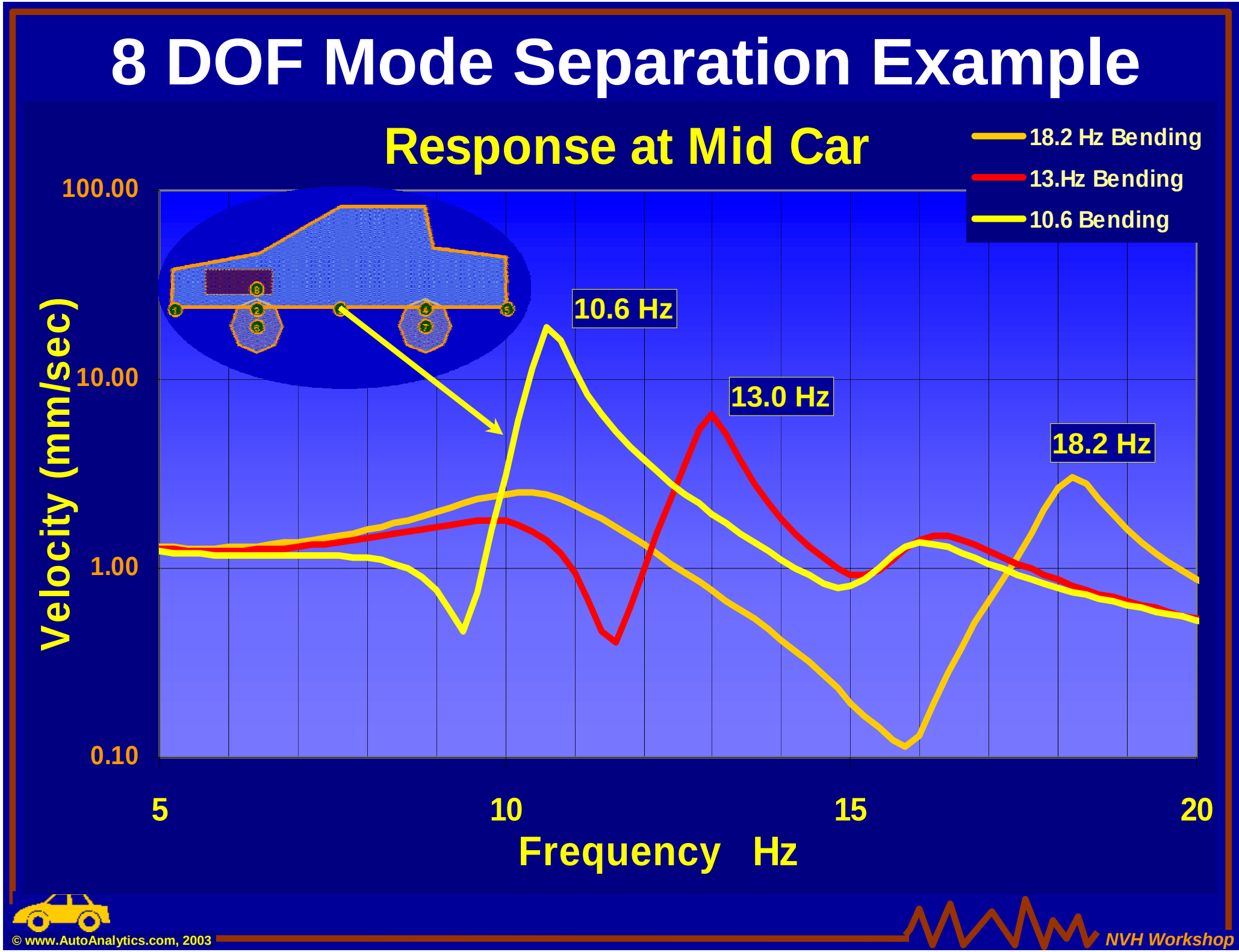
8 DOF Mode Separation Example



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Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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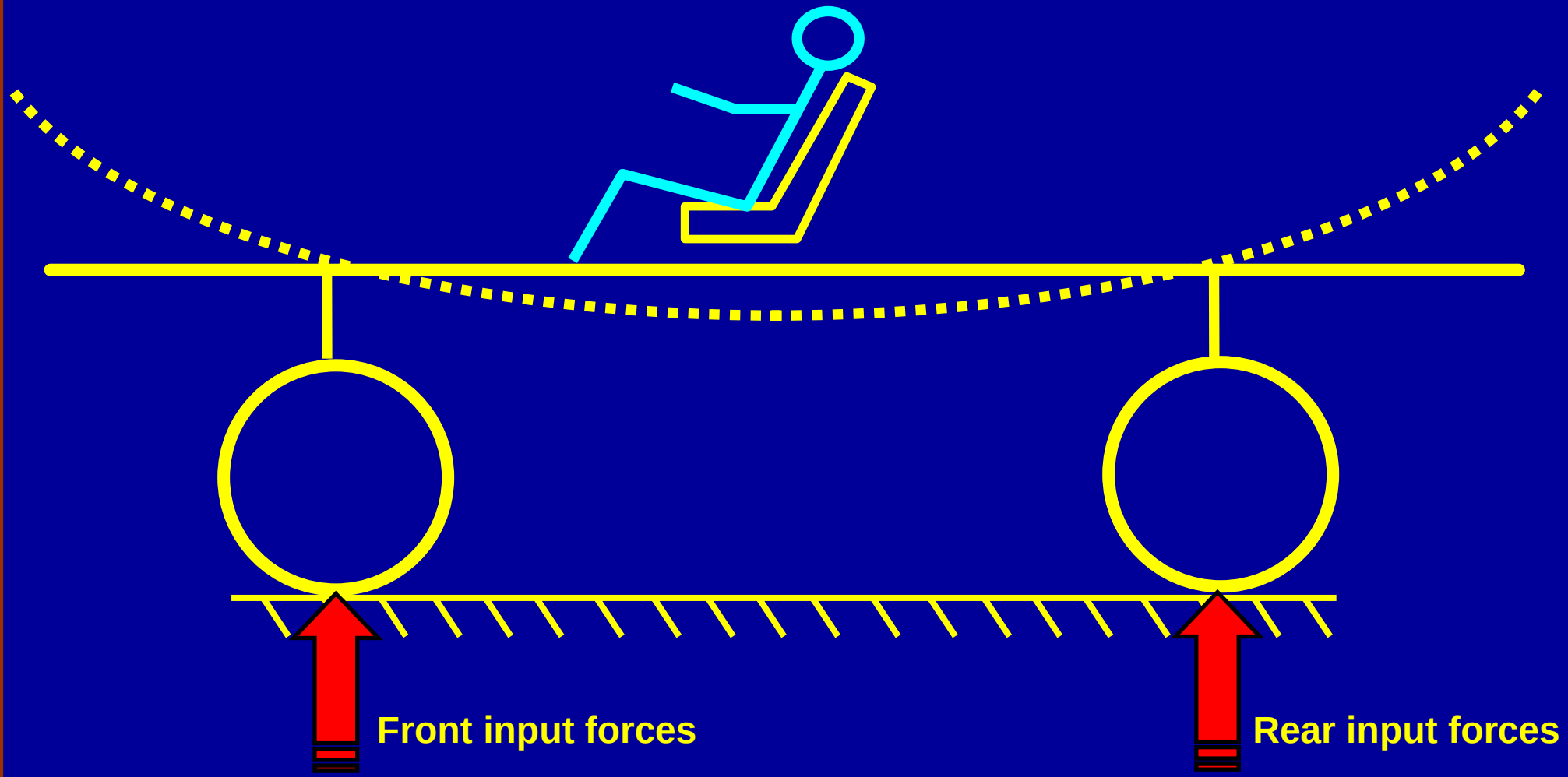
Mount at Nodal Point

First Bending: Nodal Point Mounting Example



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

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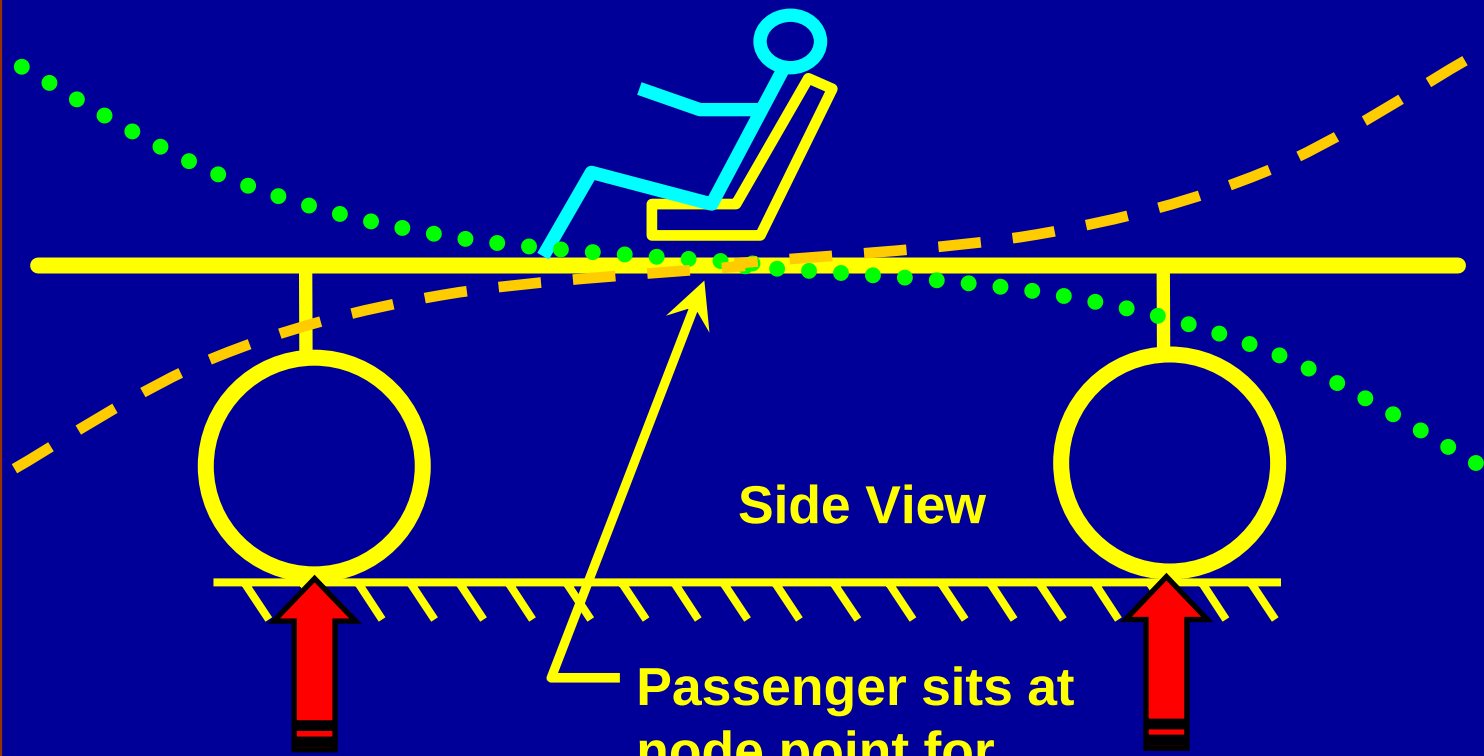
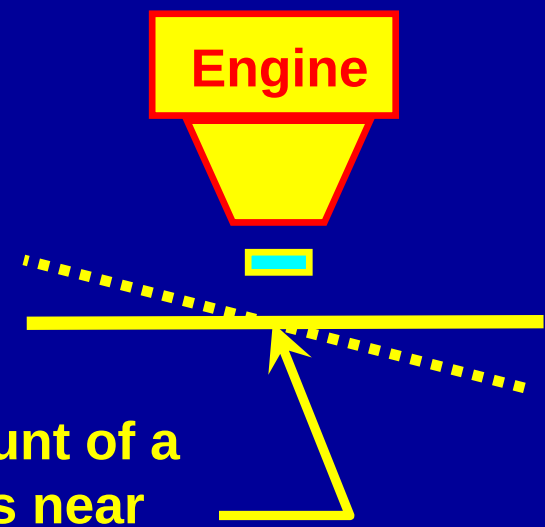
Mount at Nodal Point

First Torsion: Nodal Point Mounting Examples



Rear View



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

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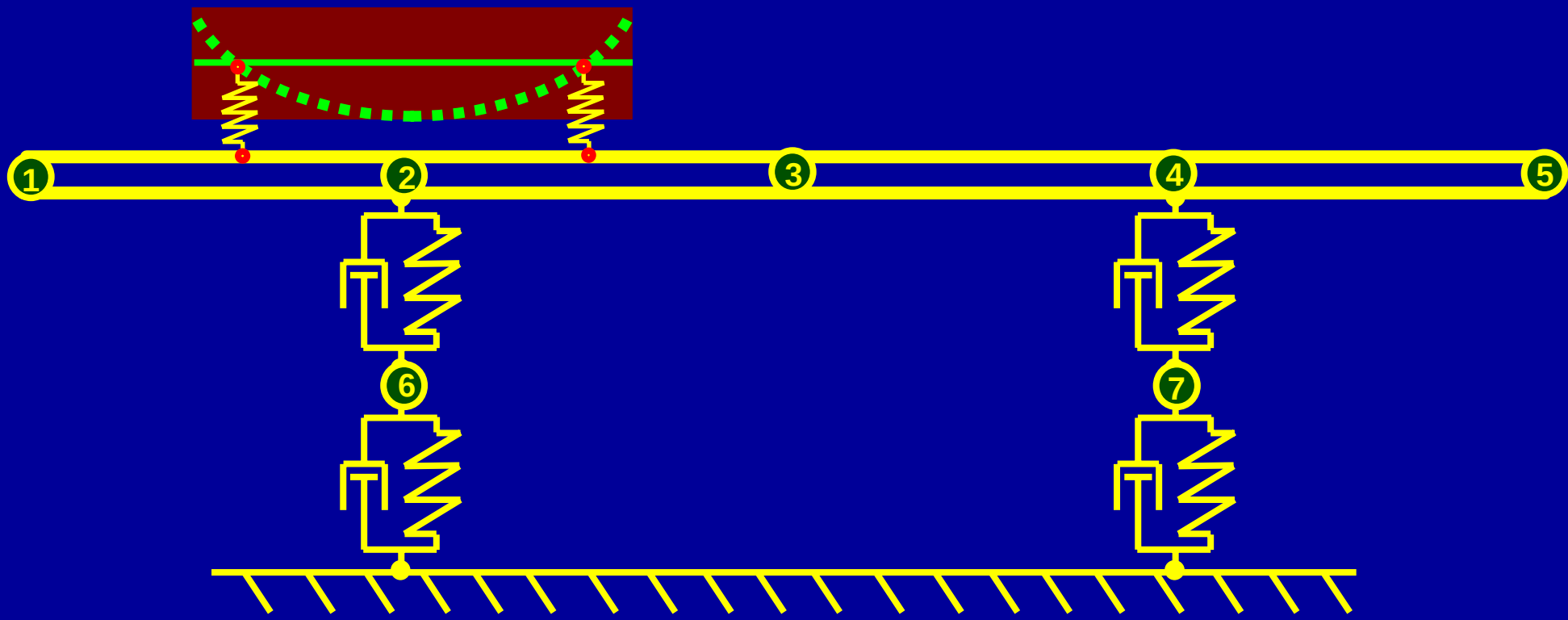
Powertrain Bending Mode Nodal Mounting



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

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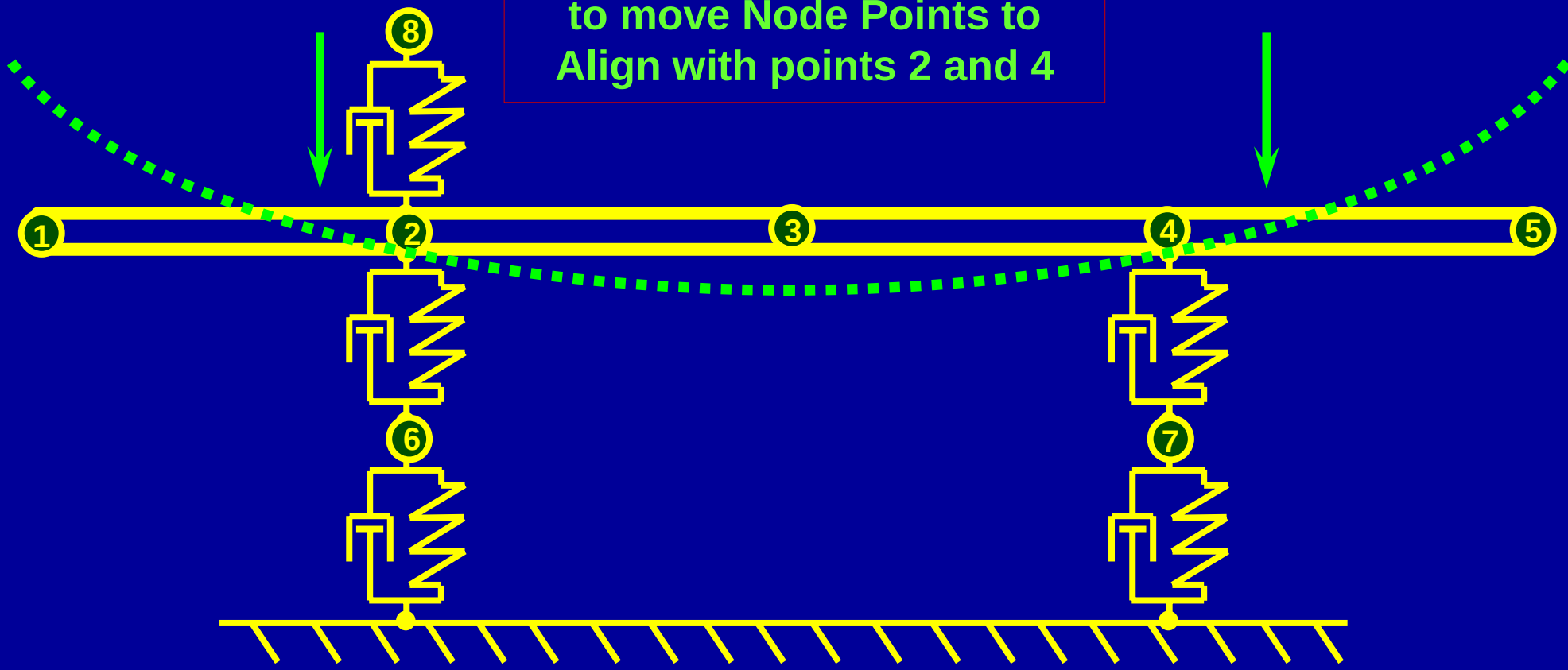
8 Degree of Freedom Vehicle NVH Model

Bending Node Alignment with Wheel Centers

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



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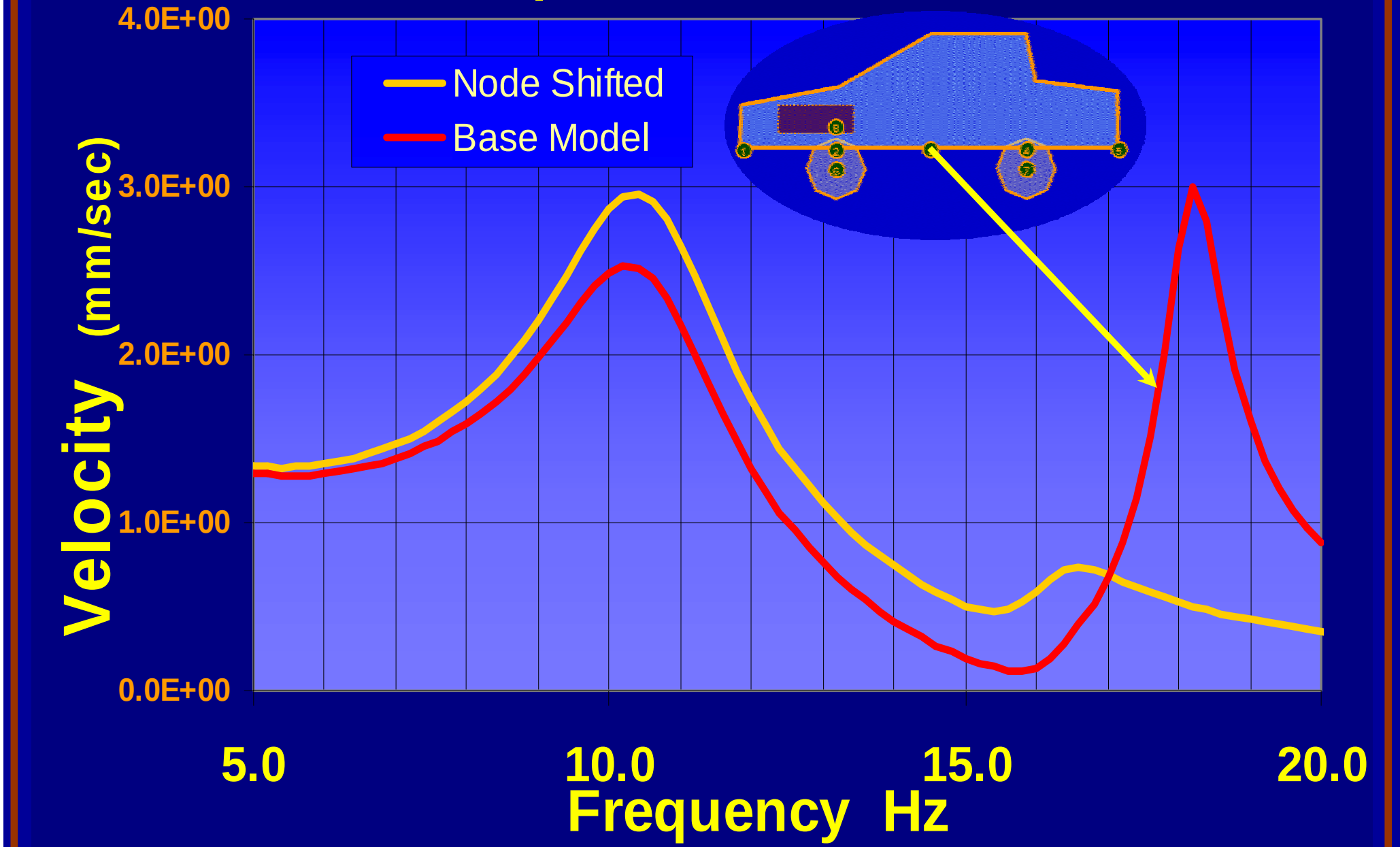
First Bending Nodal Point Alignment

Response at Mid-Car



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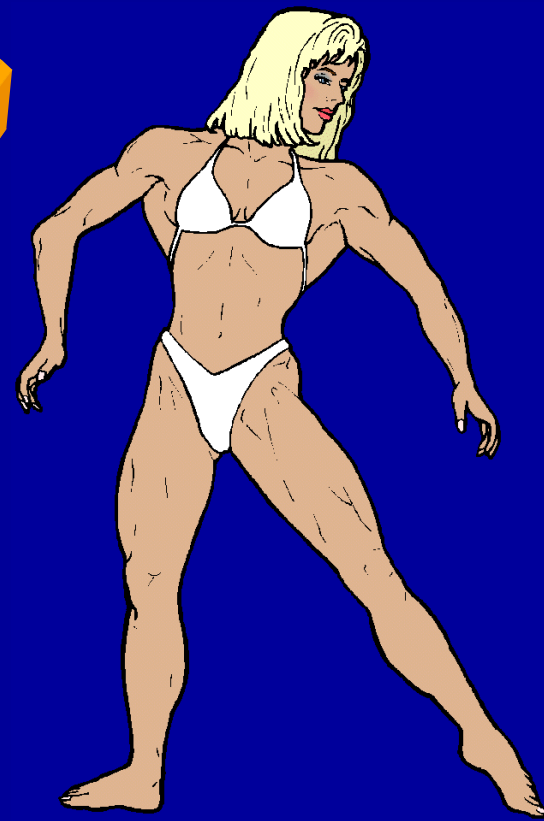
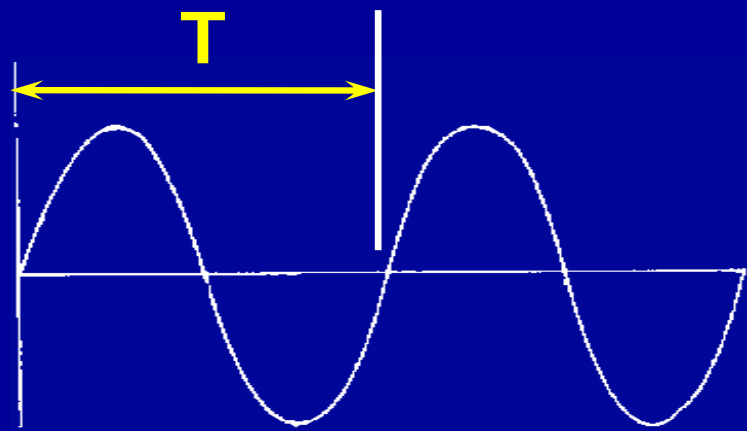
SHAPE



< is more important than >



Frequency = 1 / T

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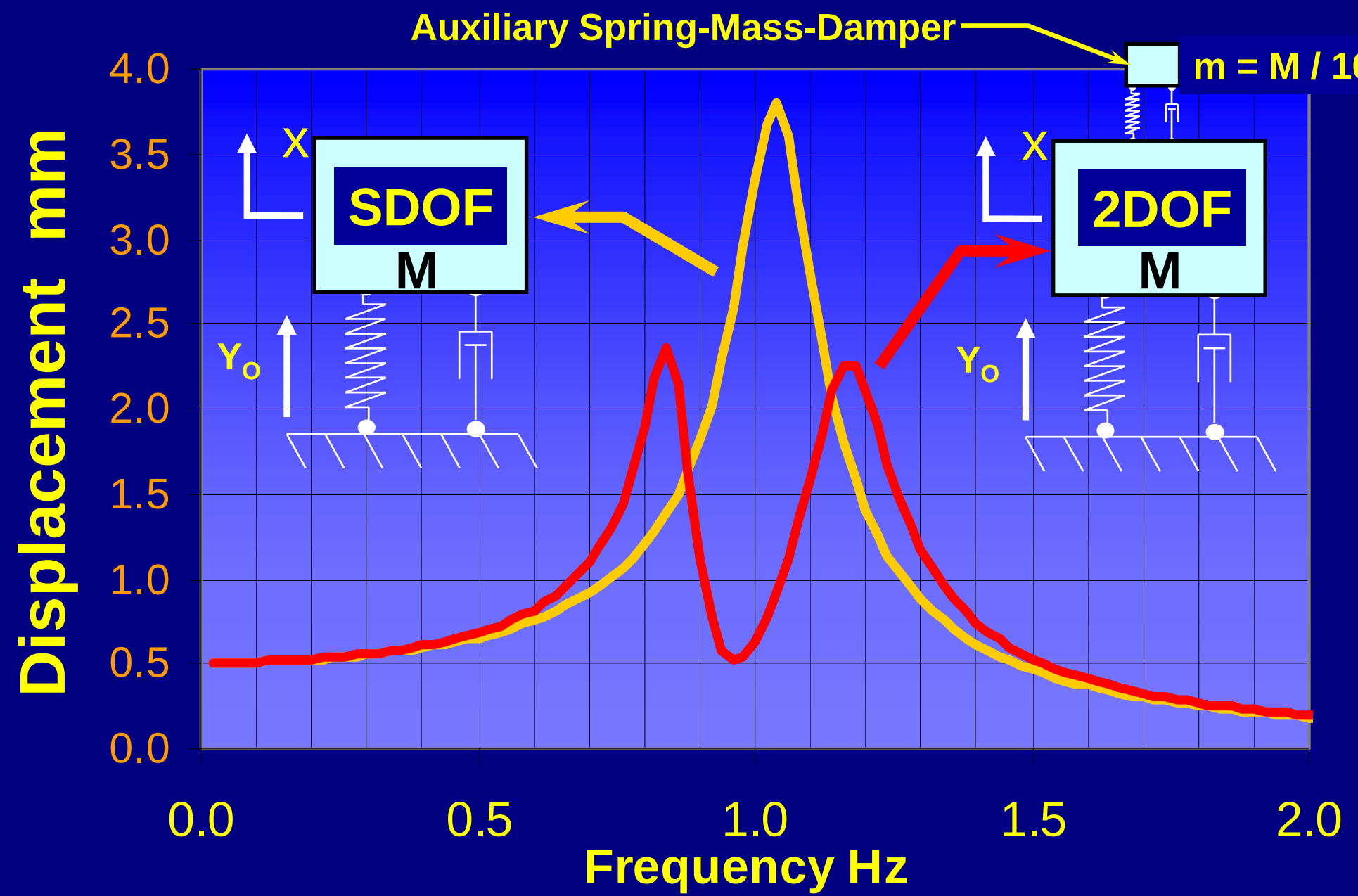
Dynamic Absorber Concept



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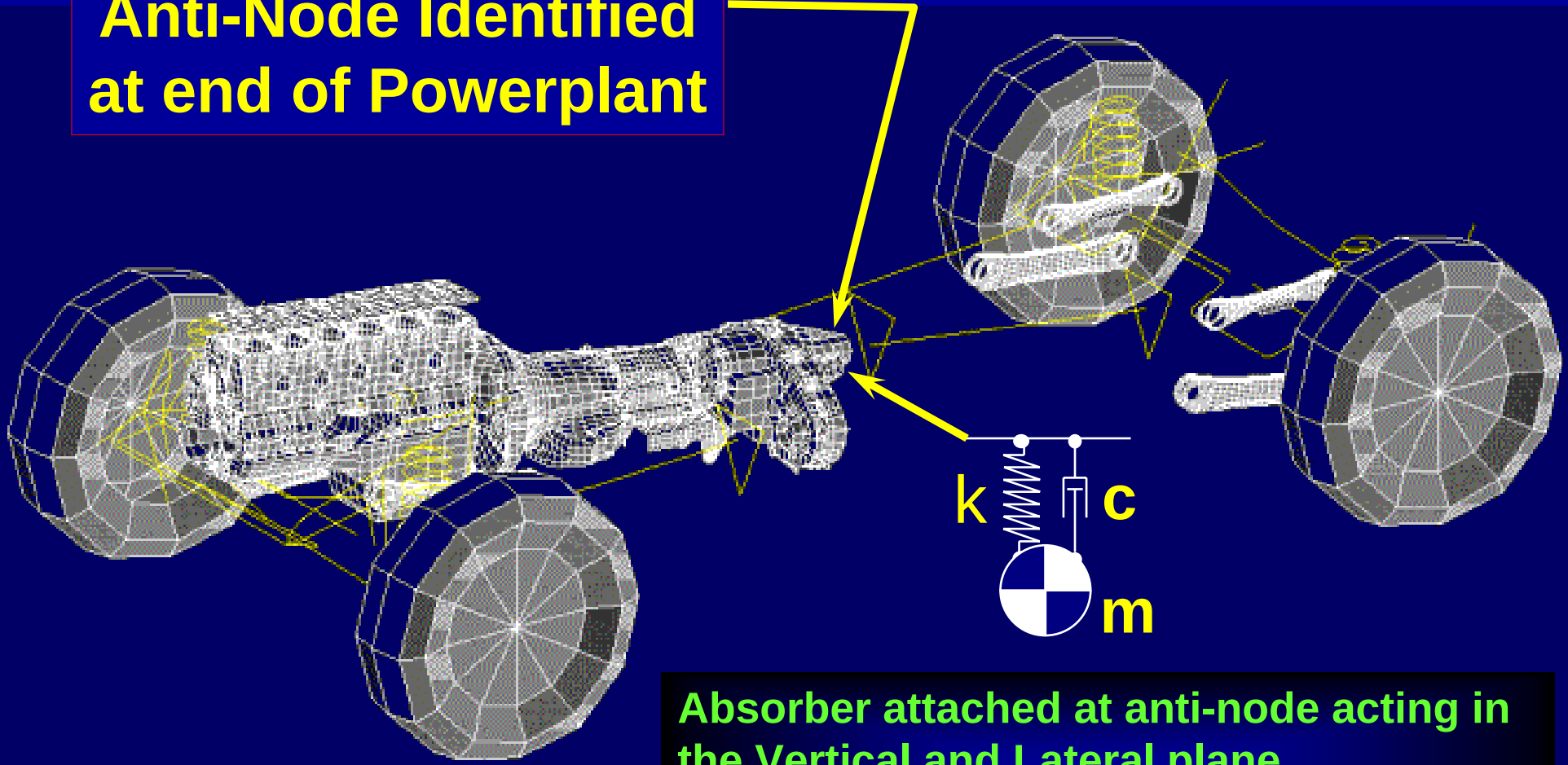
Powertrain Example of Dynamic Absorber

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]

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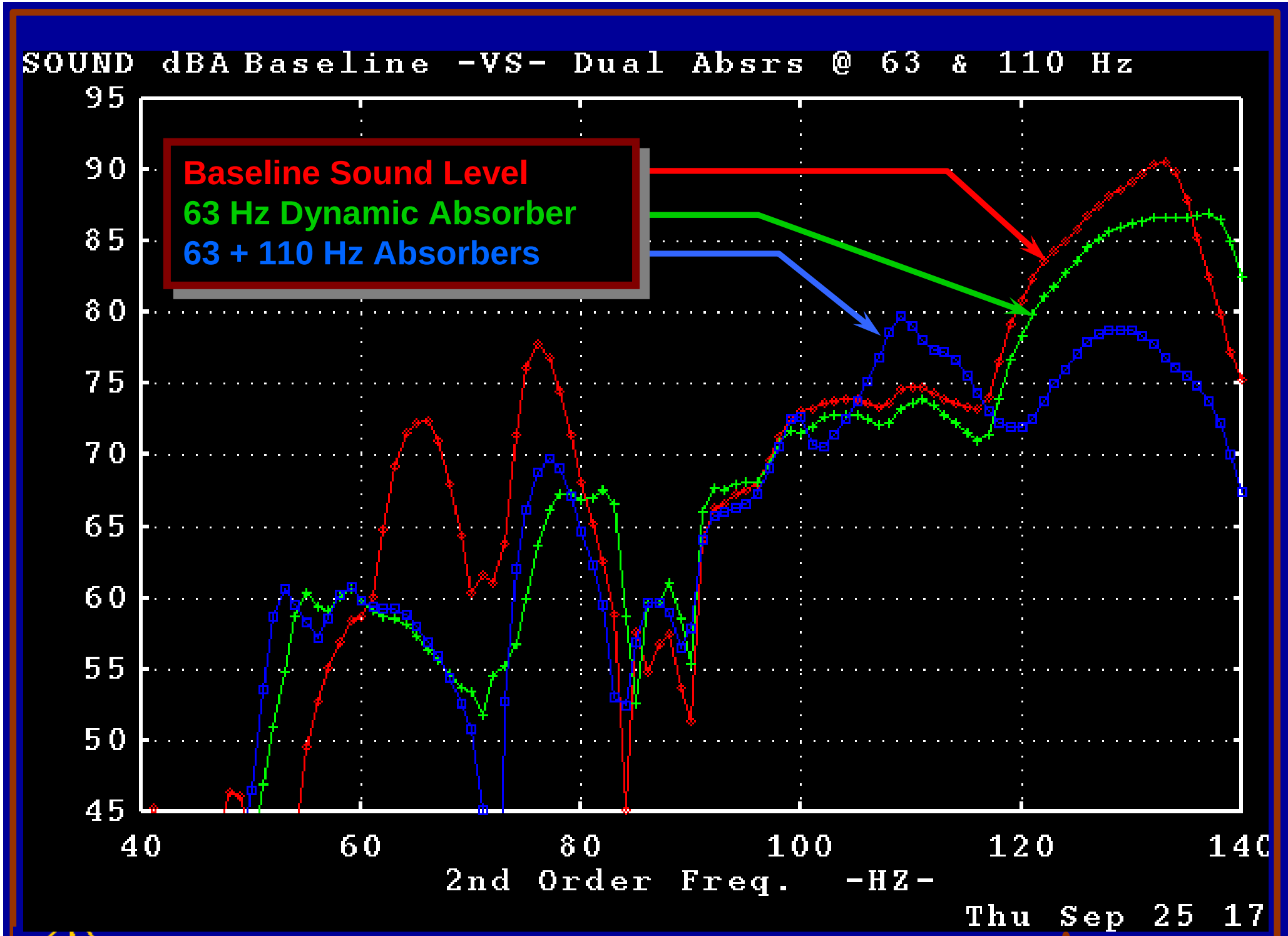

[Figure Courtesy of DaimlerChrysler Corporation]

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Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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NVH Workshop Topic Outline

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

Greg Goetchius

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NVH Workshop Topic Outline

- Introduction
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- NVH Load Conditions
- Low Frequency Basics
- *Live Noise Attenuation Demo*
- **Mid Frequency Basics** ***Sachin Gogate***
- Utilization of Simulation Models
- Closing Remarks

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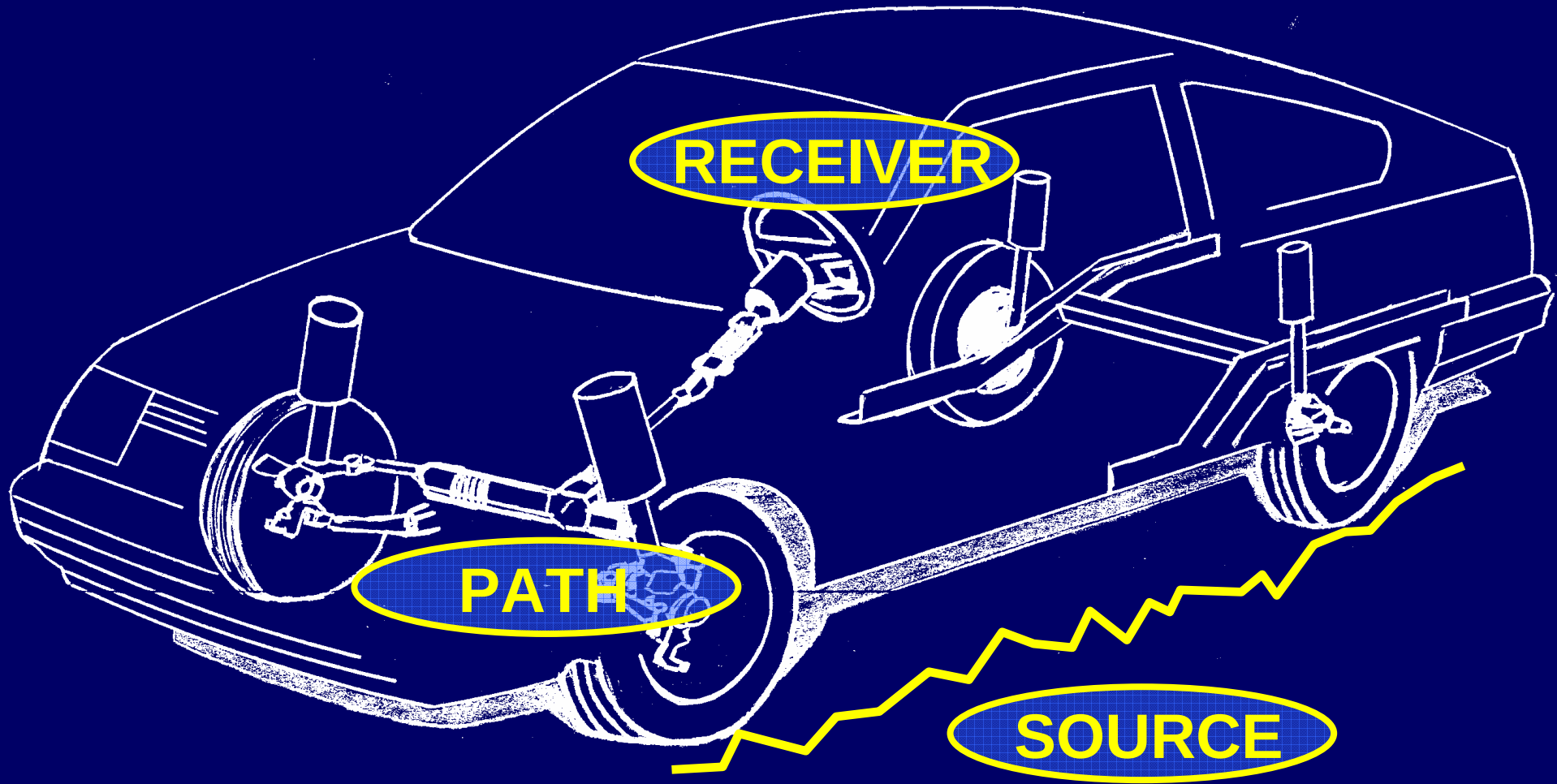
Mid Frequency NVH Fundamentals



This looks familiar!

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

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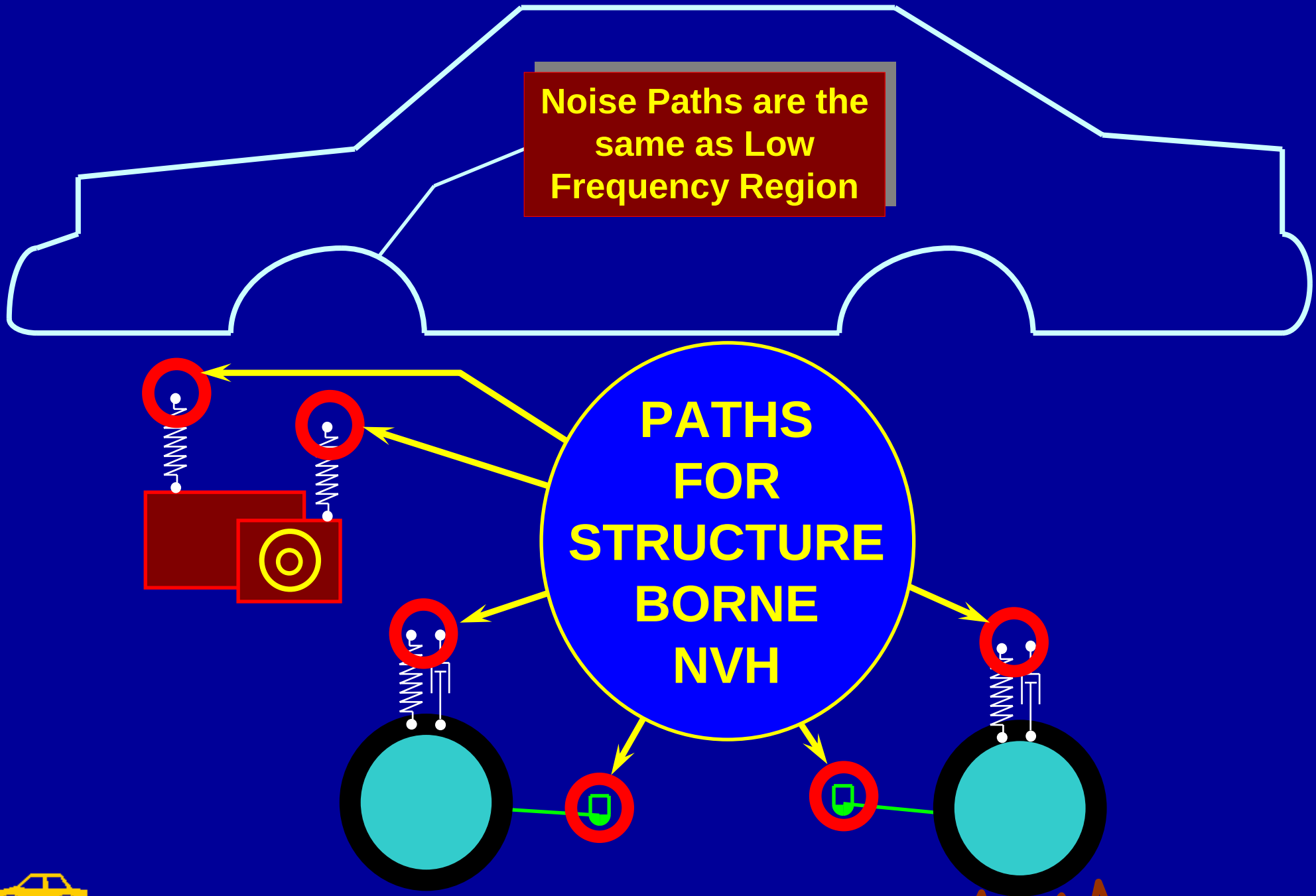
Typical NVH Pathways to the Passenger



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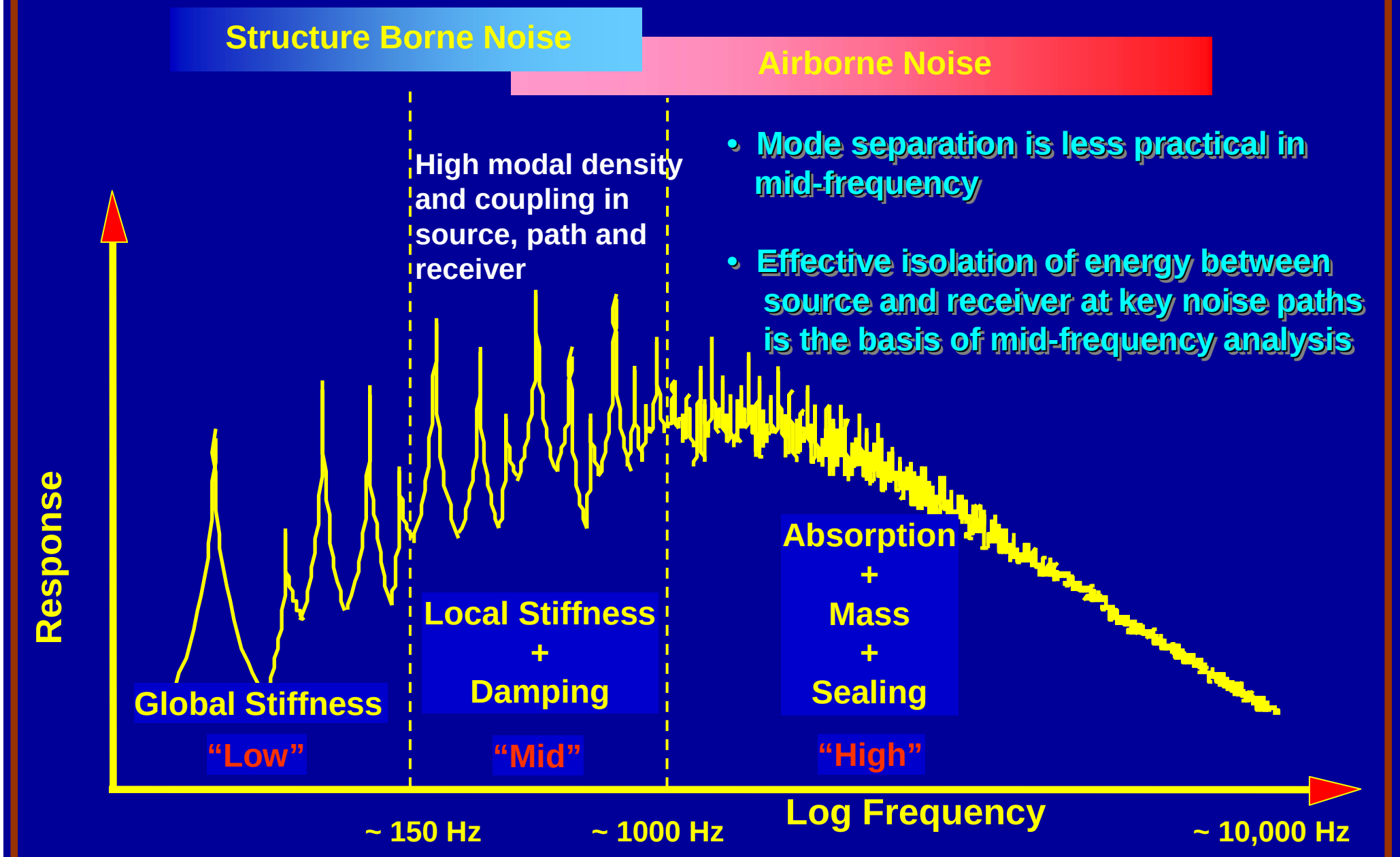
Mid-Frequency Analysis Character



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Mid-Frequency Analysis Character

- **Important characteristics of mid frequency analysis**

Effective Isolation

&

Identifying Key Noise Paths

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Mid-Frequency Analysis Character

- **Important characteristics of mid frequency analysis**

Effective Isolation

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Identifying Key Noise Paths

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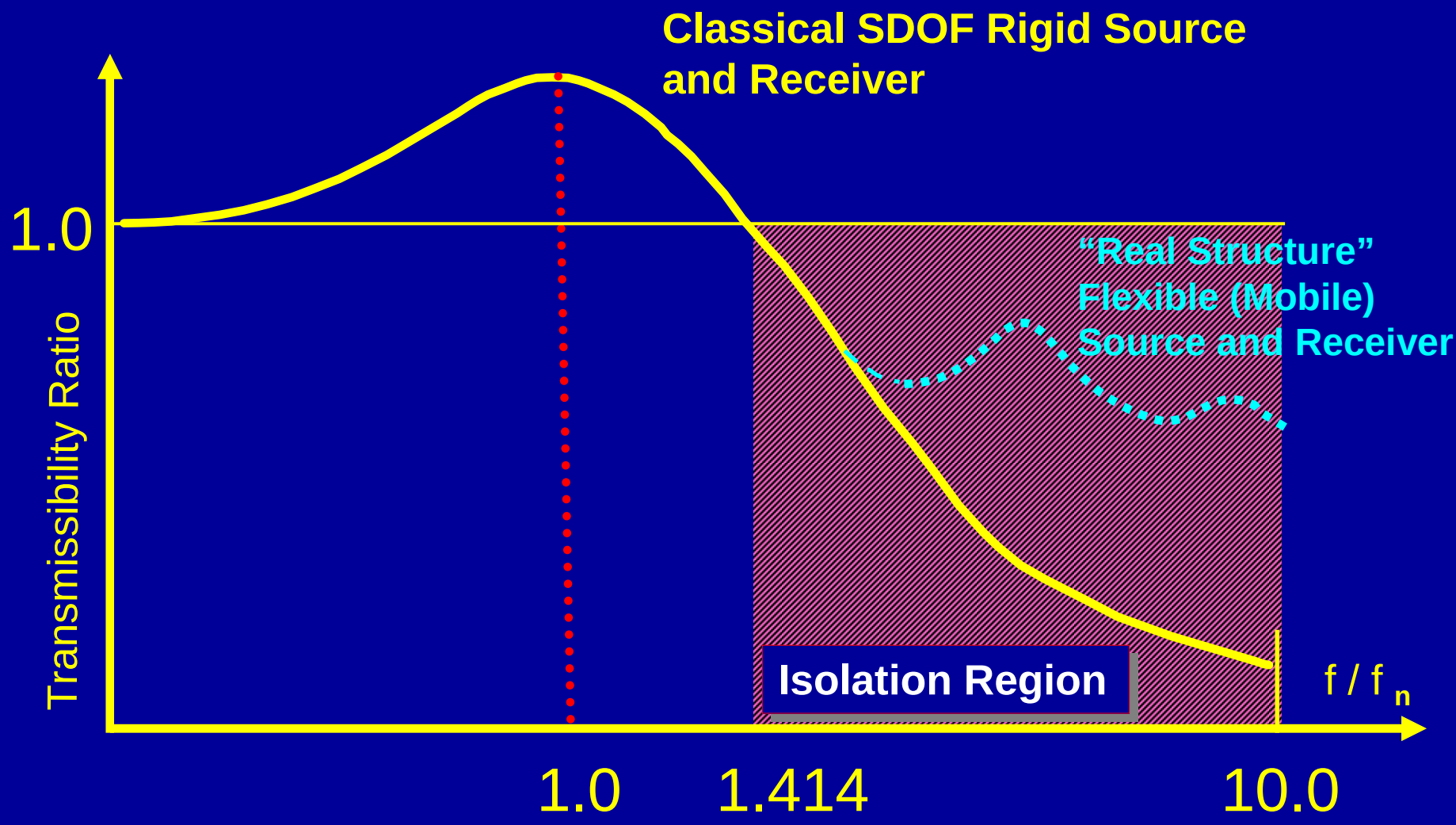
Isolation Effectiveness



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

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Mobility

- **Mobility is the ratio of velocity response at the excitation point on structure where point force is applied**

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

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

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

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

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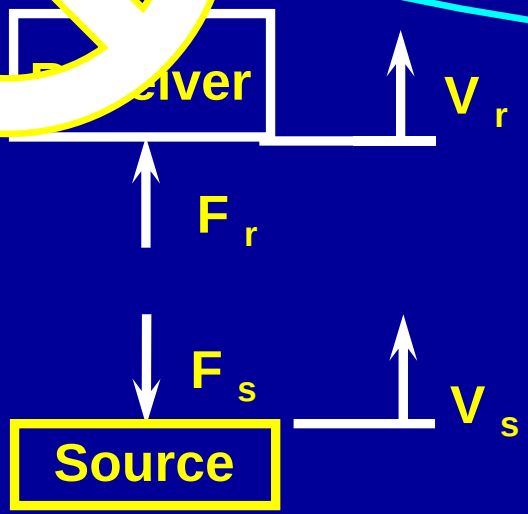
Isolation

- a theoretical model based on
rce

2003, 2005 ERRATA

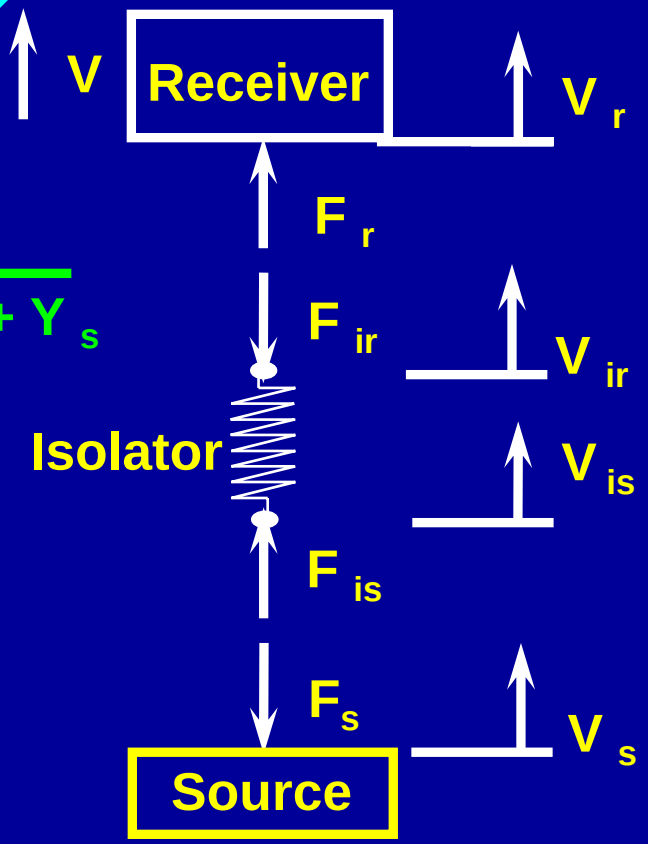
- Tr ility ratio is used to objectively define measure of isolation

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



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

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

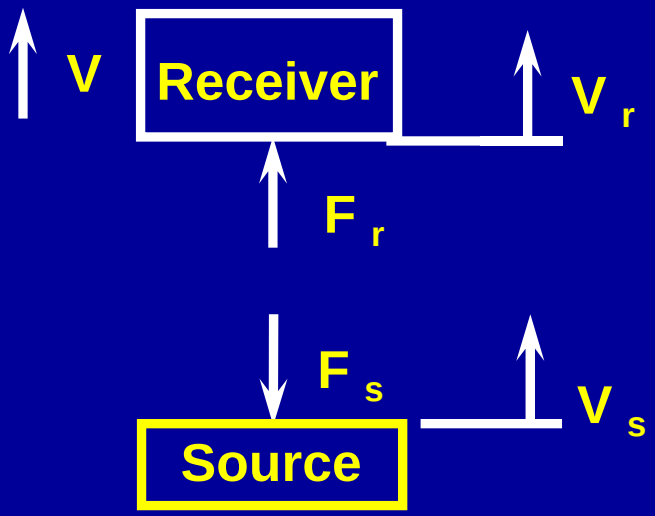


Isolation

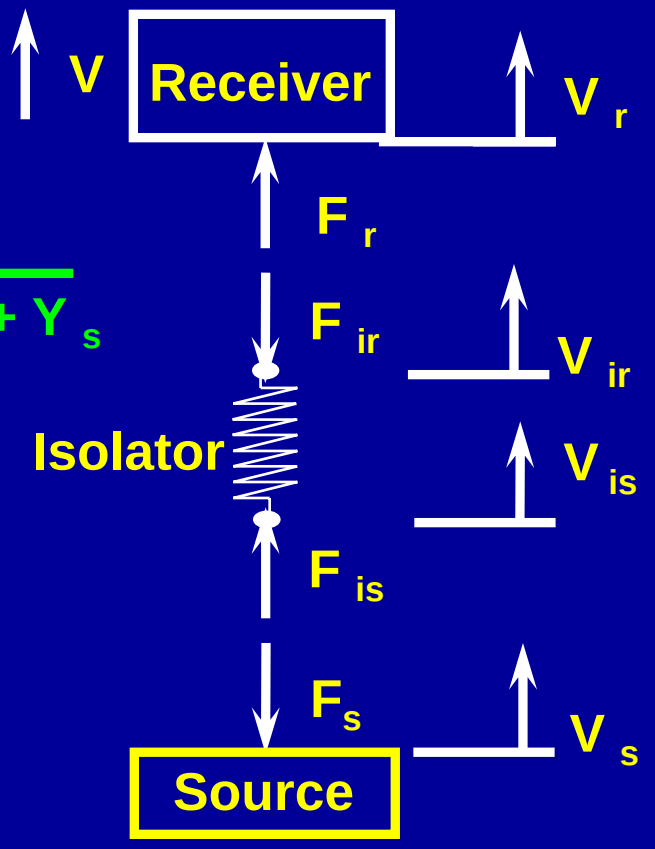
- The isolation effectiveness can be quantified by a theoretical model based on analysis of mobilities of receiver, isolator and source

- 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

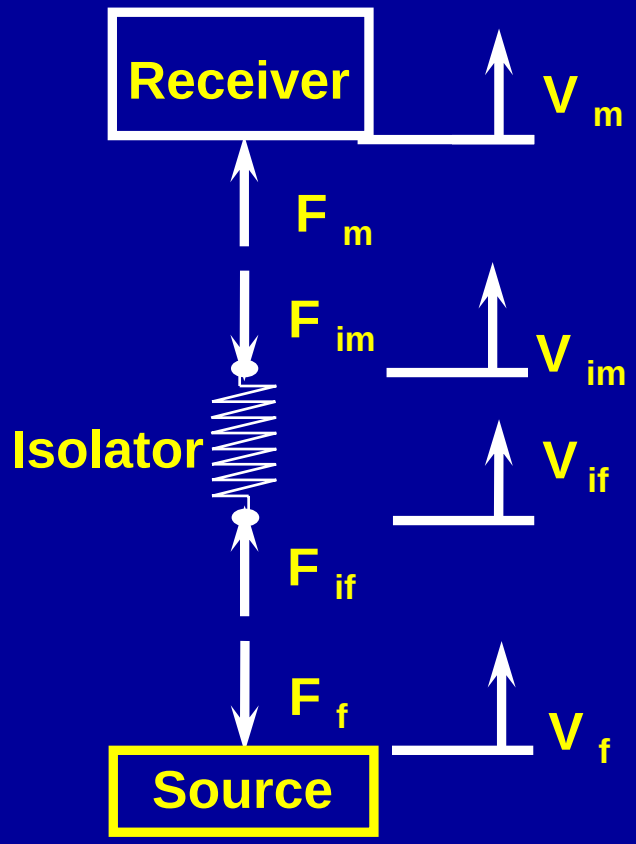
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

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

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

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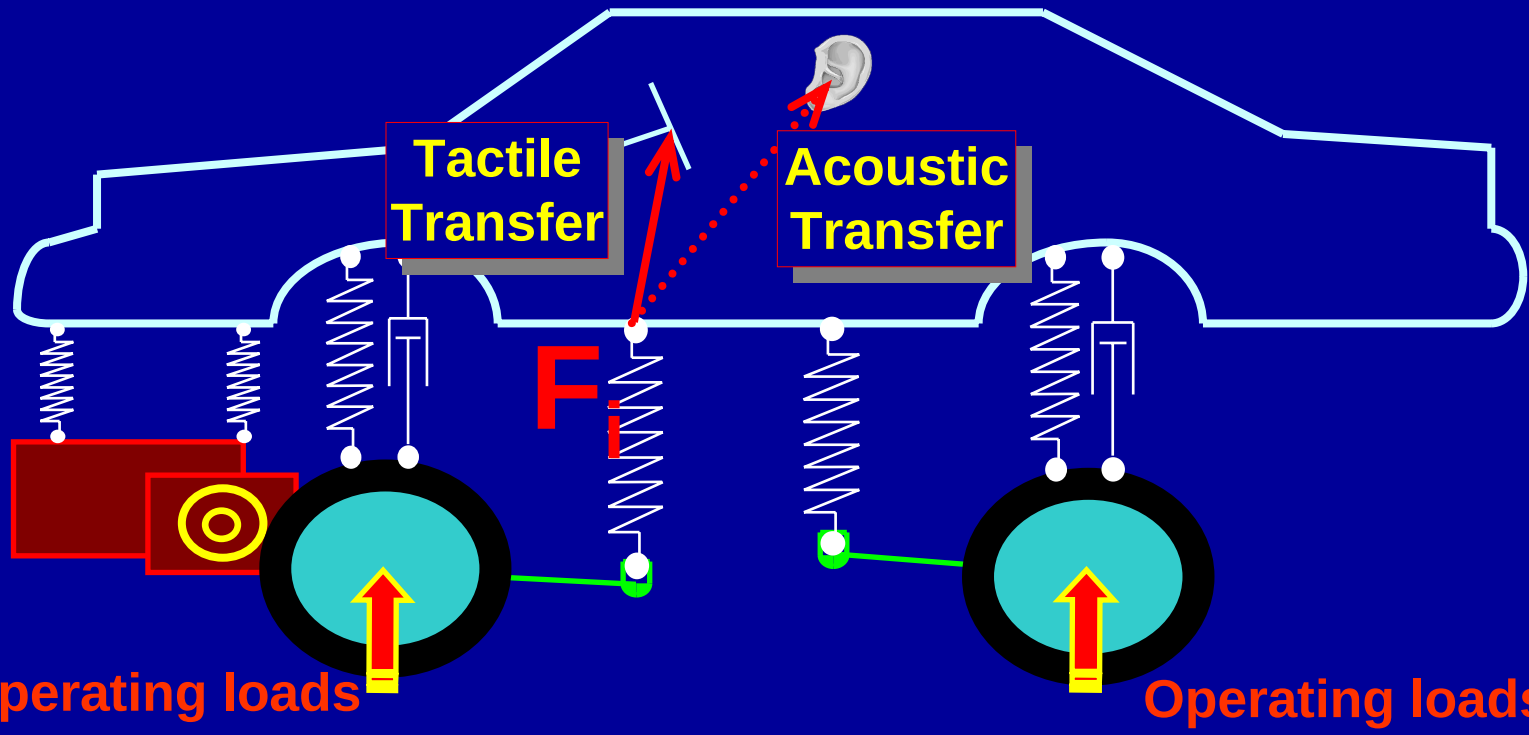
Identifying Key Noise Paths

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



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

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

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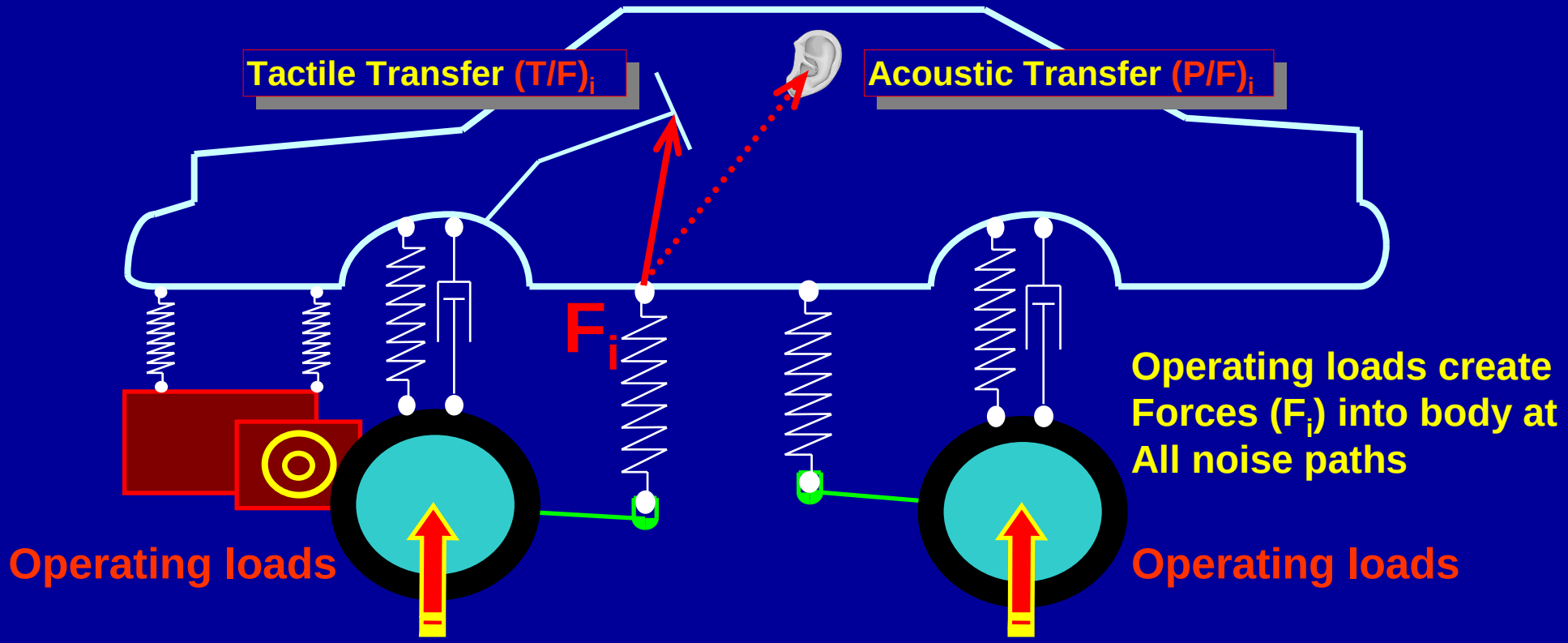
Transfer Path Analysis



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

$$R_t = \Sigma_{paths} [R_i] = \Sigma_{paths} [F_i * (R/F)_i]$$

R_i : Partial contribution of path i due to operating force

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

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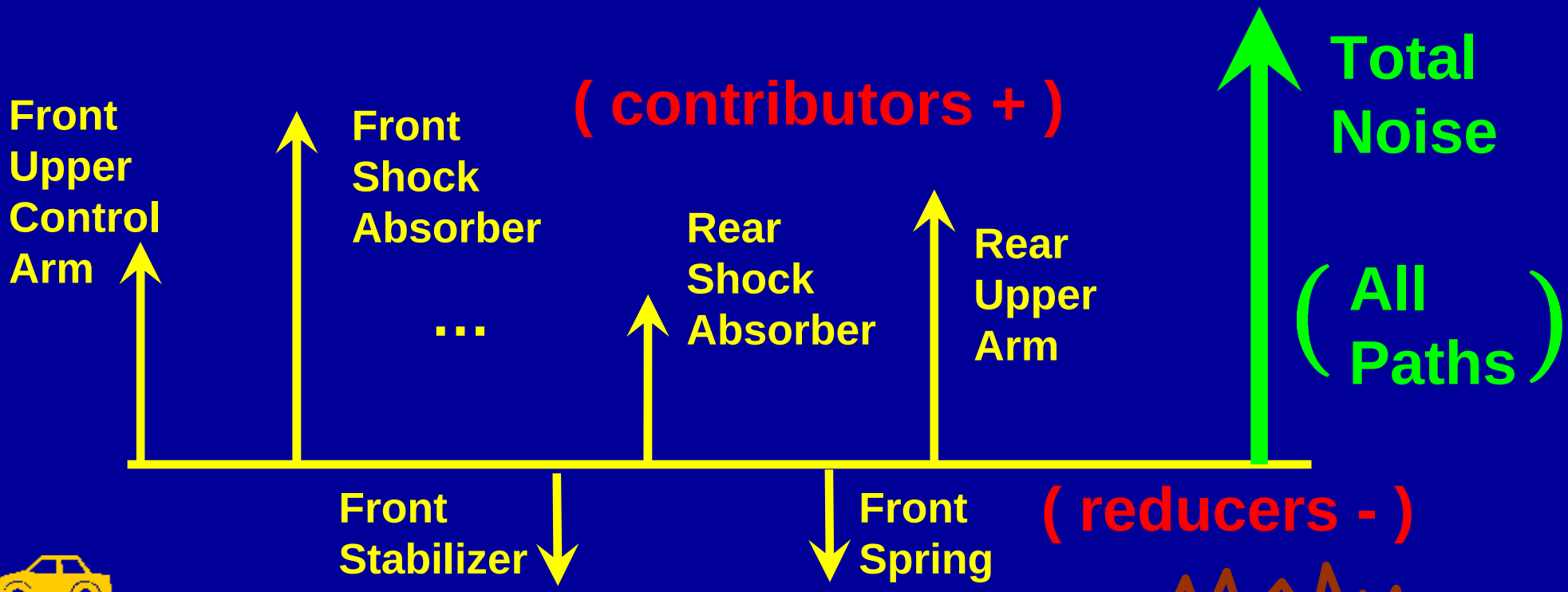
Transfer Path Analysis

- 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



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Designing for Mid Frequency

- **Important characteristics of mid frequency analysis**

Effective Isolation

&

Identifying Key Noise Paths

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Designing for Mid Frequency

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

What are these generic targets and key parameters ?

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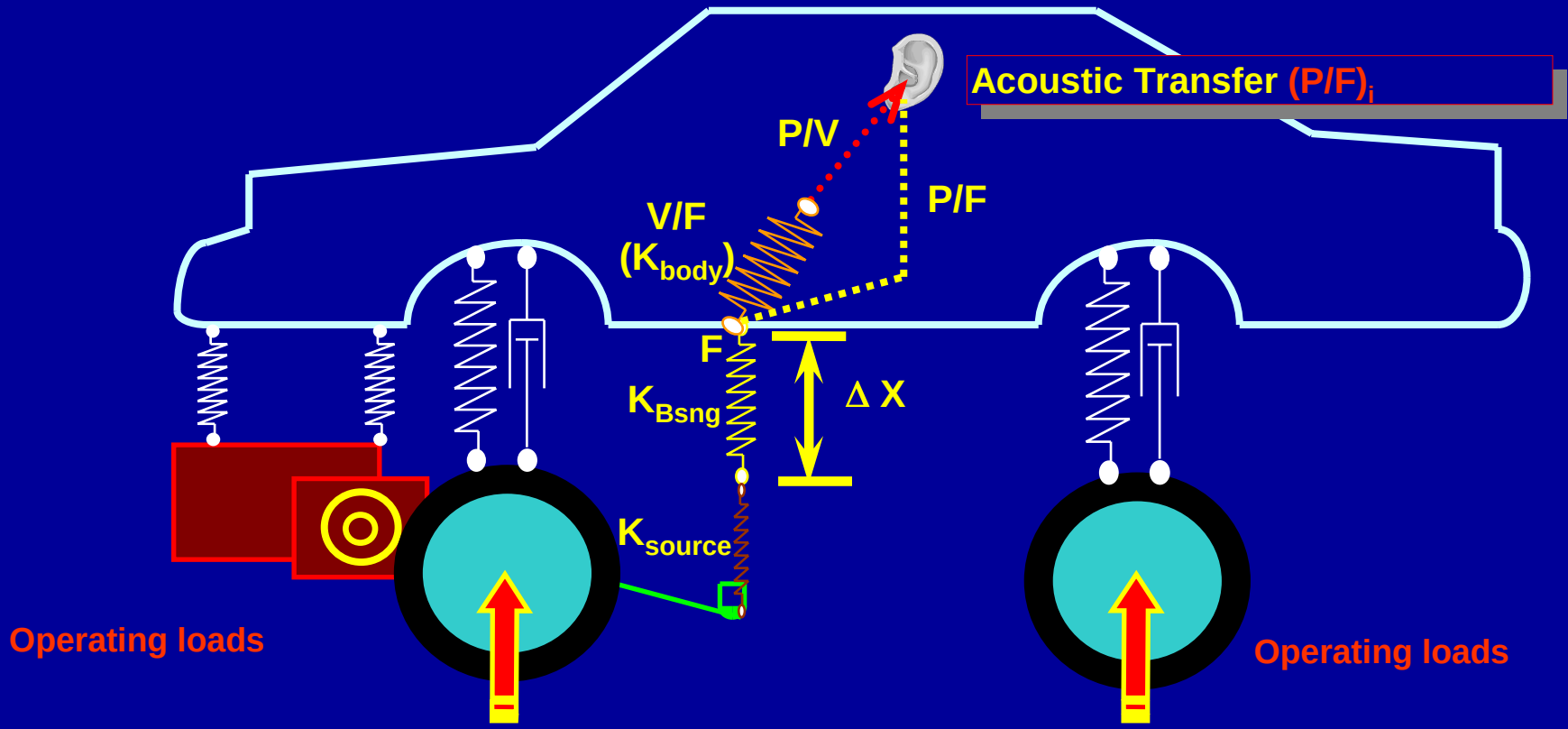
Generic Noise Path Targets



Transmissibility along a given noise path (TR_i)

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

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

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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

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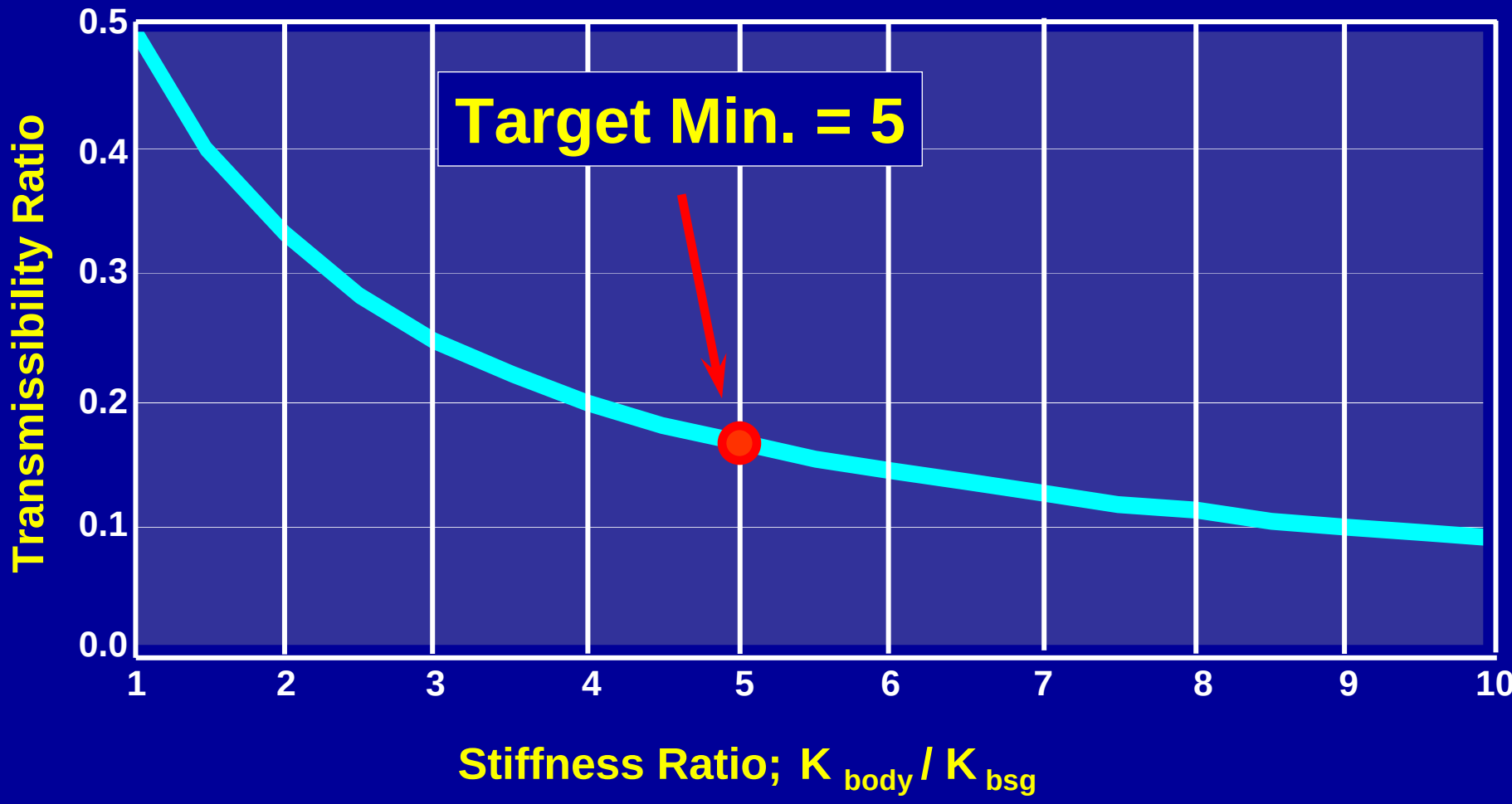
Relationship of Body-to-Bushing Stiffness ratio to Transmissibility

For infinitely stiff (immobile) source

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



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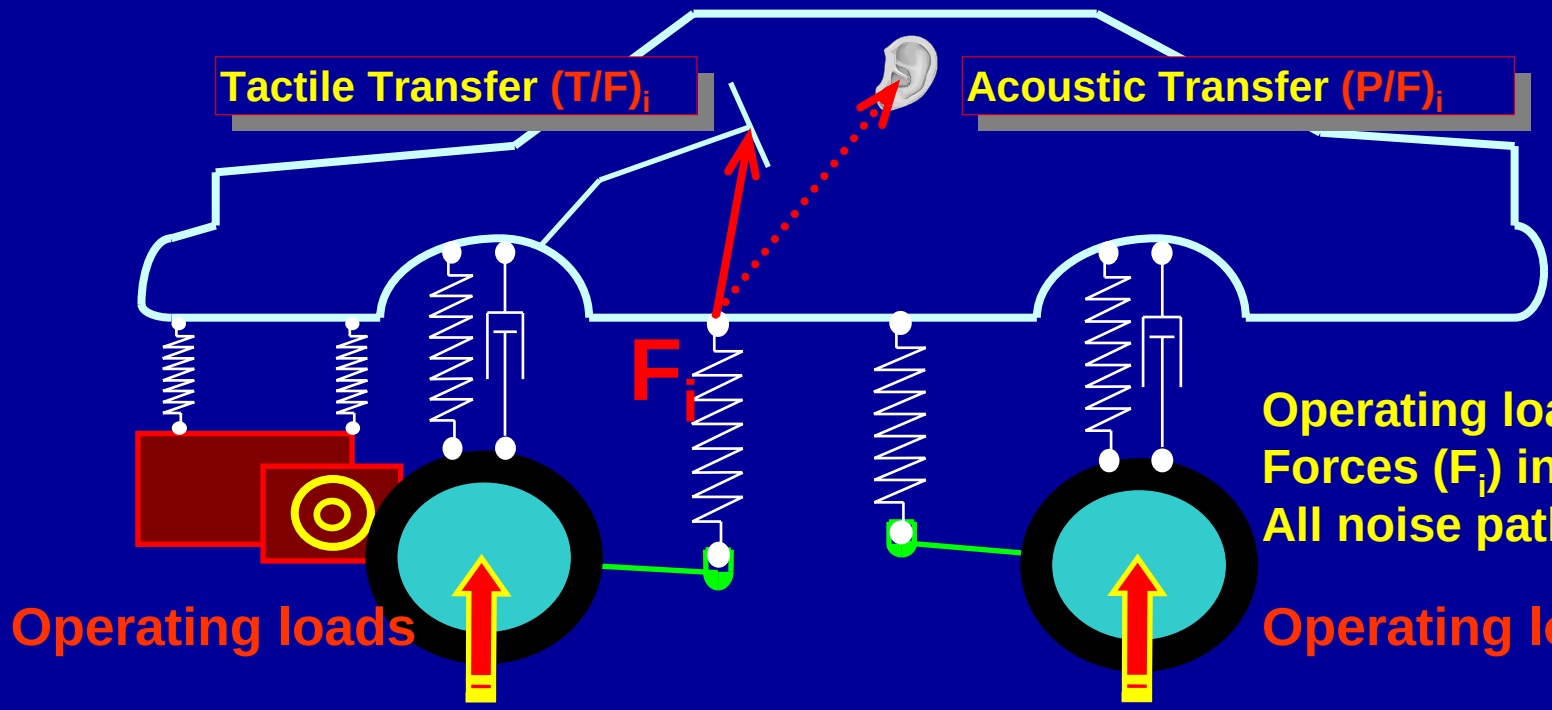
Generic Noise Path Targets



$$R_t = \Sigma_{paths} [R_i] = \Sigma_{paths} [F_i * (R/F)_i]$$
$$= \Sigma_{paths} [F_i * (R/V)_i * (V/F)_i]$$

Response R_t could be either Tactile or Acoustic Response

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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 suspension attachment to body, lowering sensitivities (P/F) or (V/F) along a path would reduce total response
- As a generic target,
  - Acoustic sensitivity (P/F) in 55-60 dBL/N range
  - Structural Mobility (V/F) less than 0.312 mm/sec/N

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Generic Noise Path Targets

How does one achieve these generic targets :

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

$$\frac{K_{source}}{K_{iso}} \sim \text{infinite}$$

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

$$\text{Structural Mobility} < 0.312\ \text{mm/sec/N}$$

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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**

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Generic Noise Path Targets

How does one achieve

$$\frac{K_{source}}{K_{iso}} \sim \text{infinite}$$

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

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

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Source Excitation Characteristics

Other means of reducing source input to paths

- Add lumped mass on component
- Use tuned absorber

} Applicable when Rigid Body Resonance of Source Side Component are present

- Bring nodal points toward path

} Applicable when flexible modes of source side component are present

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Generic Noise Path Targets

How does one achieve

$$55 < \text{Acoustic Mobility} < 60$$

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

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

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

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Generic Noise Path Targets

How does one achieve

$$55 < \text{Acoustic Mobility} < 60$$

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

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

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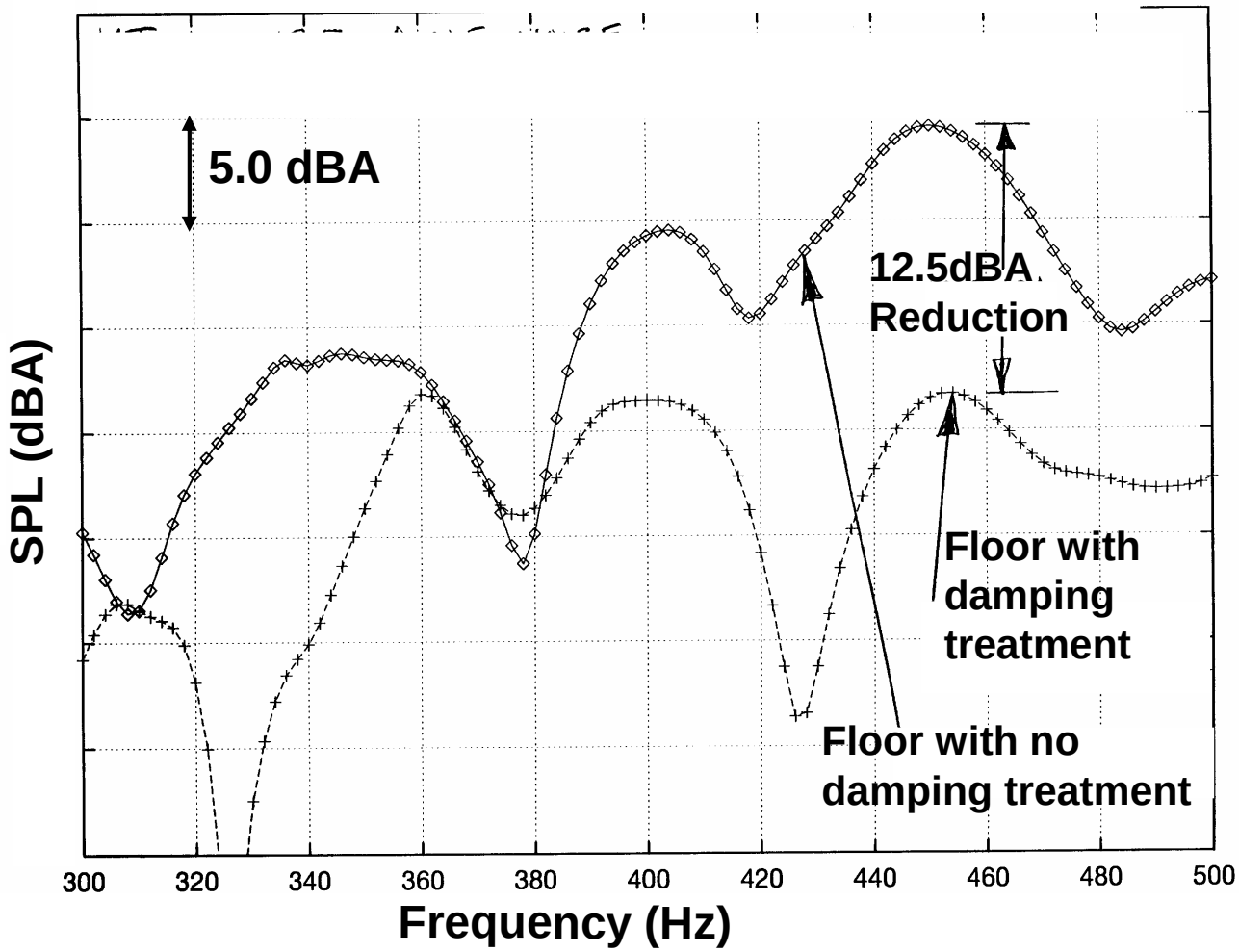
Application of Damping Treatment

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



[Figure Courtesy of DaimlerChrysler Corporation]

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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 !!

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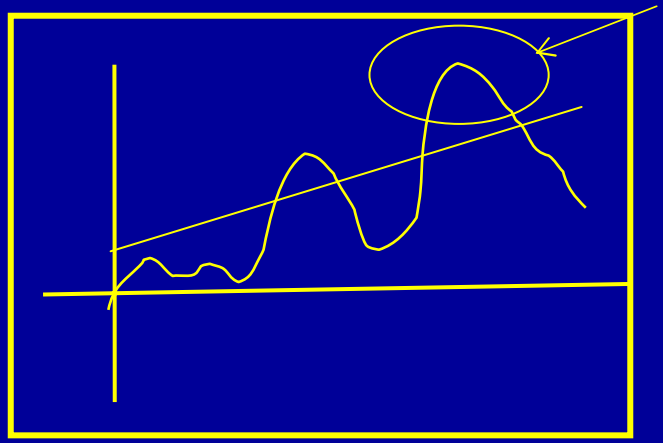
Designing for Mid Frequency

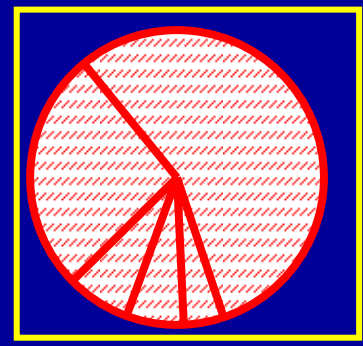
Driver's Ear Noise



Vehicle Level Response

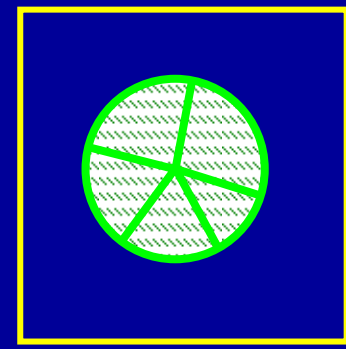
Transfer Path Analysis

Original Noise Path Contributions



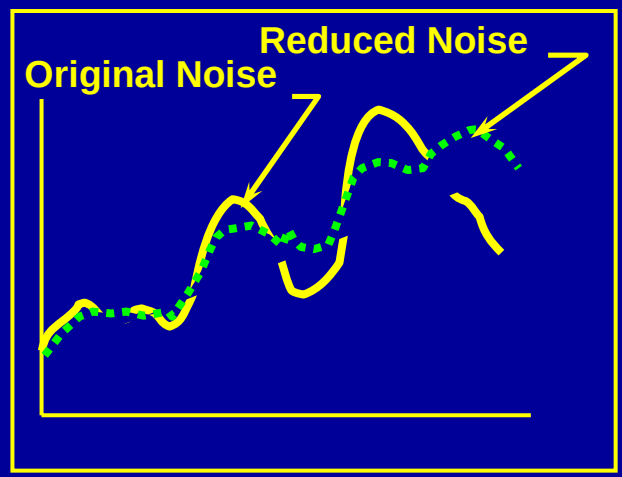
Some noise paths are more dominant than others

Original 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

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Designing for Mid Frequency

Principles to follow

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

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

- Perform path “rebalancing” to arrive at revised path targets (more strict for dominant paths) for all NVH operating load conditions.

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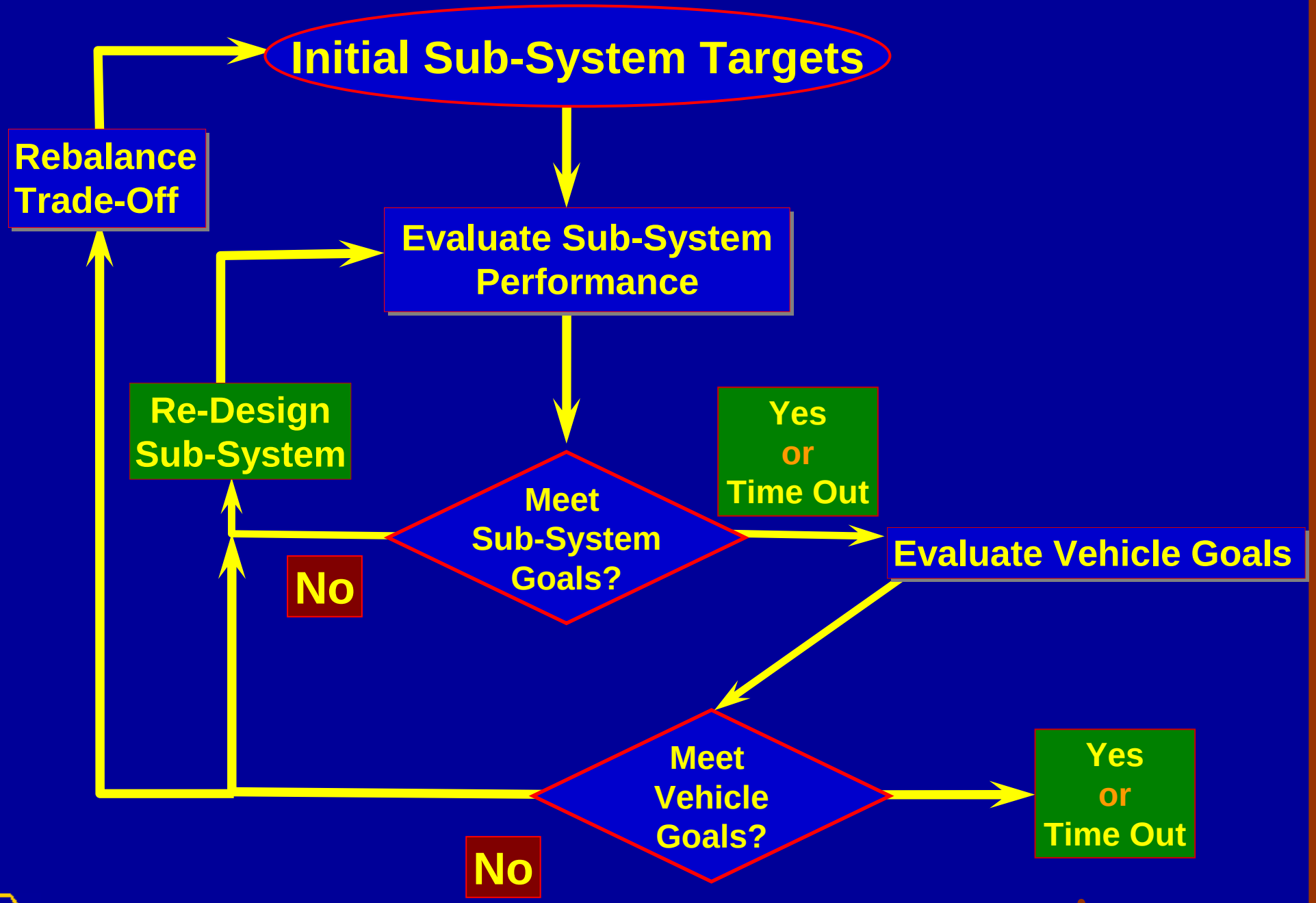
Mid Frequency NVH Goal Achievement Process



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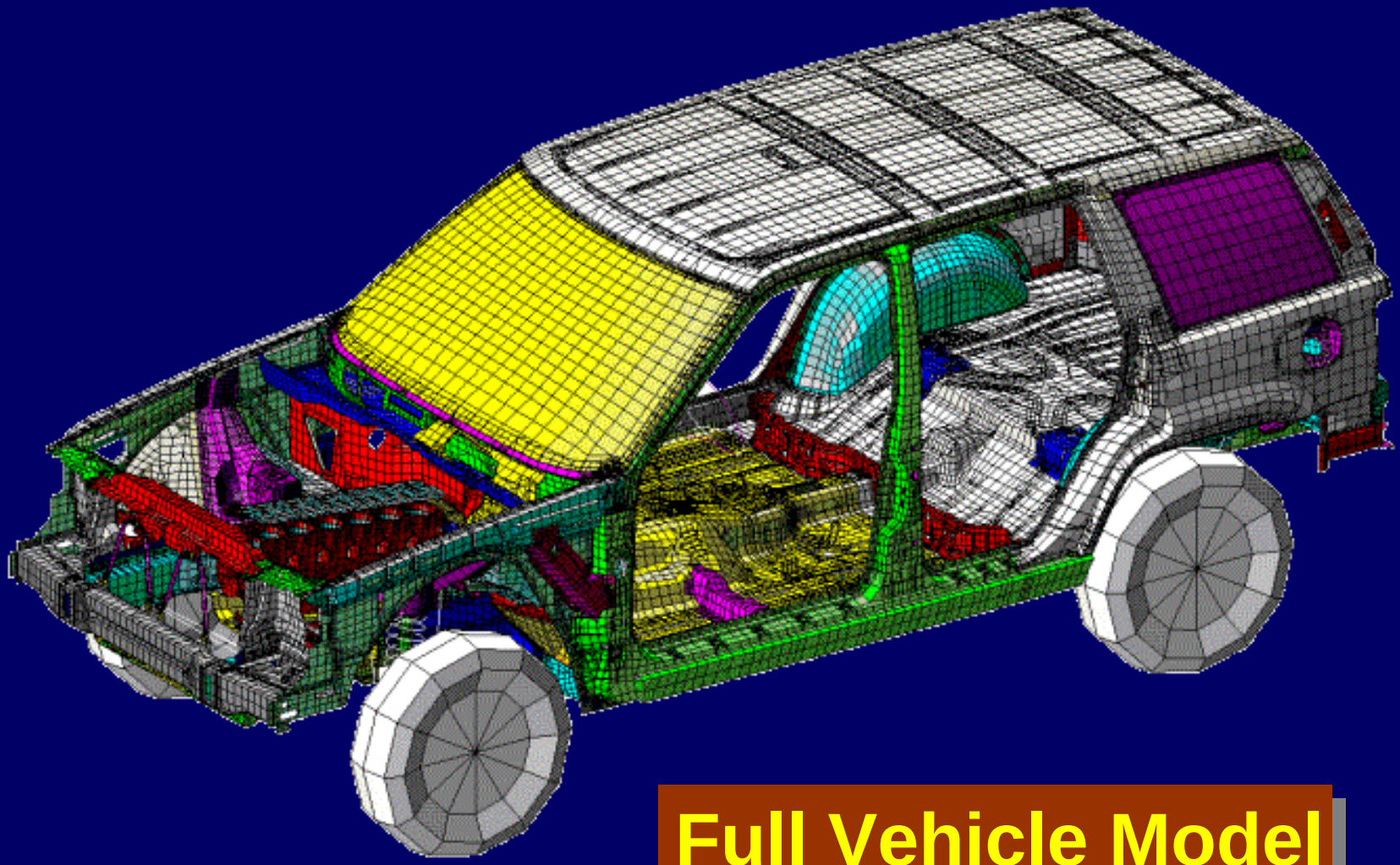
Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)



Full Vehicle Model

[Figure Courtesy of DaimlerChrysler Corporation]

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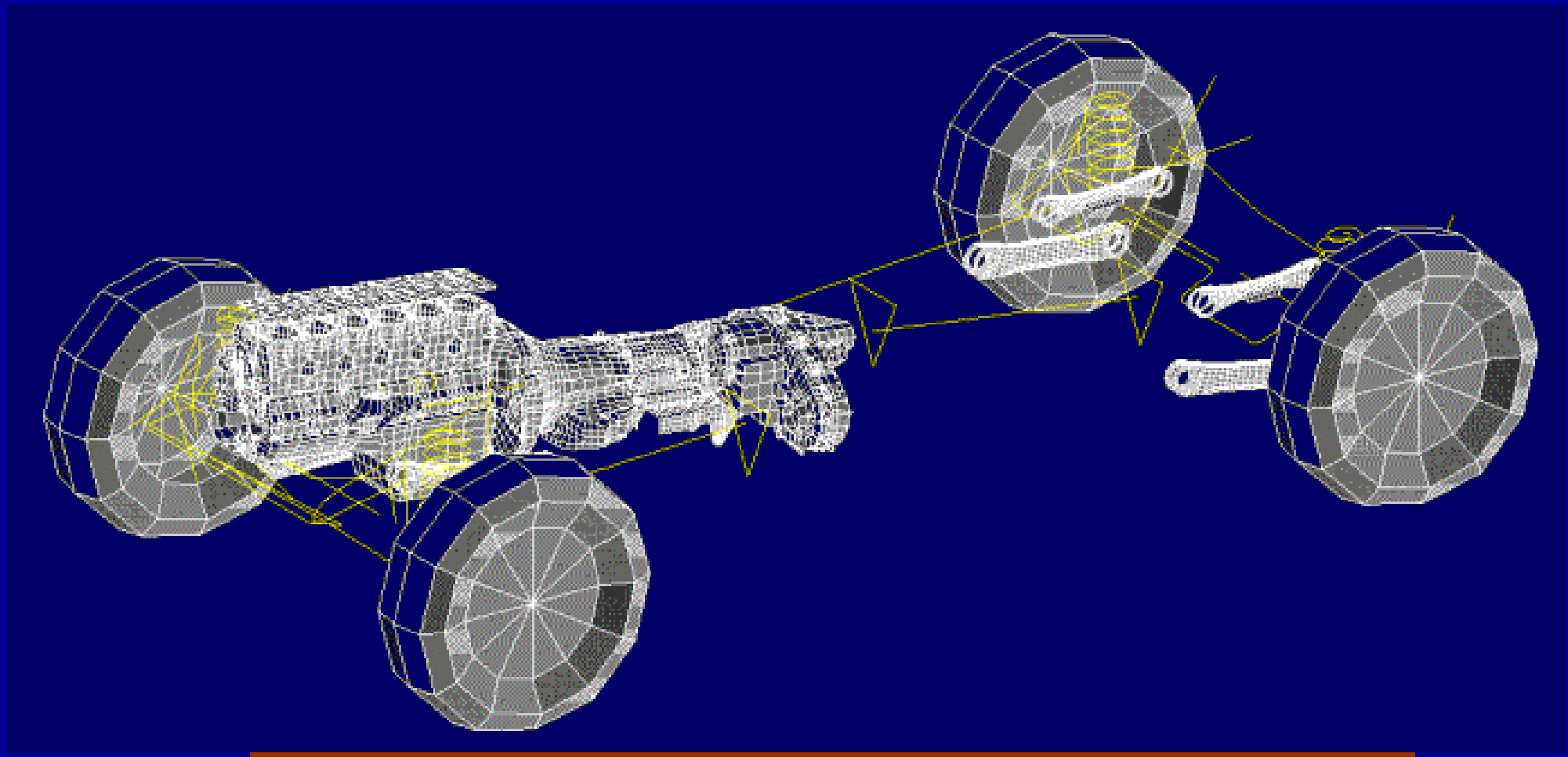
Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)



Powertrain/Axle/Suspension Model

[Figure Courtesy of DaimlerChrysler Corporation]

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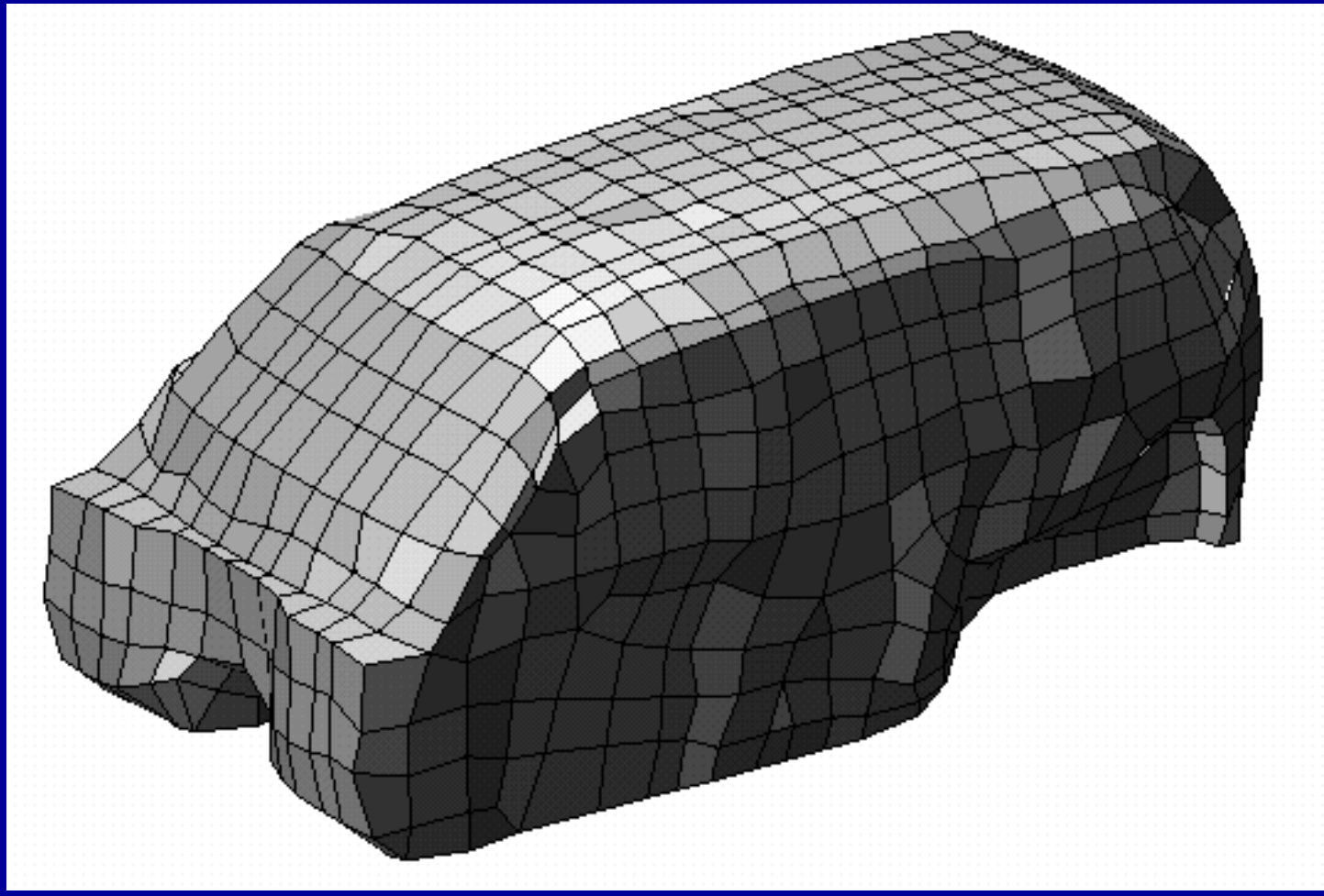
Mid Frequency NVH Improvement (Sports Utility Vehicle Example)



Acoustic Cavity Model

[Figure Courtesy of DaimlerChrysler Corporation]

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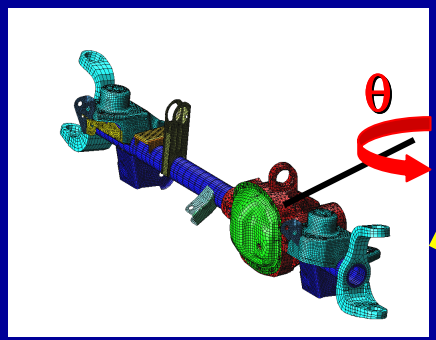
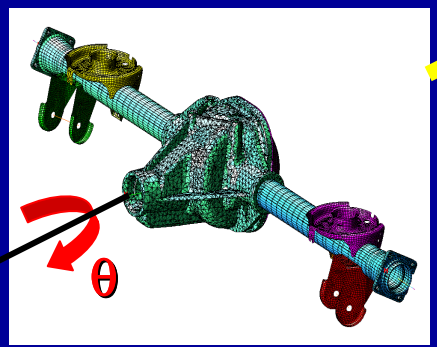
Mid-Frequency NVH improvement

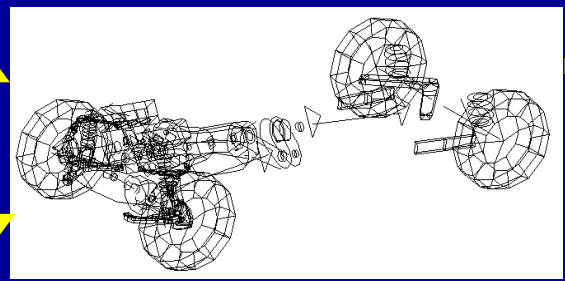
Axle Whine Example : 300-500 Hz

Front and Rear Axle
Gear-Pinion Mesh
Transmission Error

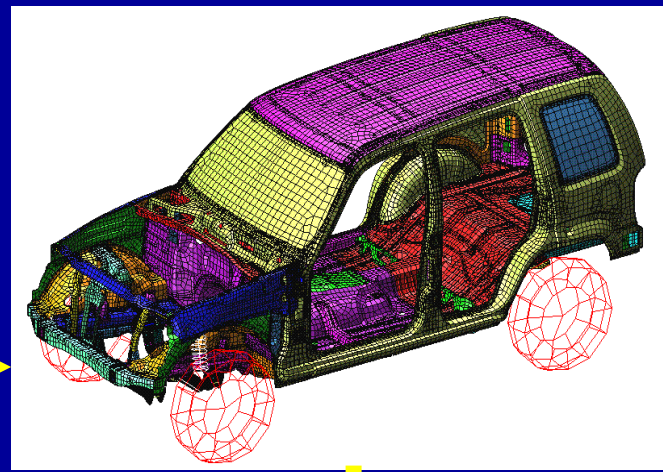


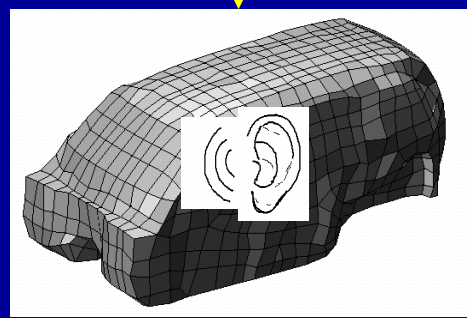


Chassis



Trimmed Body





Interior Acoustic Cavity

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

[Figure Courtesy of DaimlerChrysler Corporation]

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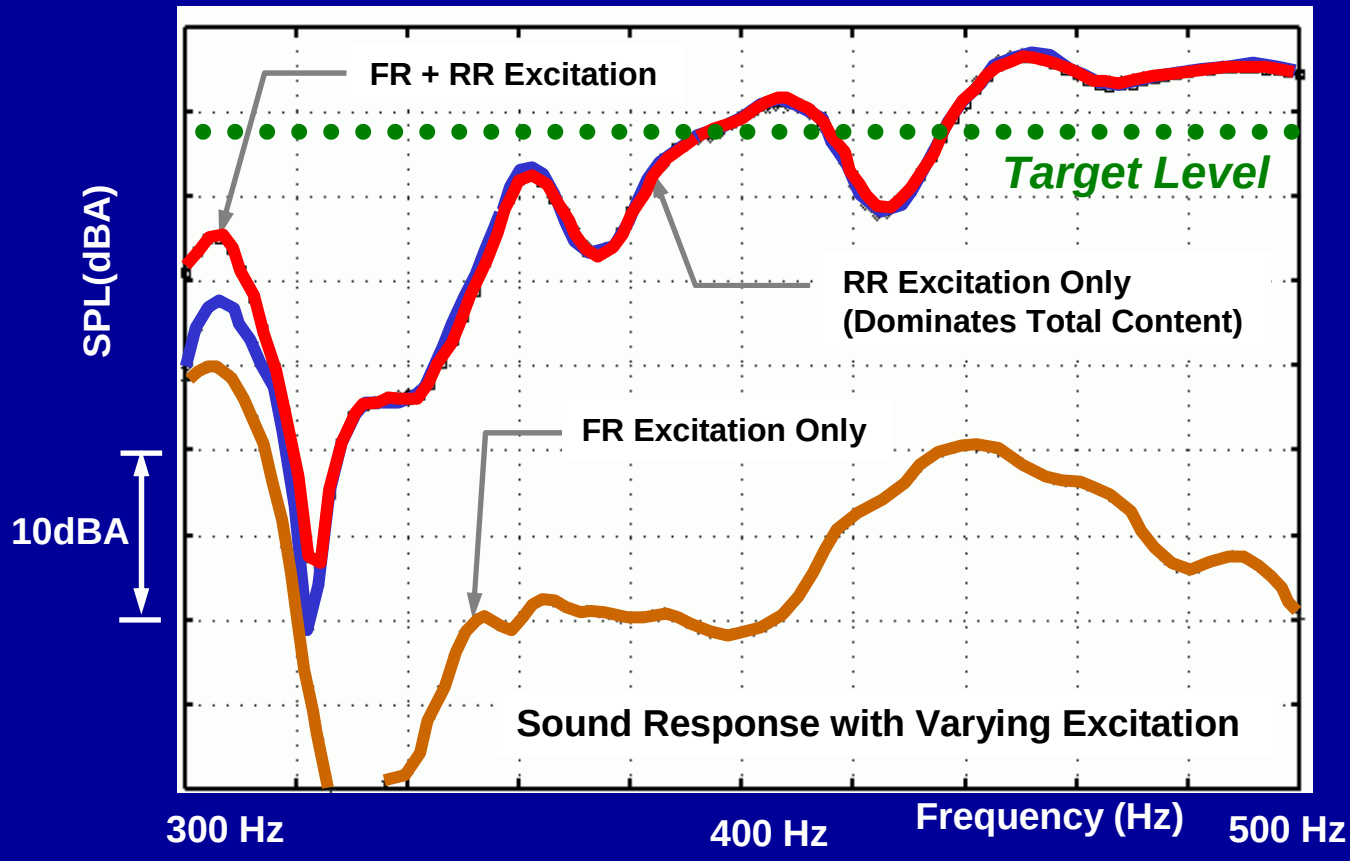
Axle Whine Example

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



[Figure Courtesy of DaimlerChrysler Corporation]

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Axle Whine Example

- **Before embarking on identifying the root cause for under-target performance at dominant noise paths, it is a good practice to perform reasonableness check on the response**
- **Steps for Reasonableness Determination**
  - **Judging the response based on system knowledge**
    - **Total response content is dominated by rear excitation. This is reasonable since vehicle has IFS and solid axle rear suspension which is harder to isolate for noise**
  - **Forced mode Animation**
    - **Operating deformed shape motion is rear axle pitching about ring gear axis. This was expected since input excitation is MTE imposed as enforced angular rotation between ring and pinion gear**
  - **Disconnect Studies**
    - **Disconnecting rear suspension noise paths (shock in particular) in pair had the most significant effect on Driver's SPL response**

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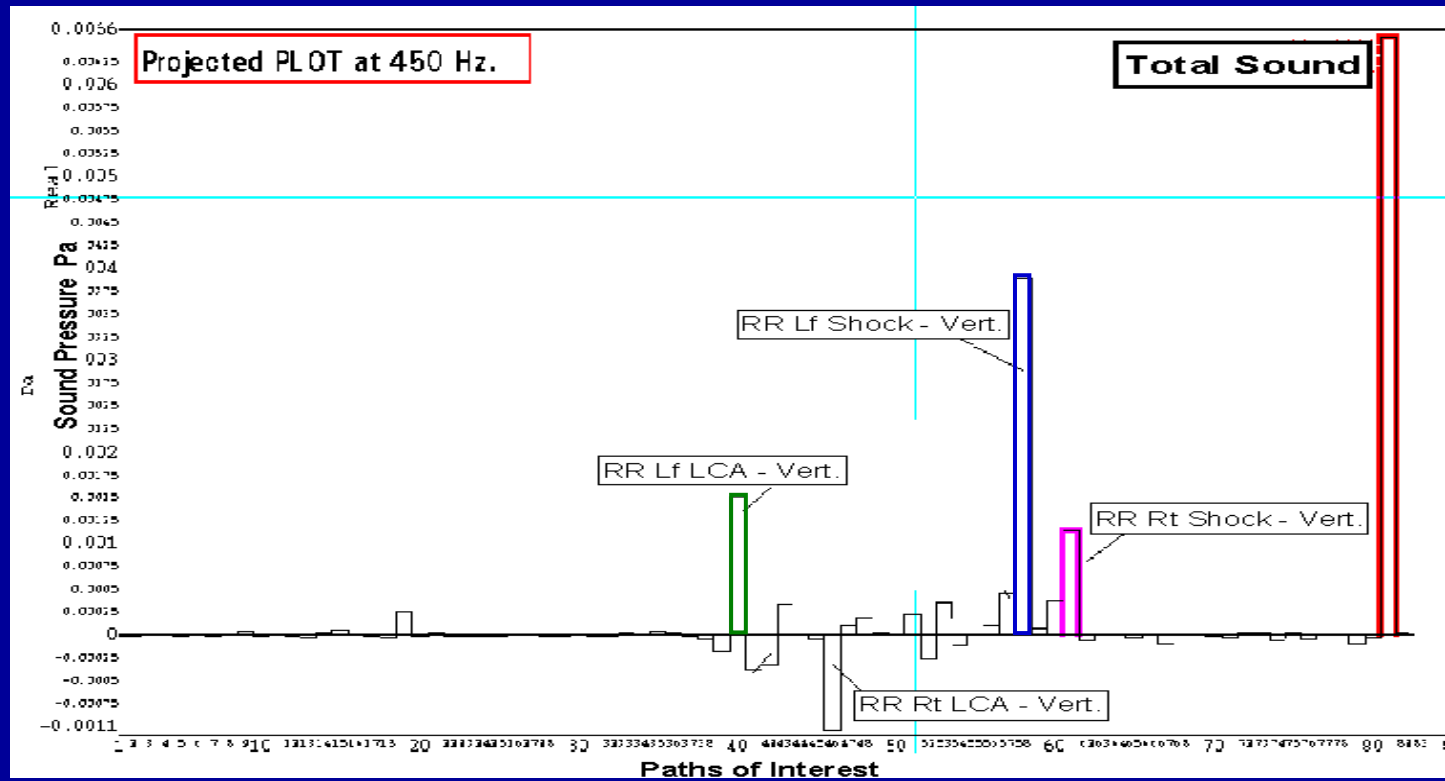
Axle Whine Example

- **Transfer Path Analysis**



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

- **The conclusion matches with reasonableness checks**

[Figure Courtesy of DaimlerChrysler Corporation]

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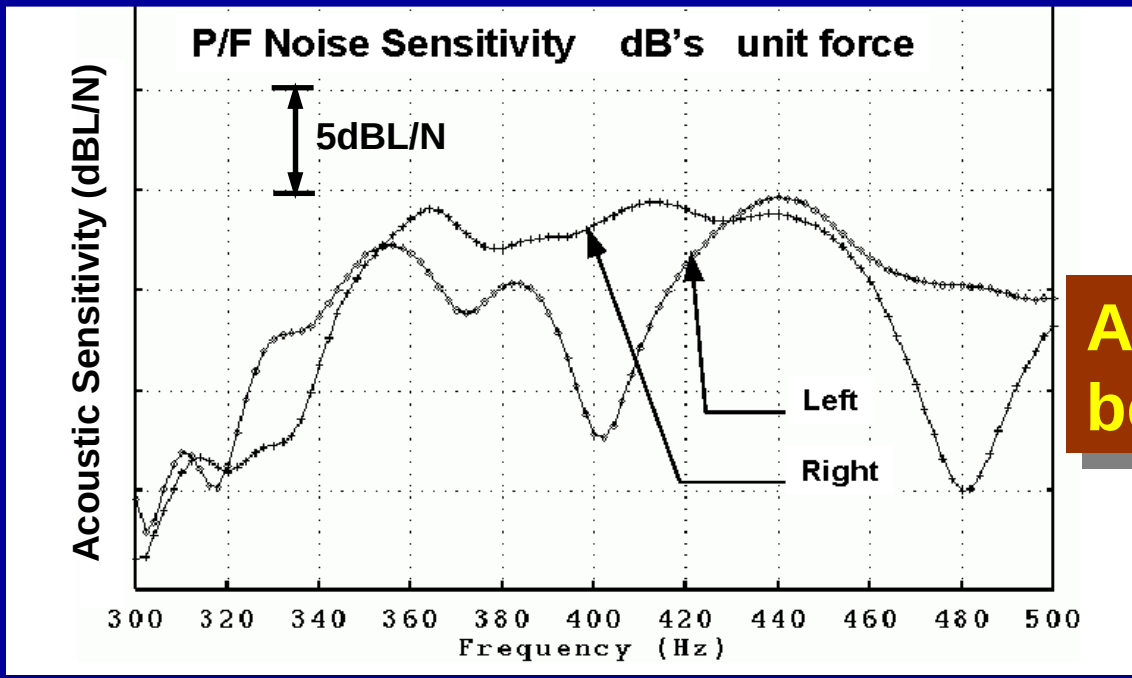
Axle Whine Example

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

$$R_t = \Sigma_{paths} [R_i] = \Sigma_{paths} [F_i * (R/F)_i]$$



Acoustic sensitivity is better than generic target

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

[Figure Courtesy of DaimlerChrysler Corporation]

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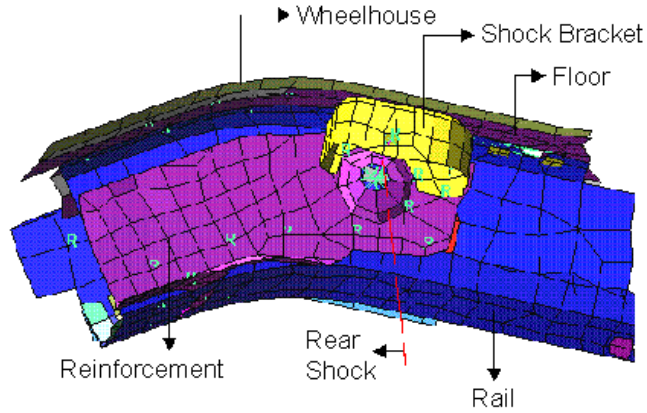
Axle Whine Example

Solution

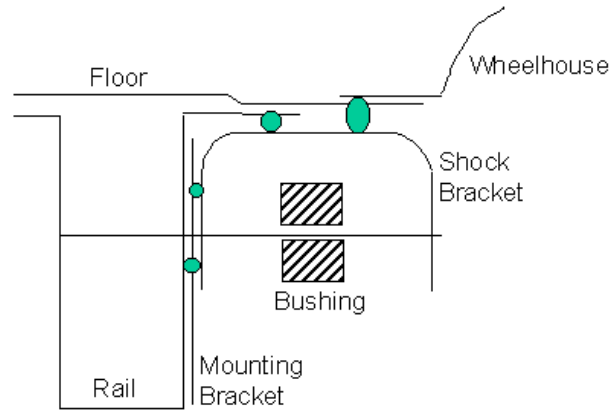
- Soften shock vertical bushings by 65%

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

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



Configuration of Shock Bracket



Rear View Section of Shock Bracket

[Figure Courtesy of DaimlerChrysler Corporation]

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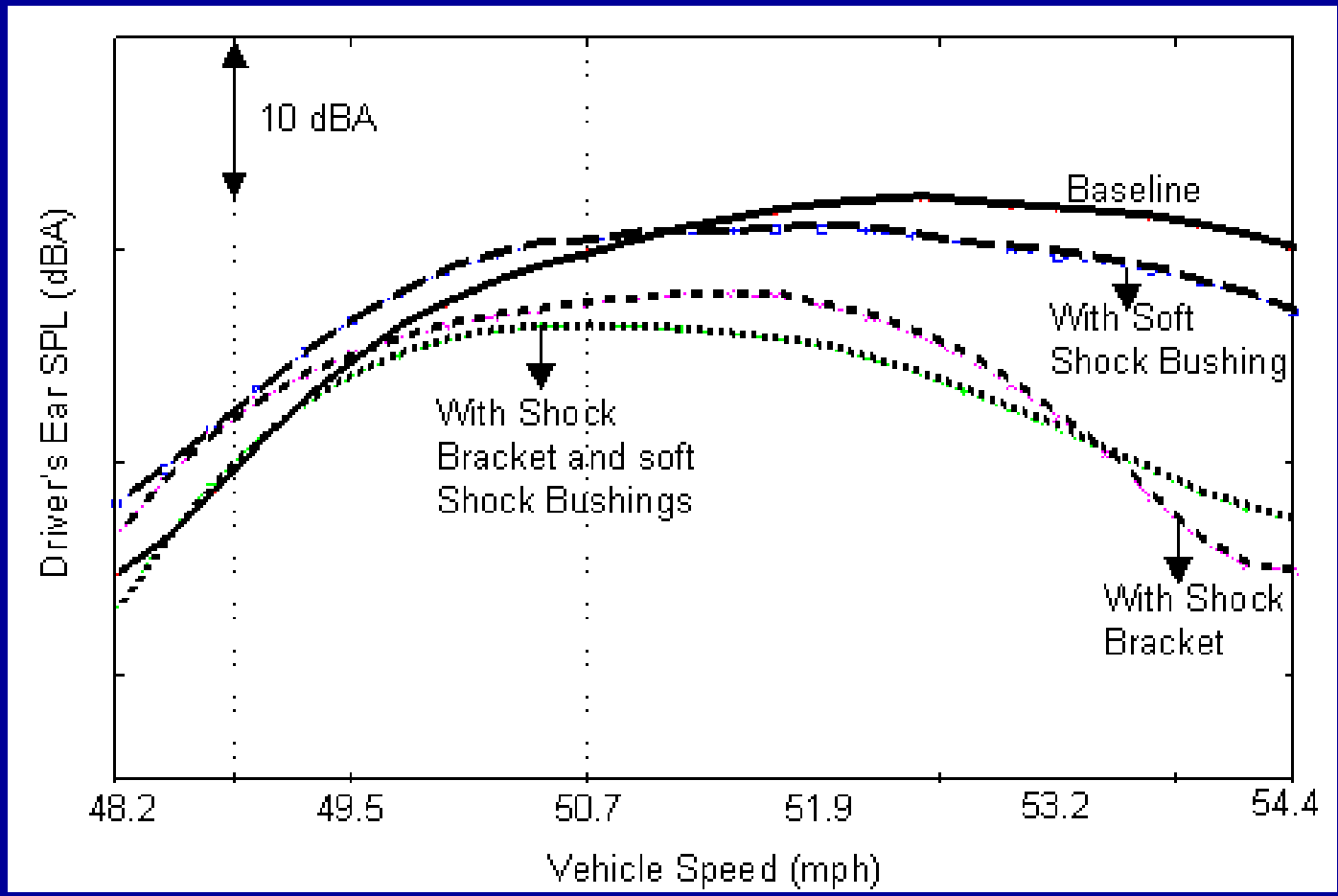
Axle Whine Example

Response Improvement due to proposed solution



[Figure Courtesy of DaimlerChrysler Corporation]

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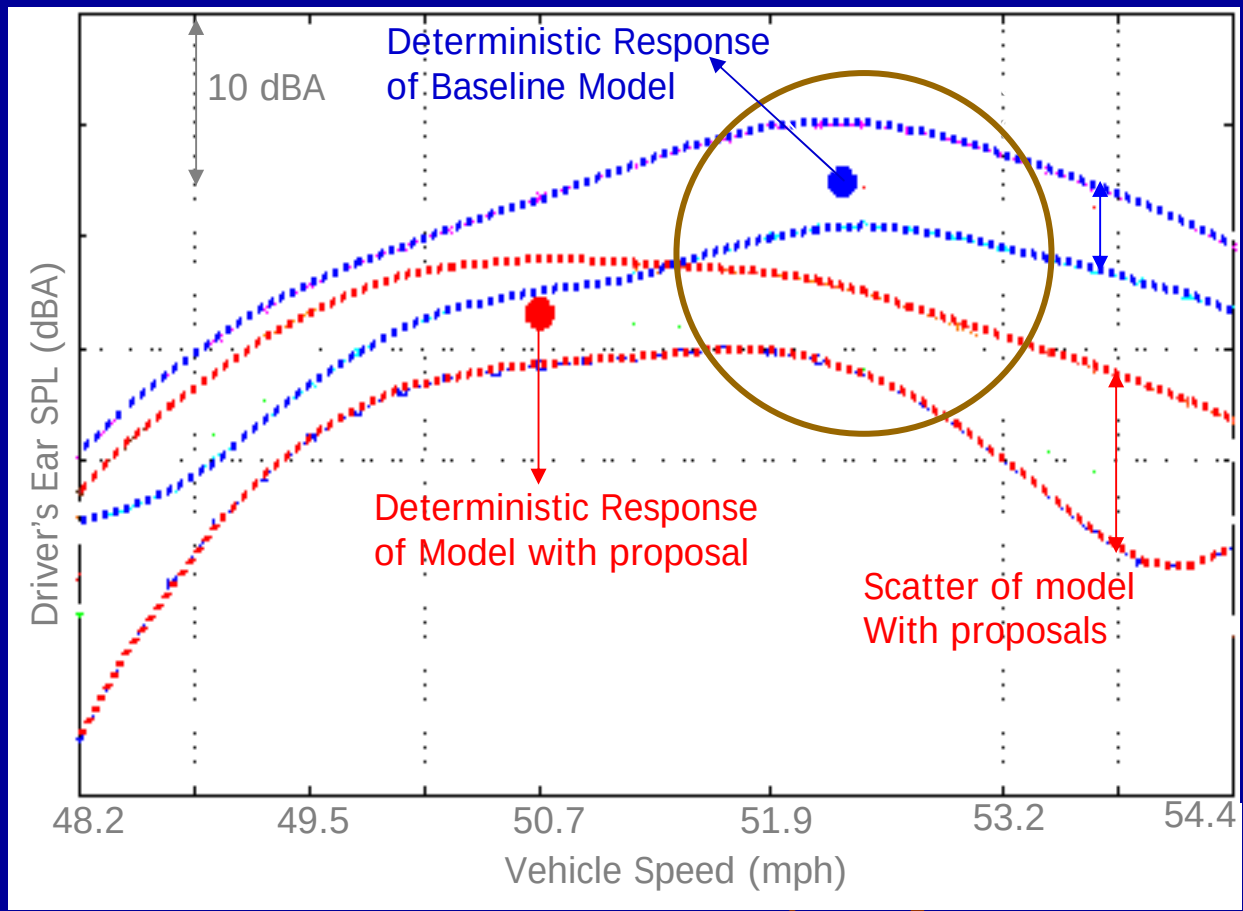
Axle Whine Example

How Robust is the proposed solution ?

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

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

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



[Figure Courtesy of DaimlerChrysler Corporation]

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Final Remarks on Mid Frequency Analysis

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

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NVH Workshop Topic Outline

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

Alan Duncan

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NVH Workshop Topic Outline

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- *Live Noise Attenuation Demo*
- Mid Frequency Basics
- **Utilization of Simulation Models**

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Utilizing NVH Simulation Models

Considerations

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

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

- Q. How do I make design decisions before hardware is available?
- ANS. Correlation must be performed on existing hardware to establish modeling methods to be applied to the future design.

  (The Reference Baseline Ref. 3)

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

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Utilizing NVH Simulation Models

Considerations

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

- Q. How wide is the band of variability?
- ANS. Don't Ask !!!

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Discussion of Product Variability

Topics

- Kompella and Bernhard Observations
- Freyman NVH Scatter Results
- Model Confidence Criteria
- Conclusions

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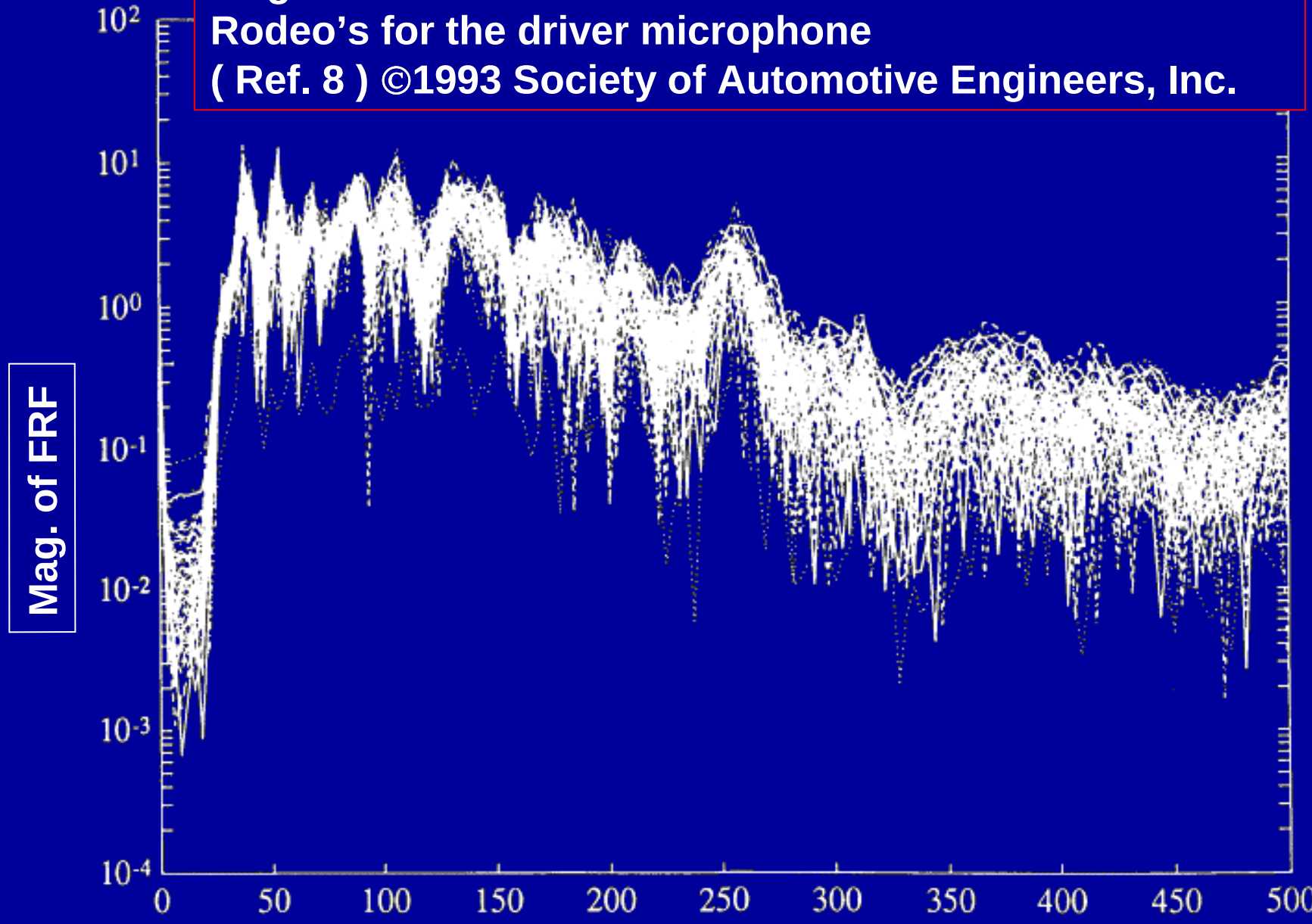
Magnitude of 99 Structure – borne FRF's for the Rodeo's for the driver microphone

(Ref. 8) ©1993 Society of Automotive Engineers, Inc.



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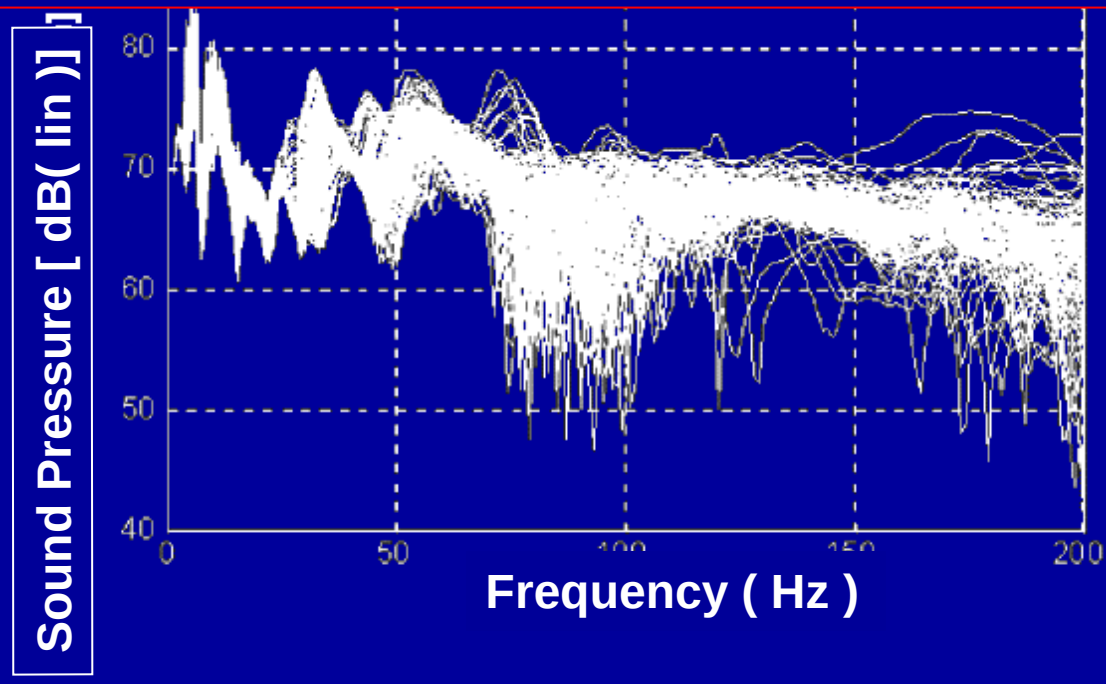
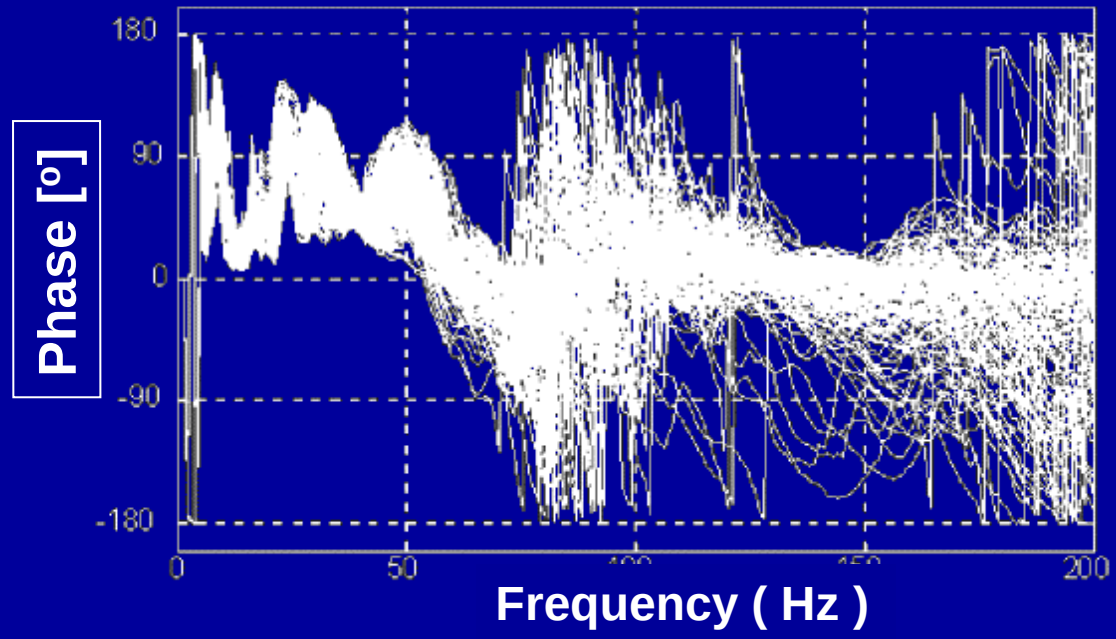
Acoustic scatter numerically determined in the vibro – acoustic behavior of a vehicle due to possible tolerances in the component area and in the production process





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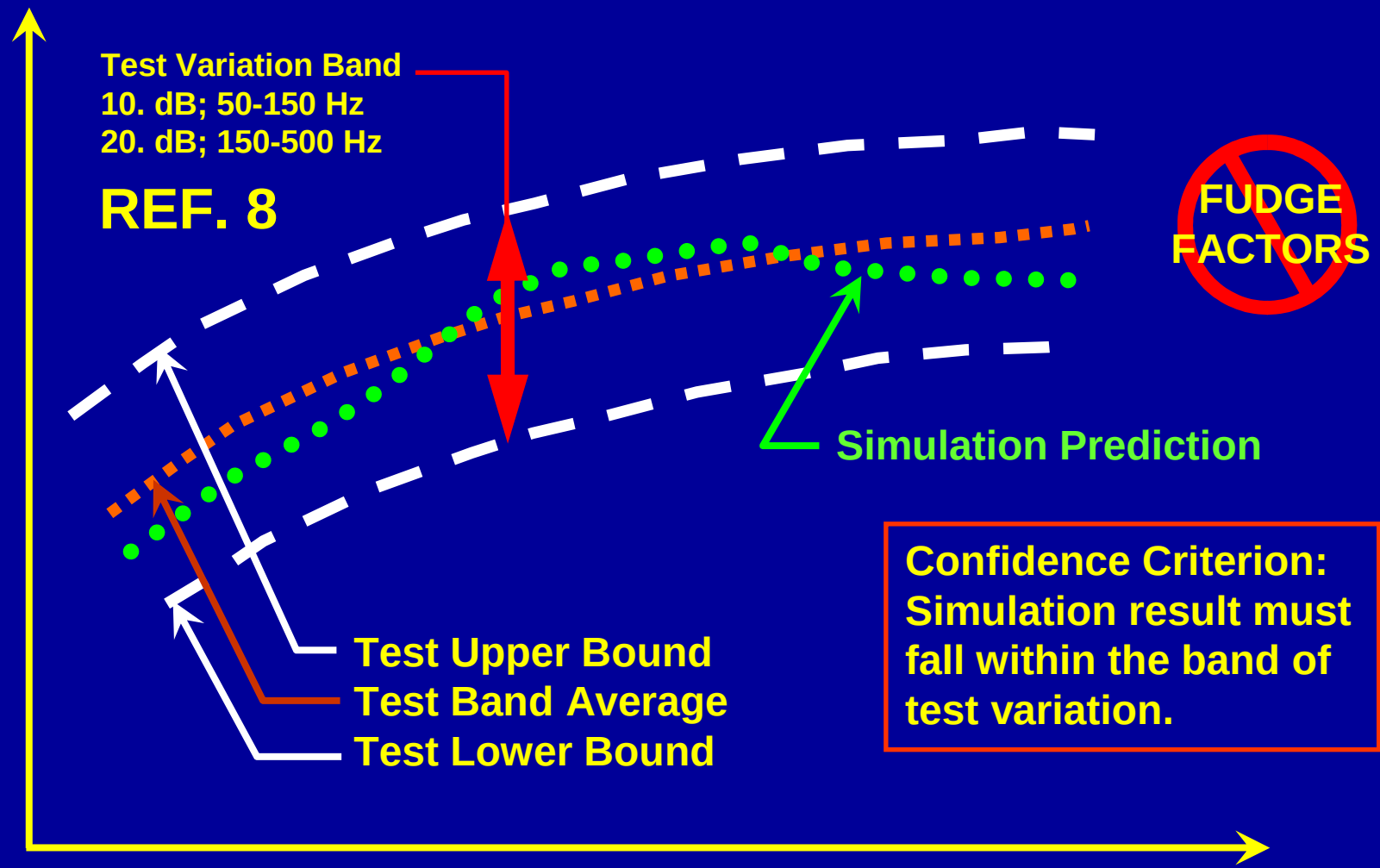
Reference Baseline Confidence Criterion

For *Operating* Response Simulations



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

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Axle Whine Example

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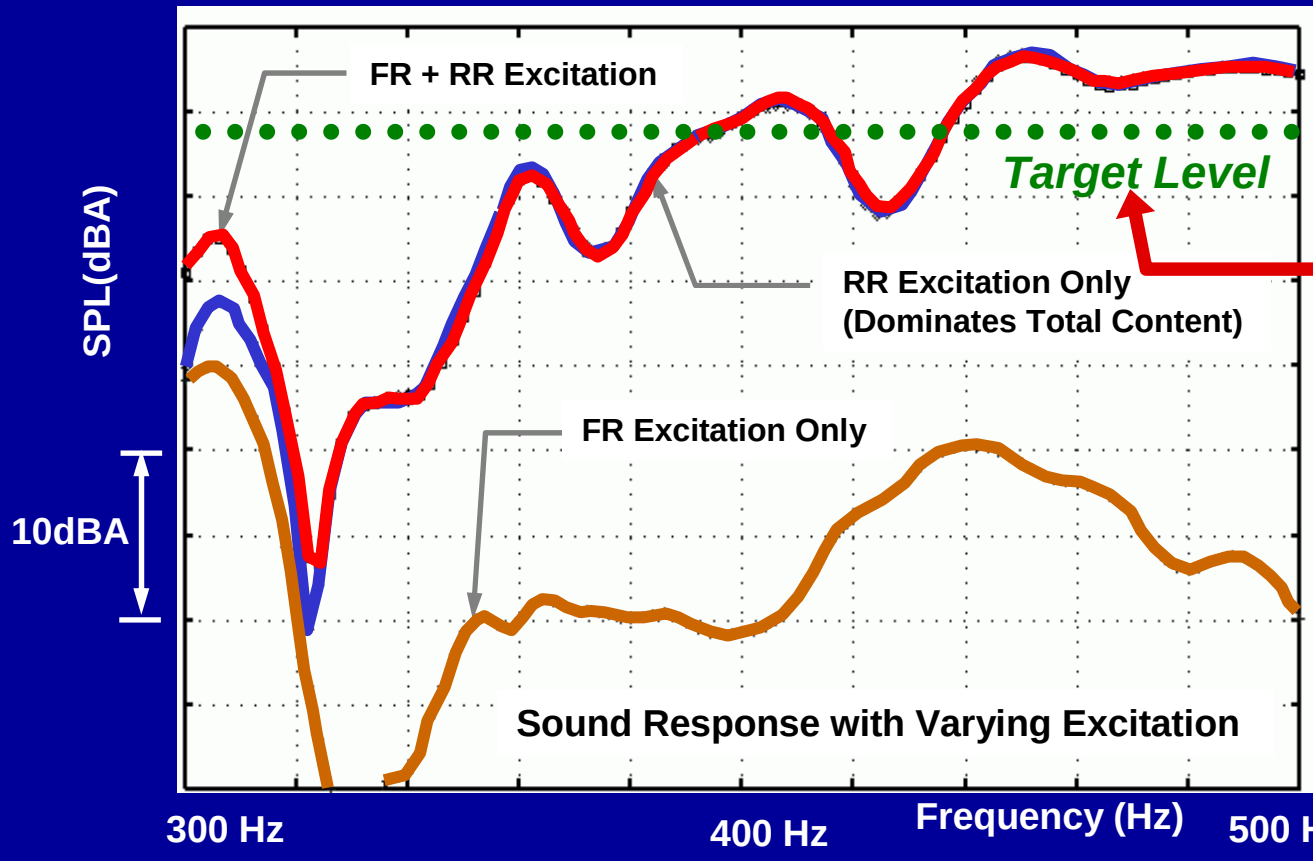
- As the design was firmed out, full vehicle analysis revealed under target performance for Driver's ear SPL response which was dominated by rear excitation



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

[Figure Courtesy of DaimlerChrysler Corporation]

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Conclusions:

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

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

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

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

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

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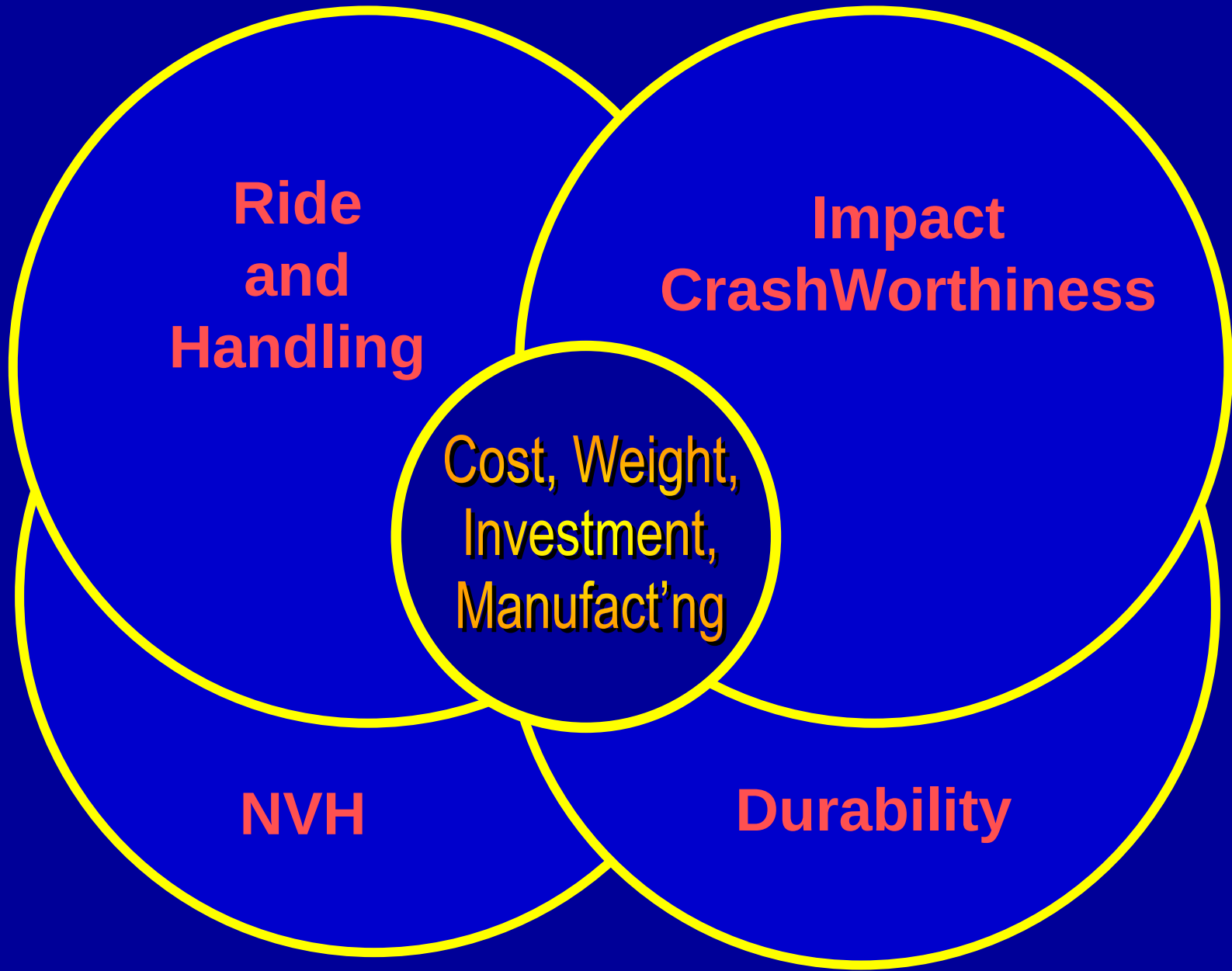
Competing Vehicle Design Disciplines



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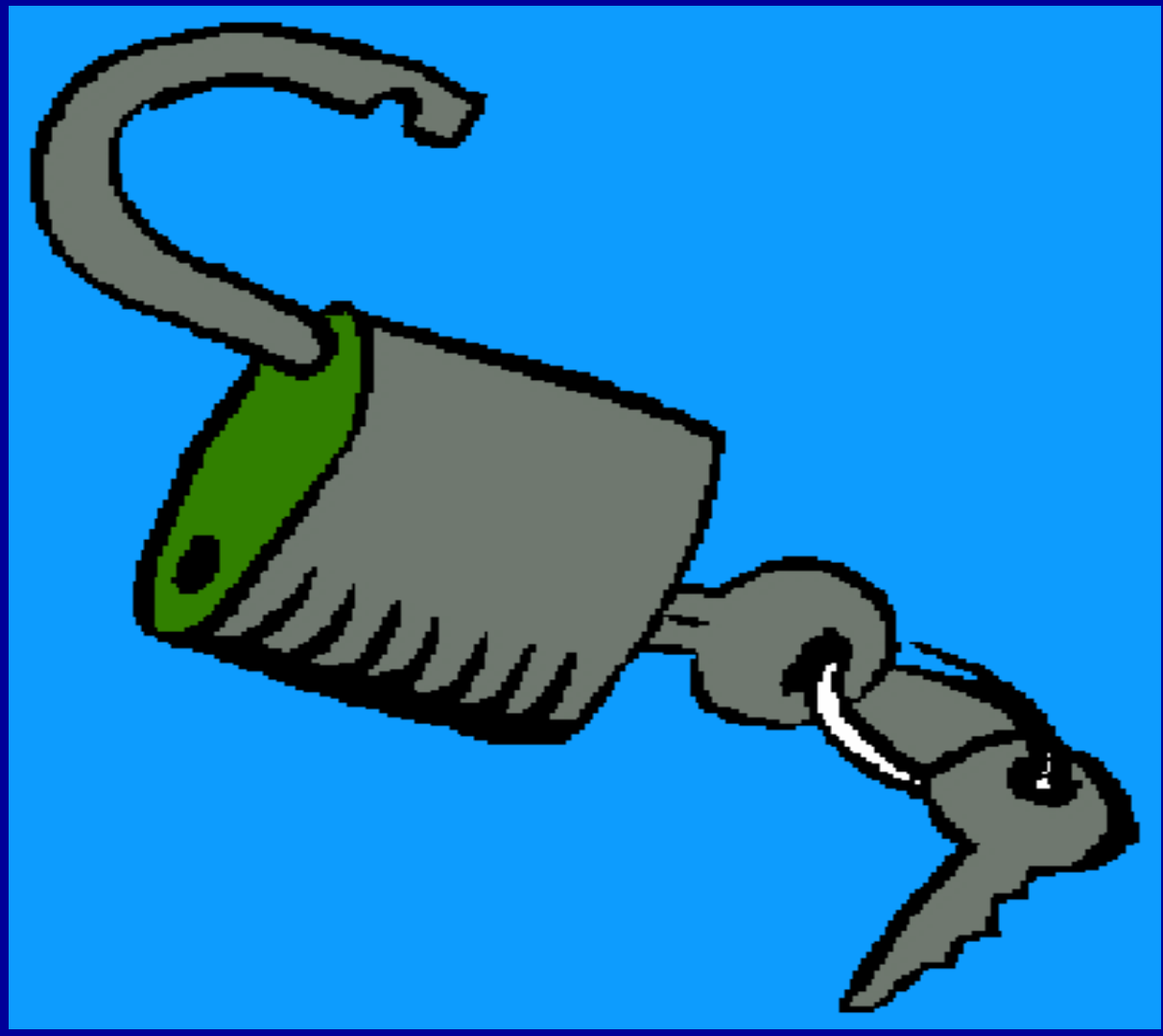
The Fundamental Secret of Structure Borne NVH Performance



Revealed here tonight !

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Fundamental Secret to Making Money in the Stock Market

Buy Low and Sell High !

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The Fundamental Secret of Structure Borne NVH Performance

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

Meets Conditions of the Attenuations Strategies

- *Minimize the Source Load*
- *Manage Mode Placement*
- *Provide Isolation*
- *Mount at Nodal Points*
- *Provide Dynamic Absorber*
- *Reduce Source - Receiver Mobility*

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The Fundamental Secret of Structure Borne NVH Performance

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

That's All Folks !

Thank You for Attending the

SAE Structure Borne NVH Workshop

Your Presenters tonight were:

Alan Duncan, Automotive Analytics, Inc.

Greg Goetchius, Material Sciences Corp.

Sachin Gogate, Altair Engineering, Inc.

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

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