Structure Borne NVH Basics

SAE 2007 NVH Conference; St. Charles, Illinois
Wednesday Evening, May 16, 2007

Presenters:

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

S. Gogate — **DaimlerChrysler Corp.**

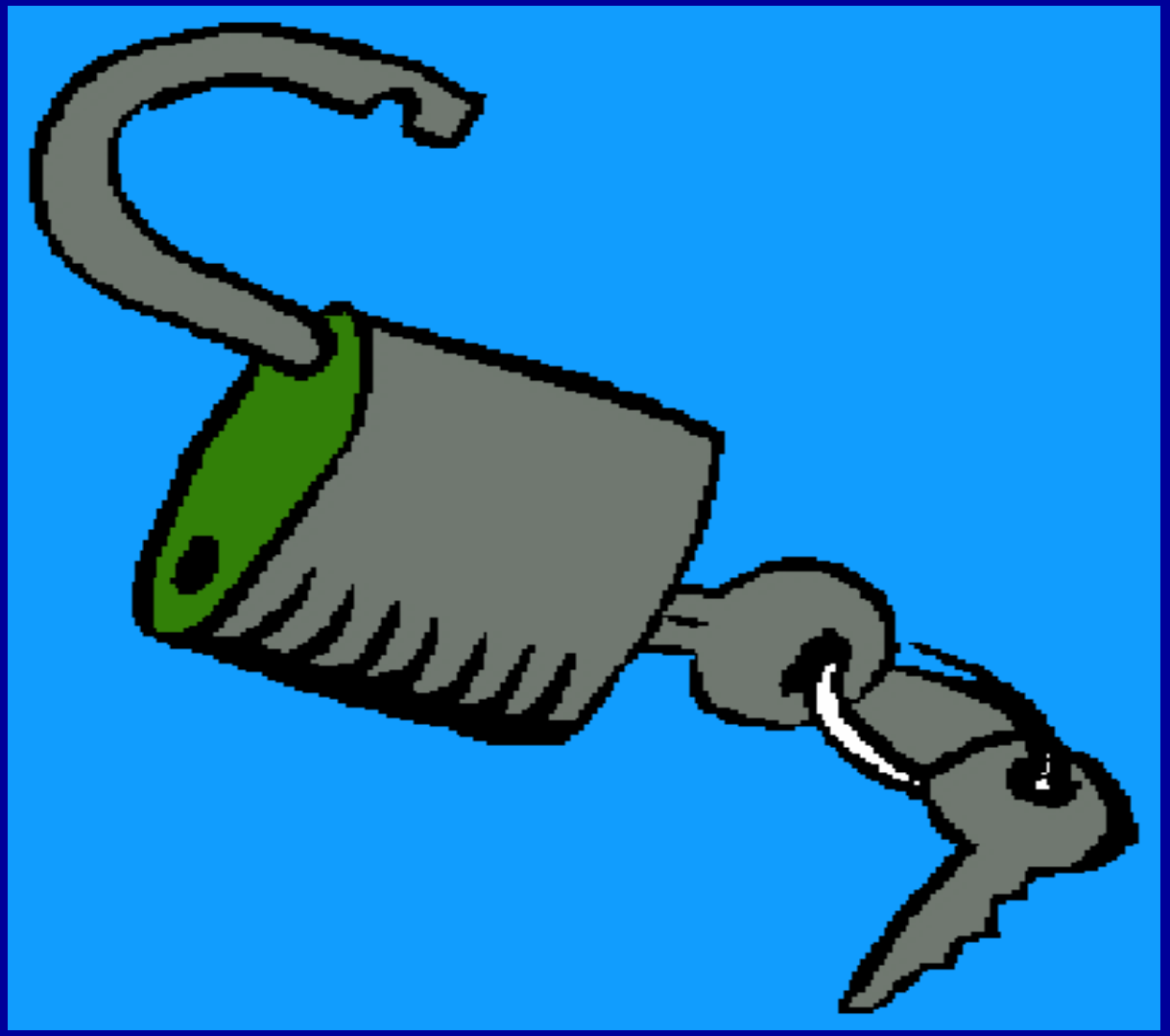
Sponsored By:

SAE Noise and Vibration Committee

NVH Workshop Topic Outline

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

The Fundamental Secret of Structure Borne NVH Performance



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

Structure Borne NVH 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

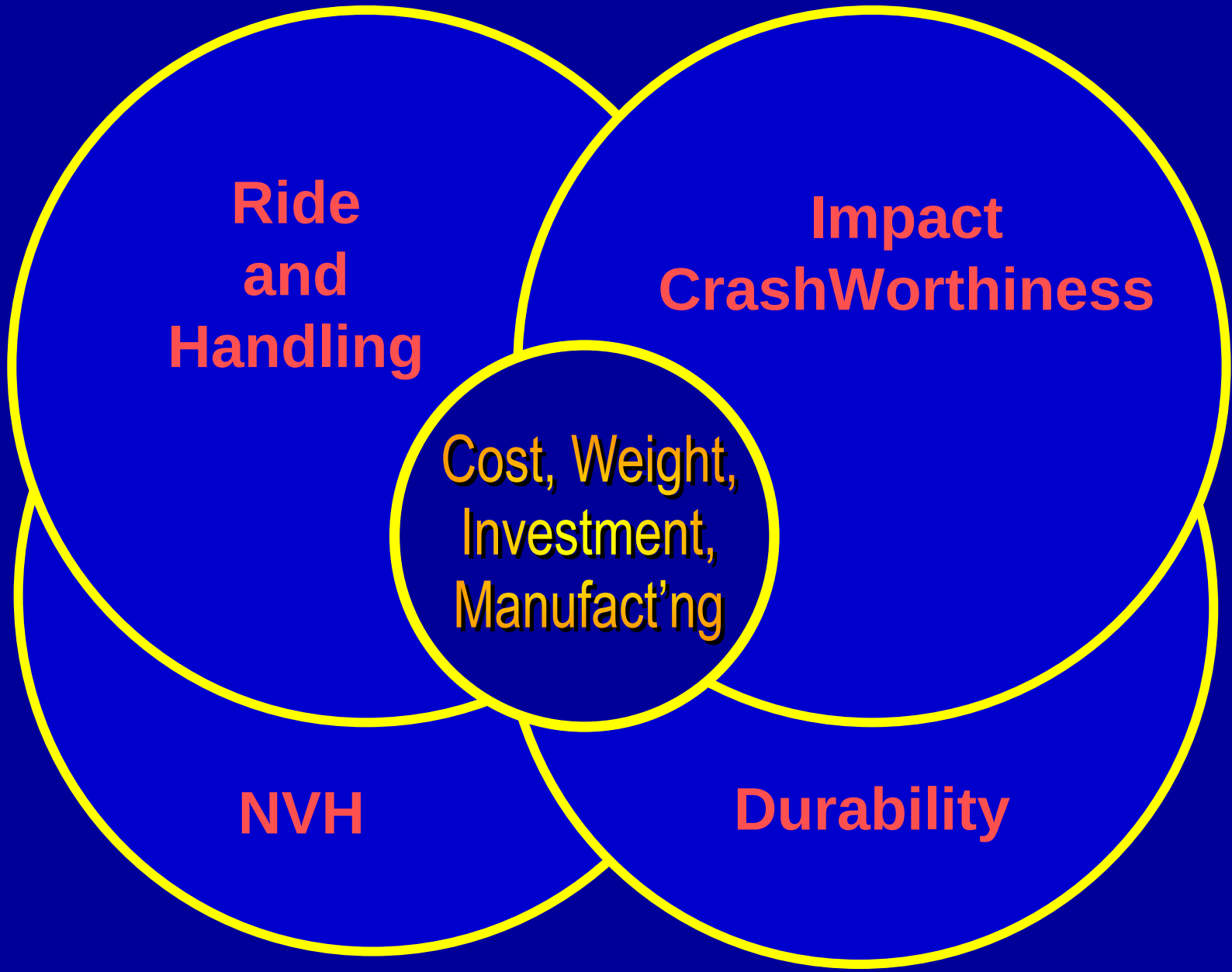
8. Kompella, M. S., and Bernhard, J., "Measurement of the Statistical Variation of Structural-Acoustic Characteristics of Automotive Vehicles", SAE No. 931272, 1993

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

NVH Workshop Topic Outline

- Introduction
- Fundamentals in NVH
- Automotive NVH Load Conditions
- Low Frequency Basics

Competing Vehicle Design Disciplines



NVH Workshop Topic Outline

- Introduction
- **Fundamentals in NVH**
- NVH Load Conditions
- Low Frequency Basics

Fundamentals in NVH

- What is N, V and H?
- Time and Frequency Relation
- Subjective to Objective Conversions
- Single Degree of Vibration and Vibration Isolation Principle
- Automotive NVH Frequency Range

What is N, V and H?

(in Automotive Context)

Based on SAEJ670e Standard (Vehicle Dynamics Committee July 1952)

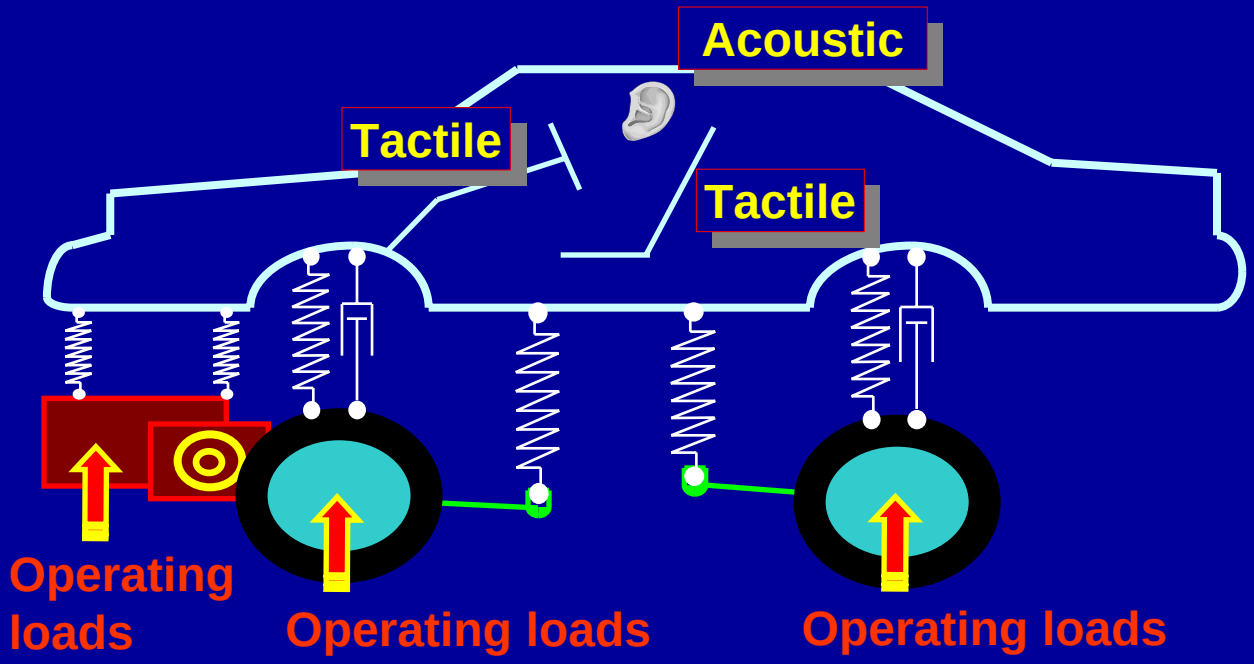
Noise : Vibration perceived audibly and characterized as sensations of pressure by the ear

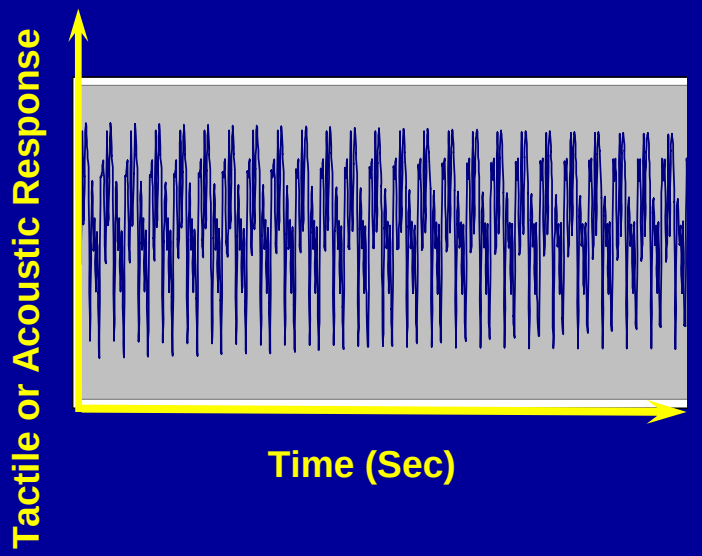
Vibration : Perceived tactually (at vehicle occupant interface points of steering column, seats, etc.

Harshness : Related to transient nature of vibration and noise associated with abrupt transition in vehicle motion. It could be perceived both tactually and audibly

Together, they define the measure of vehicle NVH Quality

Time and Frequency Relation

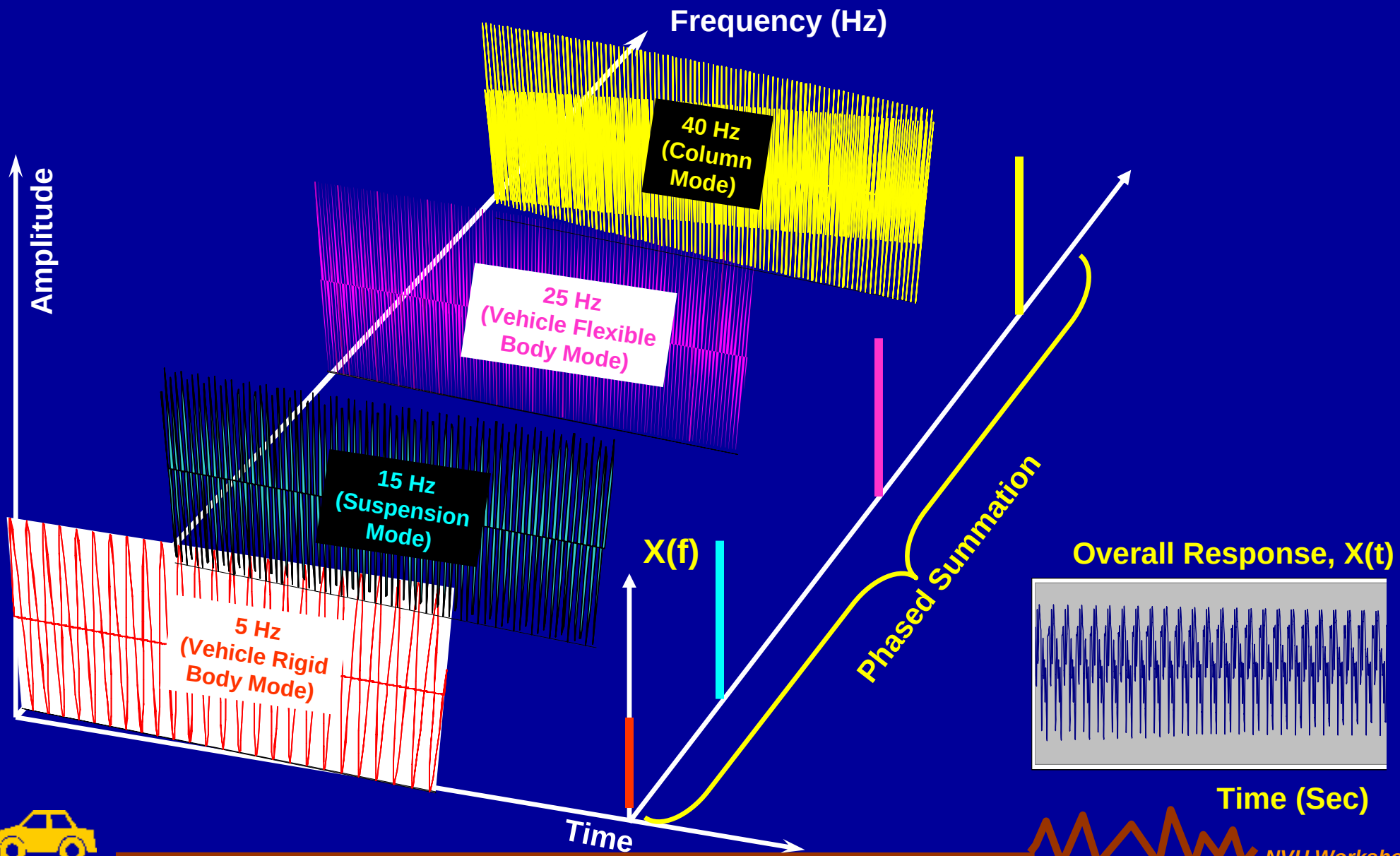




- Responses perceived in vehicle vary with time as vehicle operates under loads
- Responses are usually steady state and periodic in nature
- It is convenient and intuitive to consider responses in frequency domain while preserving the signal content

Time and Frequency Relation

- **Conversion to frequency domain lends to formulation of principles for addressing structure borne NVH**



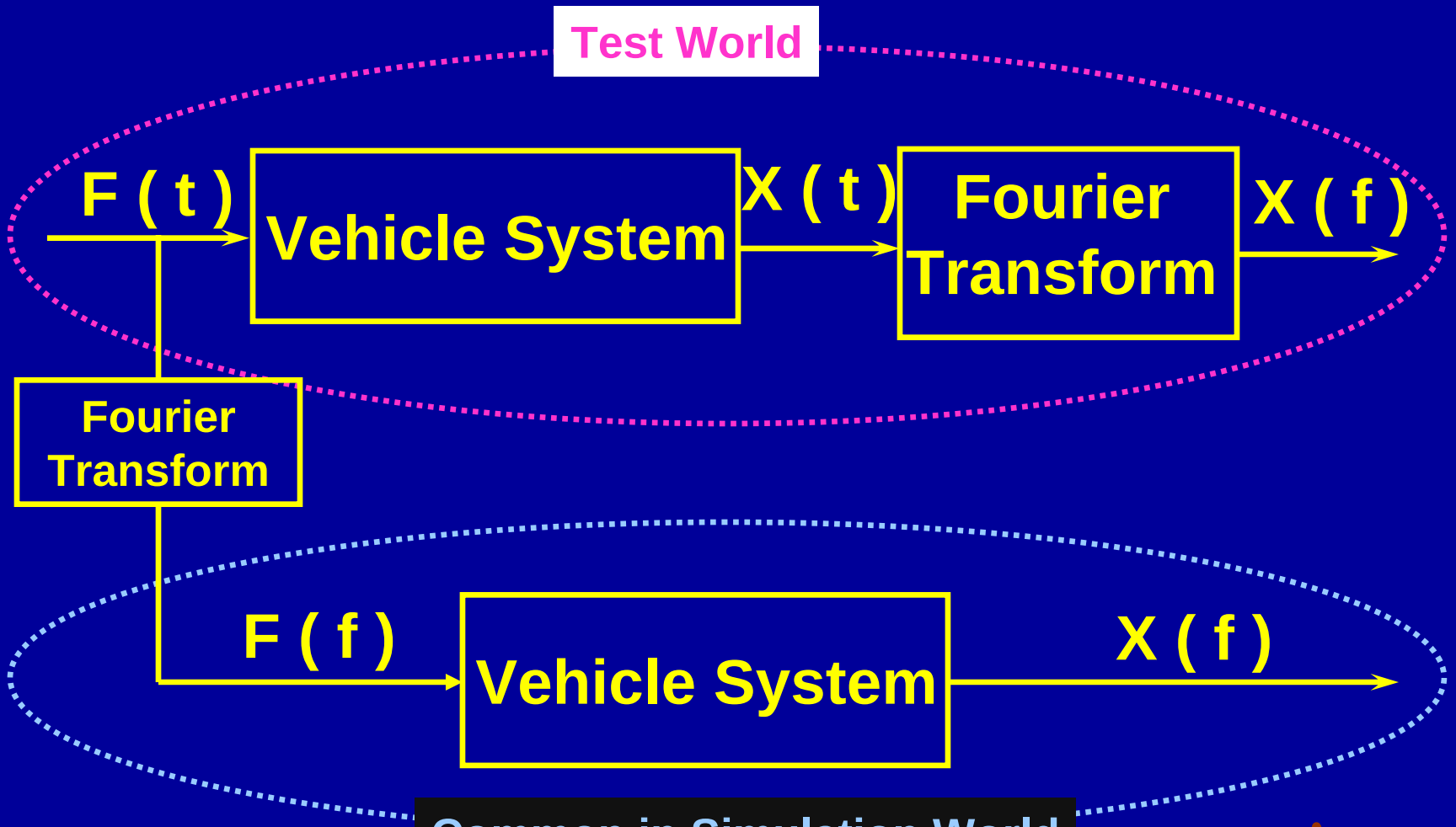
Time and Frequency Relation

- **Mathematically Speaking …….**

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

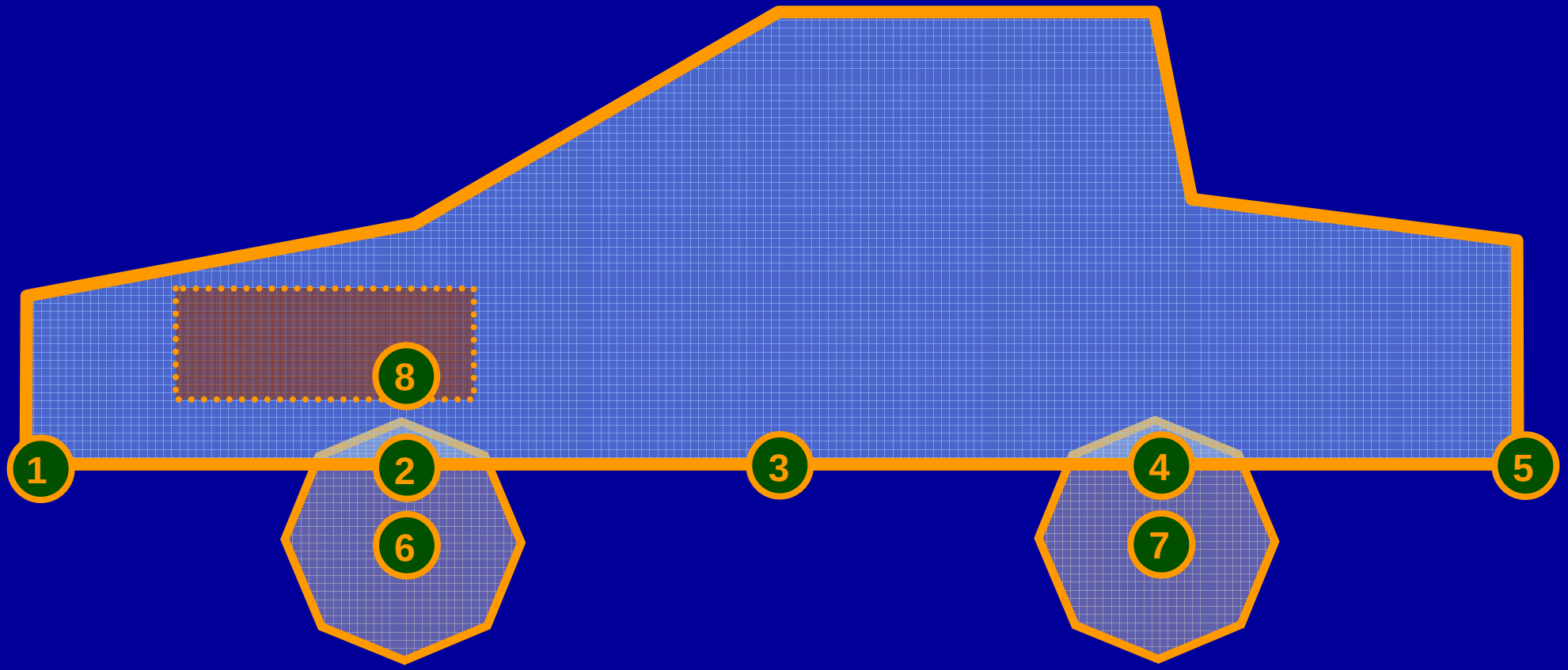
Time and Frequency Relation

- **Responses can be obtained in frequency domain either through Fourier Transform of time domain signal or directly in frequency domain**



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	

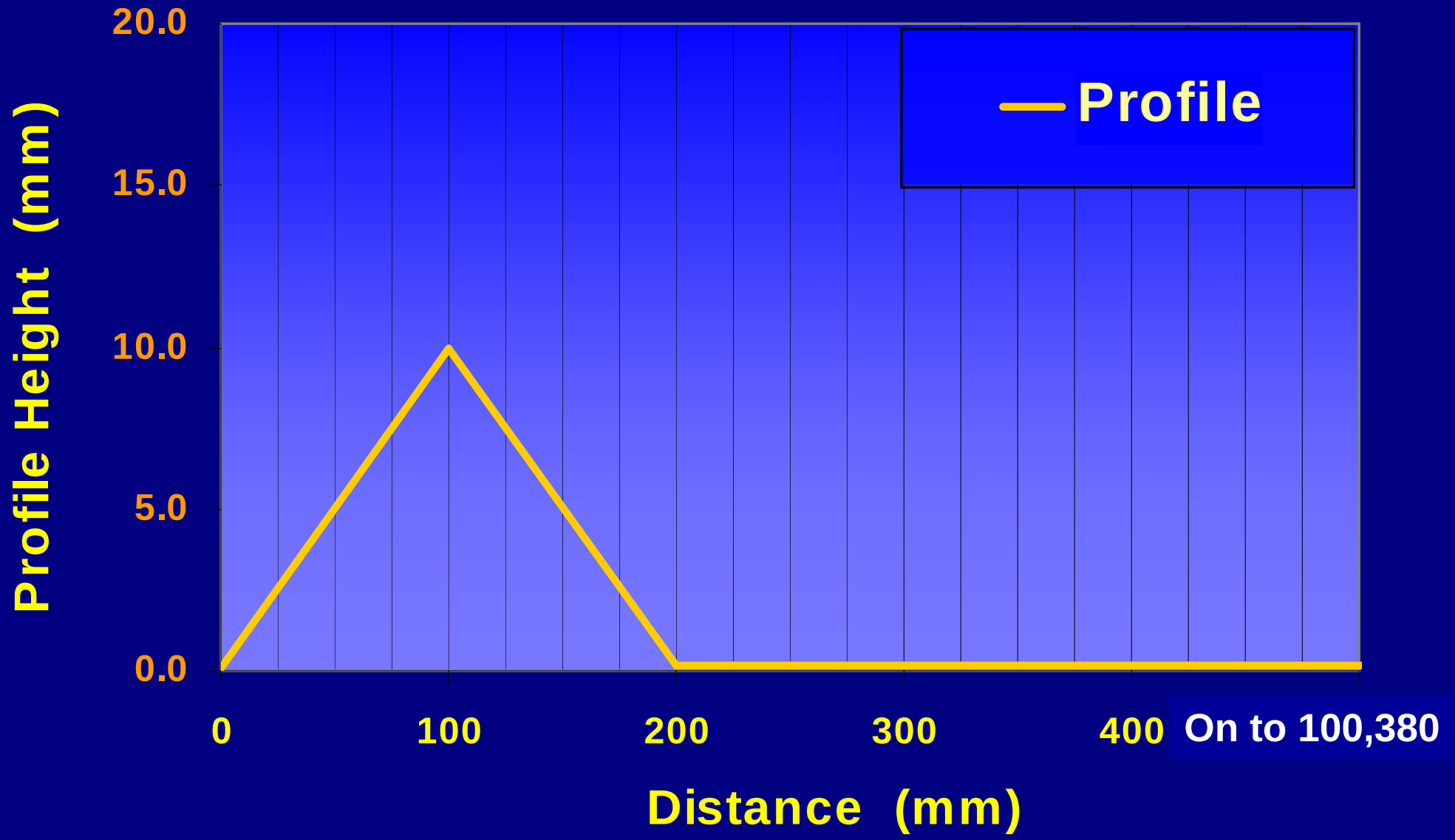
Tires	350.3	N/mm
KF	43.8	N/mm
KR	63.1	N /mm

Beam mass lumped on grids like a beam

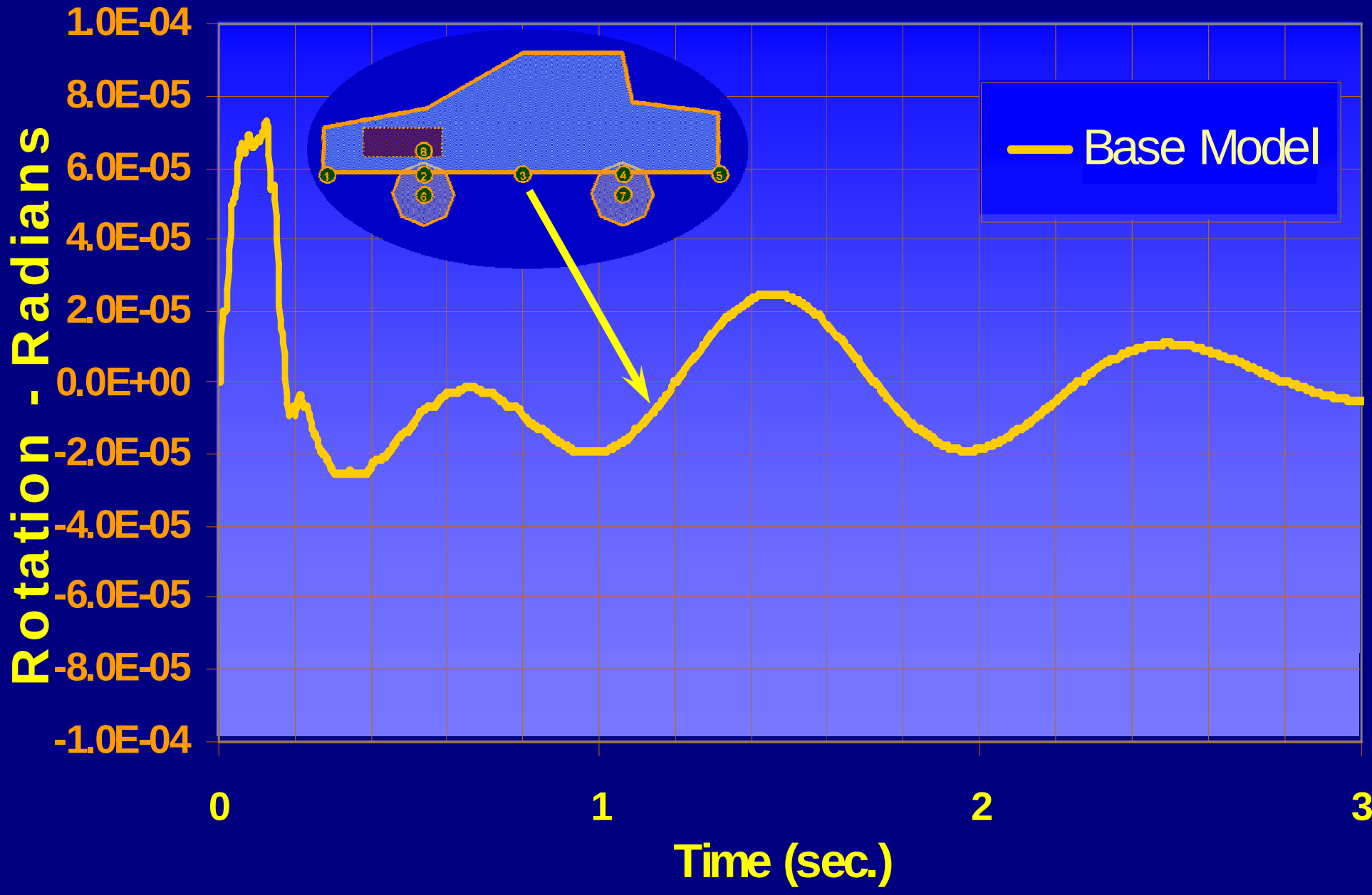
M2,3,4 =2 * M1,5

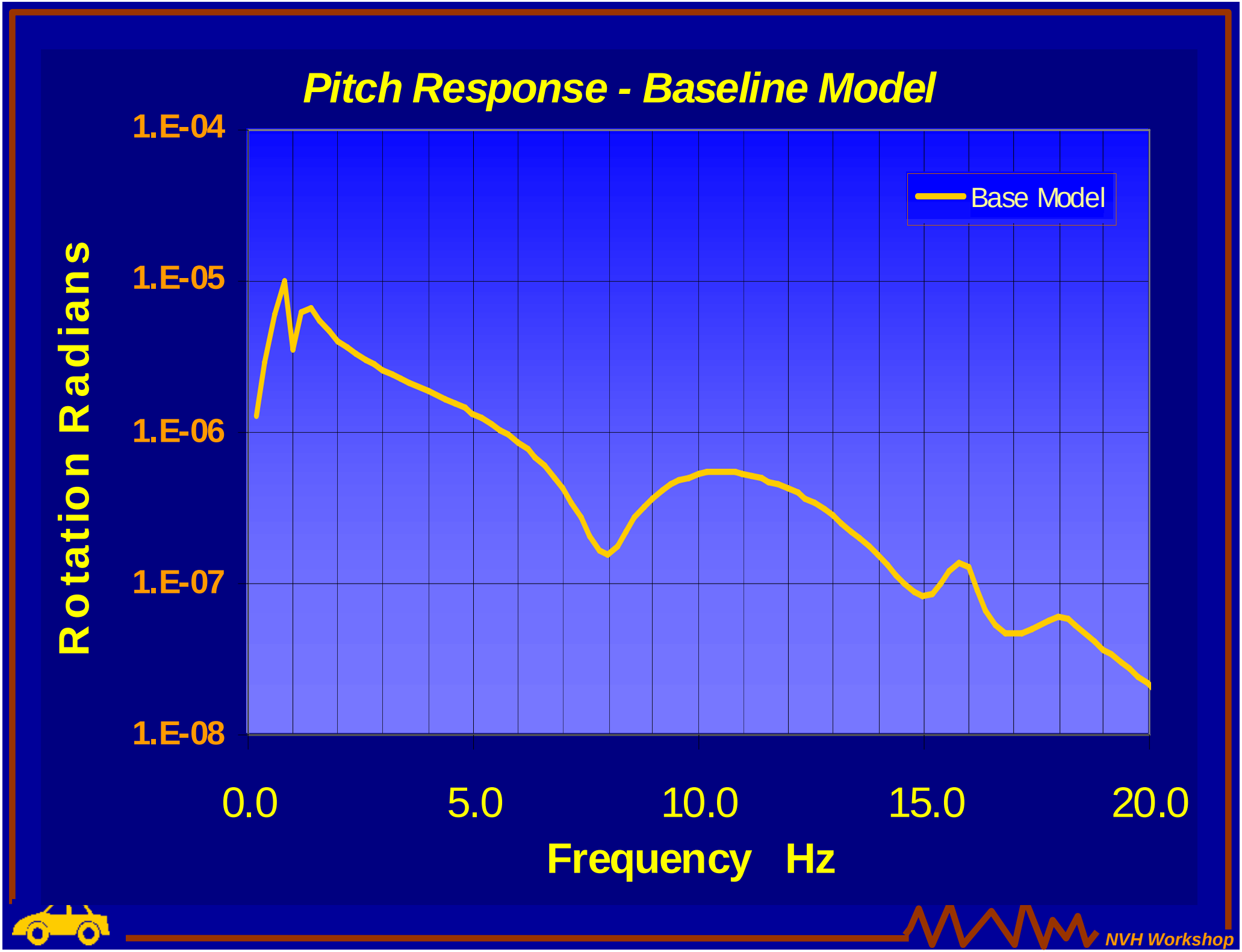
From Reference 6

Excitation Bump Profile



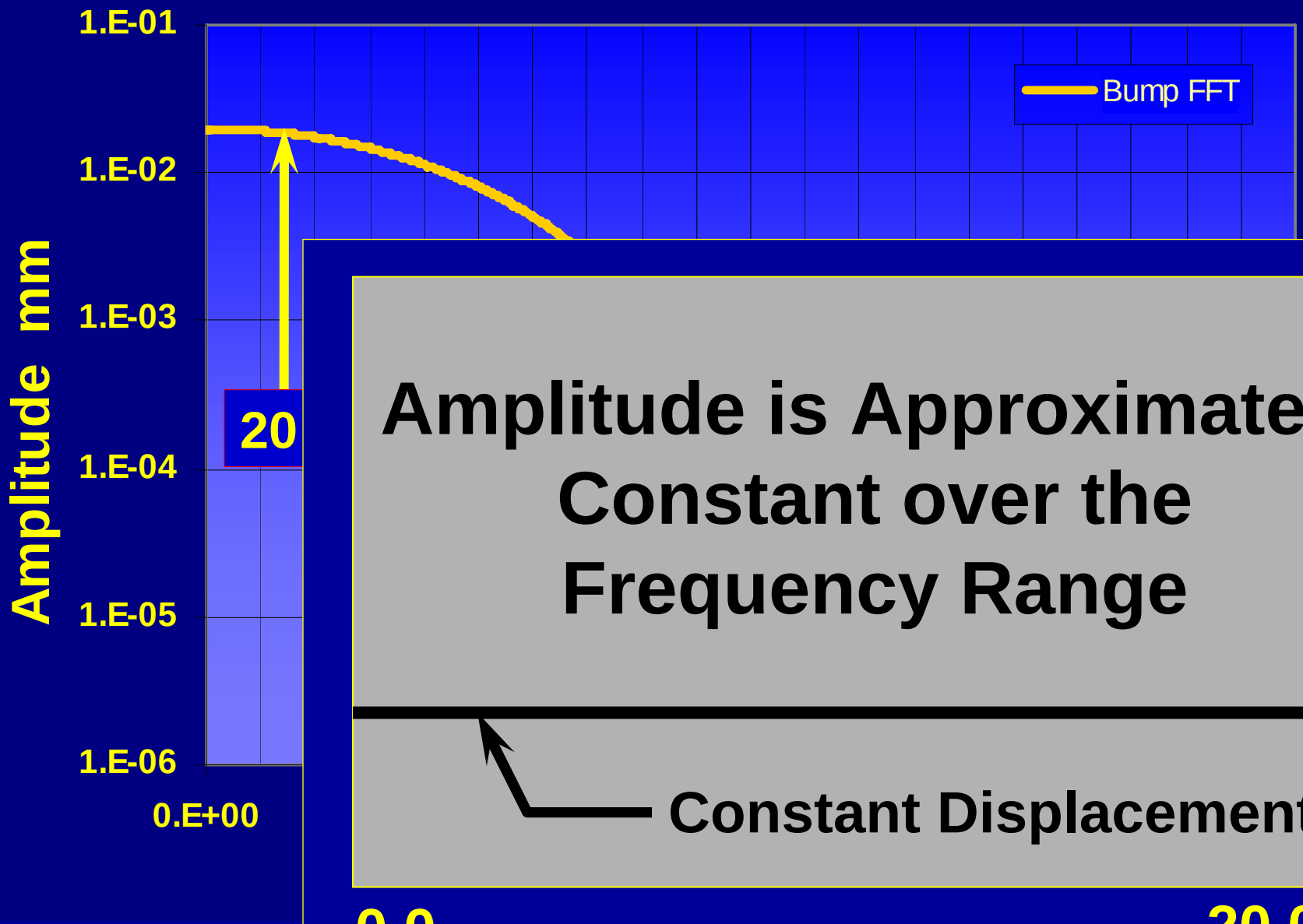
Pitch at Mid-Car DOF3

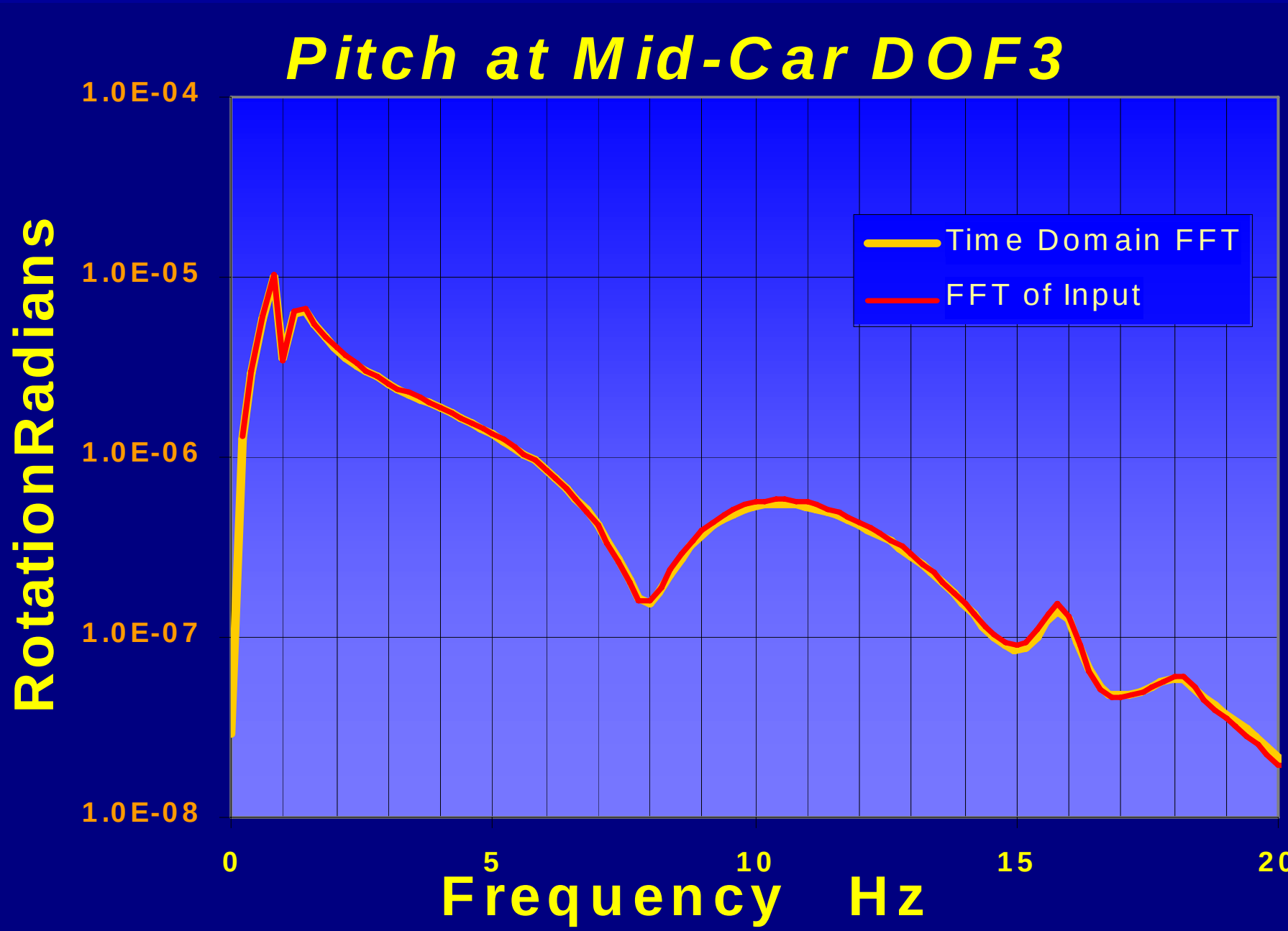



Pitch Response - Baseline Model
1.E-04
1.E-05
1.E-06
1.E-07
1.E-08
Rotation Radians
Base Model
0.0
5.0
10.0
15.0
20.0
Frequency Hz

Transform Input Force to F(f)

FFT of the Input Bump




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

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

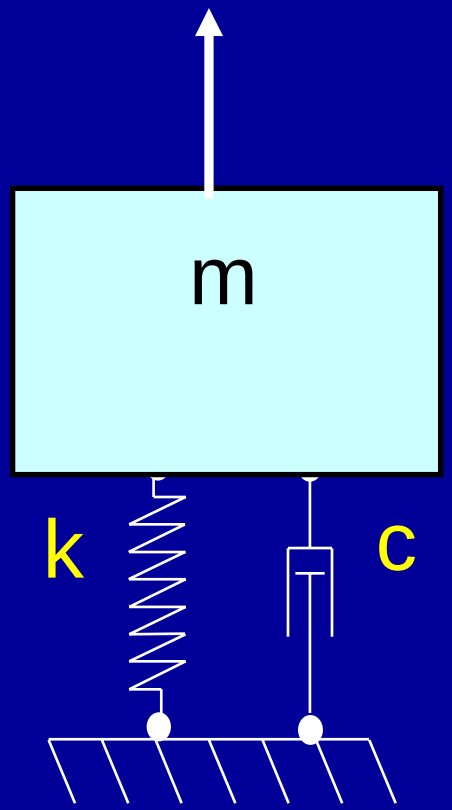
TACTILE: 50% reduction in motion

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

Single Degree of Freedom Vibration

APPLIED FORCE

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



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

Transmitted Force

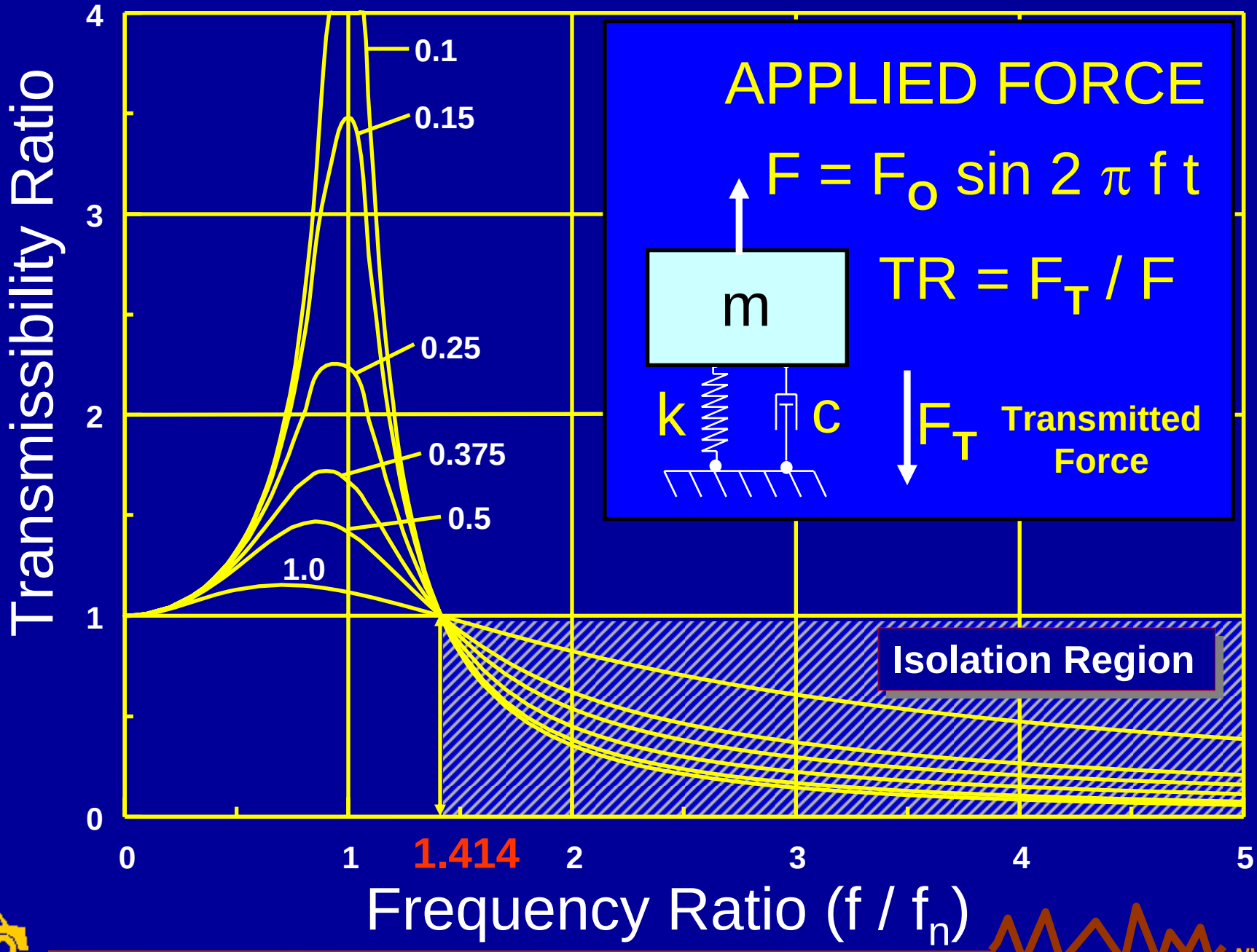
F_T

d = fraction of critical damping

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

f = operating frequency

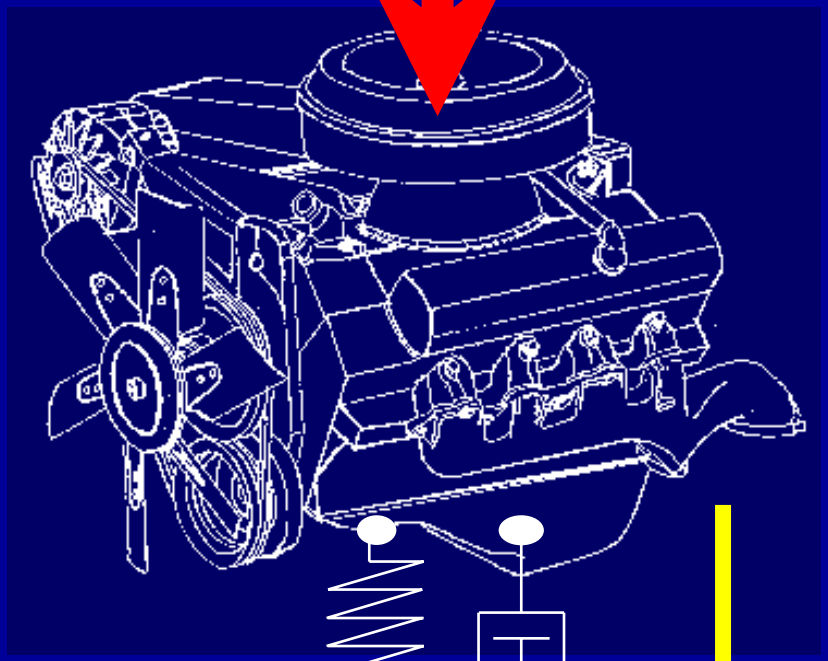
Vibration Isolation Principle



Isolation from an Applied Force

Excitation Force Coming from Engine

F_0



F_T

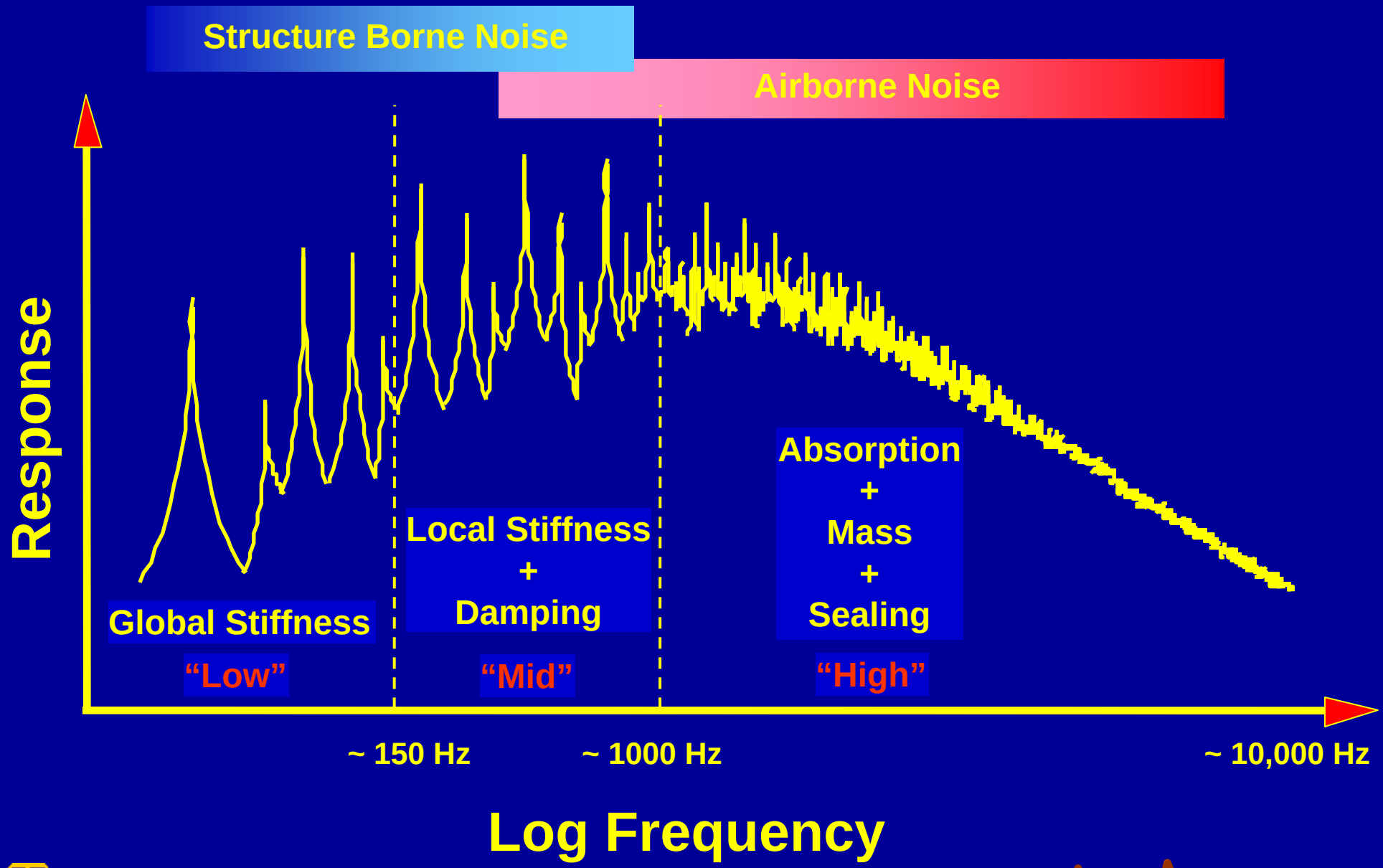
Support Forces Transmitted to Body

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.

Automotive NVH Frequency Range



NVH Workshop Outline

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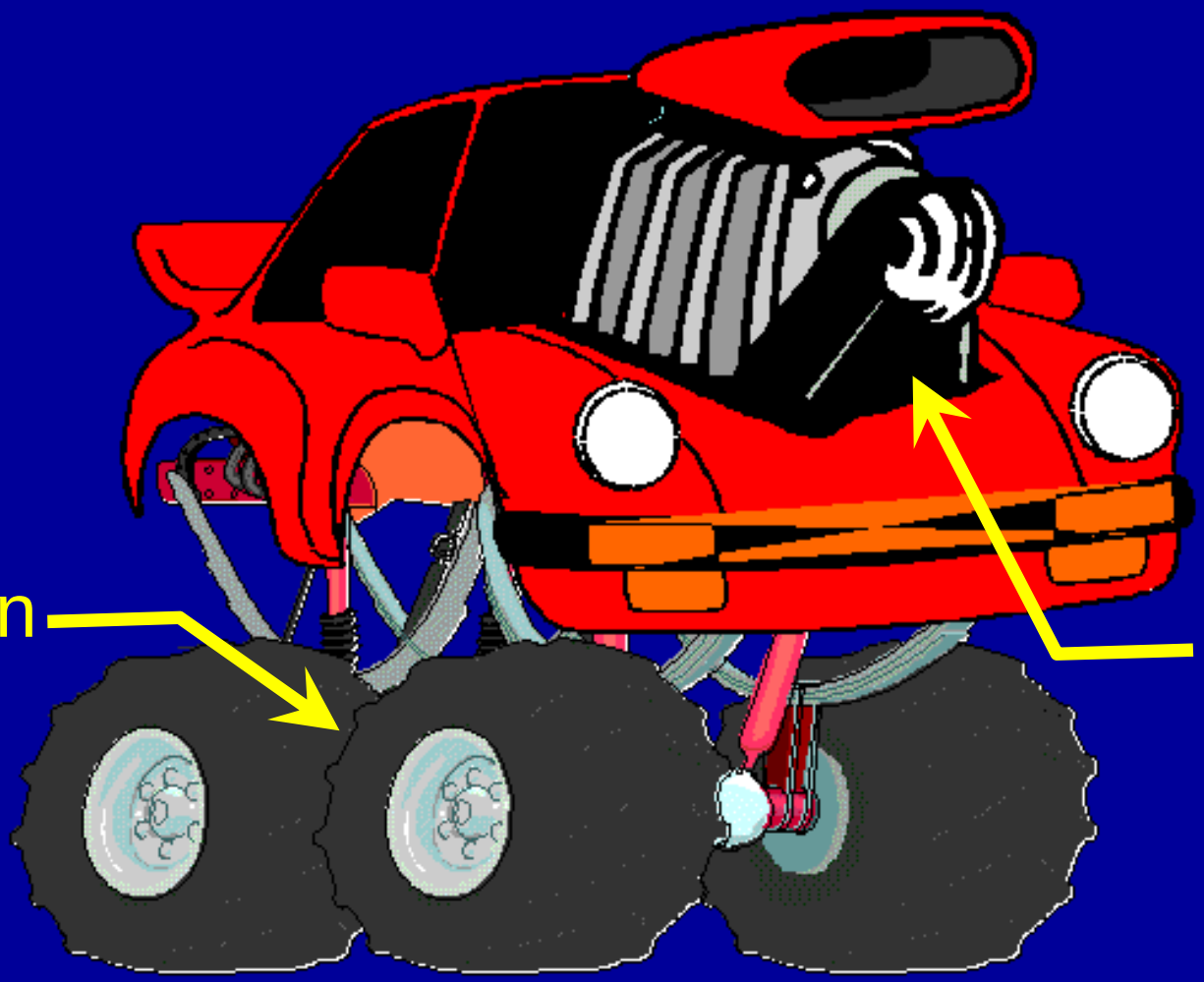
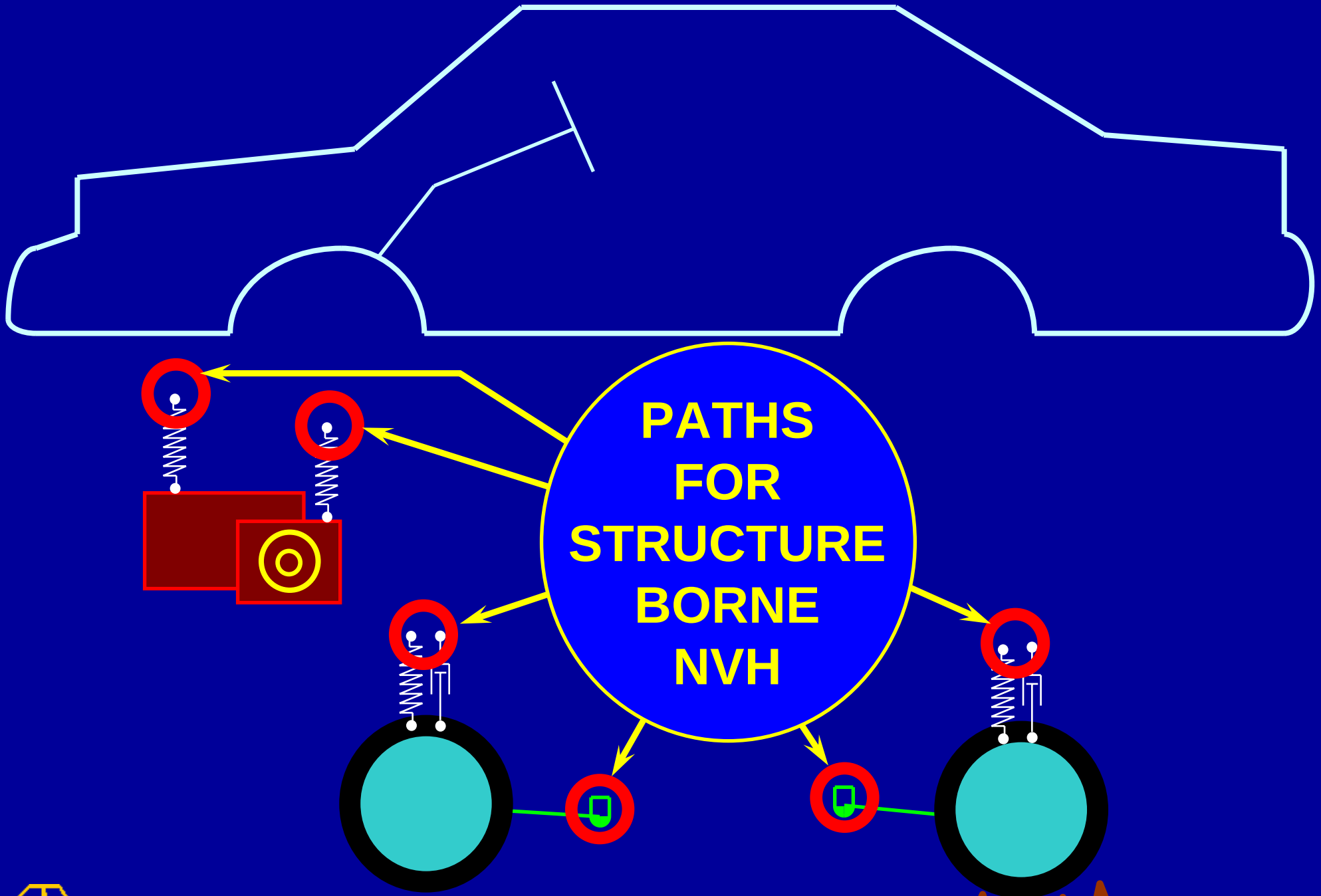
- Fundamentals in NVH
- **NVH Load Conditions**
- Low Frequency Basics

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
Powertrain 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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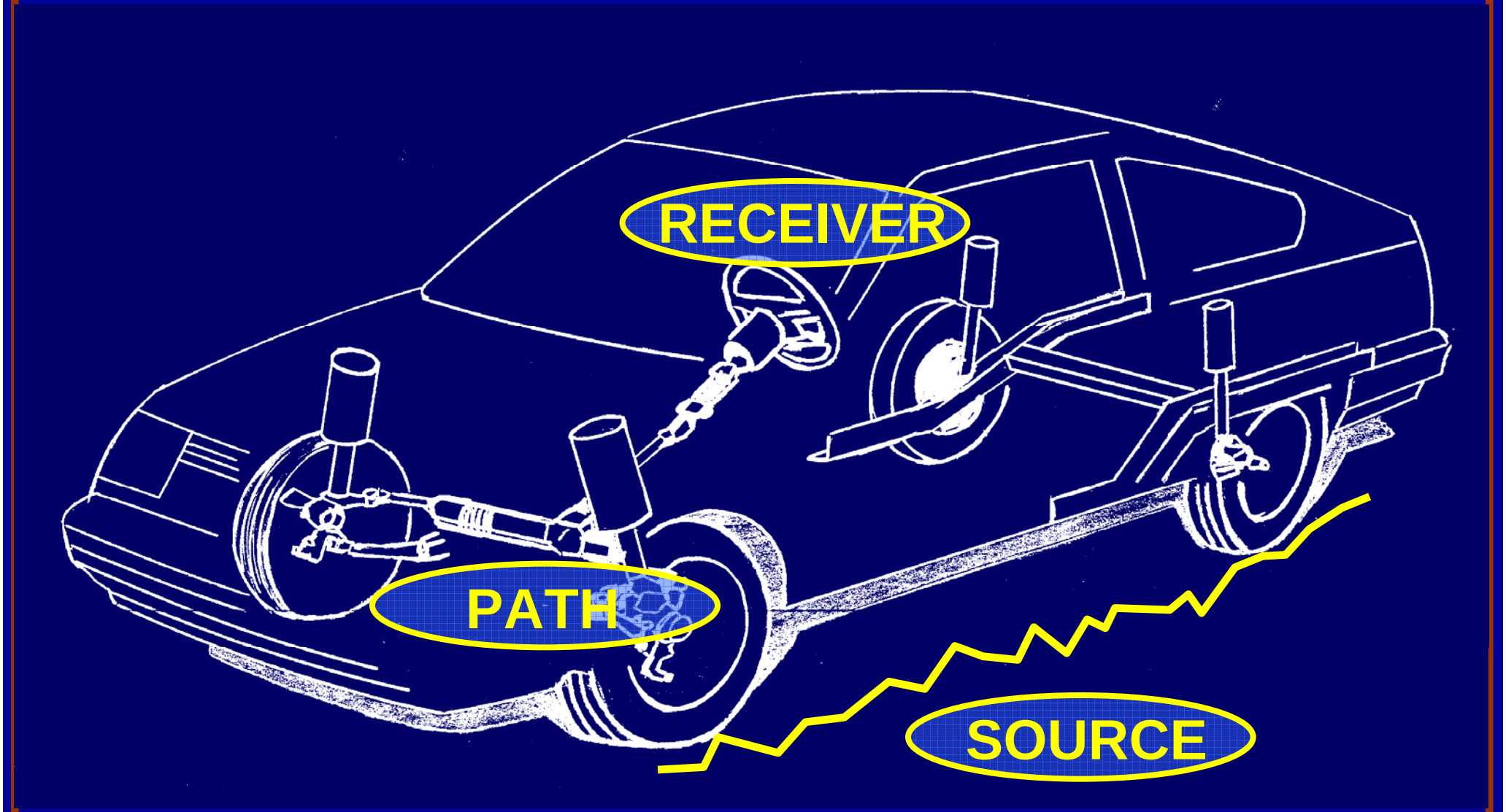
Low Frequency NVH Fundamentals

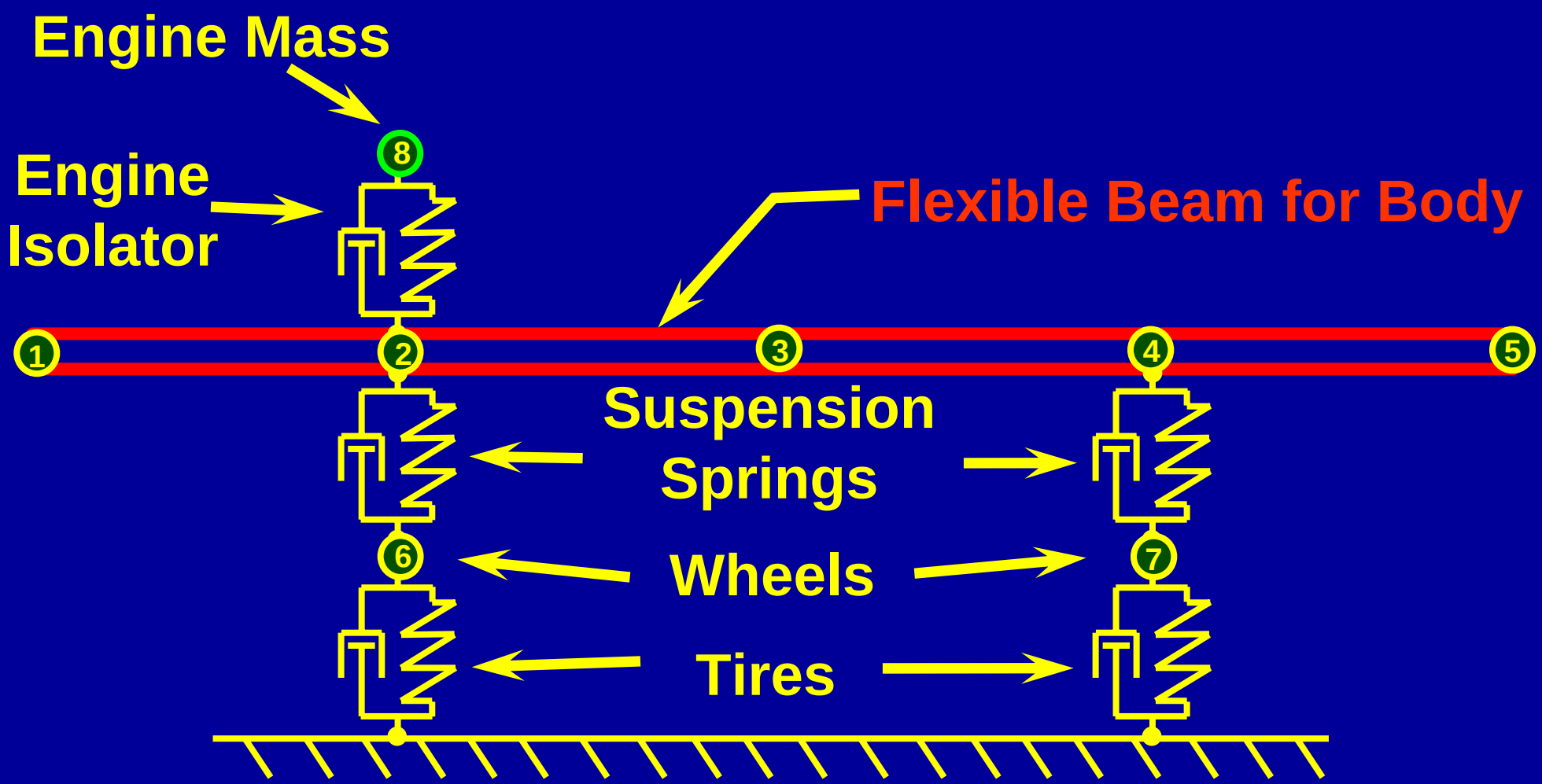


Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

8 Degree of Freedom Vehicle NVH Model



Vibration and Noise Attenuation Methods

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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.

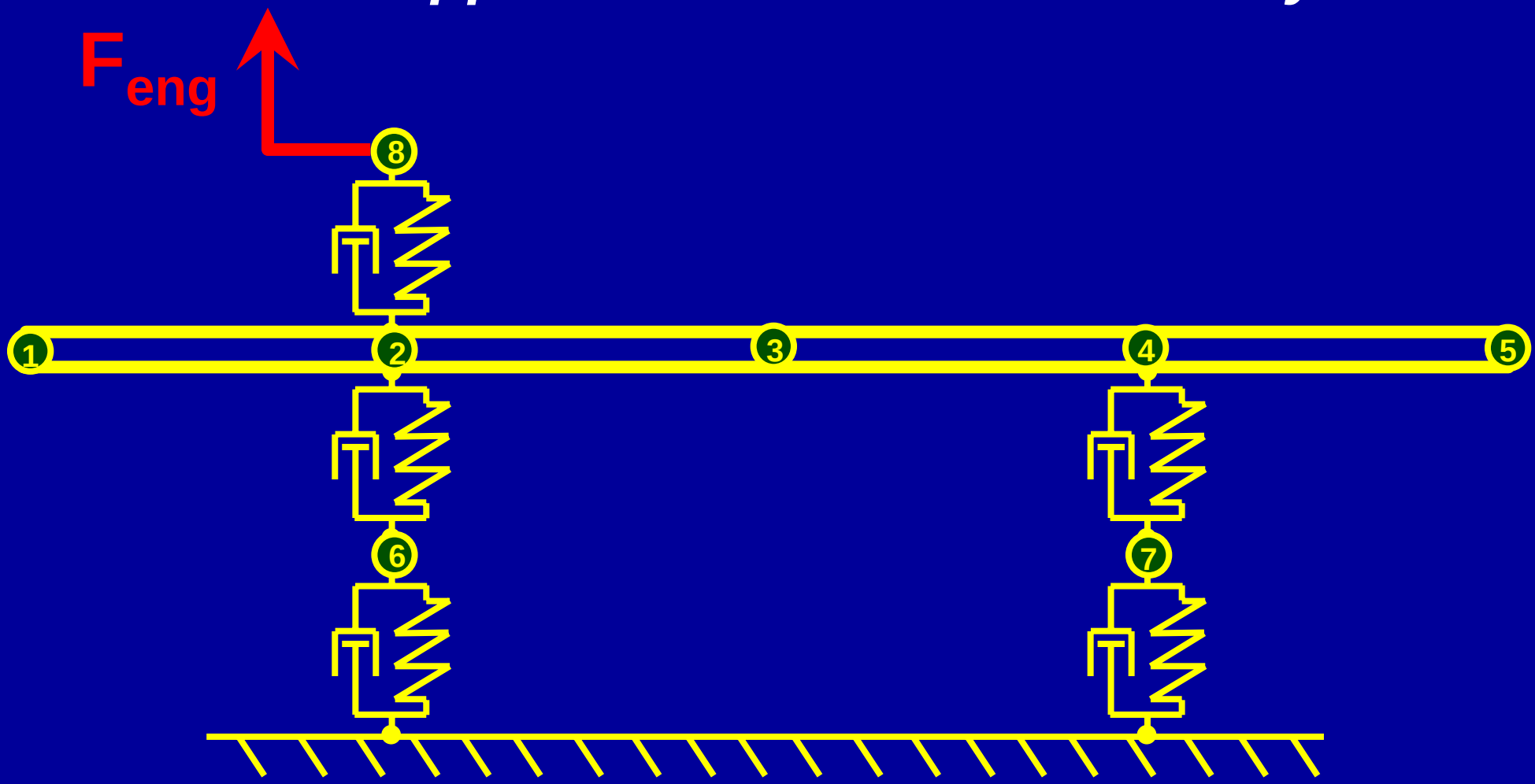
Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

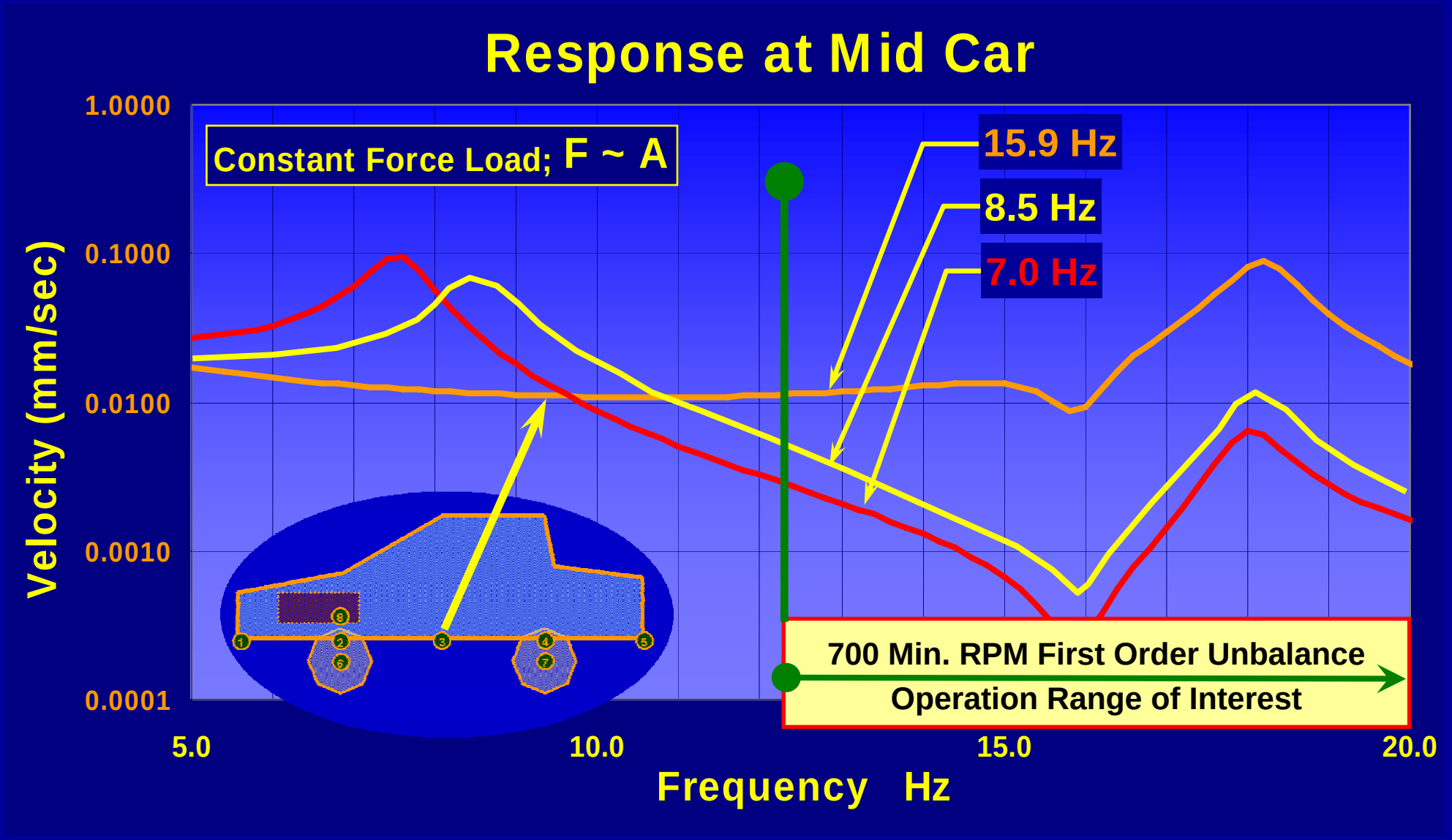
8 Degree of Freedom Vehicle NVH Model

Force Applied to Powertrain Assembly



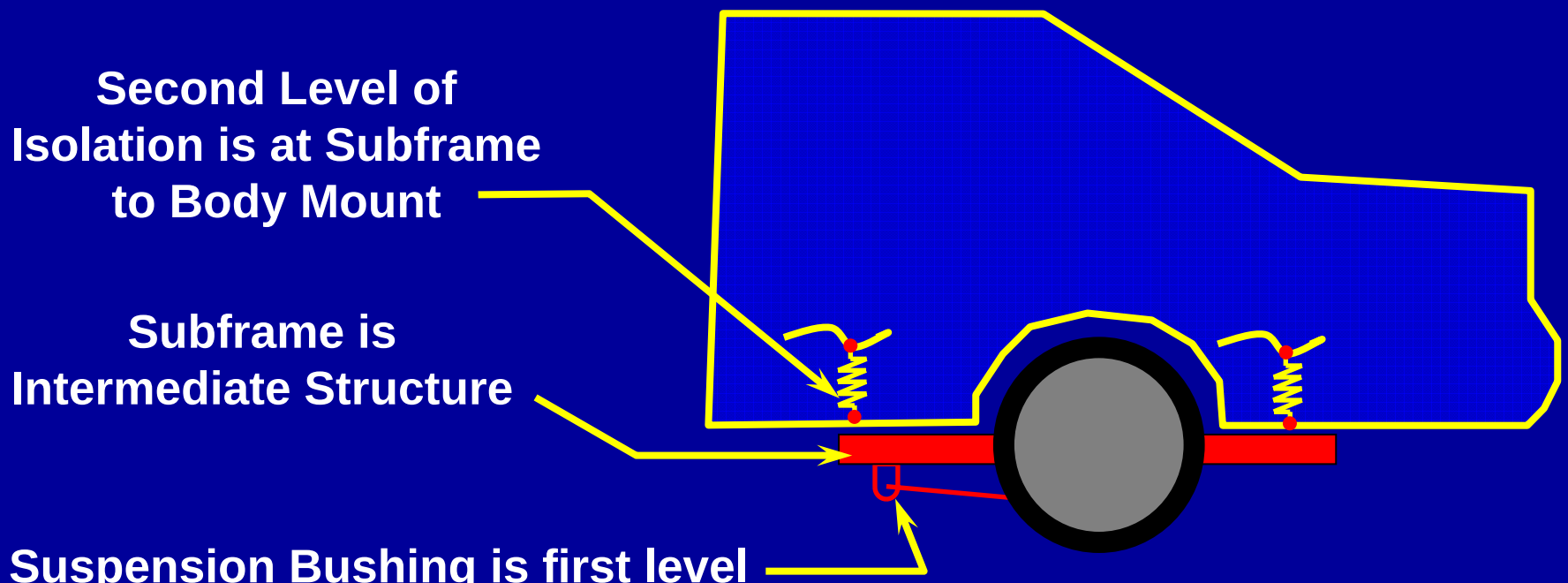
Forces at Powertrain could represent a First Order Rotating Imbalance

Engine Isolation Example



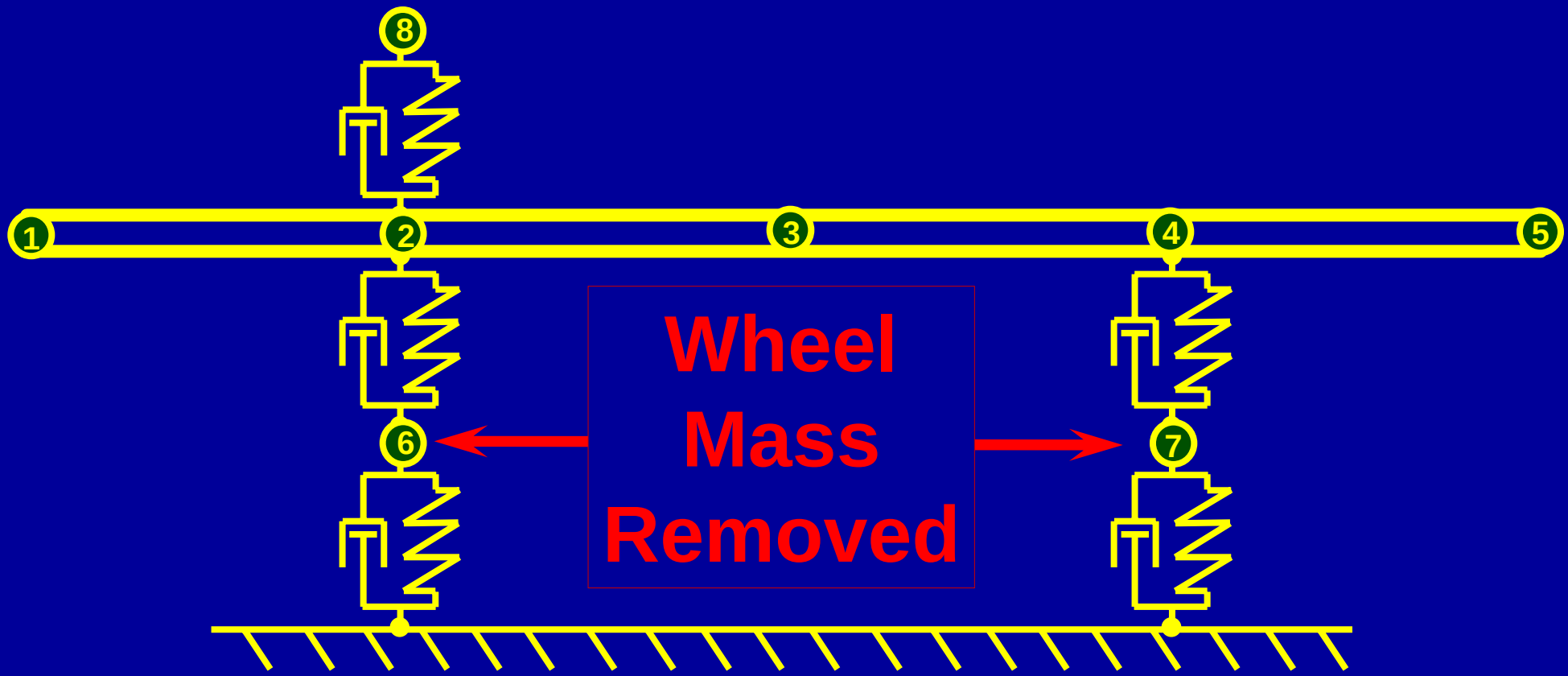
Concepts for Increased Isolation

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



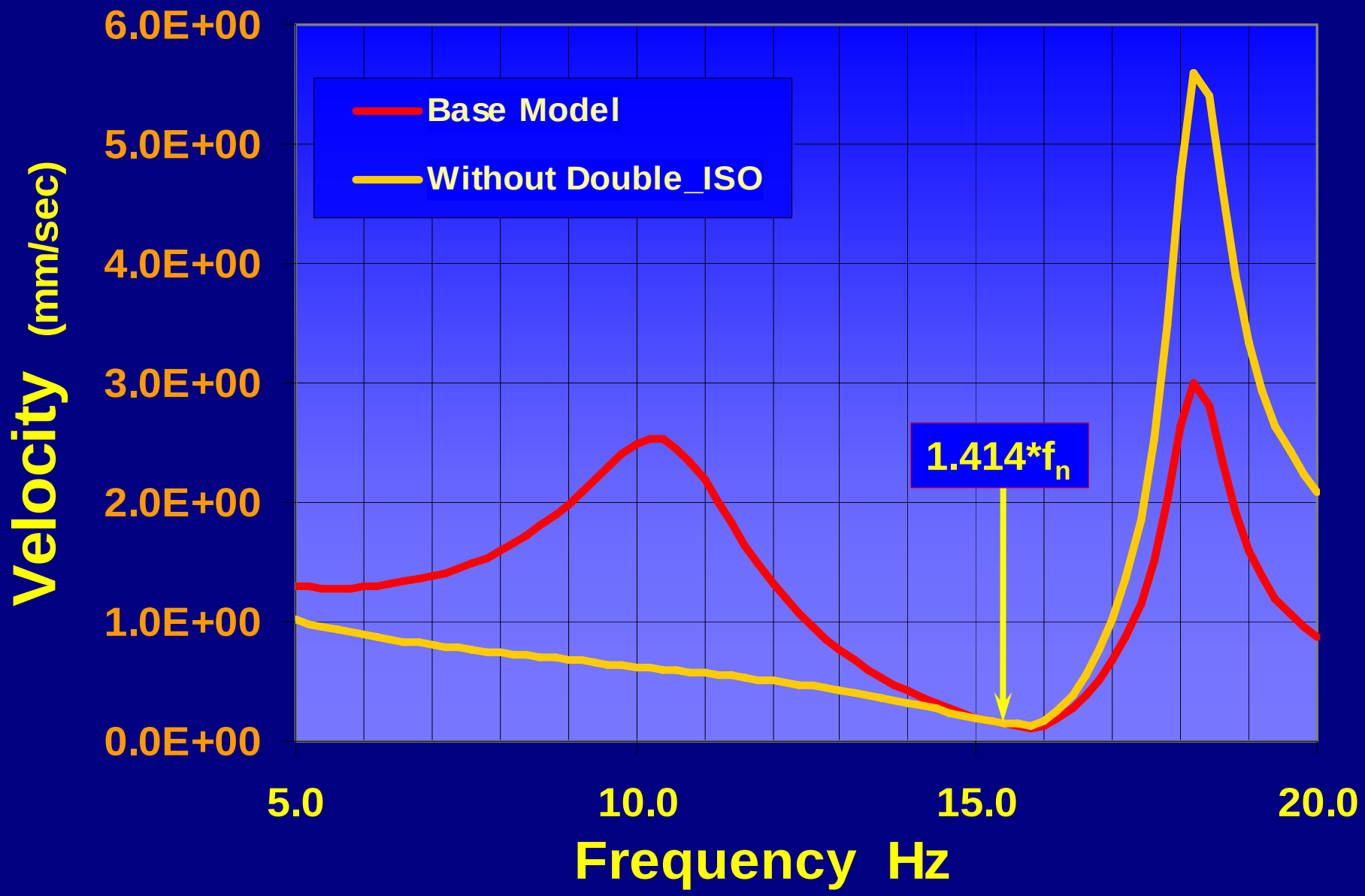
8 Degree of Freedom Vehicle NVH Model

Removed Double Isolation Effect



Double Isolation Example

Vertical Response at DOF3



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

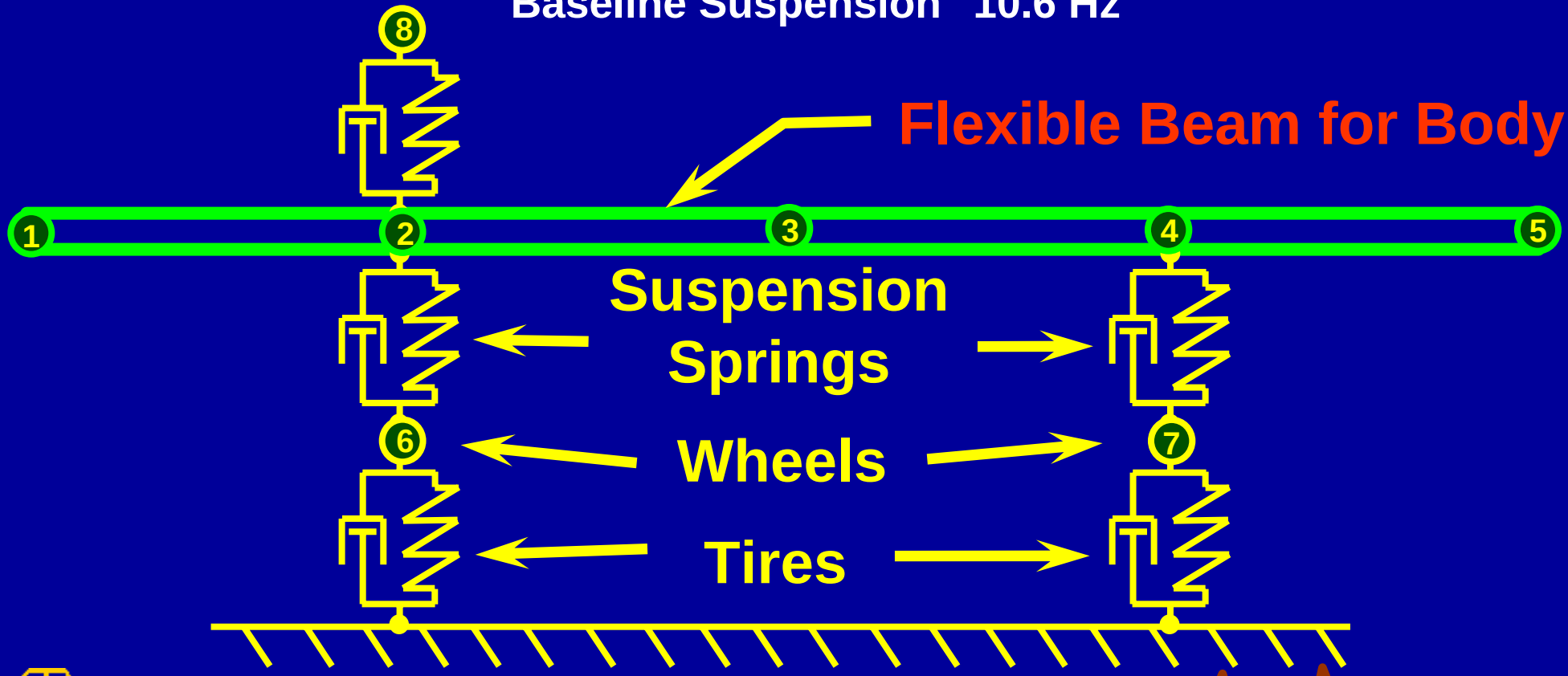
- Provide Separation between:
  - Critical modes of Sub-systems in Vehicle (e.g. Body, Suspension, Powertrain, etc.)
  - Critical modes of Sub-systems and Excitation

Need for Mode Management

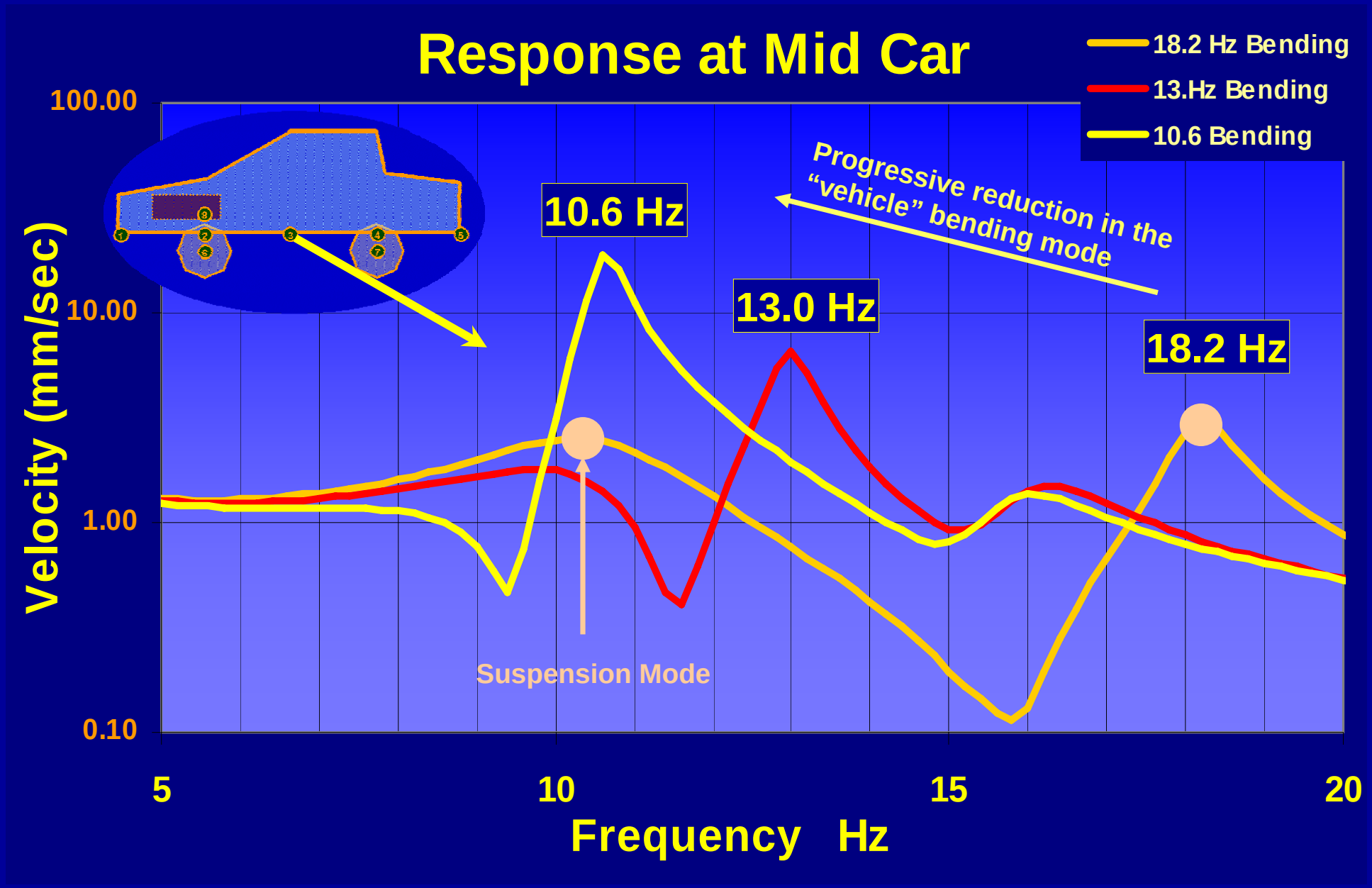
Beam Stiffness which represents the body stiffness was adjusted to align Bending Frequency with Suspension Modes and then progressively separated back to Baseline.

Baseline Bending 18.2 Hz

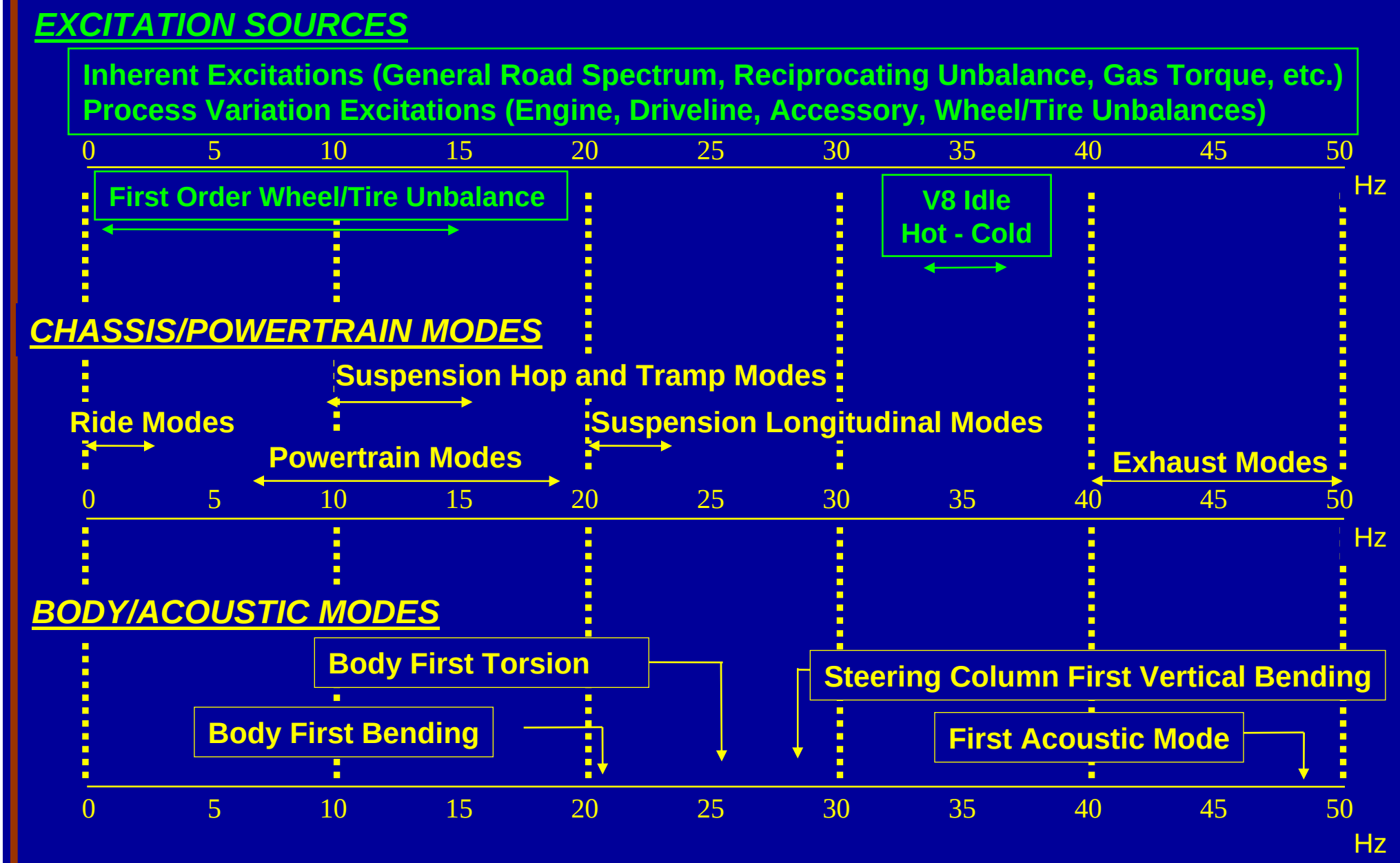
Baseline Suspension 10.6 Hz



8 DOF Mode Separation Example



Mode Management Chart

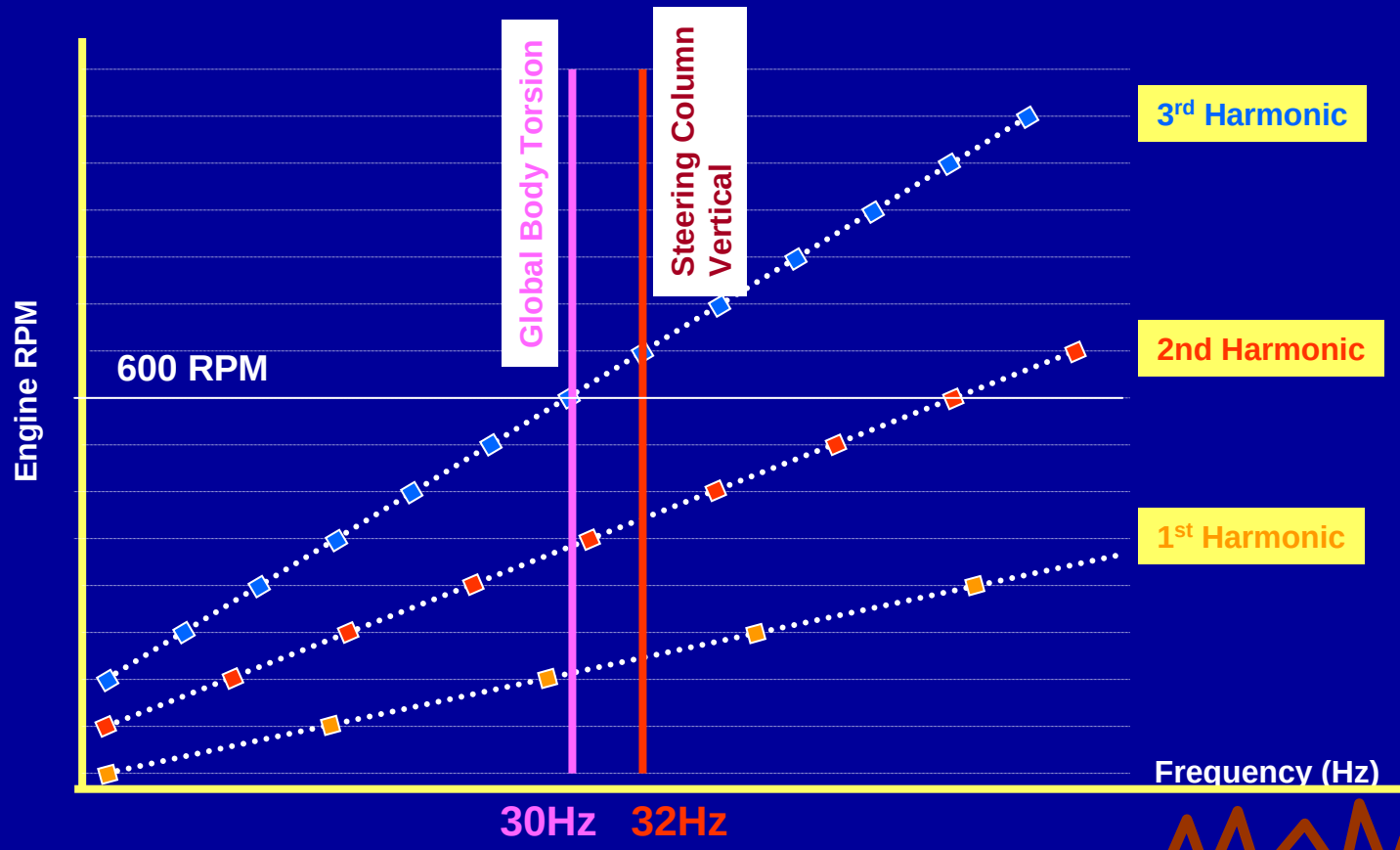


(See Ref. 1)

Campbell Diagram

- **Campbell Diagram simultaneously represents excitations and vehicle system modes on a RPM vs. Frequency axis system**

- **A more convenient way (compared to mode management charts) of understanding the interaction among modes and excitation and mapping onto ERPM and vehicle speed**



Vibration and Noise Attenuation Methods

Main Attenuation Strategies

- Reduce the Input Forces from the Source
- Provide Isolation
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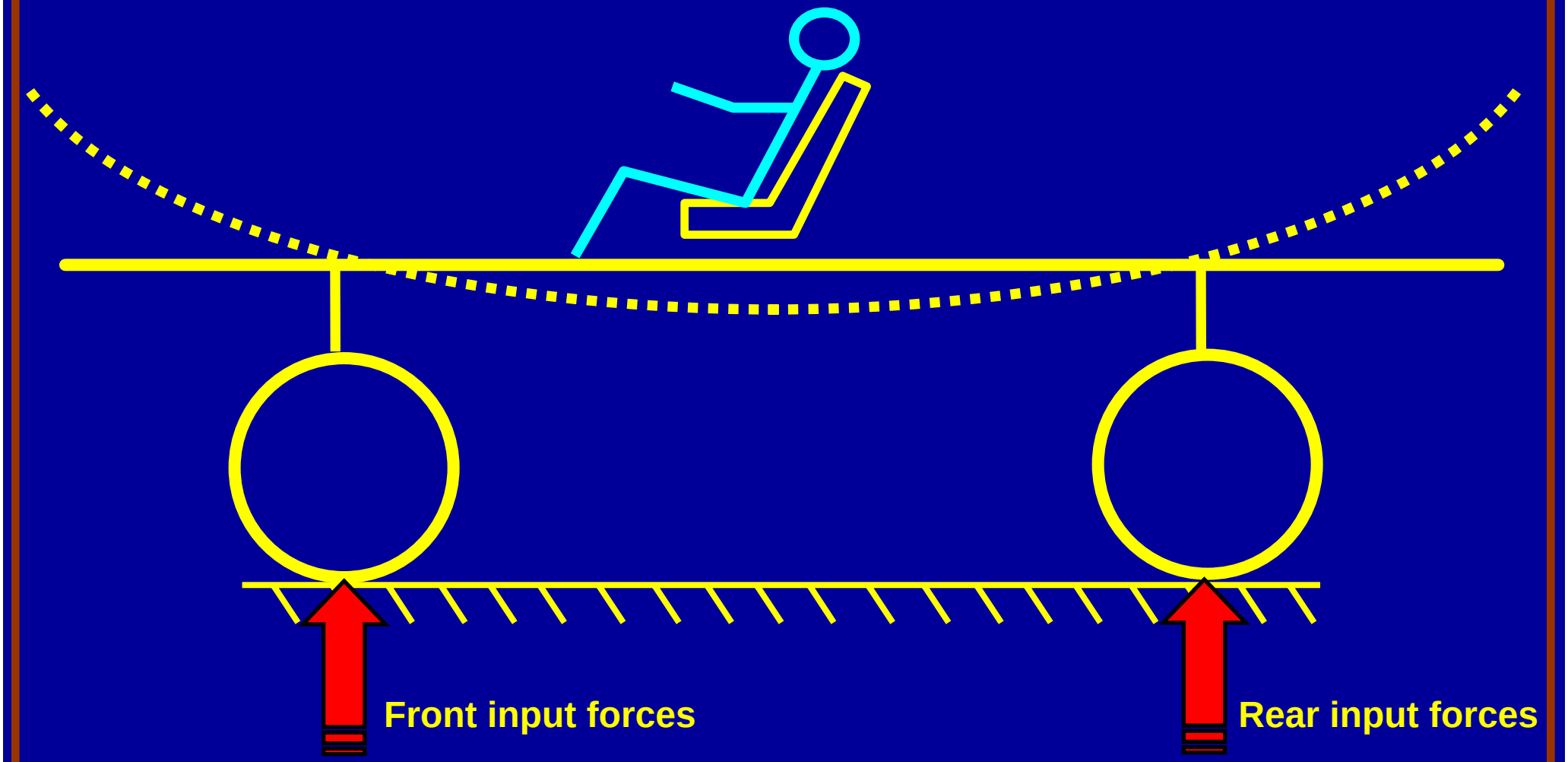
Mount at Nodal Point

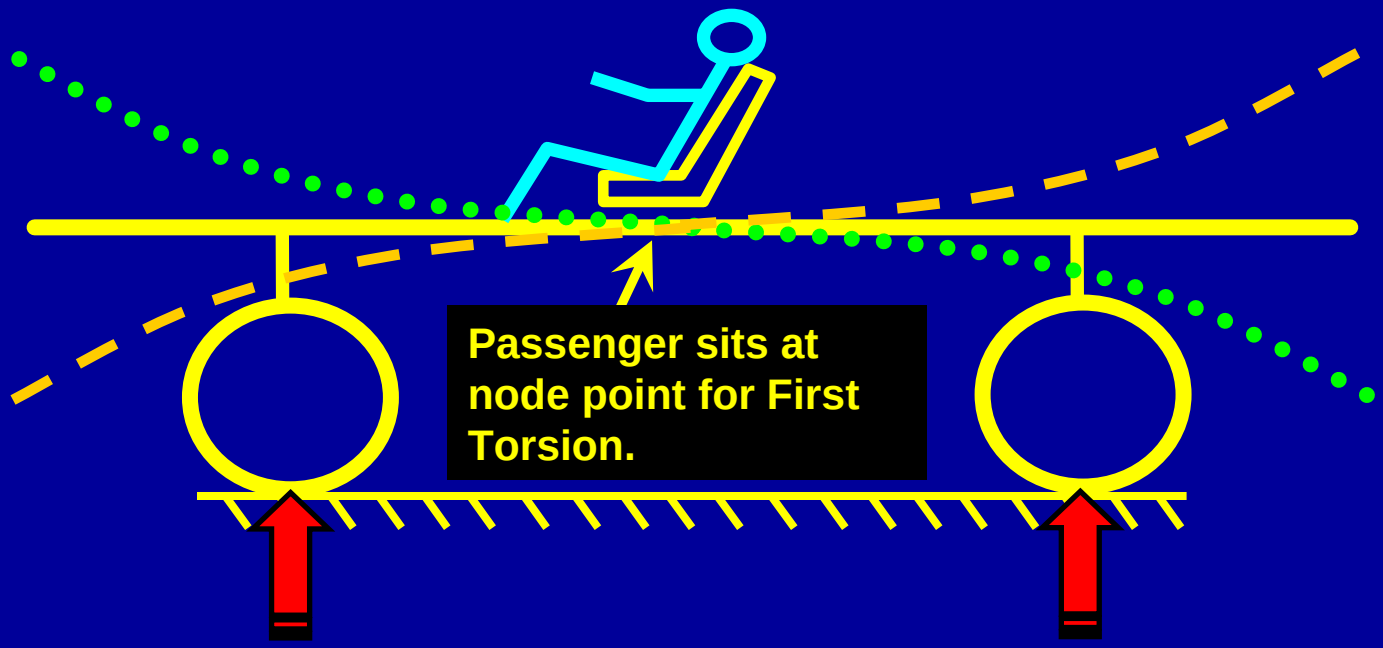
First Bending: Nodal Point Mounting Example



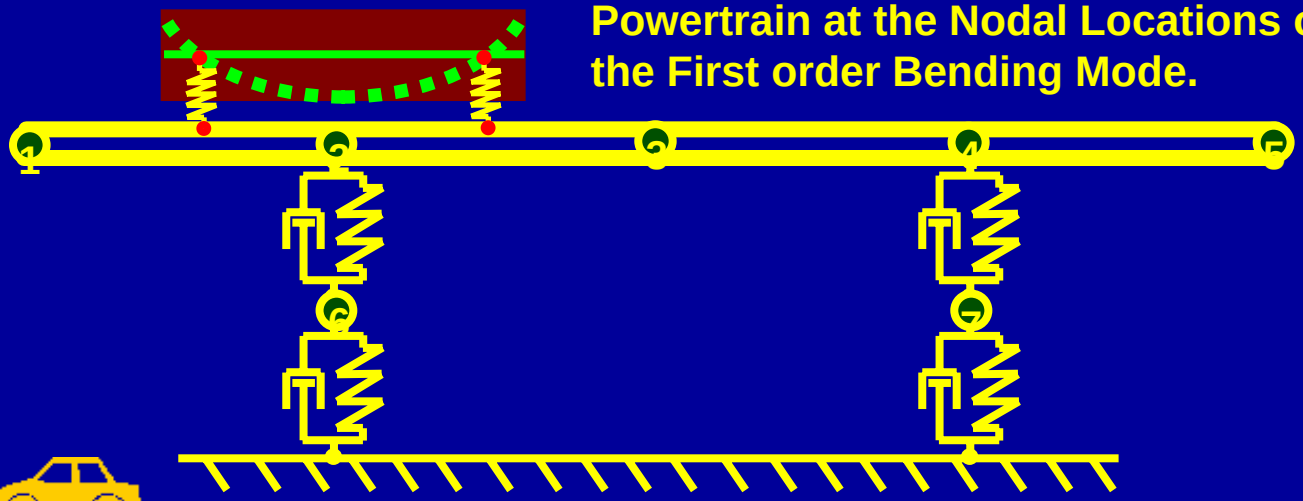
Locate wheel centers at node points of the first bending modeshape to prevent excitation coming from suspension input motion.

Mount at Nodal Point

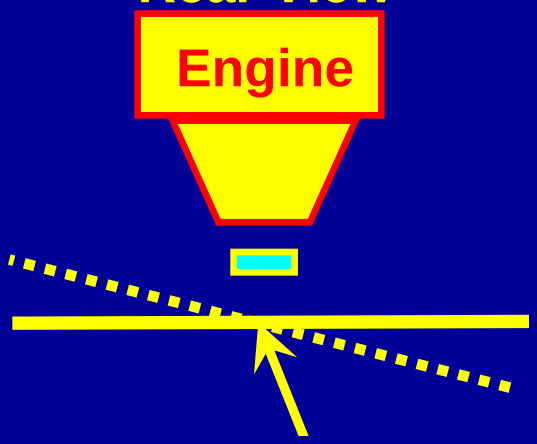
Nodal Point Mounting Examples



Mount system is placed to support Powertrain at the Nodal Locations of the First order Bending Mode.



Rear View

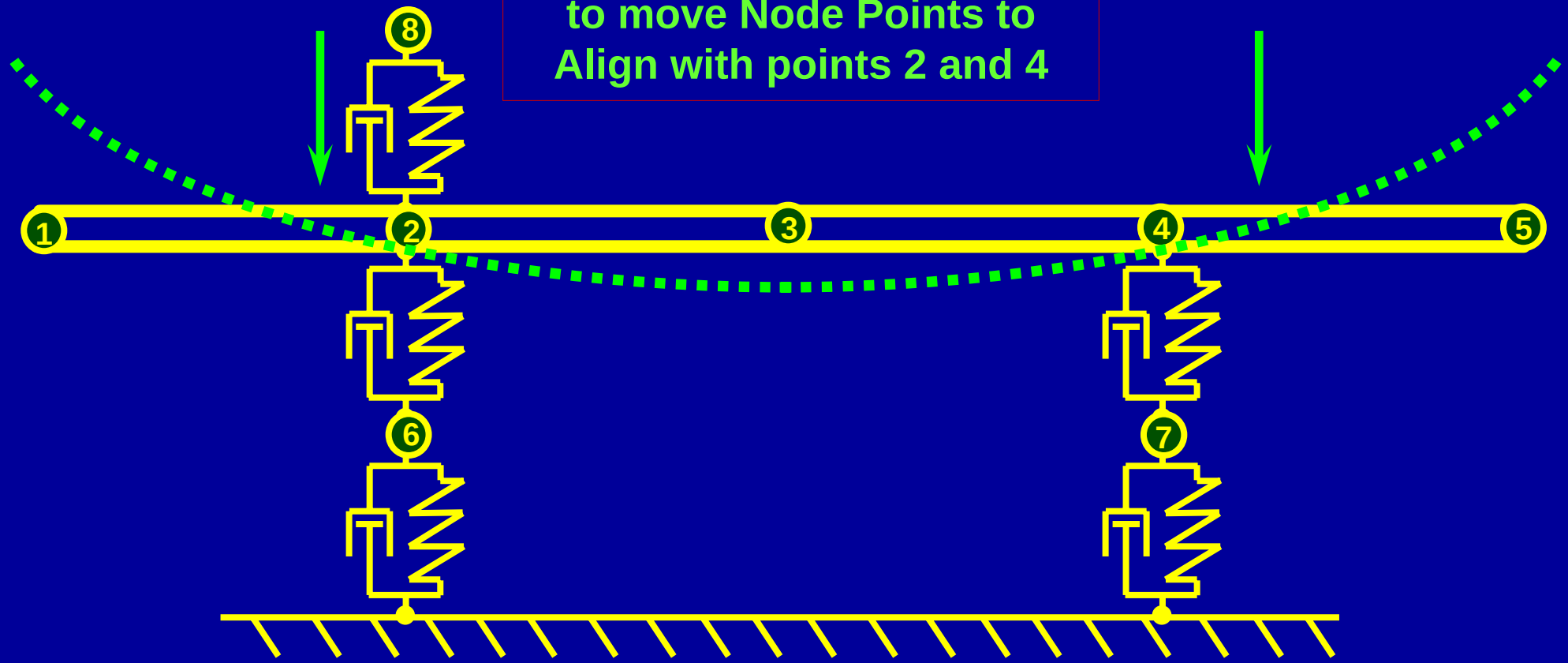


Transmission Mount of a 3 Mount N-S P/T is near the Torsion Node.

8 Degree of Freedom Vehicle NVH Model

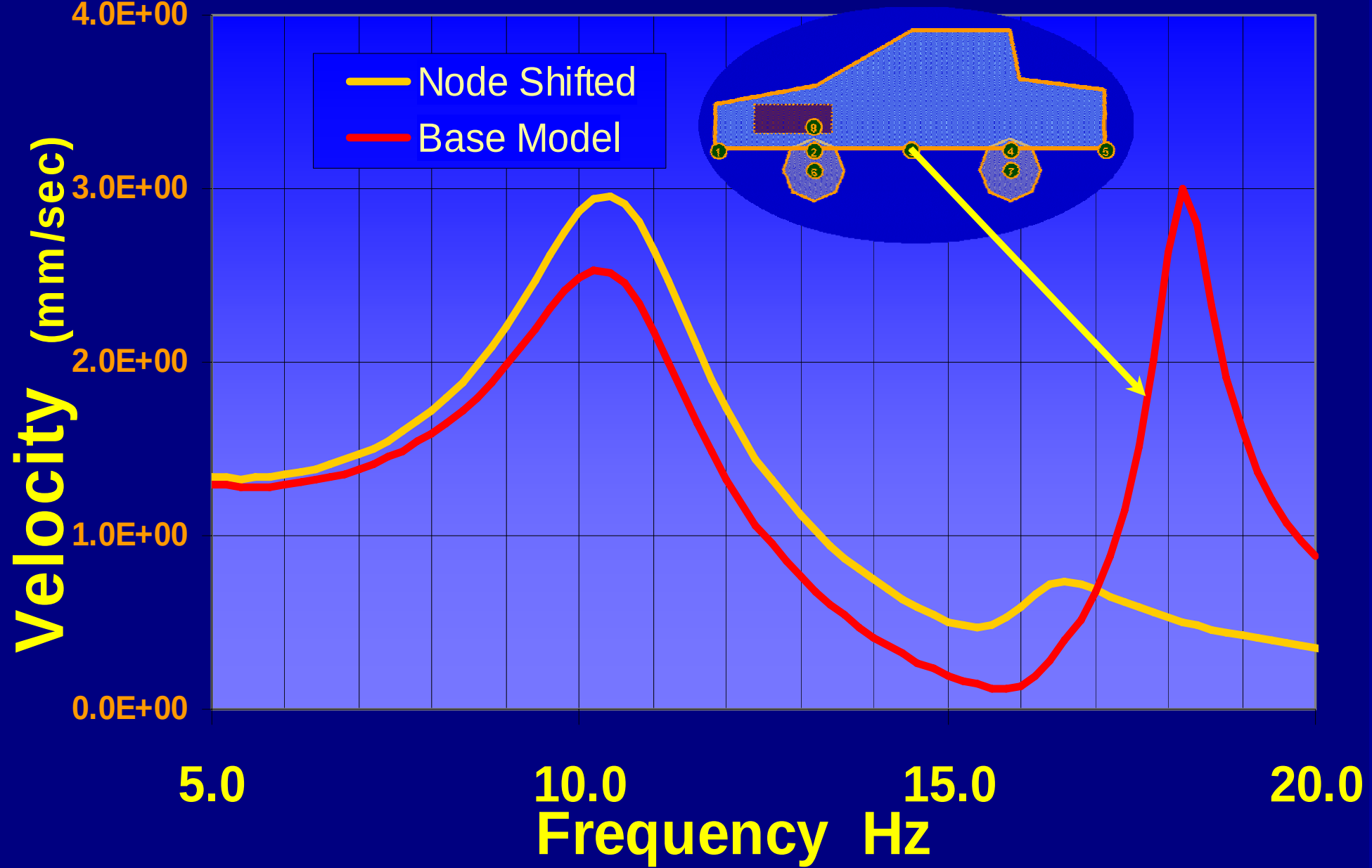
Bending Node Alignment with Wheel Centers

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



First Bending Nodal Point Alignment

Response at Mid-Car

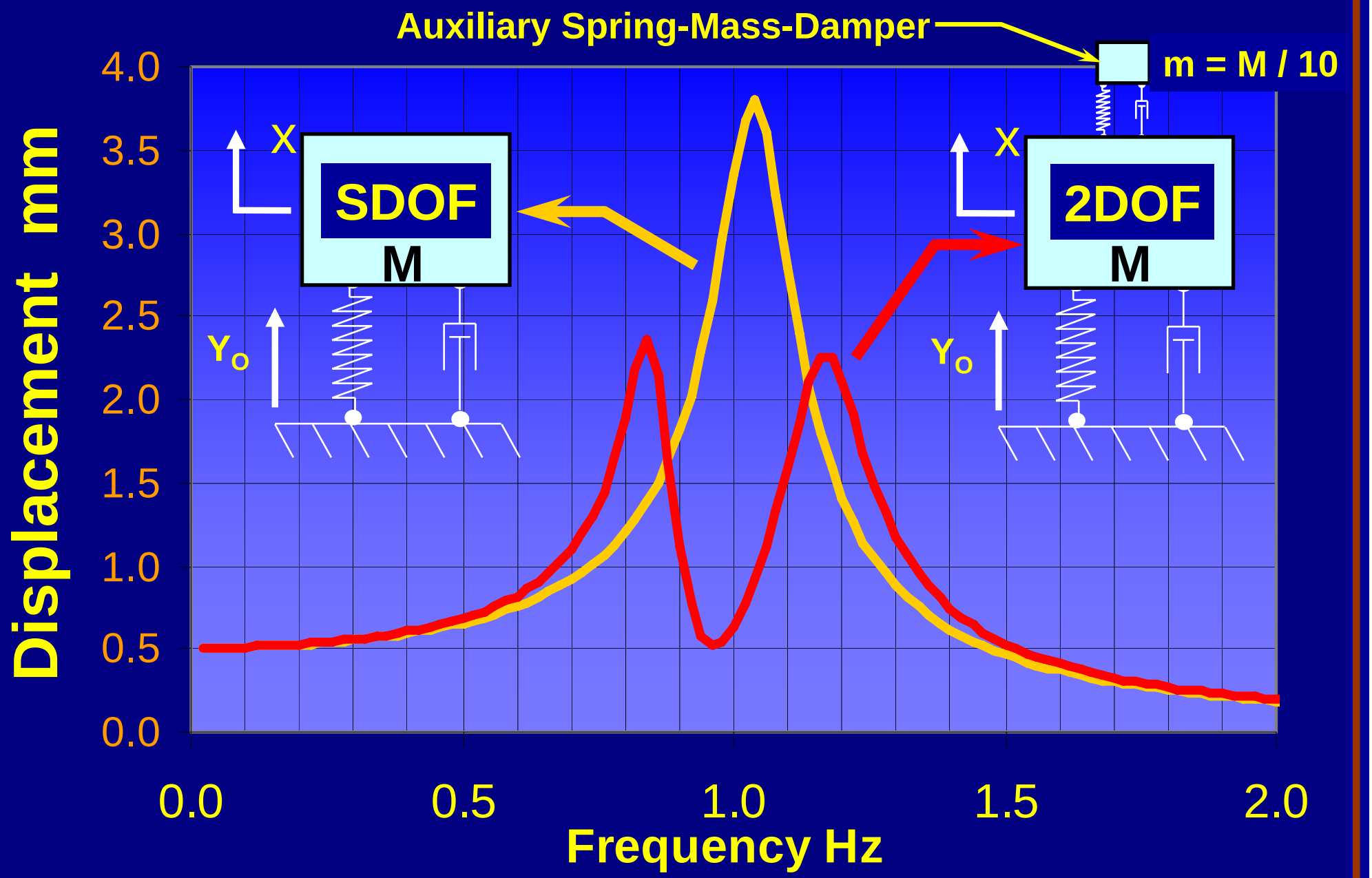


Vibration and Noise Attenuation Methods

Main Attenuation Strategies

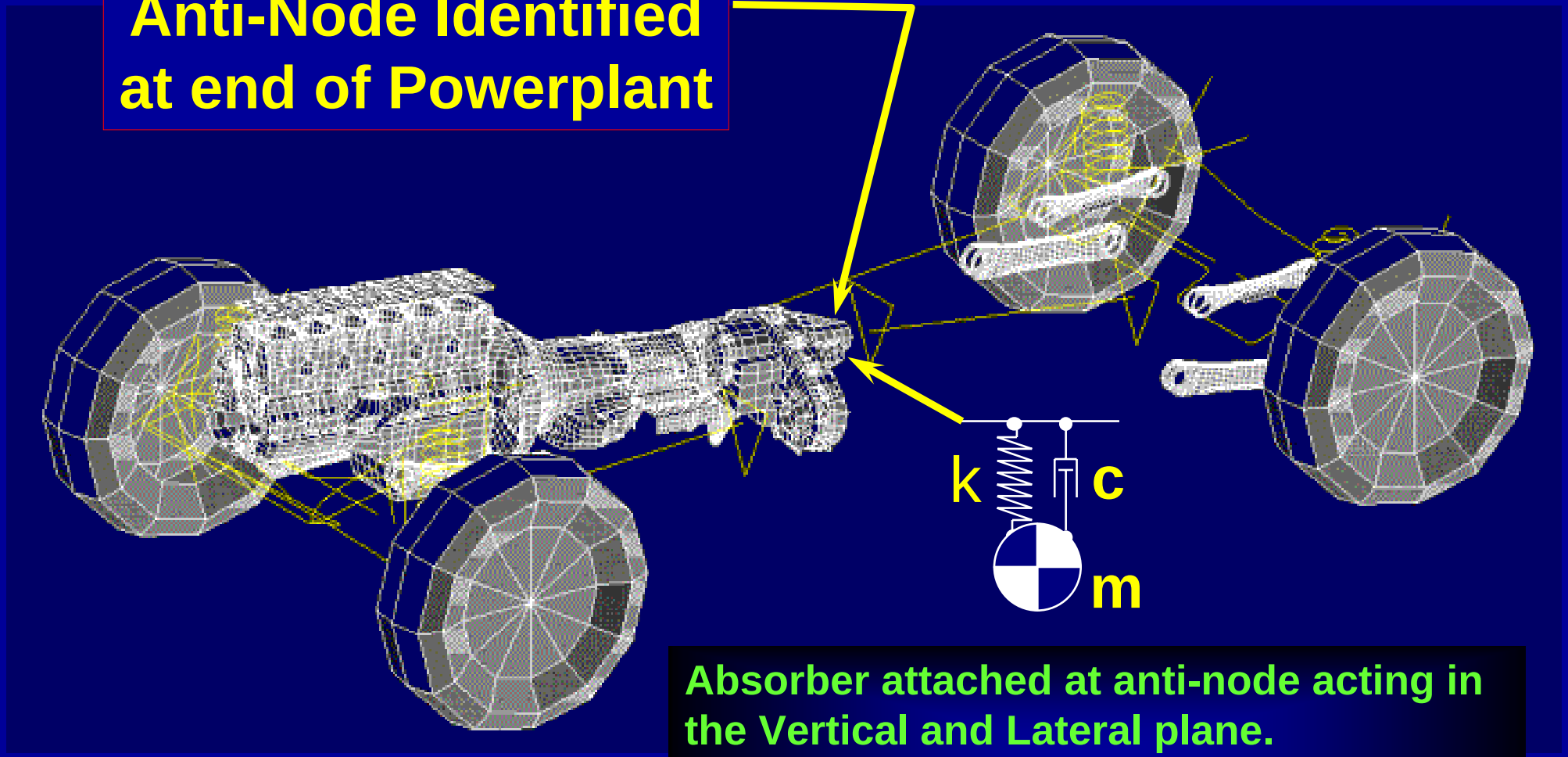
- Reduce the Input Forces from the Source
- Provide Isolation
- Mode Management
- Nodal Point Mounting
- Dynamic Absorbers

Dynamic Absorber Concept



Powertrain Example of Dynamic Absorber

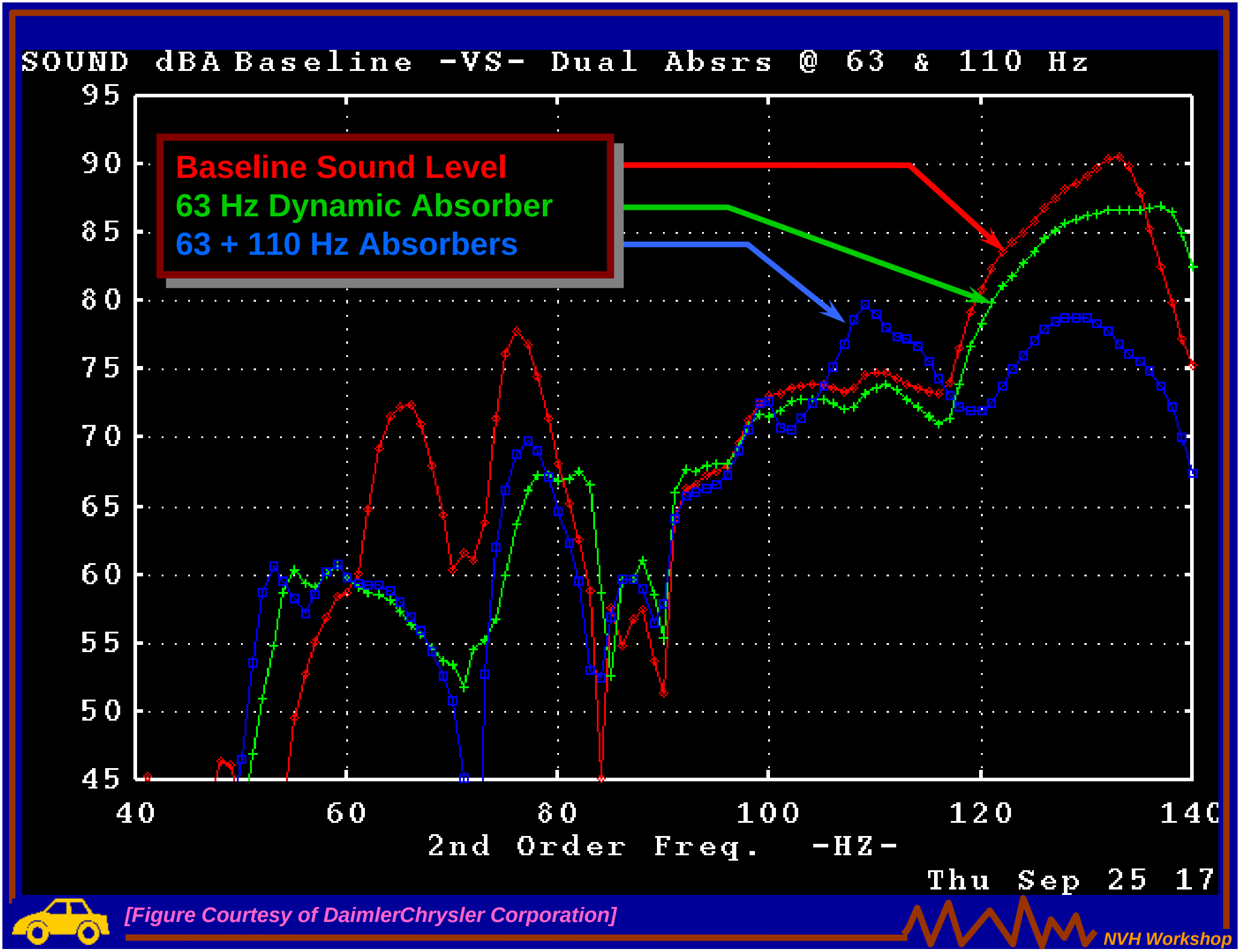
Anti-Node Identified at end of Powerplant



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

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

[Figure Courtesy of DaimlerChrysler Corporation]



[Figure Courtesy of DaimlerChrysler Corporation]

NVH Workshop

Vibration and Noise Attenuation Methods

Main Attenuation Strategies

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

The above attenuation concepts have been demonstrated using simulation model results. A similar demonstration using physical hardware would have required more cost and time. Hardware testing in low frequency range could be used as a final verification tool

NVH Workshop Topic Outline

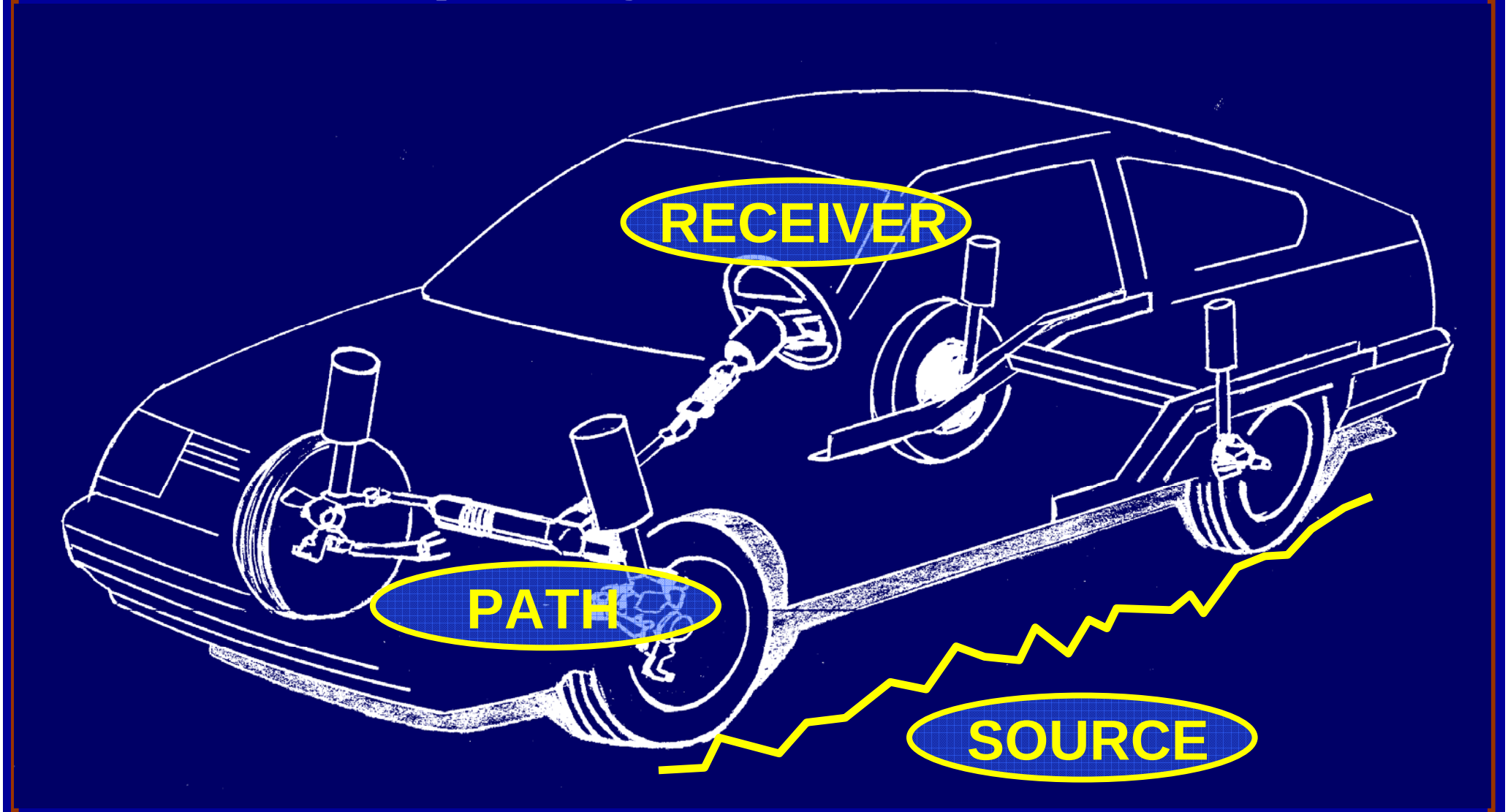
- Introduction
- Fundamentals in NVH
- Automotive NVH Load Conditions
- Low Frequency Basics
- ***Live Noise Attenuation Demo***
- Mid Frequency Basics
- Utilization of Simulation Models
- Closing Remarks

Greg Goetchius

NVH Workshop Topic Outline

- Introduction
- Fundamentals in NVH
- Automotive NVH Load Conditions
- Low Frequency Basics
- *Live Noise Attenuation Demo*
- **Mid Frequency Basics** ***Alan Duncan***
- Utilization of Simulation Models
- Closing Remarks

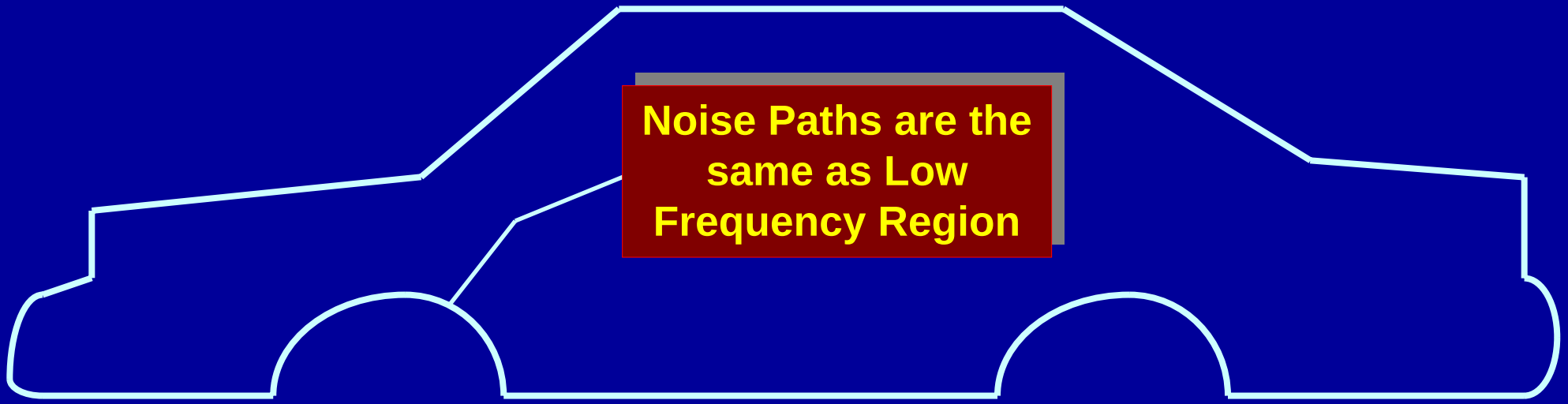
Mid Frequency NVH Fundamentals



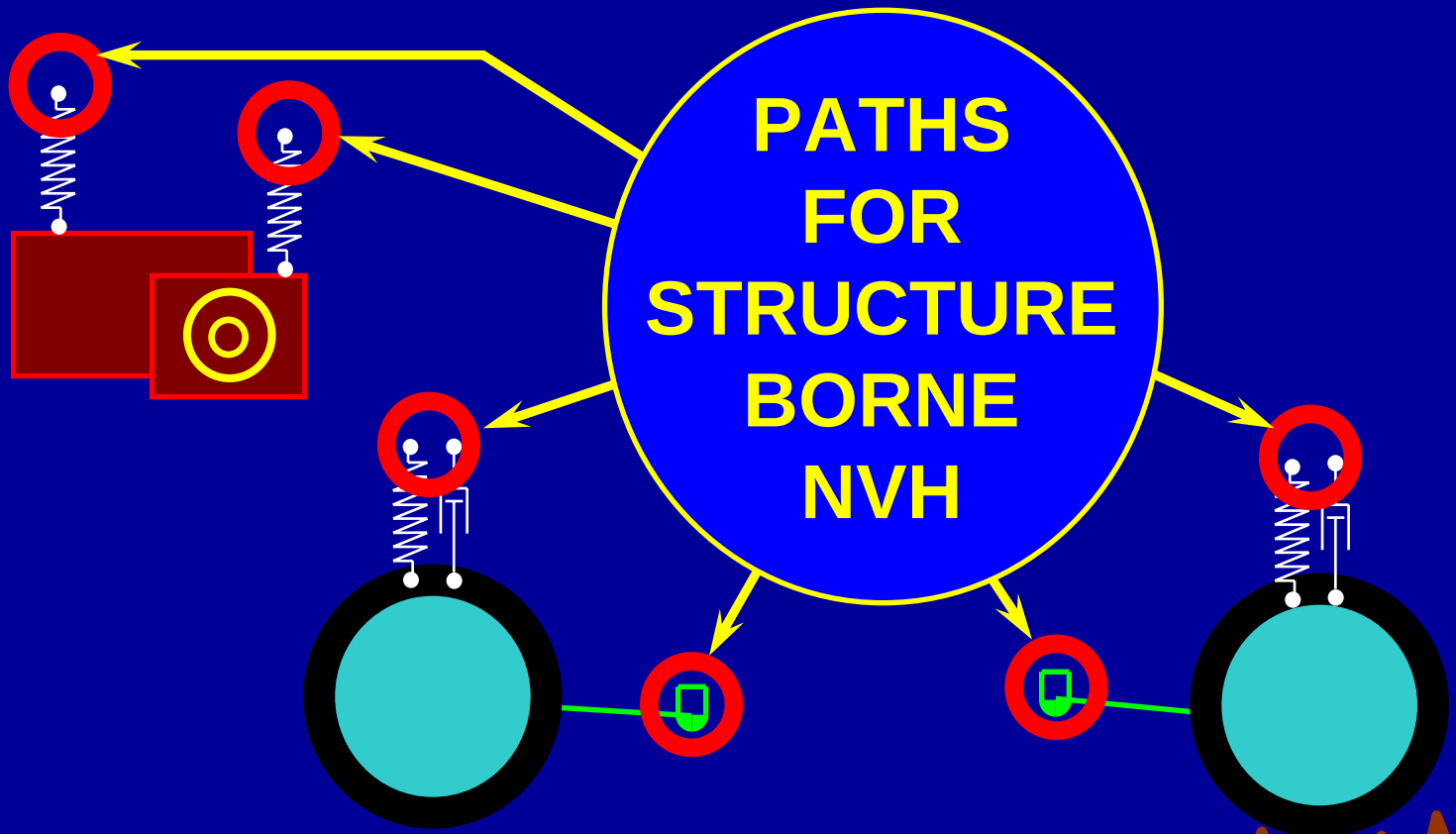
This looks familiar!

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

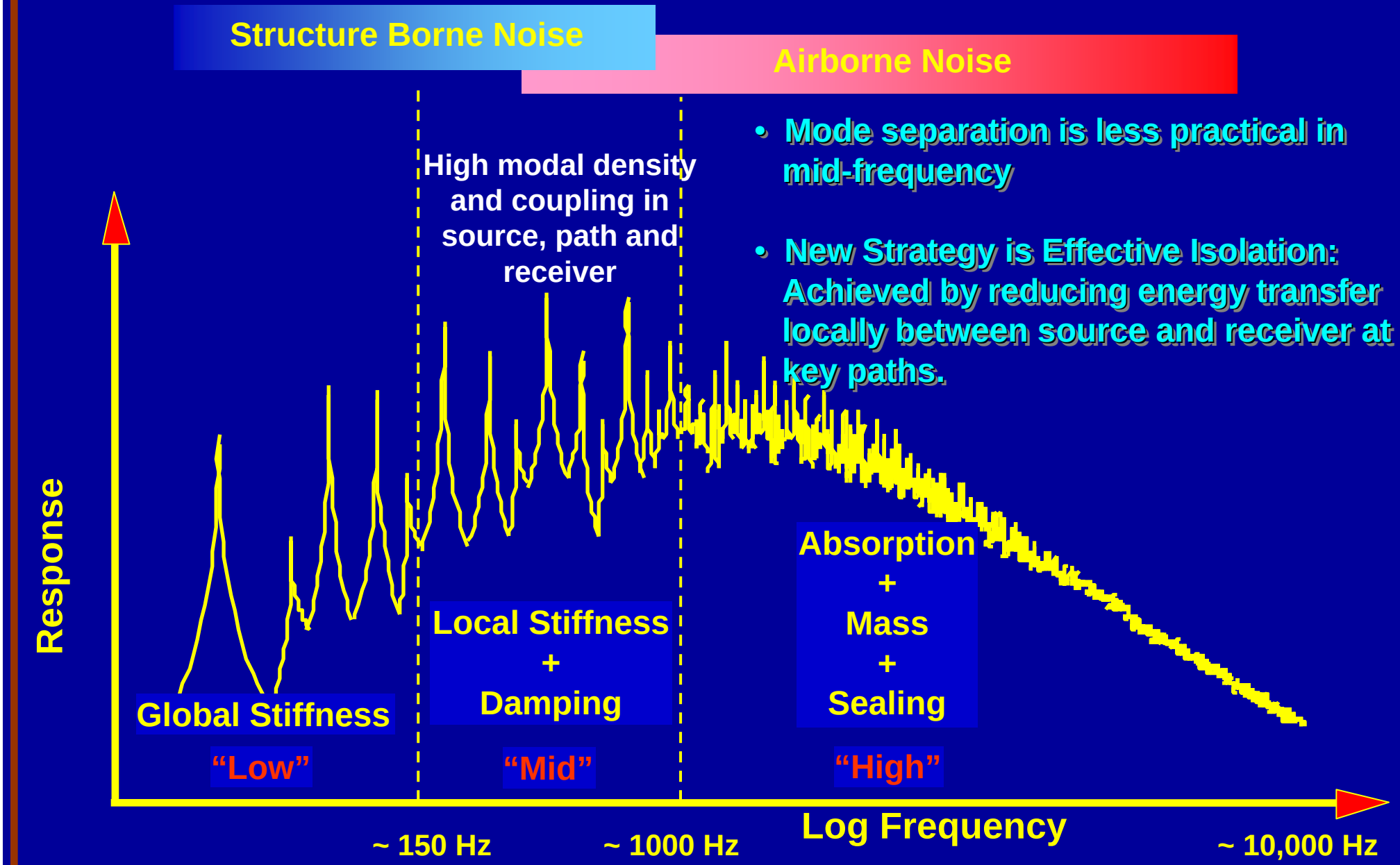
Typical NVH Pathways to the Passenger



Noise Paths are the same as Low Frequency Region



Mid-Frequency Analysis Character



- Mode separation is less practical in mid-frequency
- New Strategy is Effective Isolation: Achieved by reducing energy transfer locally between source and receiver at key paths.

Mid-Frequency Analysis Character

- **Important characteristics of mid frequency analysis**

Effective Isolation

&

Identifying Key Noise Paths

Mid-Frequency Analysis Character

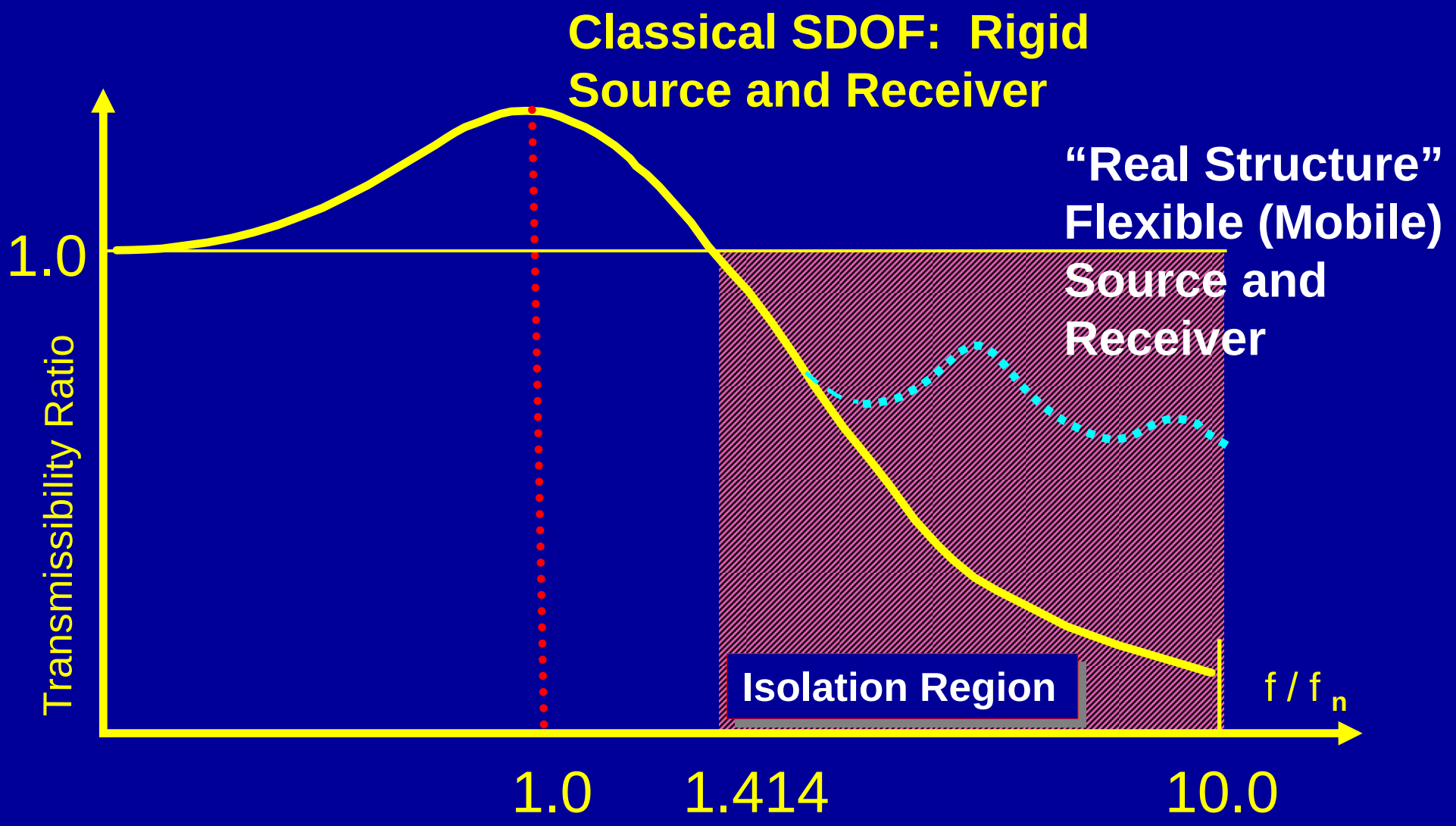
- **Important characteristics of mid frequency analysis**

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

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

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

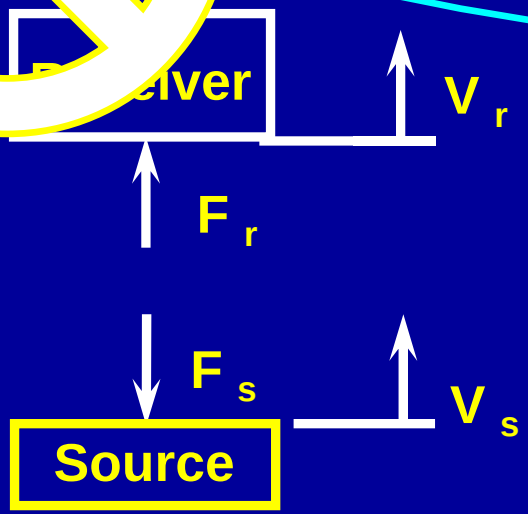
Isolation

- a theoretical model based on
rce

2003, 2005 ERRATA

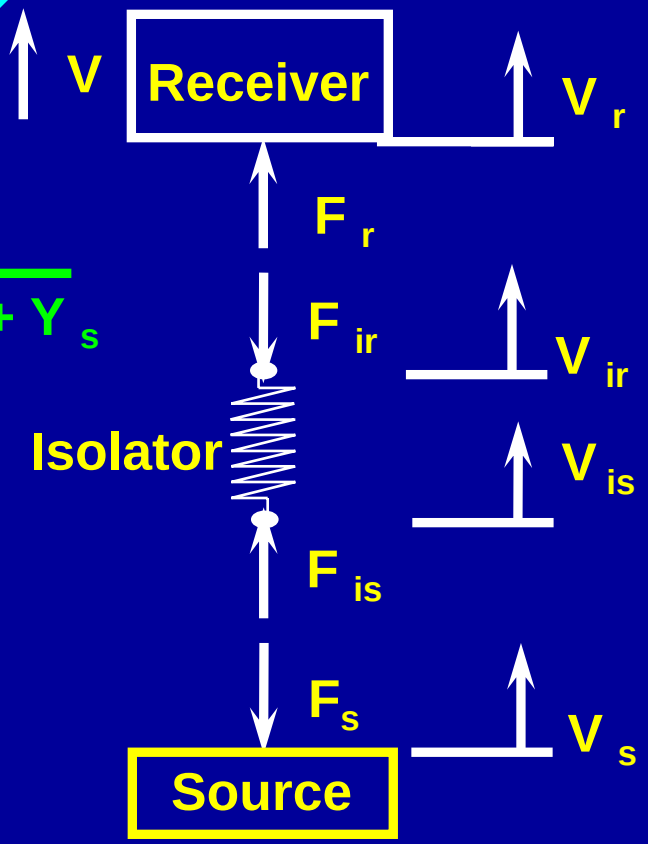
- Tr ility ratio is used to objectively define measure of isolation

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



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

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

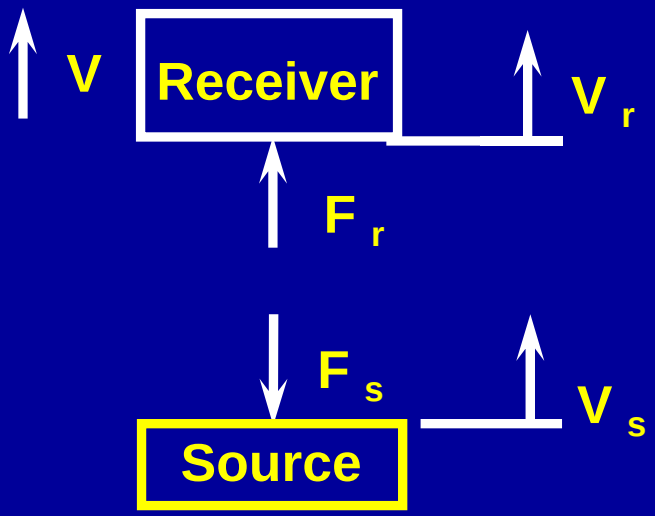


Isolation

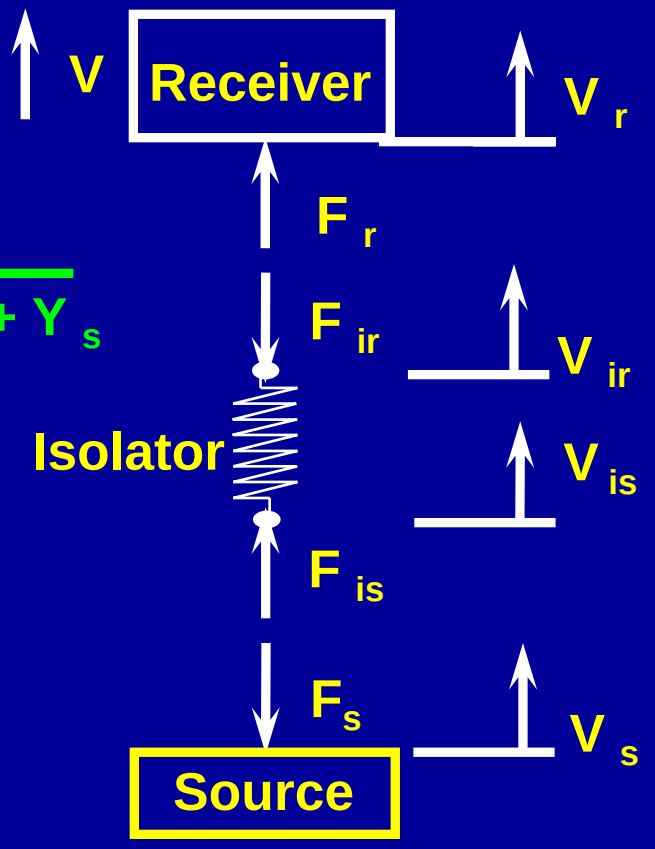
- The isolation effectiveness can be quantified by a theoretical model based on analysis of mobilities of receiver, isolator and source

- 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

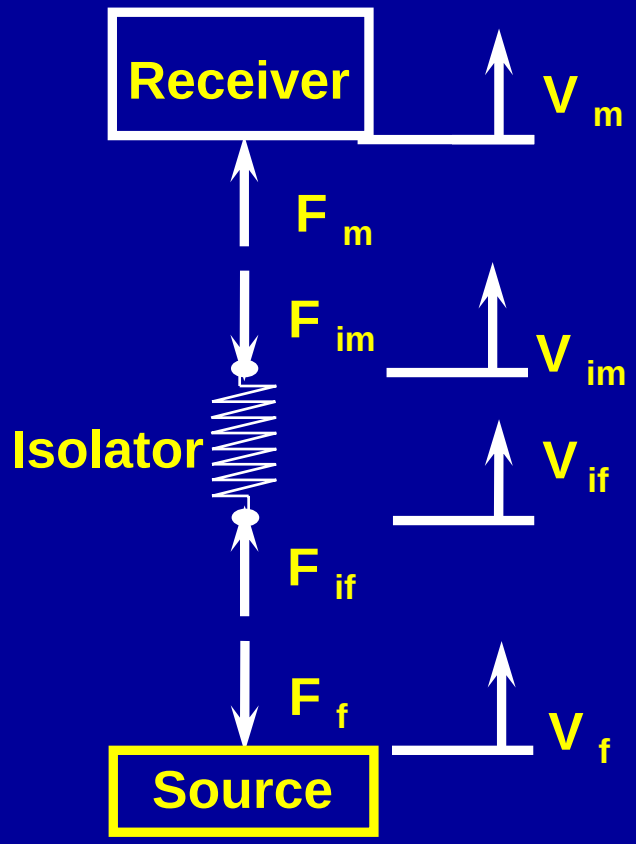
$$TR = \frac{\text{Force from source with an isolator}}{\text{Force from source without an isolator}}$$

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

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**

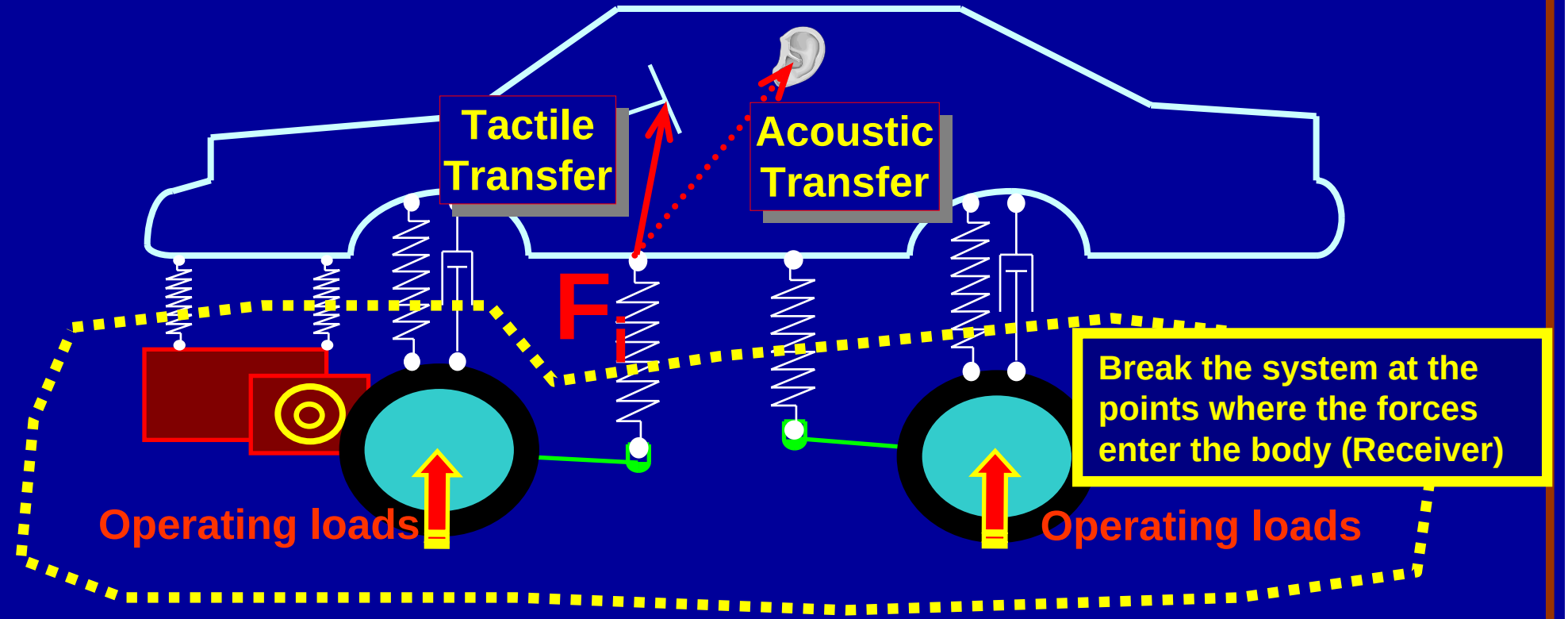
Effective Isolation

&

Identifying Key Noise Paths

Identifying Key NVH Paths

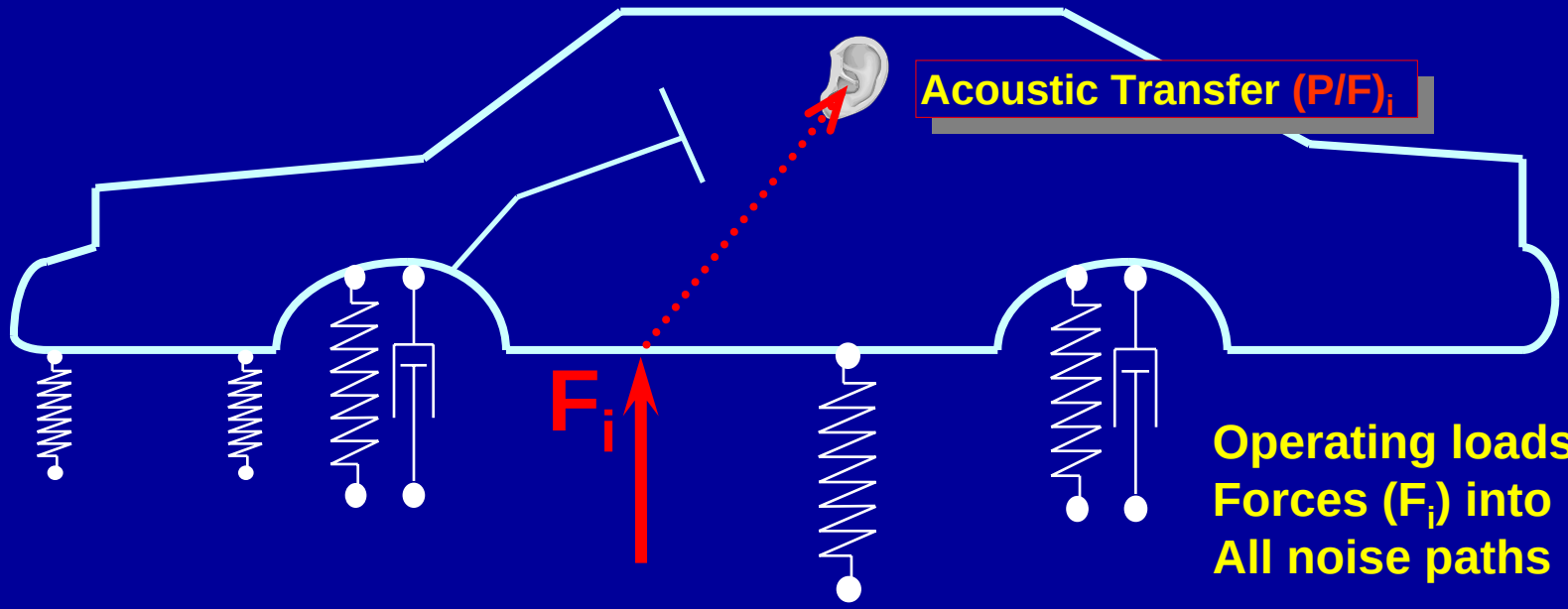
Key NVH paths are identified by Transfer Path Analysis (TPA)



- 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



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

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

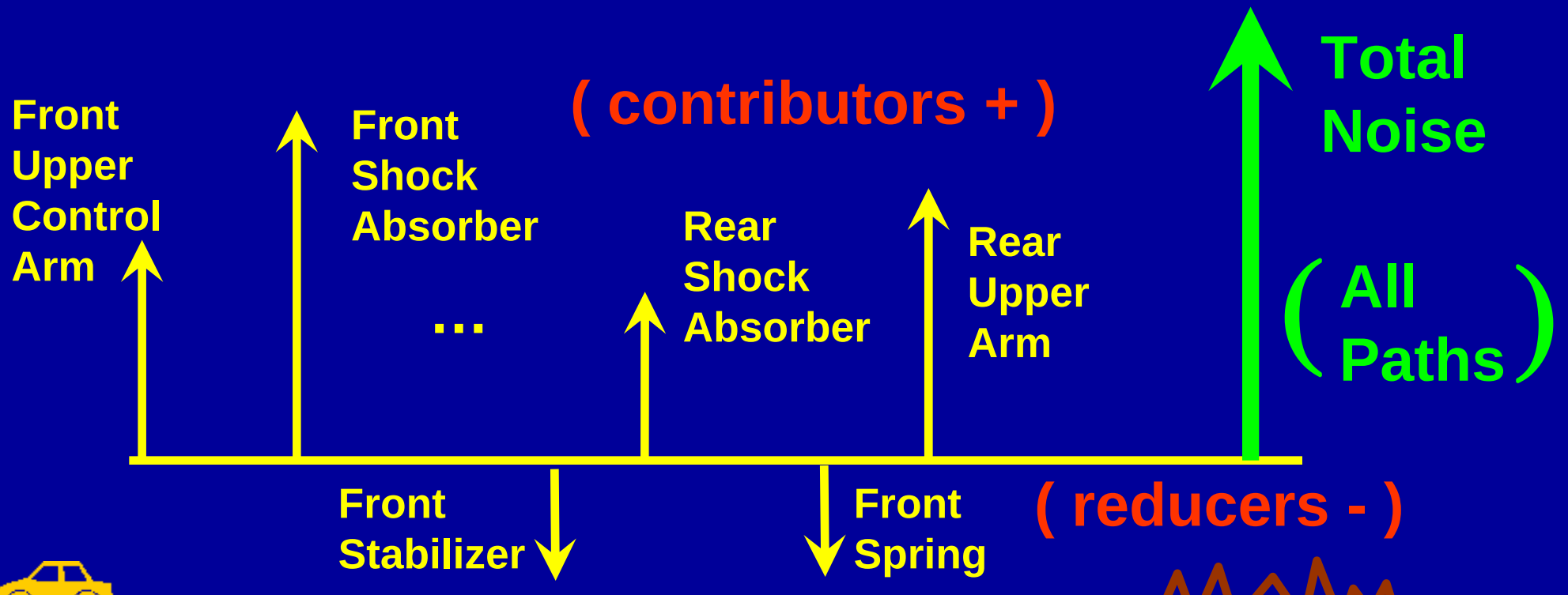
$$P_t = \Sigma_{paths} [P_i] = \Sigma_{paths} [(P/F)_i * F_i]$$

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

&

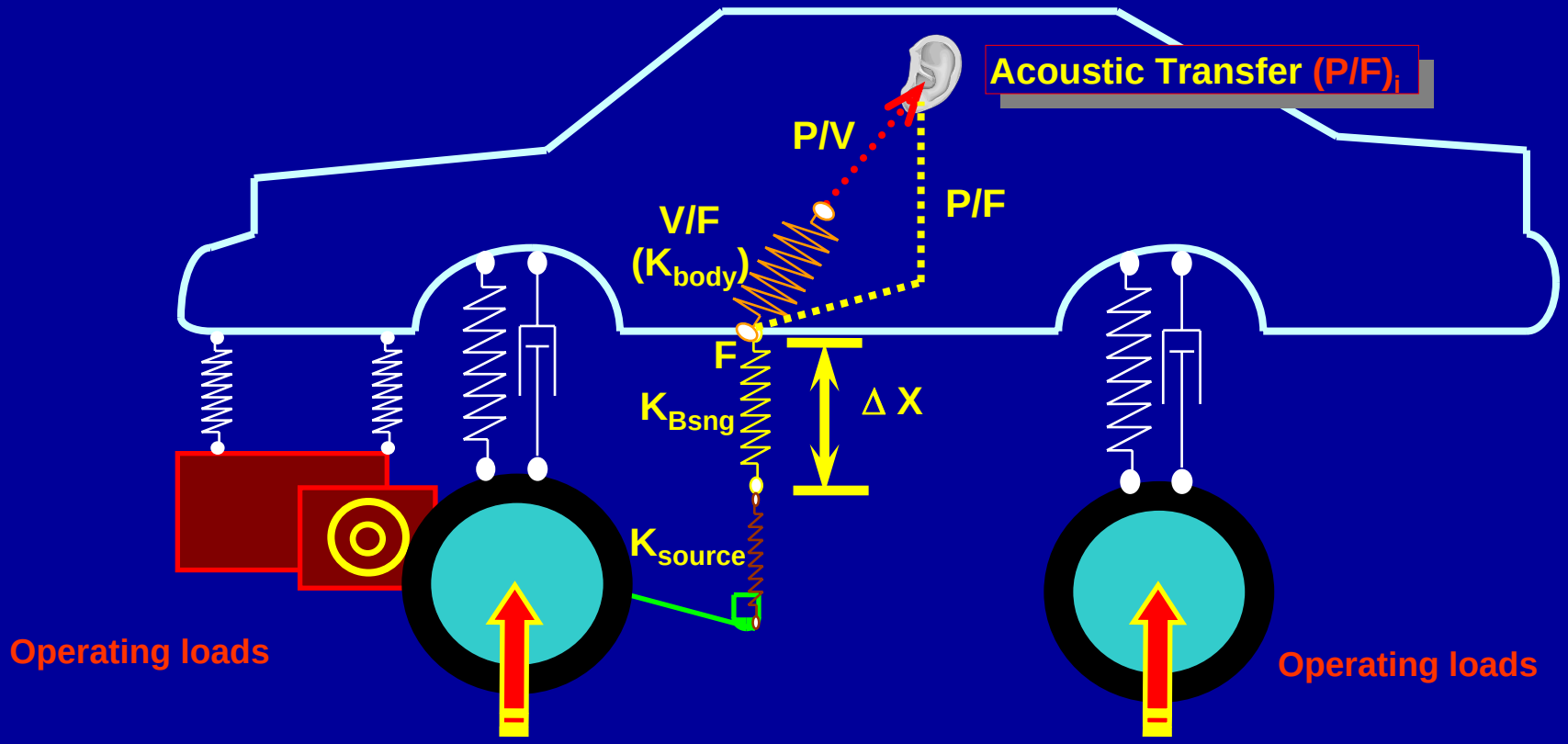
Identifying Key Noise Paths

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

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

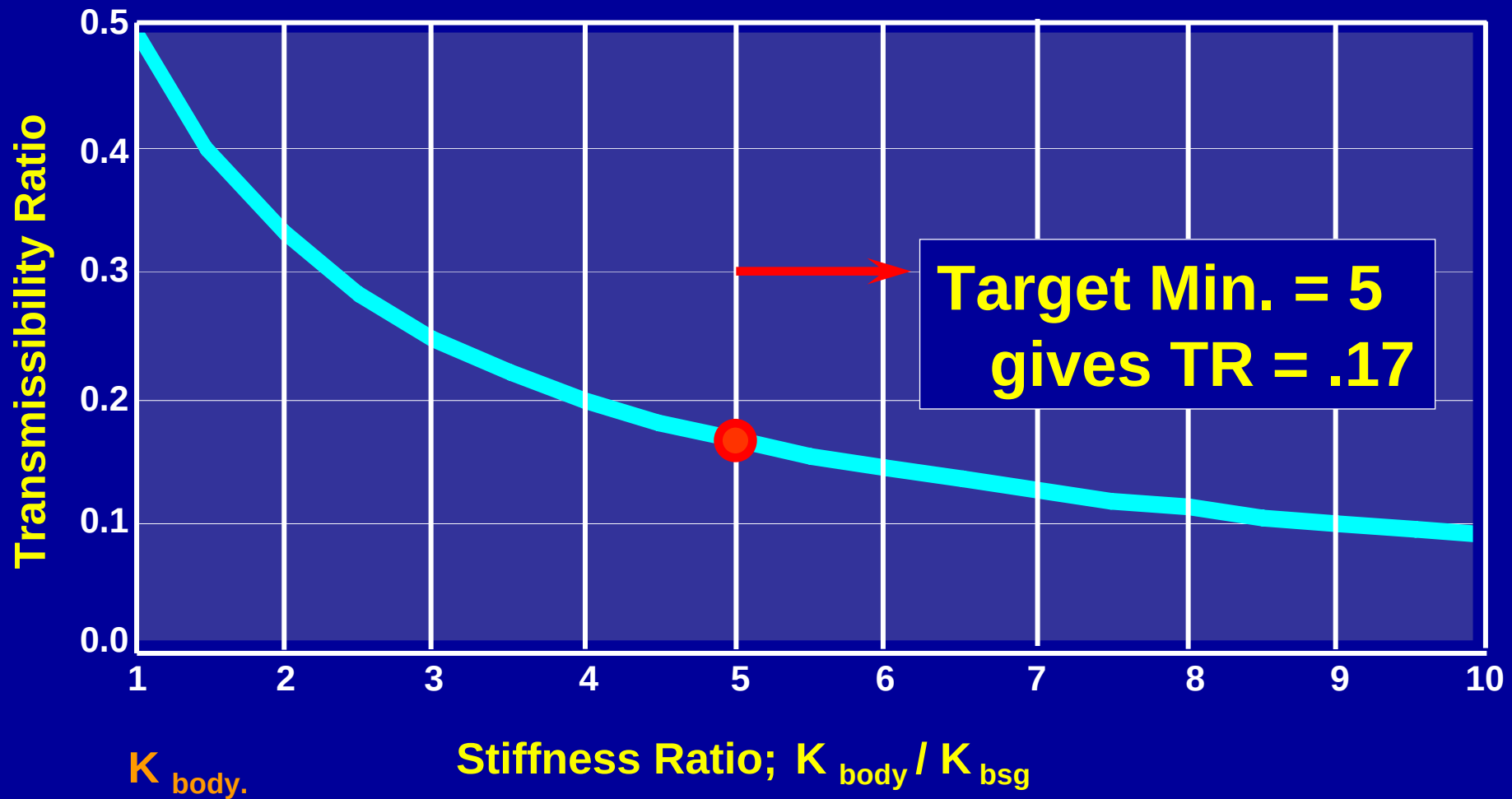
$\frac{K_{body}}{K_{iso}}$ ↓ \ $\frac{K_{source}}{K_{iso}}$ →	1.0	5.0	Infinite
1.0	0.67	0.54	0.50
5.0	0.54	0.28	0.17
Infinite	0.50	0.17	0.00

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

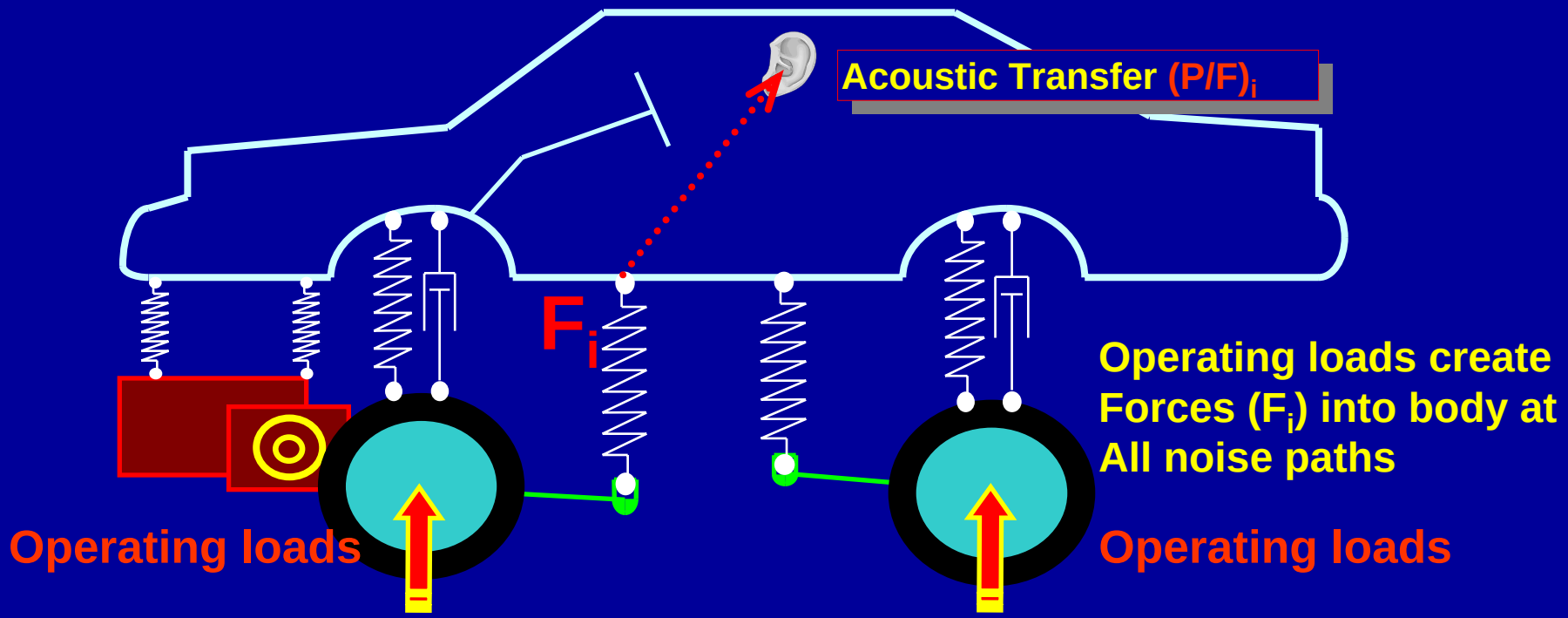
Relationship of Body-to-Bushing Stiffness Ratio to Transmissibility

For an infinite 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 0.2 - 0.3 mm/sec/N**

Generic Noise Path Targets

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

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

Acoustic Sensitivity < 50 - 60 dBL/N

Structural Mobility < .2 to .3 mm/sec/N

How does one achieve these generic targets ?

Generic Noise Path Targets

How does one achieve

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

Structural Mobility < .2 - .3

- **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

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

- **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:

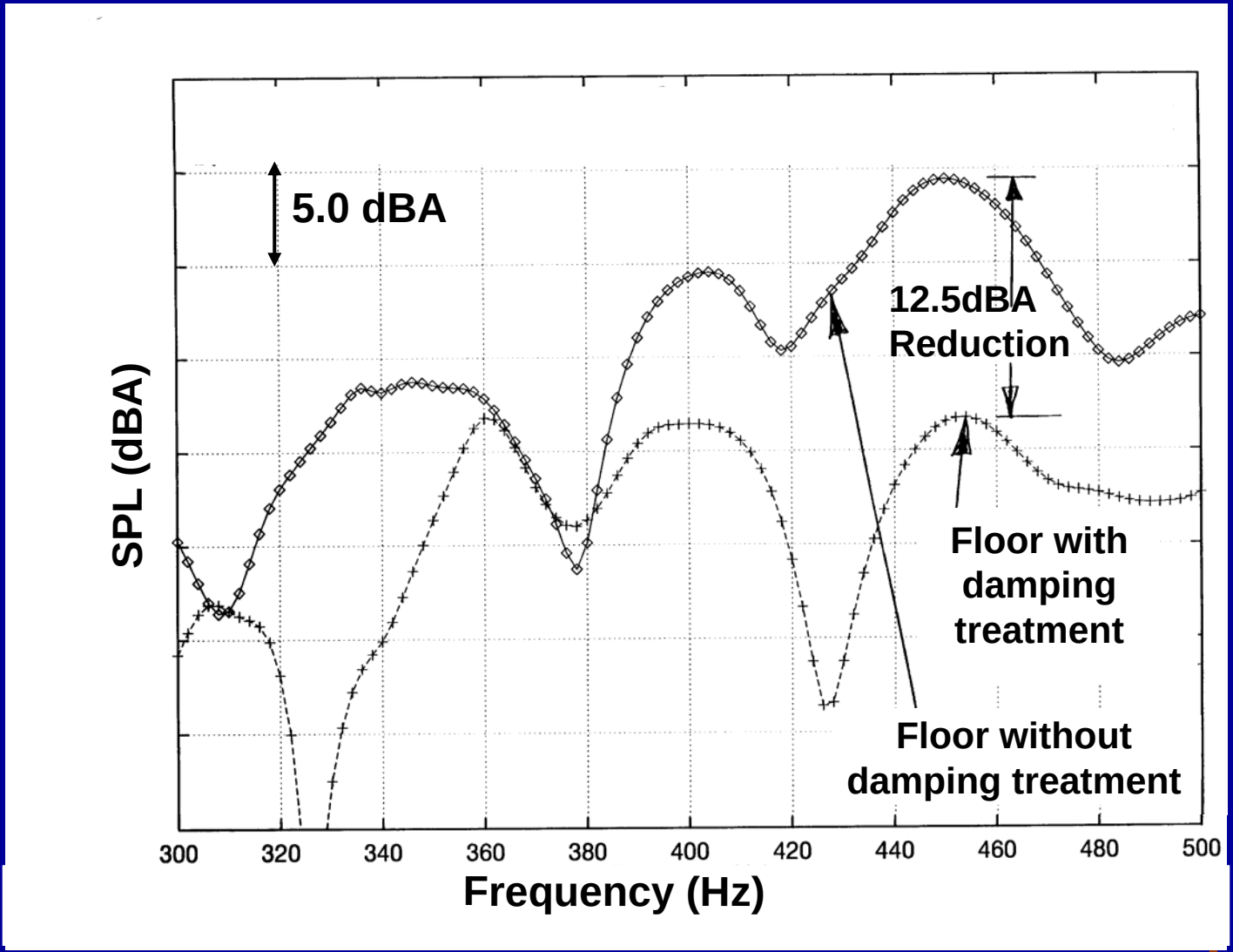
Acoustic Sensitivity	<	50 - 60 dBL/N

- **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

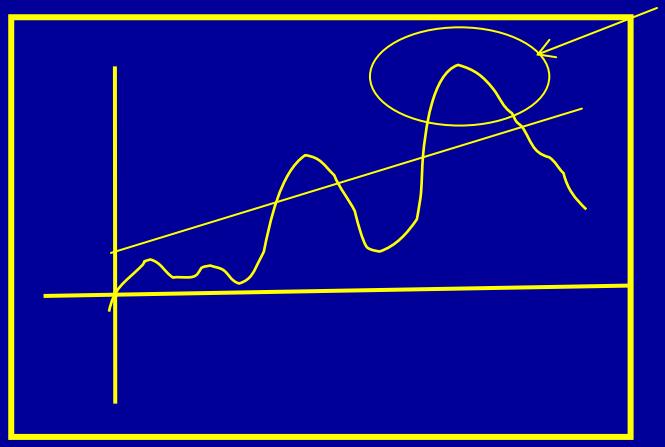
While designing a new vehicle, generic targets are set for key parameters along *all noise paths* in order to achieve effective isolation.

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

Probably Not !!

Designing for Mid Frequency

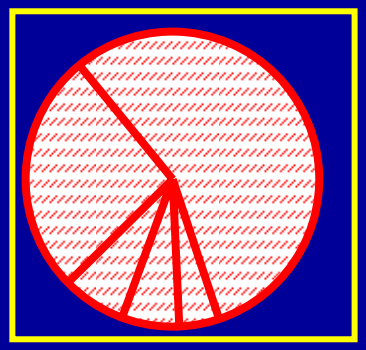
Driver's Ear Noise



Vehicle Level Response

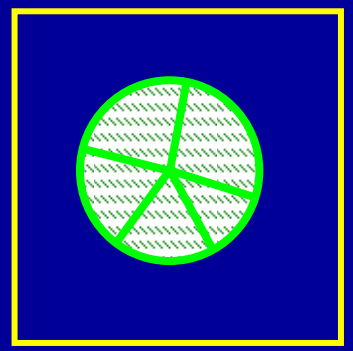
Transfer Path Analysis

Original Noise Path Contributions



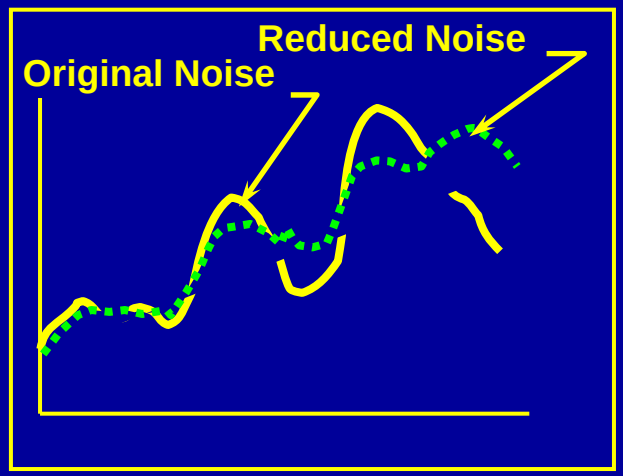
Some noise paths are more dominant than others

Improved Noise Path Contributions



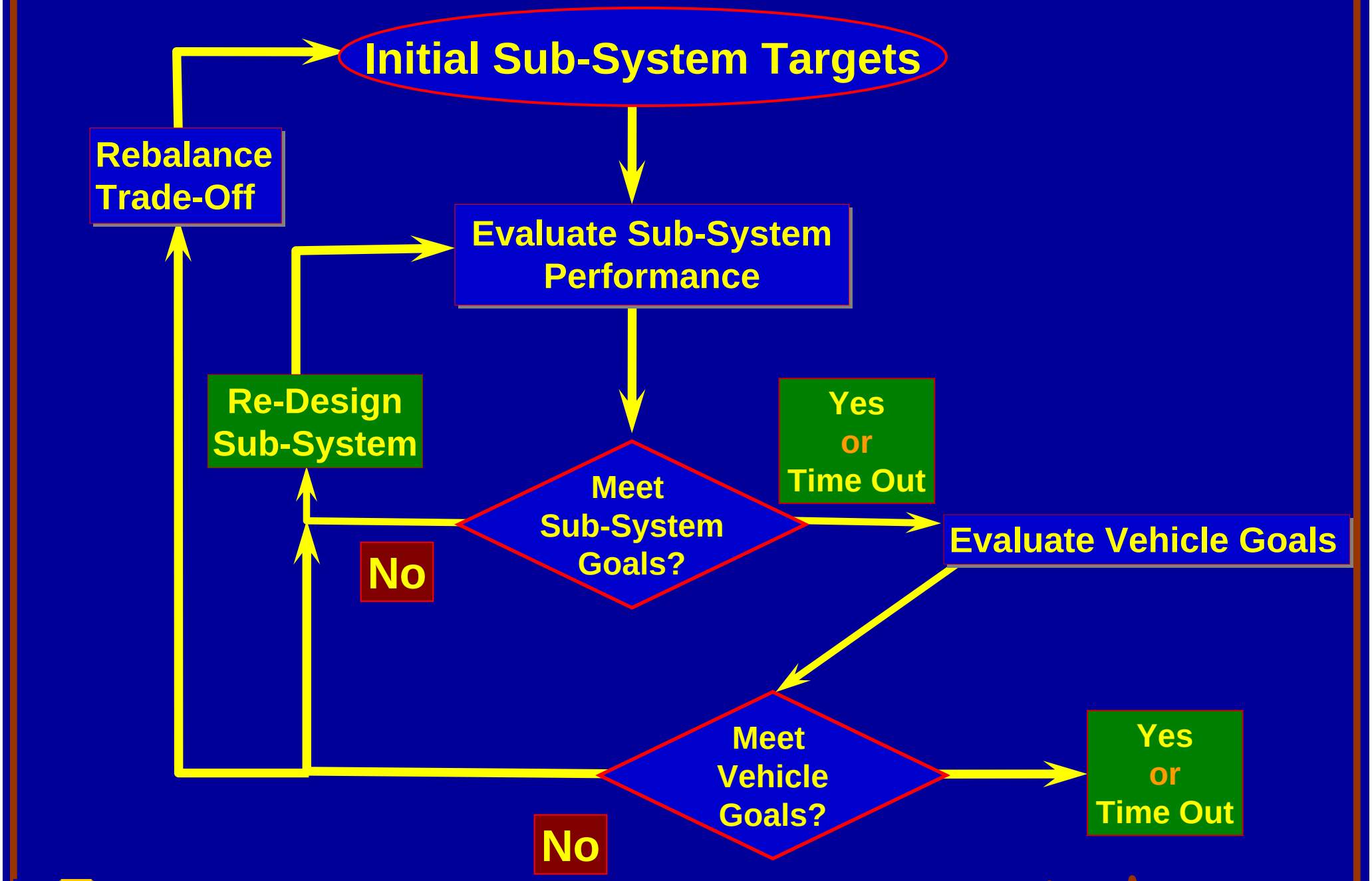
Impose more strict requirements for these dominant paths and relax requirements for other paths to achieve more "rebalanced" noise

Driver's Ear Noise



Vehicle Response to Meet NVH Targets

Mid Frequency NVH Goal Achievement Process



Designing for Mid Frequency

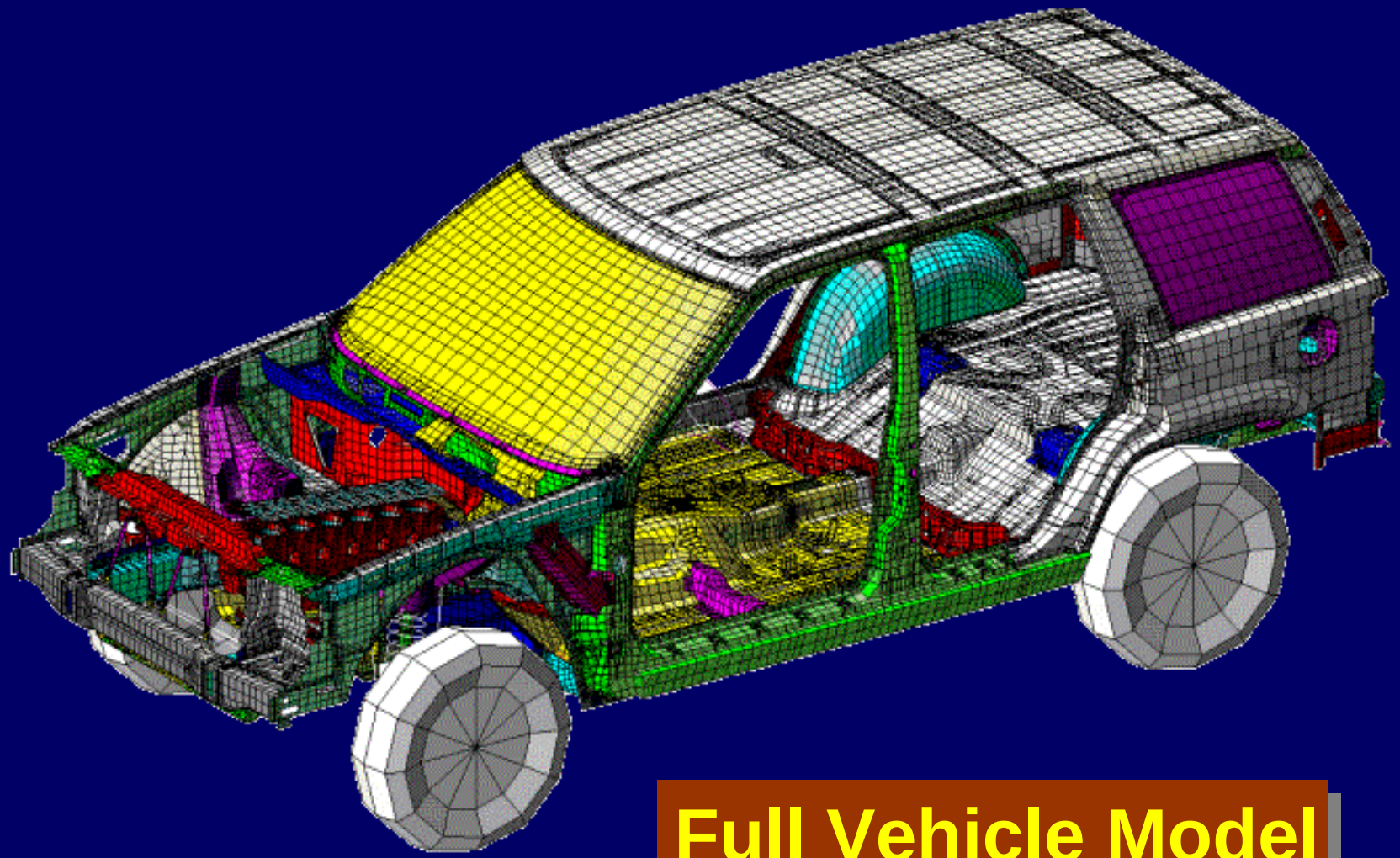
Principles to follow

- 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.

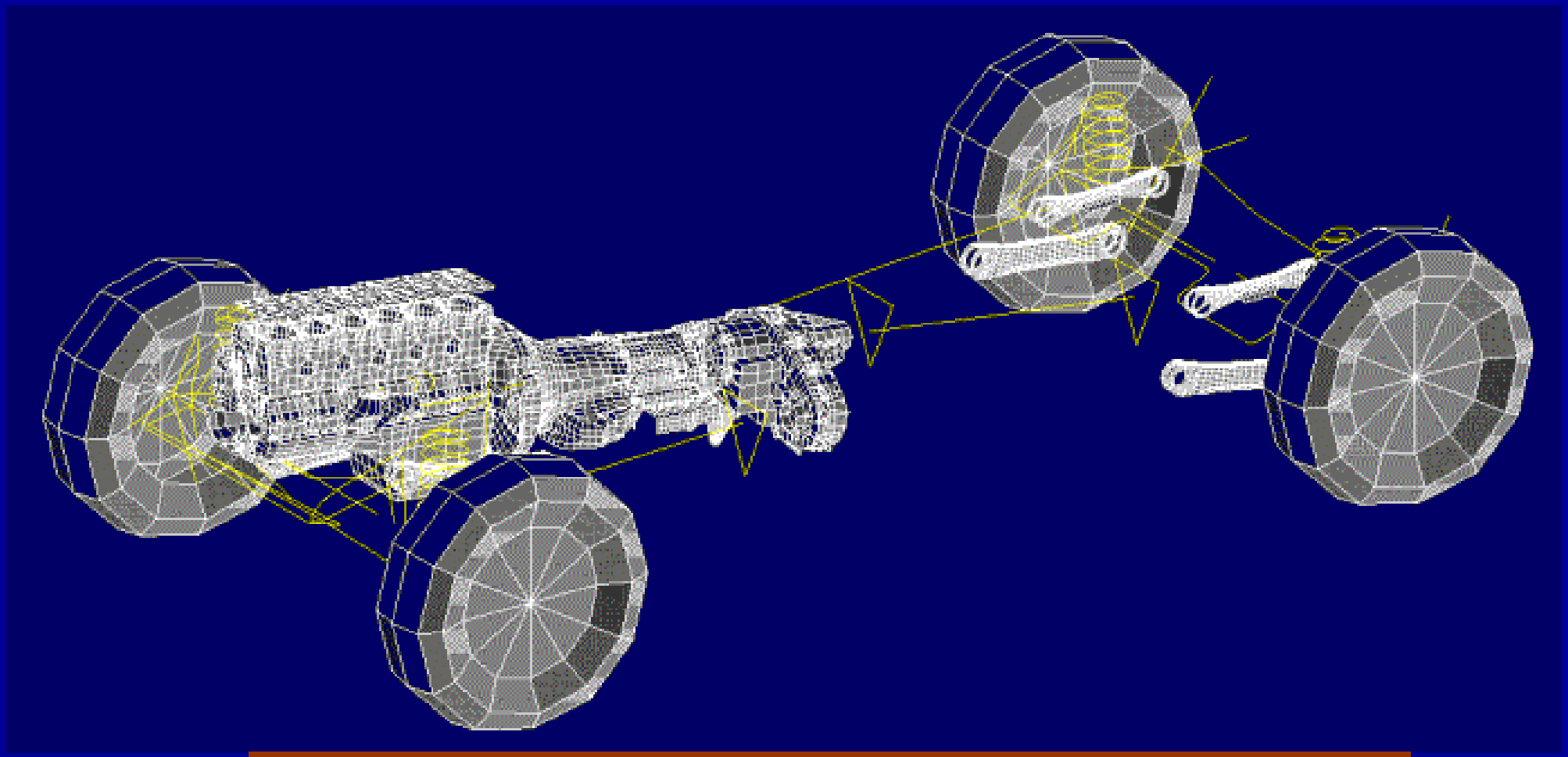
Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)



Full Vehicle Model

[Figure Courtesy of DaimlerChrysler Corporation]

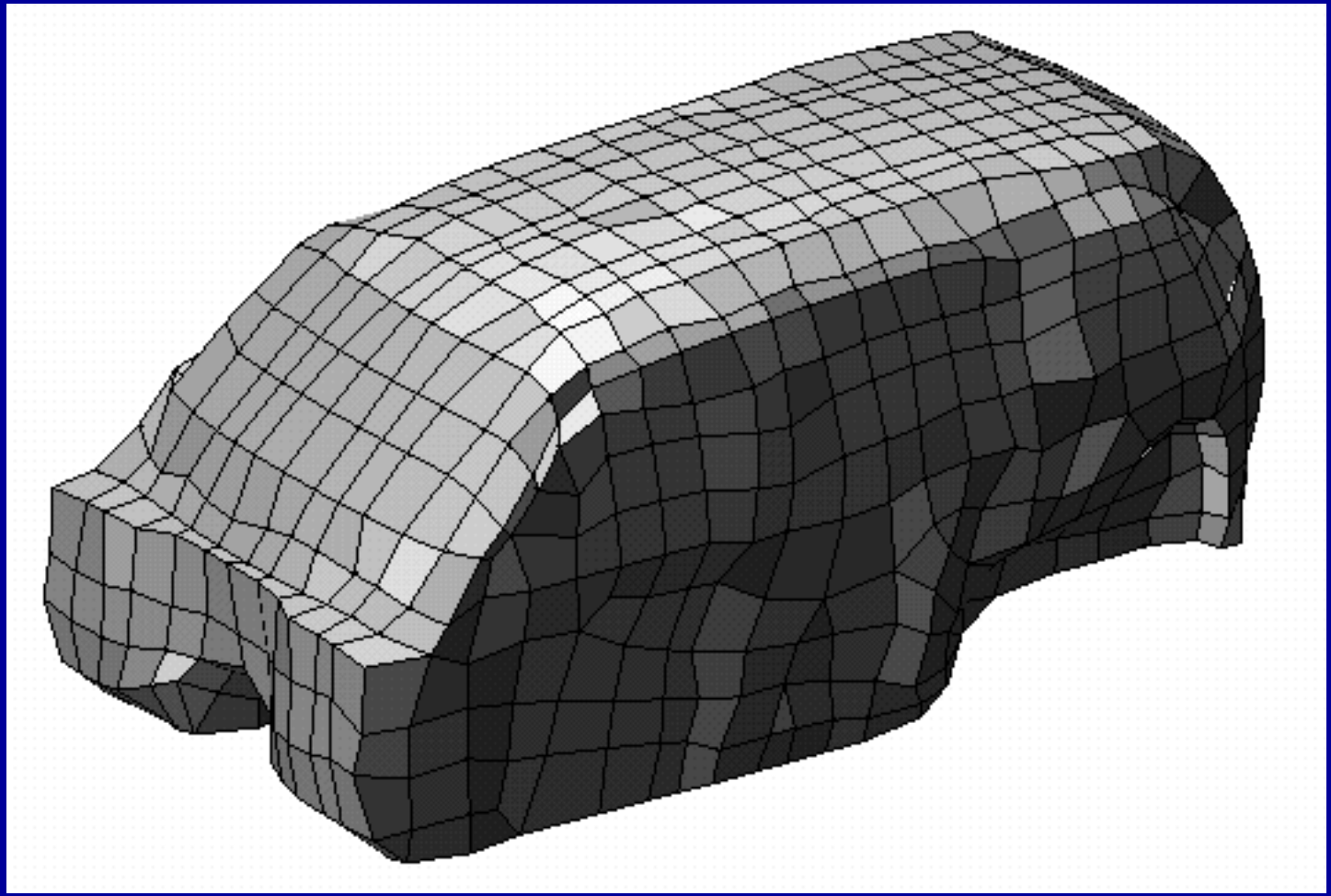
Mid Frequency NVH Improvement
(Sports Utility Vehicle Example)



Powertrain/Axle/Suspension Model

[Figure Courtesy of DaimlerChrysler Corporation]

Mid Frequency NVH Improvement (Sports Utility Vehicle Example)



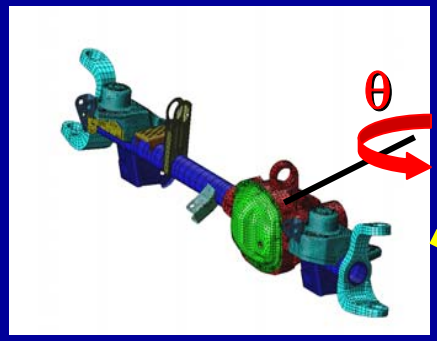
Acoustic Cavity Model

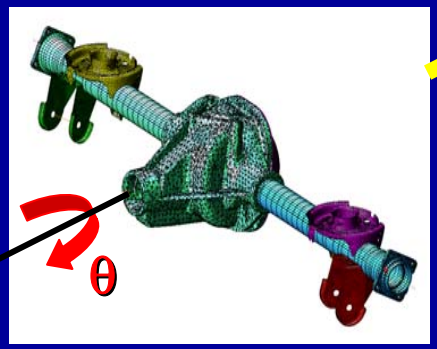
[Figure Courtesy of DaimlerChrysler Corporation]

Mid-Frequency NVH improvement

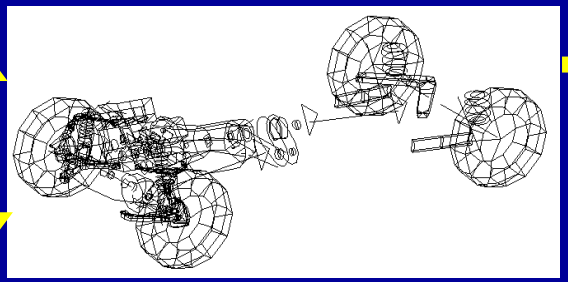
Axle Whine Example : 300-500 Hz

Front and Rear Axle
Gear-Pinion Mesh
Transmission Error

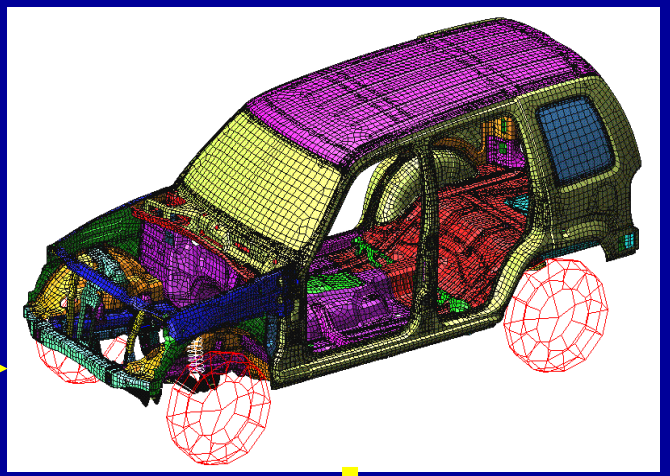


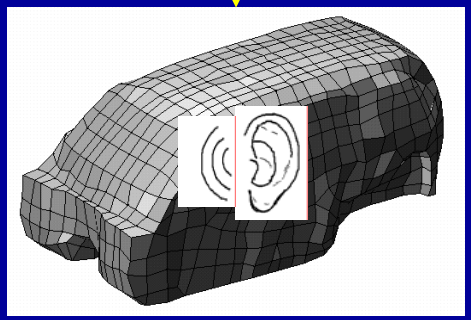


Chassis



Trimmed Body





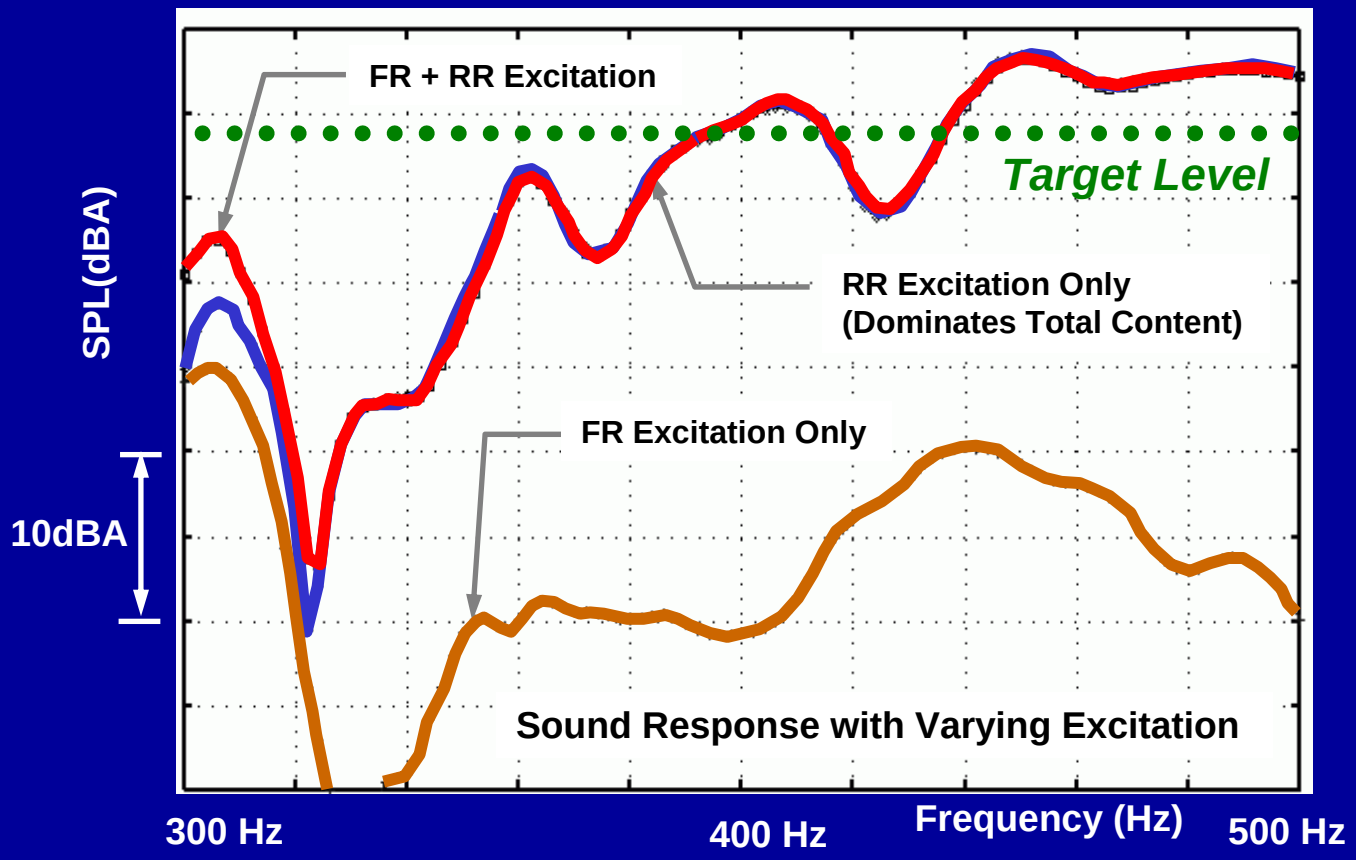
Interior Acoustic Cavity

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

[Figure Courtesy of DaimlerChrysler Corporation]

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



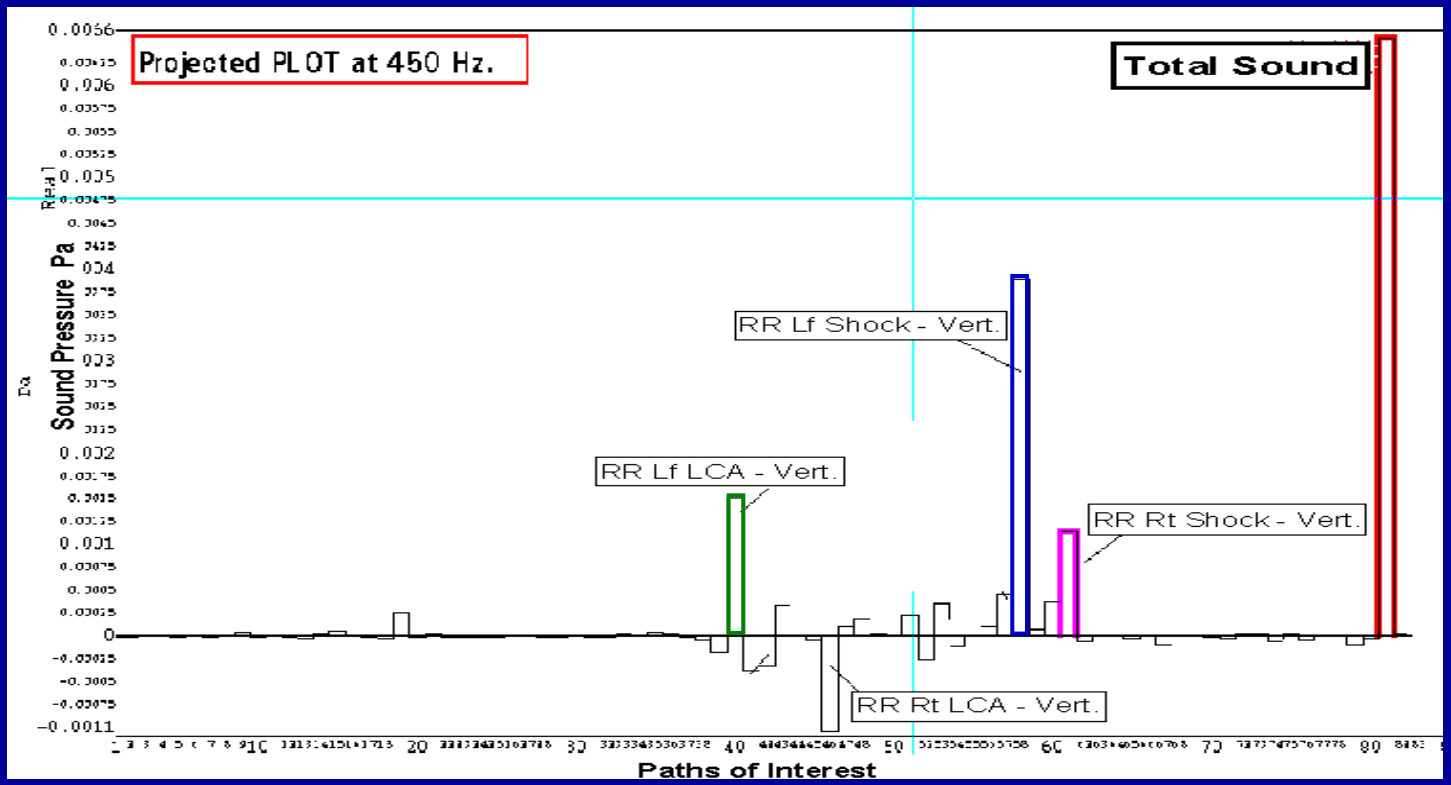
[Figure Courtesy of DaimlerChrysler Corporation]

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 a 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) had the most significant effect on Driver's SPL response**

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**

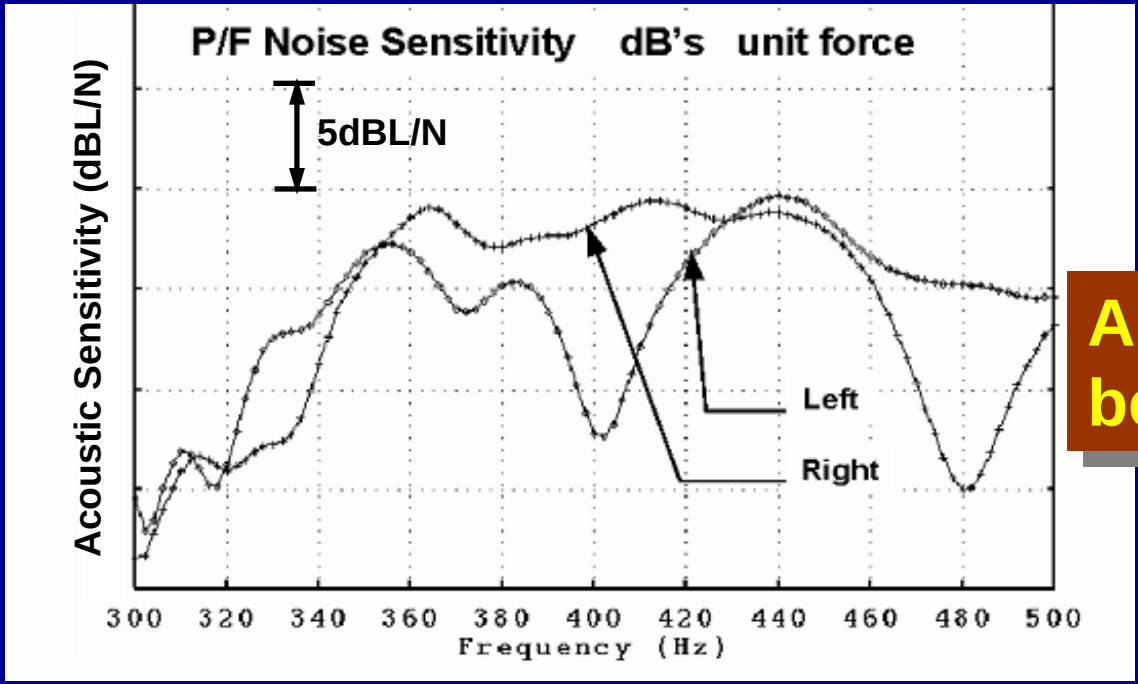
- **The conclusion matches with reasonableness checks**

[Figure Courtesy of DaimlerChrysler Corporation]

Axle Whine Example

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

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



Acoustic sensitivity was 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]

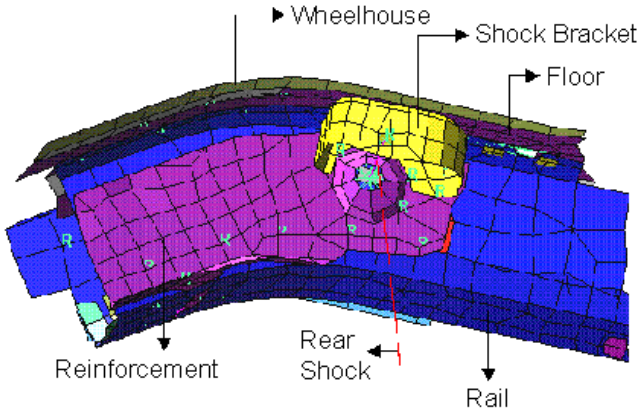
Axle Whine Example

Solution

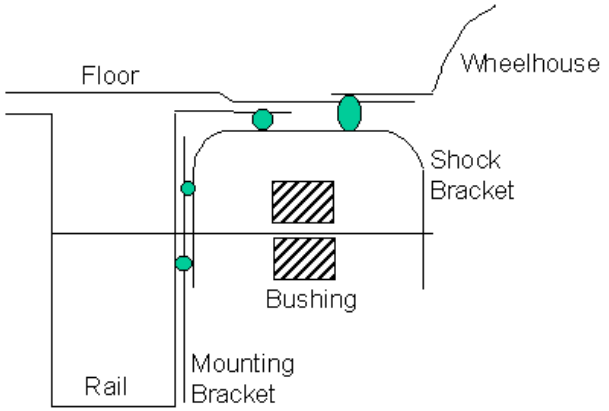
- Soften shock vertical bushings by 65%

- To balance this against Ride and 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

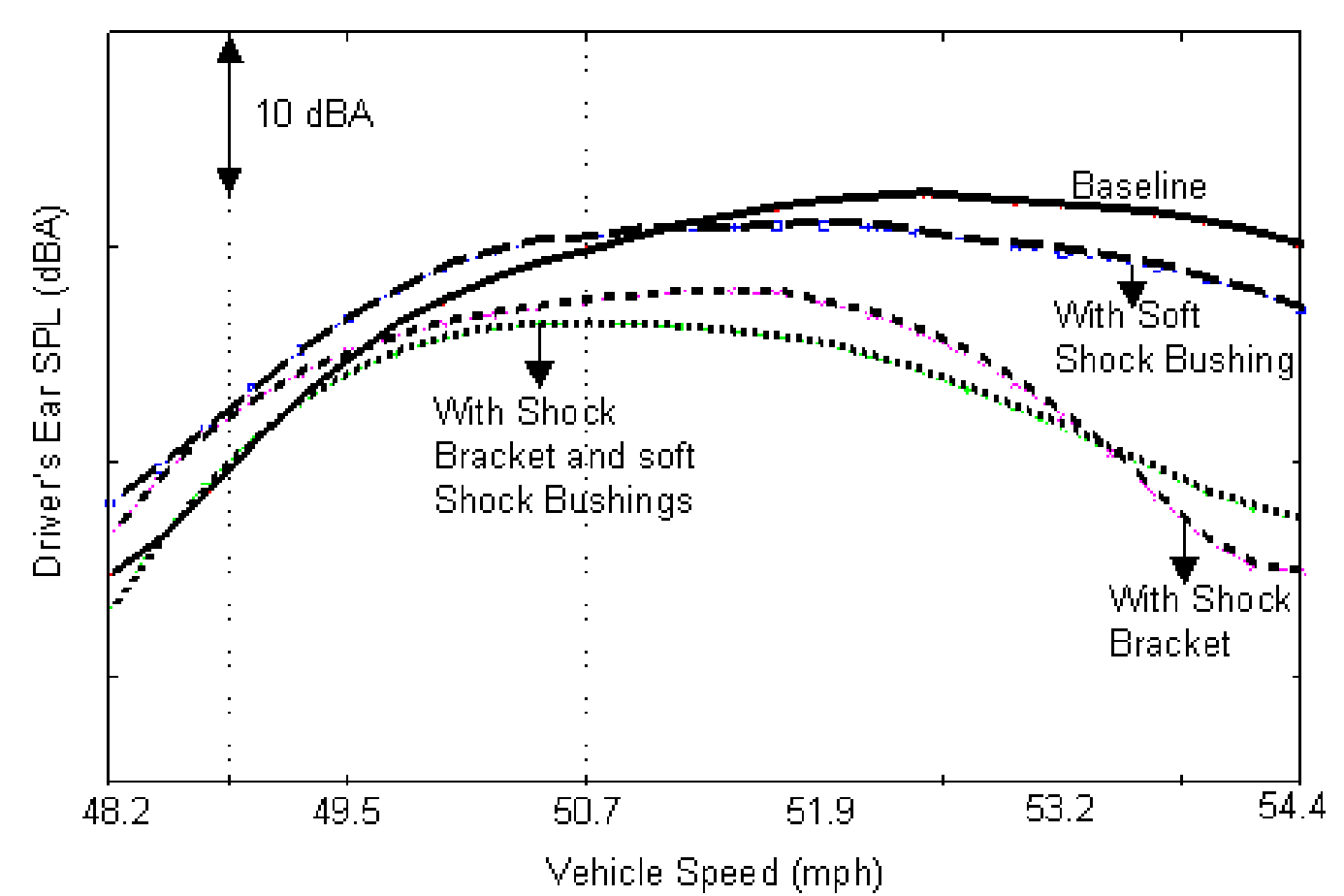


Rear View Section of Shock Bracket

[Figure Courtesy of DaimlerChrysler Corporation]

Axle Whine Example

Response Improvement due to proposed solution



[Figure Courtesy of DaimlerChrysler Corporation]

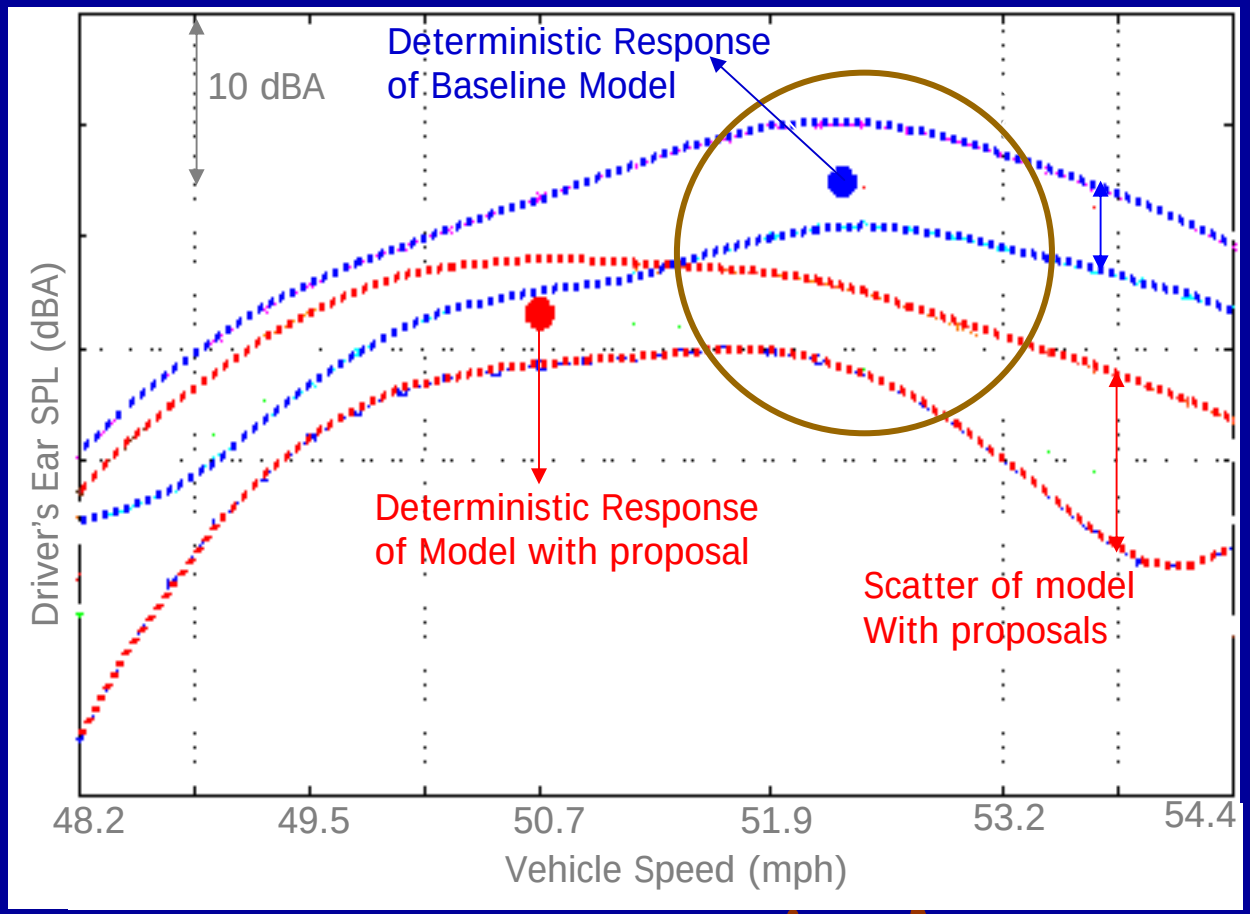
Axle Whine Example

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]

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 requirements against other functionalities (Ride and Handling, Impact)
- It is important to understand the robustness of design recommendations

NVH Workshop Topic Outline

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

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 and correlation criteria 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.

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.

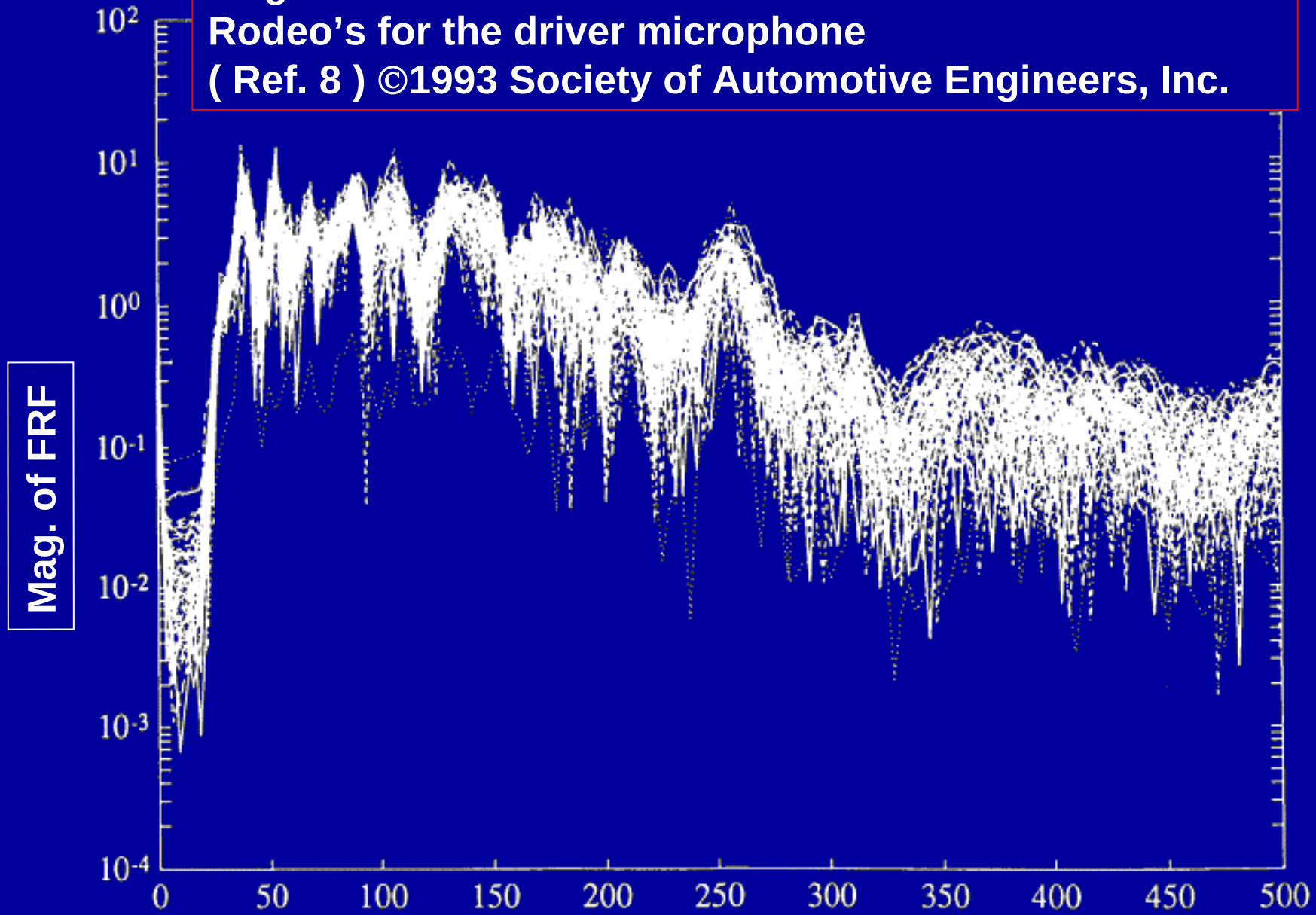
- Q. How wide is the band of variability?
- ANS. Let's EXPLORE it !!!!

Discussion of Product Variability

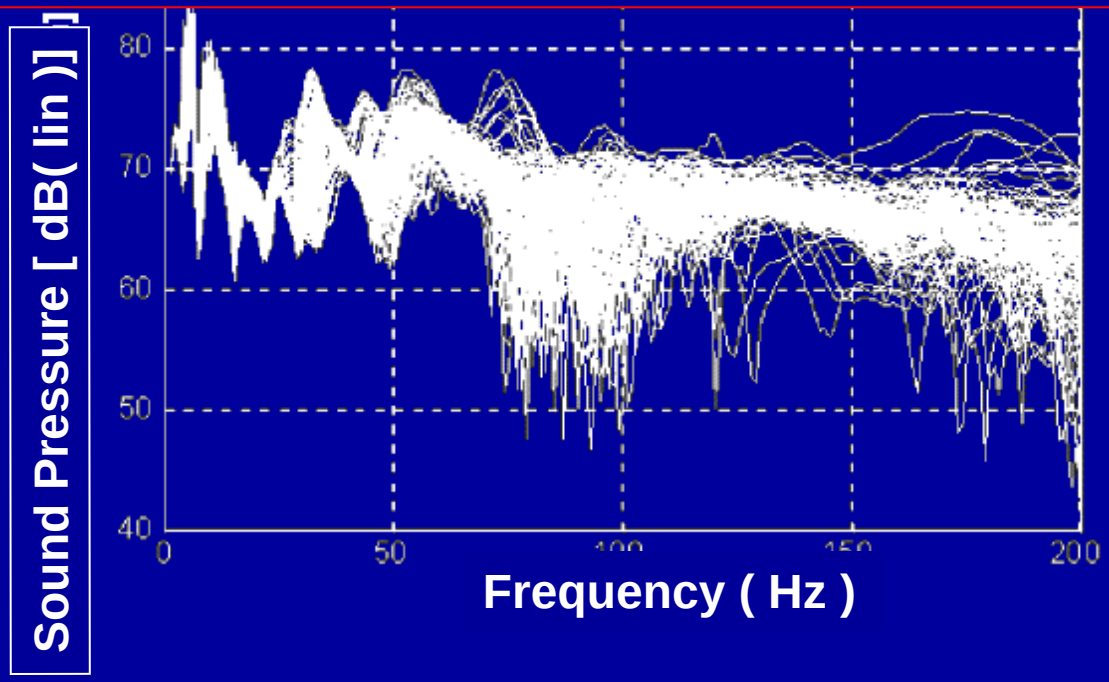
Topics

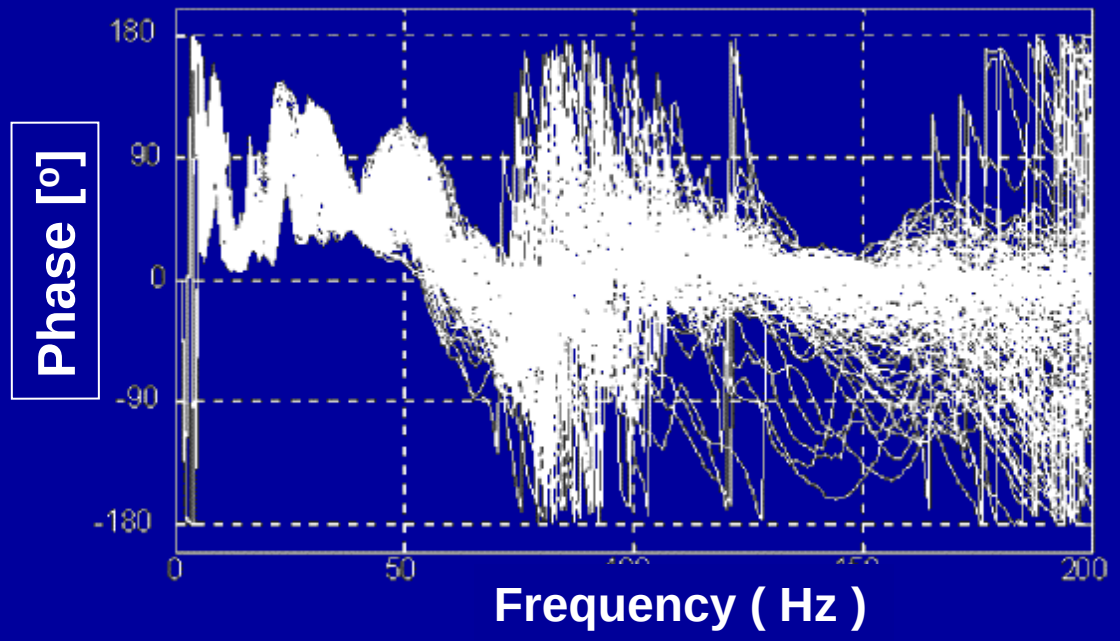
- Kompella and Bernhard Observations
- Freyman NVH Scatter Results
- Model Confidence Criteria
- Conclusions

Magnitude of 99 Structure – borne FRF's for the Rodeo's for the driver microphone
(Ref. 8) ©1993 Society of Automotive Engineers, Inc.



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

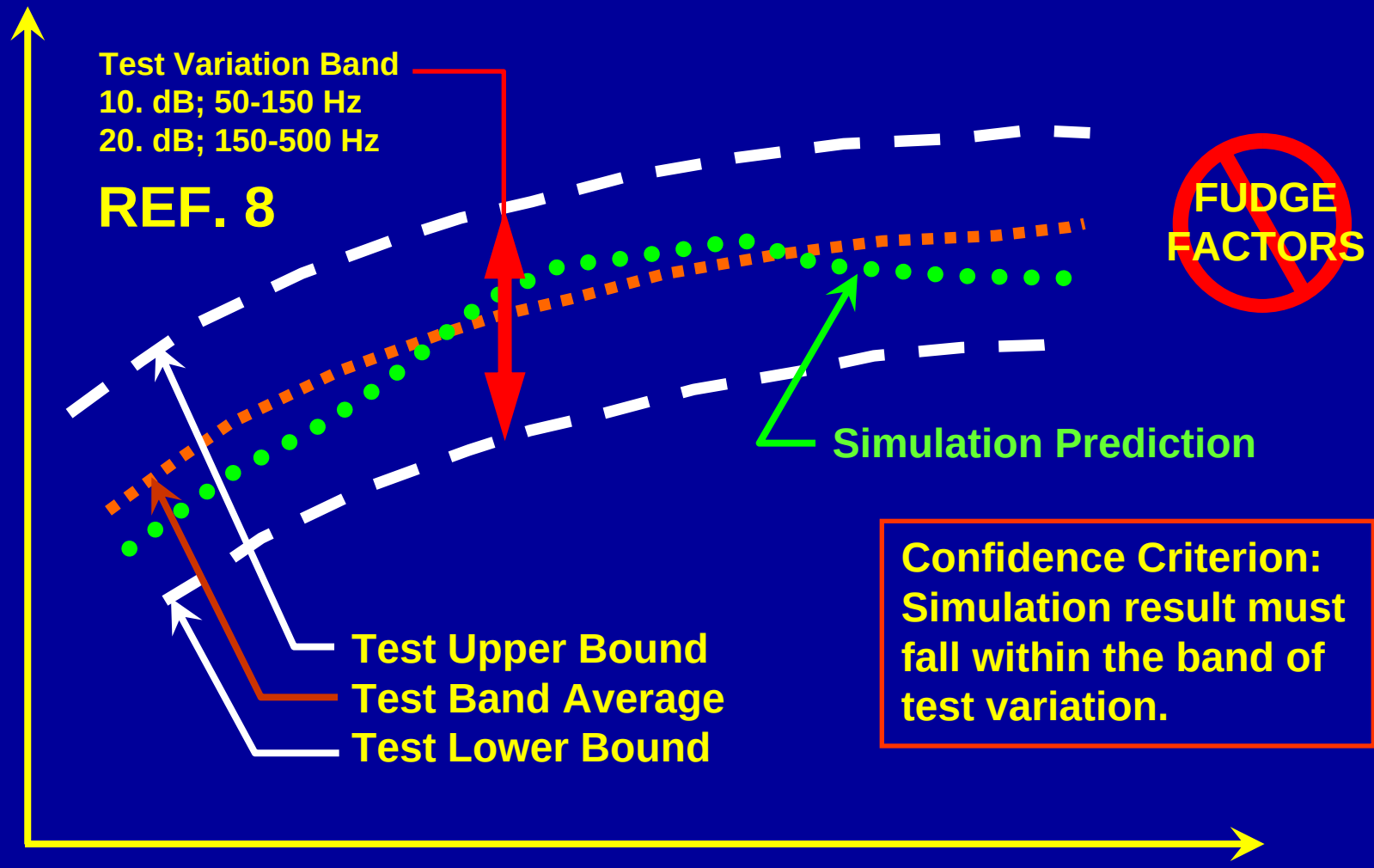




(Ref. 9) ©2000 Society of Automotive Engineers, Inc.

Reference Baseline Confidence Criterion

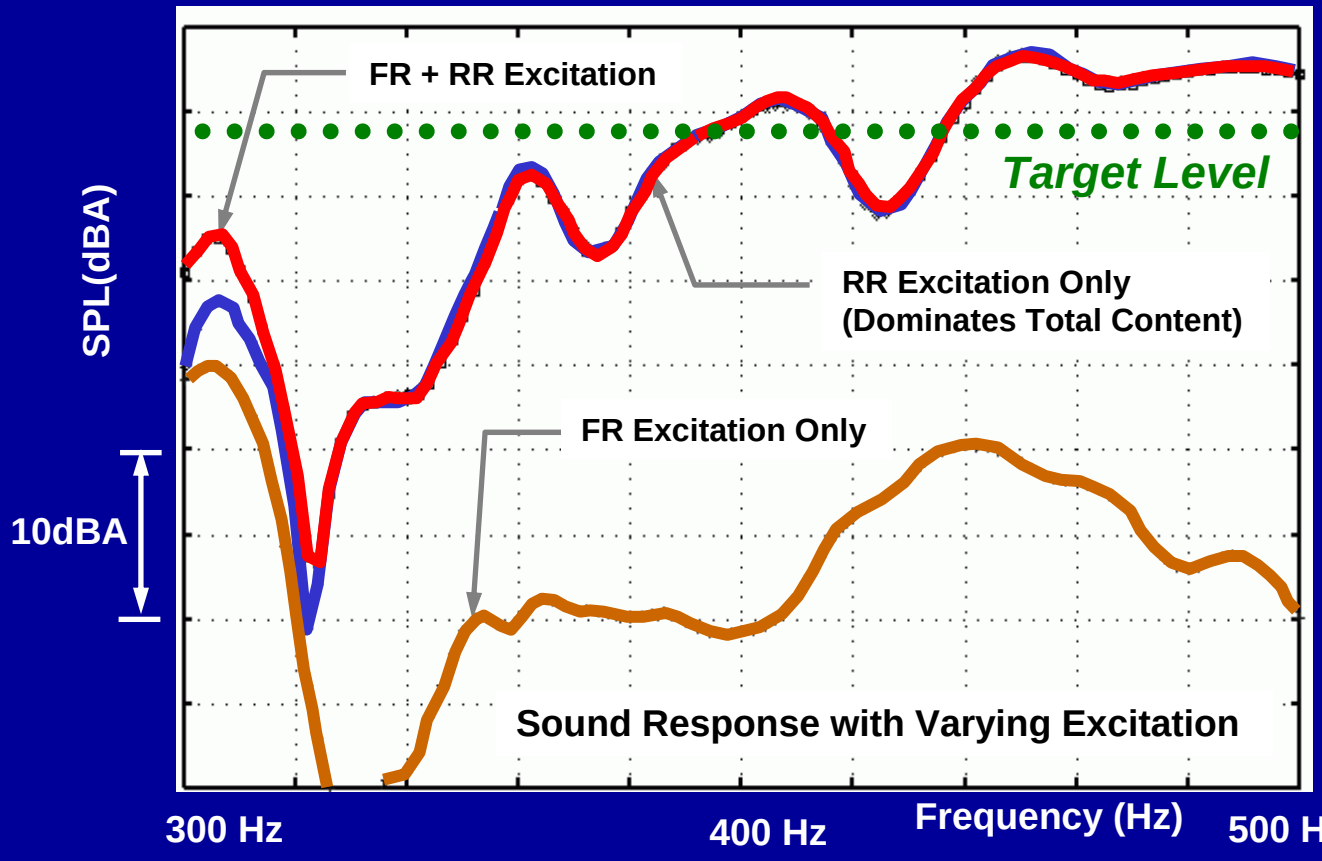
For *Operating* Response Simulations



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

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]

Conclusions:

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

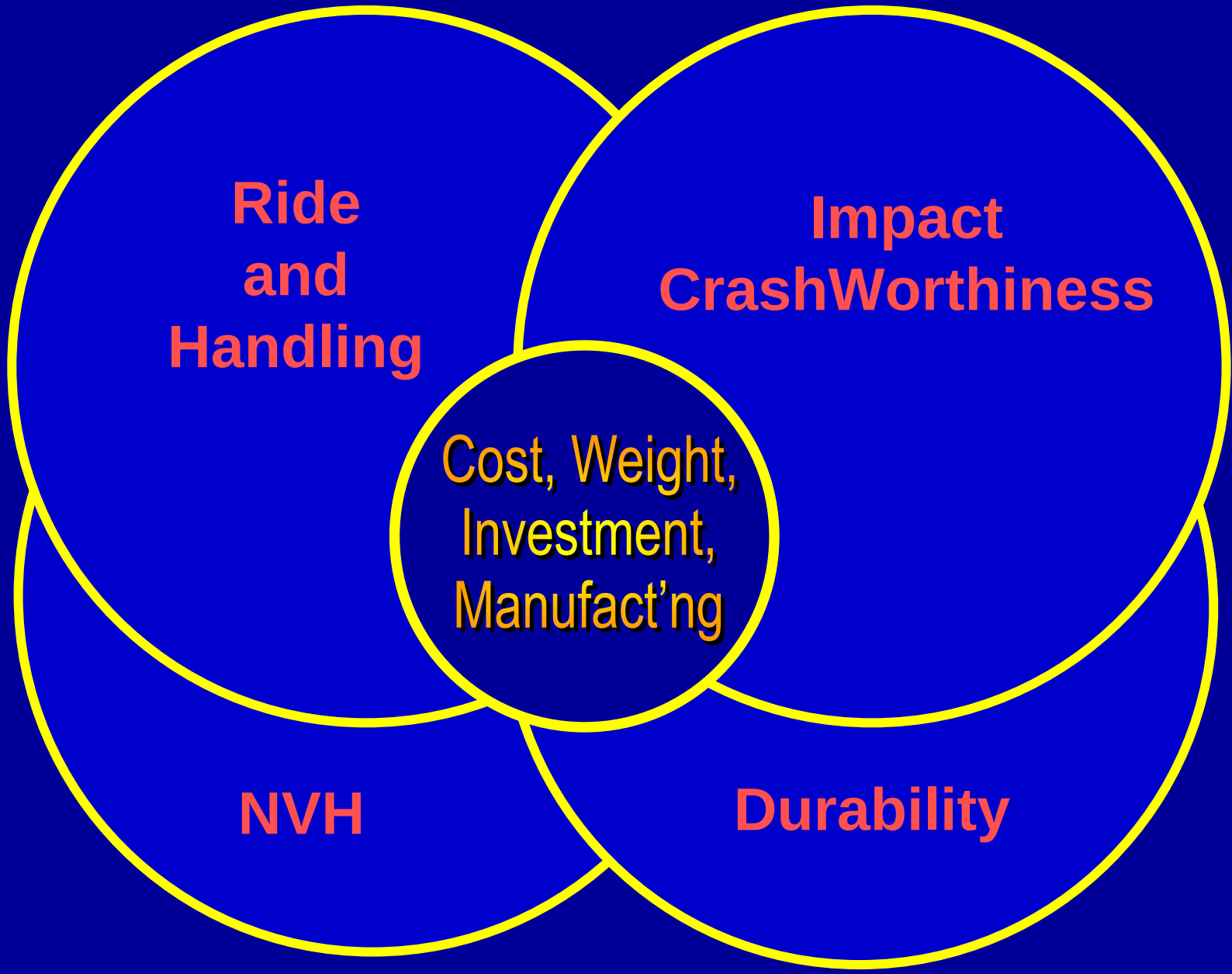
Correlation should be considered as being within the band of variability whether test or simulation.

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

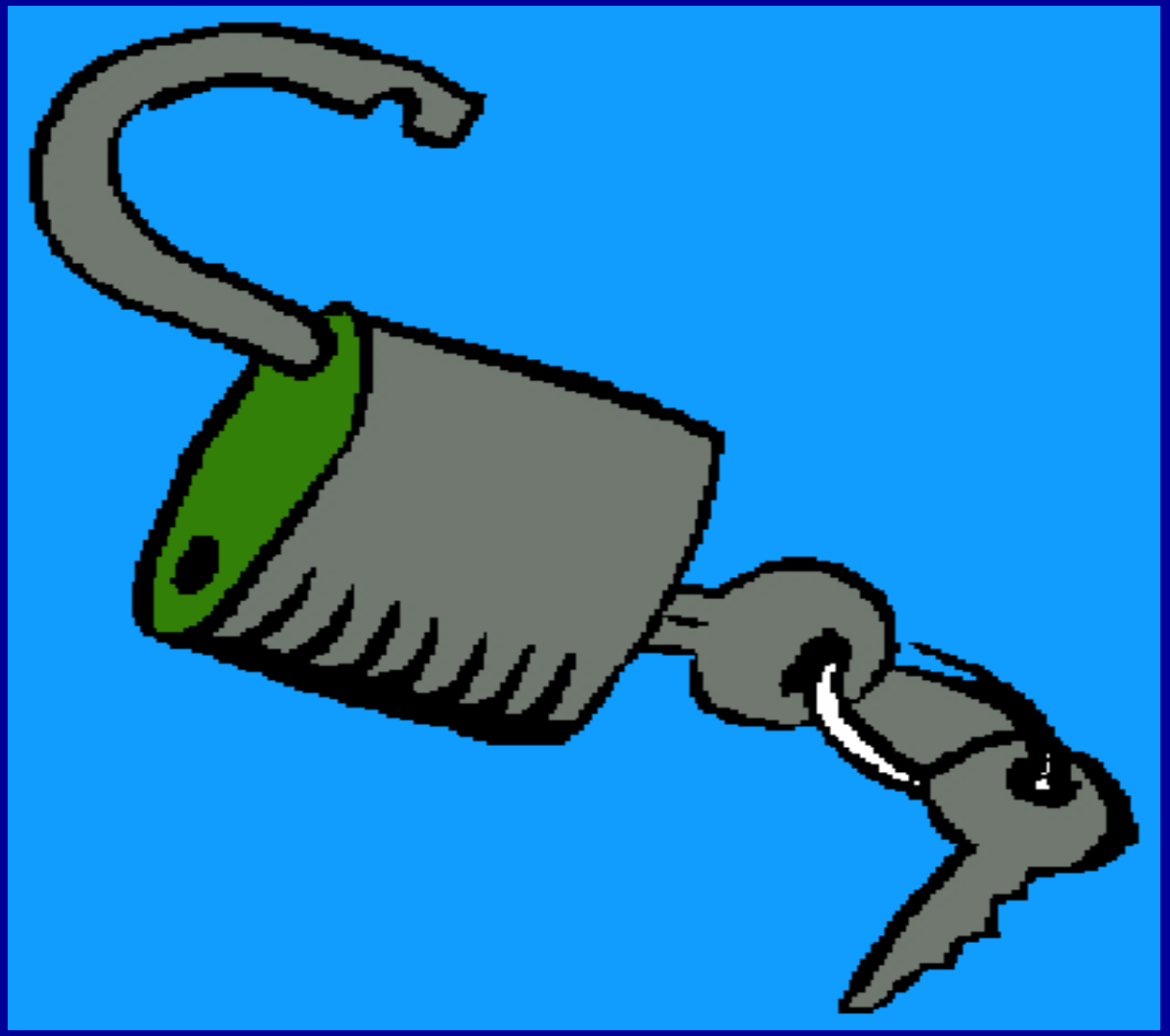
- ✓ **It uses the same bandwidth as Kompella (Ref. 8), determined from simple FRF's, while the criteria is for operating responses which are subject to additional variation in the operating loads.**
- ✓ **It assumes that one test will generate the mean response level in the band subject to the condition 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.

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.

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*

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.

1st Corollary

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

2nd Corollary

Effectively Isolate the remaining source from the receivers at Key paths

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.

That's All Folks !

Thank You for Attending the

2007 Structure Borne NVH Workshop

Your Presenters today were:

Alan Duncan, Material Sciences Corp.

Greg Goetchius, Material Sciences Corp.

Sachin Gogate, DaimlerChrysler Corp.

Visit: www.AutoAnalytics.com/papers.html (or www.sae.org) to download the 2007 Structure Borne NVH Workshop

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