Automotive Design for NVH Control

Professional Course PPT with Concepts, Derivations, Practice, and Industry Application

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April 2026

Course Roadmap

Course Outcomes

After completing this module, students should be able to:

- Define NVH quantitatively using frequency, amplitude, and sound pressure metrics.
- Derive transmissibility and natural frequency relations for isolation systems.
- Explain the role of BIW stiffness, mount tuning, and damping materials in vehicle refinement.
- Interpret practical NVH data such as FRF, spectra, mode shapes, and transmissibility curves.
- Relate classroom vibration theory to automotive development, testing, and CAE practice.
- Solve introductory design problems on engine mounts, bushings, and structural tuning.

What is NVH?

Definition

Noise, Vibration, and Harshness (NVH) is the engineering study of unwanted sound, structural oscillation, and human discomfort arising in a vehicle system.

- **Noise:** airborne acoustic response, measured in dB.
- **Vibration:** mechanical oscillation, measured as displacement, velocity, or acceleration.
- **Harshness:** subjective severity of impacts, shocks, or rough transients.

Core aim: reduce objectionable response without compromising mass, cost, durability, and performance.

Why NVH is Critical in the Automotive Industry

- Customer perception of vehicle quality is strongly influenced by idle shake, door closing feel, and cabin quietness.
- Poor torsional compliance can lead to shake, squeaks, rattles, and degraded handling.
- Engine mount tuning is routinely used in industry to reduce vibration transmitted to driver seat rail and body structure.
- EV development has shifted focus toward tire noise, motor whine, inverter tone, and HVAC noise because masking from combustion engines is reduced.

Industry studies and technical sources report that body modes, engine mount stiffness tuning, and whole-body vibration evaluation are central to automotive NVH development [web:19][web:18][web:26].

Source–Path–Receiver Concept

- **Source:** engine firing, gear mesh, road-tire interaction, aerodynamic turbulence, pumps, compressor, motor electromagnetic force.
- **Path:** BIW, subframe, bushings, mounts, suspension links, body cavities, air leakage paths.
- **Receiver:** driver and passenger as acoustic and tactile receivers.

Engineering Rule

An NVH problem becomes manageable when it is decomposed into source reduction, path interruption, and receiver mitigation.

SPR Diagram with Control Actions

Source
Engine, tire, gear, aero

Path
Body, mounts, air, trims

Receiver
Seat, steering,

Balance, combustion control, gear micro-geometry

Stiffening, damping, isolation, sealing

Acoustic trim, A
seat isolatio

NVH Metrics Used in Practice

Metric	Unit	Use	Typical Tool
Sound pressure level	dB	Cabin and pass-by noise	Microphone
Acceleration RMS	m/s^2 or g	Ride and shake severity	Accelerometer
FRF	response/input	Structural diagnosis	Modal test
Order spectrum	order vs rpm	Rotating source tracking	Order analysis
Transmissibility	ratio	Isolation efficiency	Mount/bushing test

Human Exposure and Whole-Body Vibration

- ISO 2631-1 defines methods for evaluating periodic, random, and transient whole-body vibration.
- The standard considers health, comfort, perception, and motion sickness under vehicle-induced vibration exposure.
- In automotive practice, seat rail and floor vibration are often interpreted with human comfort considerations in mind.

ISO 2631-1 is explicitly intended for motions transmitted through supporting surfaces in vehicles and machinery [web:18][web:24].

Body-in-White as the Main Structural Path

- The BIW is the dominant structure-borne transfer path from source subsystems to the passenger cabin.
- Its first torsional and first bending modes are key NVH indicators in early vehicle development.
- Reduced stiffness lowers modal frequencies and may increase shake and boom response.
- Body stiffness also influences squeak-rattle risk and dynamic handling behavior.

Technical literature links stiffness drops in BIW to first torsional and first bending modes, which govern global structural response [web:17][web:26].

Equation of Motion for Vehicle Structures

The linear multi-degree structural model is:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F(t)\}$$

where:

- $[M]$ = mass matrix,
- $[C]$ = damping matrix,
- $[K]$ = stiffness matrix,
- $\{F(t)\}$ = excitation vector.

Practical interpretation

NVH engineering aims to control $[K]$, $[C]$, and path coupling so that critical eigenfrequencies do not coincide with dominant excitation orders.

Natural Frequency Relation

For a single-degree-of-freedom model:

$$m\ddot{x} + cx + kx = 0$$

For undamped free vibration, with $c = 0$:

$$m\ddot{x} + kx = 0$$

Assume harmonic solution $x = X\sin(\omega_n t)$. Substituting gives:

$$-m\omega_n^2 X\sin(\omega_n t) + kX\sin(\omega_n t) = 0$$

Therefore,

$$\omega_n = \sqrt{\frac{k}{m}}$$

Engineering Meaning of $\omega_n = \sqrt{k/m}$

- Increase stiffness k to move resonance upward.
- Increase mass m to move resonance downward.
- The equation is the starting point for panel tuning, mount design, and structural reinforcement decisions.
- Any practical modification must consider simultaneous change in stiffness, mass, damping, packaging, and manufacturability.

Global and Local Modes in BIW

Global modes

- Whole-body torsion
- Whole-body bending
- Affect overall vehicle shake and boom

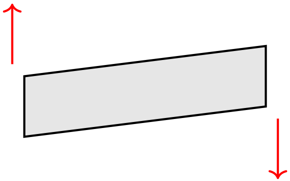
Local modes

- Roof panel, floor panel, shock tower, dash panel
- Affect local noise radiation and mount boundary stiffness
- Important for targeted optimization

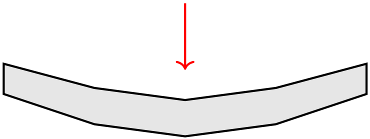
Literature on vehicle body dynamics emphasizes distinguishing global and local modes because both govern dynamic stiffness behavior [web:17].

Mode Shape Illustration

Torsional Mode



Bending Mode



Representative BIW Modal Data from Literature

Mode	Frequency	Observation
Local torsional mode	26.72 Hz	Front impact bar twisting reported in body-frame study
First global bending mode	35.31 Hz	Rear roof and rear floor deformation reported
Modes 1–10 example range	30.49–78.51 Hz	Global and local body modes reported in example NVH study

Example published values show body modes commonly appearing in the low tens of Hz, which is why structural tuning is a major design task [web:23][web:20].

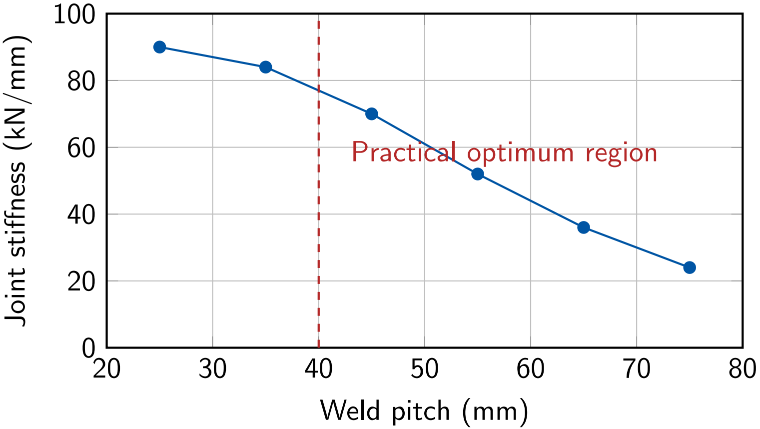
How Stiffening is Achieved in Industry

- Closed sections and ring structures around door apertures.
- Reinforcements at suspension towers, subframe mounts, and dash cross-members.
- Beads, ribs, embossments, and curvature increase panel bending rigidity.
- Structural adhesive and weld-bonding improve both stiffness continuity and damping.
- Roof rail and floor tunnel reinforcements are common practical interventions.

Weld Pitch and Joint Design

- Spot weld pitch directly influences local joint stiffness.
- Excessive pitch creates flexibility and reduces load path continuity.
- Too many welds increase manufacturing cost and may introduce thermal distortion.
- Weld-bonding is increasingly preferred where NVH refinement and crash performance must coexist.

Joint Stiffness Trend with Weld Pitch



Practice Slide: Structural Tuning Example

Given: A panel has mass $m = 4$ kg and stiffness $k = 90 \times 10^3$ N/m.
Required: Find the original natural frequency and the new frequency if stiffness increases by 25%. **Solution:**

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{90 \times 10^3}{4}} = 150 \text{ rad/s}$$

$$f_n = \frac{\omega_n}{2\pi} = 23.87 \text{ Hz}$$

If $k_{new} = 1.25k$, then

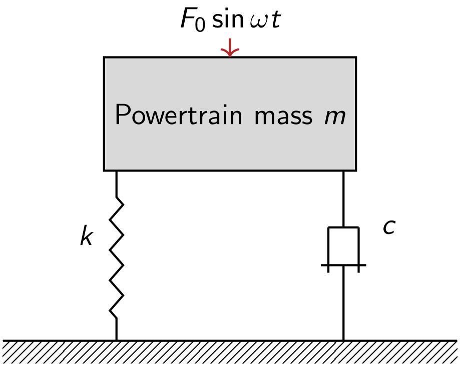
$$f_{new} = \frac{1}{2\pi}\sqrt{\frac{1.25 \times 90 \times 10^3}{4}} = 26.68 \text{ Hz}$$

Interpretation: local reinforcement shifts resonance upward by about 2.81 Hz.

When Isolation is Needed

- If source reduction is insufficient, the transmission path must be interrupted.
- Engine mounts, transmission mounts, exhaust hangers, cabin mounts, and suspension bushings all act as isolators.
- The goal is low transmitted force with acceptable static deflection and durability.

Mass–Spring–Damper Idealization of a Mount



Derivation of Transmissibility

For harmonic excitation, the equation is:

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin \omega t$$

The steady-state displacement amplitude is:

$$X = \frac{F_0/k}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}$$

where

$$r = \frac{\omega}{\omega_n}, \qquad \zeta = \frac{c}{2\sqrt{km}}$$

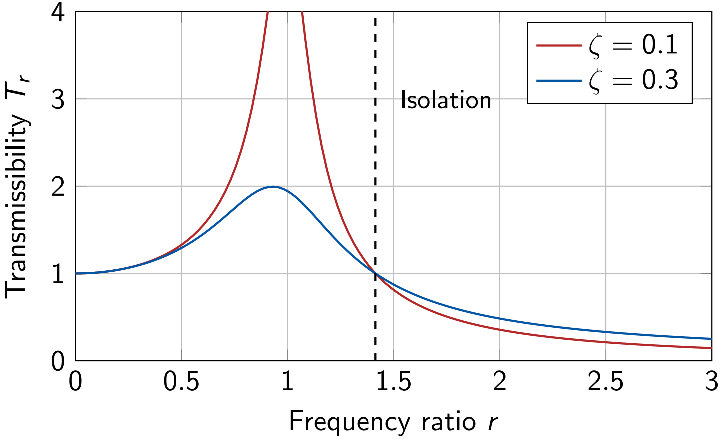
The transmitted force contains spring and damper components, giving:

$$T_r = \sqrt{\frac{1 + (2\zeta r)^2}{(1 - r^2)^2 + (2\zeta r)^2}}$$

Interpretation of Transmissibility

- $T_r > 1$: transmitted force is amplified.
- $T_r = 1$: no net benefit.
- $T_r < 1$: effective isolation region.
- Isolation starts at approximately $r > \sqrt{2}$ for the force transmissibility model.
- Increased damping reduces resonance peak but may slightly worsen high-frequency isolation.

Transmissibility Curves for Two Damping Ratios



Hands-on Problem: Mount Frequency Selection

Given: Powertrain mass $m = 180$ kg, required mount natural frequency $f_n = 12$ Hz.
Required: Find equivalent mount stiffness k_{eq}.

$$\omega_n = 2\pi f_n = 2\pi(12) = 75.40 \text{ rad/s}$$

$$k_{eq} = m\omega_n^2 = 180(75.40)^2 = 1.02 \times 10^6 \text{ N/m}$$

Interpretation: the combined mount set should provide approximately 1.02×10^6 N/m equivalent stiffness to achieve a 12 Hz rigid-body mode.

Engine Mount Design in Industry

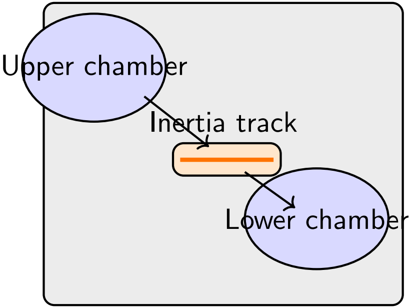
- Automotive teams tune mount stiffness and orientation to place rigid-body powertrain modes in acceptable frequency bands.
- DOE and multi-body/FE methods are used to identify the most influential stiffness directions for reducing transmitted vibration.
- Published work reports improvement in driver seat rail vibration after mount stiffness optimization using mixed FE and multi-body approaches.

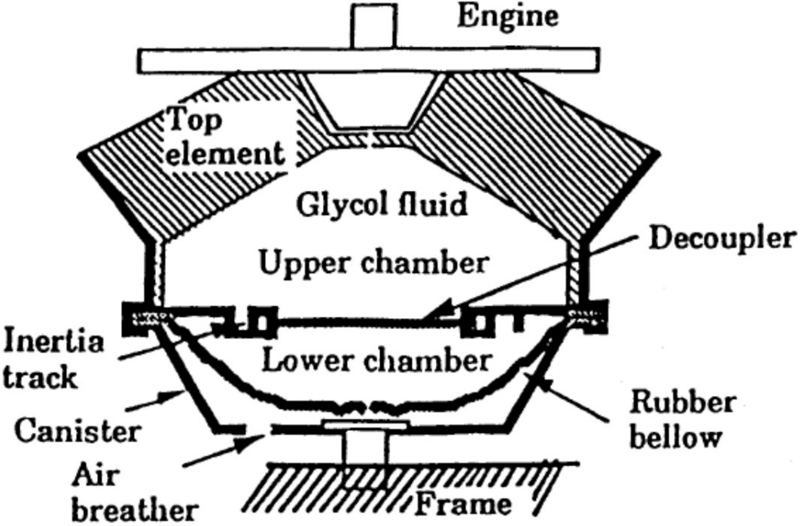
Technical literature describes mount stiffness tuning through DOE, FE, and multi-body simulation for reduced transmitted vibration [web:19].

Elastomeric, Hydraulic, and Active Mounts

Mount type	Strength	Limitation	Typical use
Elastomeric	Simple, durable	Fixed properties	Economy vehicles
Hydraulic	Frequency-dependent damping	More complex	Passenger cars, premium NVH
Active / semi-active	Real-time adaptation	Cost and controls	Premium and advanced platforms

Hydraulic Mount Functional Diagram




Engine
Top
element
Glycol fluid
Upper chamber
Decoupler
Inertia
track
Lower chamber
Rubber
bellow
Canister
Air
breather
Frame

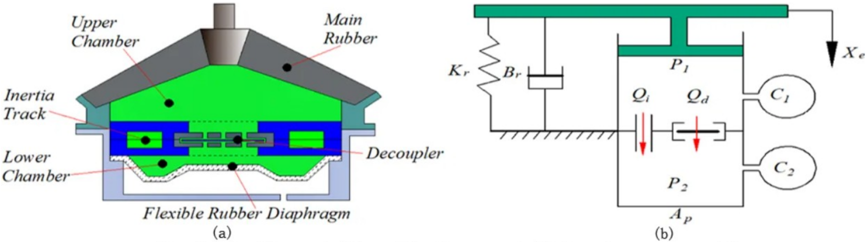


Fig. 1 Hydraulic mount (a) profile diagram and (b) lumped parameter model

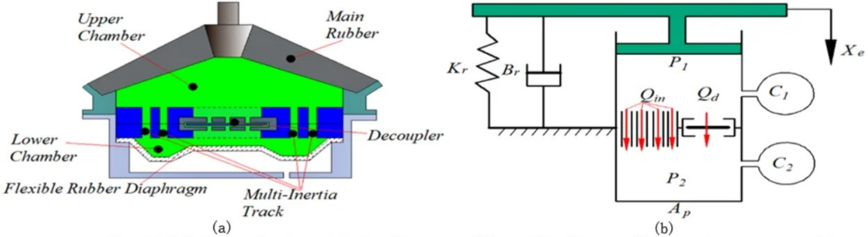
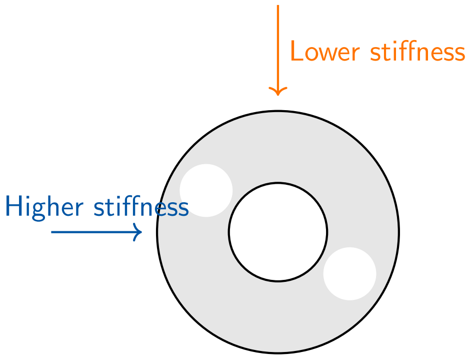


Fig. 10 Multi-inertia channel hydraulic mount (a) profile diagram (b) lumped parameter model

Bushings and Directional Stiffness

- Suspension bushings are designed to be compliant in some directions and stiff in others.
- Soft longitudinal stiffness improves impact isolation and ride comfort.
- High lateral stiffness supports handling, alignment retention, and braking stability.
- Voids and bonded elastomer geometry create anisotropic stiffness behavior.

Voided Bushing Conceptual Diagram



Damping Treatments in Vehicle Design

- Constrained-layer damping reduces panel vibration amplitude.
- Mastic pads are used at locally active regions to suppress panel resonance.
- Structural adhesive adds both stiffness continuity and additional damping.
- Damping is particularly useful when frequency shift alone is insufficient or impractical.

Acoustic Countermeasures Used in Industry

Barrier treatments

- Dash insulator
- Mass-loaded vinyl
- Acoustic glazing

Absorptive treatments

- Foam and fiber absorbers
- Headliner absorbers
- Carpet and underlay systems

Practical distinction: barriers block transmission, absorbers reduce reflected sound energy, damping reduces structural vibration.

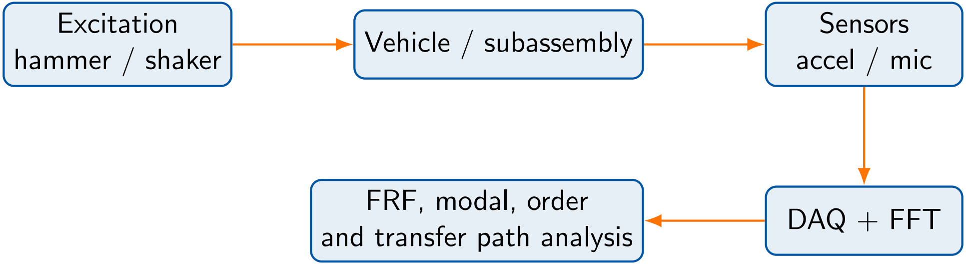
Experimental NVH Workflow in Automotive Companies

1. Benchmark and define target metrics.
2. Conduct modal test of BIW, subframe, or mount system.
3. Measure FRF and identify problematic modes or transfer paths.
4. Run order analysis for rotating sources such as engine, motor, gear, and tire.
5. Modify stiffness, damping, or isolation and retest.
6. Correlate test results with FE / MBD simulation.

Modal Testing and FRF Concept

- An impact hammer or shaker provides known excitation force.
- Accelerometers or microphones measure system response.
- Frequency Response Function (FRF) relates output response to input excitation.
- Peaks in FRF indicate resonances; phase behavior supports mode identification.

Typical NVH Measurement Chain



CAE in Modern NVH Development

- Finite element models are used for modal, harmonic, acoustic, and random vibration studies.
- Multi-body dynamics supports mount tuning, rigid-body mode placement, and load path study.
- Published engine mount work combines constrained modal analysis, dynamic stiffness simulation, and test validation.
- Random vibration responses and RMS accelerations are often compared with ISO-based comfort considerations.

Published studies report FE-based mount analysis, dynamic stiffness prediction, random vibration analysis, and experimental correlation [web:16][web:22].

Actual Use Cases in Automotive Industry

- **Idle shake reduction:** optimize engine mount stiffness and bracket stiffness.
- **Road boom control:** shift BIW bending and floor modes away from tire-road excitation bands.
- **EV tonal noise suppression:** reduce motor/inverter whine transmission and cabin radiation.
- **Seat comfort refinement:** measure and limit floor and seat rail vibration considering whole-body exposure.
- **Prototype troubleshooting:** use transfer path analysis to locate dominant transmission routes.

Case Study Format for Students

Problem statement: steering wheel shake at idle.

Possible causes:

- Mount rigid-body mode close to excitation order.
- Bracket local flexibility.
- Dash cross-member or steering column support resonance.

Actions:

- Recalculate mount stiffness and damping.
- Increase bracket local stiffness.
- Measure FRF from engine mount to steering column.

Hands-on Tutorial Exercise for Classroom

Exercise: A mount-isolated machine has $m = 60$ kg, $k = 1.2 \times 10^5$ N/m, and damping ratio $\zeta = 0.15$. If excitation frequency is 30 Hz, determine whether isolation occurs. **Steps:**

1. Compute $\omega_n = \sqrt{k/m} = 44.72$ rad/s.
2. Find $f_n = \omega_n/(2\pi) = 7.12$ Hz.
3. Compute ratio $r = f/f_n = 30/7.12 = 4.21$.
4. Since $r > \sqrt{2}$, the system lies in the isolation region.

What Recruiters Expect from NVH Engineers

- Understanding of vibration fundamentals, modal analysis, and frequency-domain interpretation.
- Familiarity with accelerometers, microphones, shakers, DAQ, and FFT-based analysis.
- Ability to work with FE models, mount tuning, and test-correlation activities.
- Practical appreciation of ride comfort, sound quality, durability, and regulatory constraints.
- Cross-functional collaboration with body, powertrain, chassis, CAE, and test teams.

Summary

- NVH combines vibration theory, structural dynamics, acoustics, material damping, and human perception.
- Natural frequency, damping ratio, and transmissibility form the analytical core of isolation design.
- BIW modes, mount tuning, and acoustic treatments are central to real vehicle refinement.
- Automotive practice relies on both CAE and experiment for diagnosis and optimization.
- Hands-on calculations and test interpretation are essential for mastering NVH engineering.

Recommended Reading and Standards

- M. P. Norton, *Fundamentals of Noise and Vibration Analysis for Engineers*.
- Vehicle NVH and engine mount technical papers from SAE and related journals.
- ISO 2631-1 for whole-body vibration evaluation in vehicles.
- FE and modal testing literature on BIW and mount dynamic behavior.

Thank You

Automotive NVH Control: Practice-Oriented Course Deck

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Q1. Engine Mount Stiffness Design

Question: A powertrain of mass 210 kg is to be supported such that its vertical rigid-body natural frequency is 11 Hz. Determine: (i) equivalent system stiffness, (ii) stiffness of each mount if 3 identical mounts share the load equally. **Solution:**

$$\omega_n = 2\pi f_n = 2\pi(11) = 69.12 \text{ rad/s}$$

$$k_{eq} = m\omega_n^2 = 210(69.12)^2 = 1.003 \times 10^6 \text{ N/m}$$

$$k_{mount} = \frac{k_{eq}}{3} = 3.34 \times 10^5 \text{ N/m}$$

Answer: Equivalent stiffness $= 1.003 \times 10^6$ N/m, stiffness per mount $= 3.34 \times 10^5$ N/m.

Q2. Force Transmissibility of a Mount

Question: For $m = 160$ kg, $k = 8.5 \times 10^5$ N/m, $\zeta = 0.18$, and excitation frequency $f = 28$ Hz, find f_n, r, and transmissibility T_r. **Solution:**

$$\omega_n = \sqrt{\frac{k}{m}} = 72.89 \text{ rad/s}, \qquad f_n = \frac{72.89}{2\pi} = 11.60 \text{ Hz}$$

$$r = \frac{28}{11.60} = 2.414$$

$$T_r = \sqrt{\frac{1 + (2 \times 0.18 \times 2.414)^2}{(1 - 2.414^2)^2 + (2 \times 0.18 \times 2.414)^2}} = 0.224$$

Answer: $f_n = 11.60$ Hz, $r = 2.414$, $T_r = 0.224$; hence the system is in the isolation region.

Q3. Panel Stiffening and Resonance Shift

Question: A floor panel has $m = 5$ kg and $k = 120 \times 10^3$ N/m. Reinforcement increases stiffness by 35% and adds 0.6 kg mass. Find original and new natural frequencies. **Solution:**

$$f_{n1} = \frac{1}{2\pi}\sqrt{\frac{120 \times 10^3}{5}} = 24.65 \text{ Hz}$$

$$k_2 = 1.35(120 \times 10^3) = 162 \times 10^3 \text{ N/m}, \qquad m_2 = 5.6 \text{ kg}$$

$$f_{n2} = \frac{1}{2\pi}\sqrt{\frac{162 \times 10^3}{5.6}} = 27.06 \text{ Hz}$$

$$\%\Delta f = \frac{27.06 - 24.65}{24.65} \times 100 = 9.78\%$$

Answer: Original frequency $= 24.65$ Hz, new frequency $= 27.06$ Hz, increase $= 9.78\%$.

Q4. Bushing Isolation Under Harmonic Road Input

Question: A bushing has $k = 2.4 \times 10^5$ N/m, supported mass $m = 32$ kg, damping ratio $\zeta = 0.12$, and excitation frequency $f = 20$ Hz. Find f_n, r, and transmissibility. **Solution:**

$$\omega_n = \sqrt{\frac{2.4 \times 10^5}{32}} = 86.60 \text{ rad/s}, \qquad f_n = 13.78 \text{ Hz}$$

$$r = \frac{20}{13.78} = 1.451$$

$$T_r = \sqrt{\frac{1 + (2 \times 0.12 \times 1.451)^2}{(1 - 1.451^2)^2 + (2 \times 0.12 \times 1.451)^2}} = 1.783$$

Answer: $f_n = 13.78$ Hz, $r = 1.451$, $T_r = 1.783$; the system is close to threshold and still amplifies force.

Q5. Modal Separation and Design Check

Question: A BIW has first torsional mode at 27 Hz and first bending mode at 31 Hz. After modification, torsional mode rises by 9% and bending mode rises by 6%. Find modified frequencies and modal separation. **Solution:**

$$f_{t,new} = 27(1.09) = 29.43 \text{ Hz}$$

$$f_{b,new} = 31(1.06) = 32.86 \text{ Hz}$$

$$\Delta f_{old} = 31 - 27 = 4 \text{ Hz}, \qquad \Delta f_{new} = 32.86 - 29.43 = 3.43 \text{ Hz}$$

Answer: Modified torsional mode = 29.43 Hz, bending mode = 32.86 Hz, separation changes from 4.00 Hz to 3.43 Hz.

Q6. Static Deflection of an Engine Mount Set

Question: An engine of mass 180 kg is mounted on 4 identical mounts. If each mount has vertical stiffness 75×10^3 N/m, find total static deflection of the system under engine weight.
Solution:

$$k_{eq} = 4(75 \times 10^3) = 3.0 \times 10^5 \text{ N/m}$$

$$W = mg = 180(9.81) = 1765.8 \text{ N}$$

$$\delta = \frac{W}{k_{eq}} = \frac{1765.8}{3.0 \times 10^5} = 5.89 \times 10^{-3} \text{ m}$$

Answer: Static deflection = 5.89 mm.

Q7. Damping Coefficient from Damping Ratio

Question: For a mount-isolated mass $m = 95$ kg and stiffness $k = 3.8 \times 10^5$ N/m, the damping ratio is $\zeta = 0.22$. Determine the viscous damping coefficient c. **Solution:**

$$c_c = 2\sqrt{km} = 2\sqrt{(3.8 \times 10^5)(95)} = 12016.7 \text{ N s/m}$$

$$c = \zeta c_c = 0.22(12016.7) = 2643.7 \text{ N s/m}$$

Answer: Damping coefficient $c = 2.64 \times 10^3$ N s/m.

Q8. Resonance Check at Engine Idle

Question: An engine idles at 900 rpm. A mount mode exists at 15 Hz. Determine the excitation frequency at idle and comment on resonance risk. **Solution:**

$$f = \frac{900}{60} = 15 \text{ Hz}$$

The excitation frequency exactly equals the mount mode frequency. **Answer:** Excitation frequency $= 15$ Hz; strong resonance risk exists at idle, so mount tuning is required.

Q9. FRF-Based Dynamic Amplification Estimate

Question: In a modal test, input hammer force amplitude at resonance is 80 N and accelerometer response magnitude is measured as 32 m/s^2. Determine the acceleration FRF magnitude at resonance. **Solution:**

$$|H(\omega)| = \frac{\text{response}}{\text{input}} = \frac{32}{80} = 0.40\ (\text{m/s}^2)/\text{N}$$

Answer: FRF magnitude at resonance $= 0.40\ (\text{m/s}^2)/\text{N}$.

Q10. Seat Vibration RMS Reduction

Question: Before mount redesign, seat rail vibration RMS is 0.92 m/s^2. After redesign it becomes 0.54 m/s^2. Determine the percentage reduction in vibration level. **Solution:**

$$\% \text{ reduction} = \frac{0.92 - 0.54}{0.92} \times 100 = 41.30\%$$

Answer: RMS vibration reduction = 41.3%.