Session-5

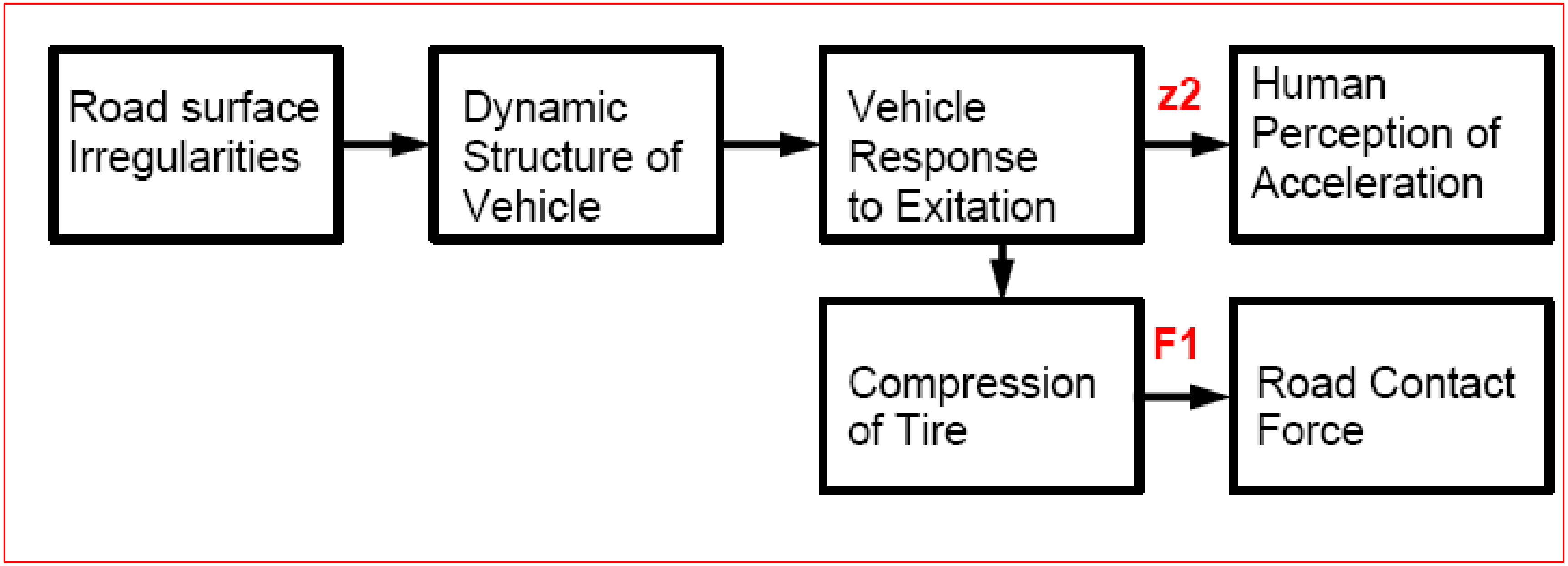
Friday, 18-12-2009; 11.30 AM – 1.00 PM

Excitation Sources and Vehicle Responses

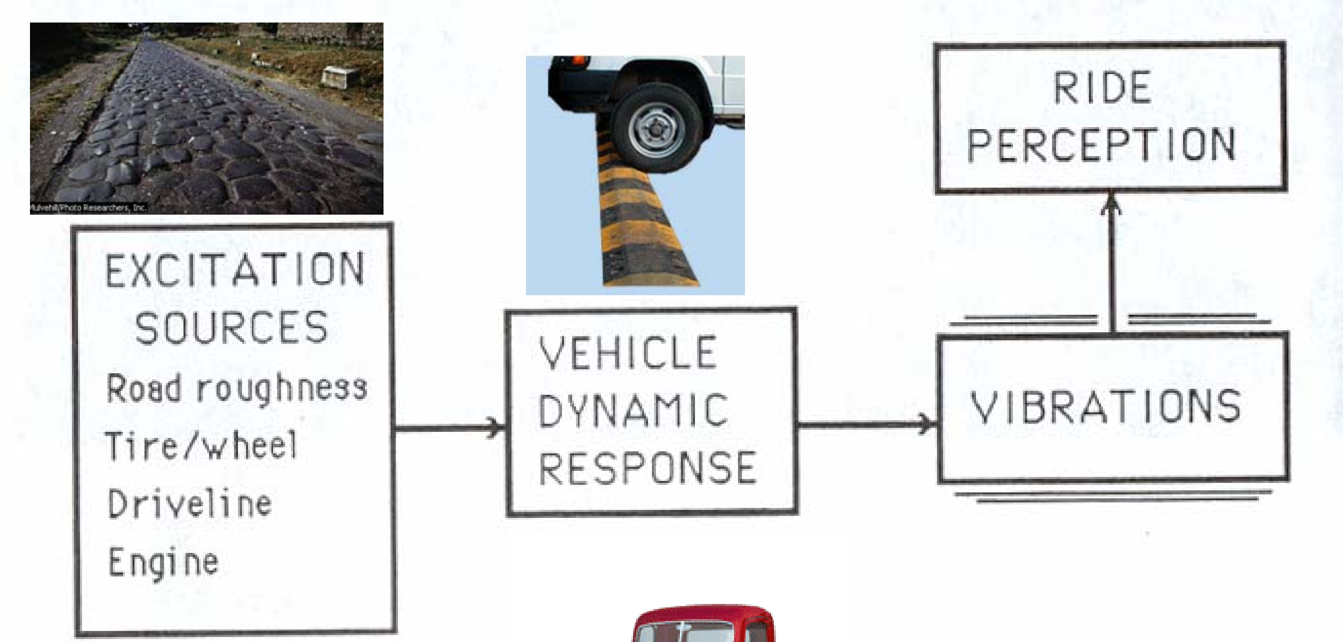
Ride Dynamic System

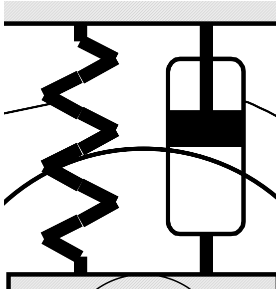
- Vehicle is a dynamic system. It exhibits vibration in response to excitation inputs
- The vehicle vibration response determine the magnitude and direction of vibration imposed on the passenger compartment, and determines the passenger's perception of the vehicle
- To understand ride, it is required to understand,
  1. Ride excitation sources
  2. Basic mechanics of vehicle vibration response
  3. Human perception and tolerance of vibrations

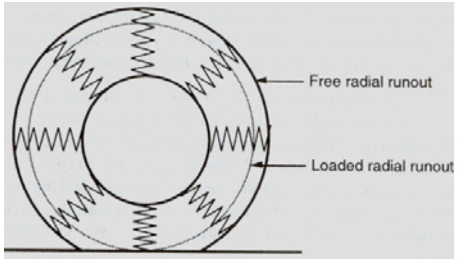
Ride Dynamic System

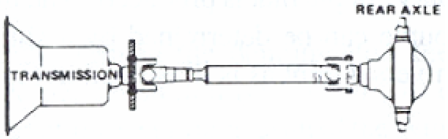


Ride Dynamic System









Road Surfaces

Classification	Example	Description	Design Speed
Arterial		Provides the highest level of service at the greatest speed for the longest uninterrupted distance, with some degree of access control.	30-60 mph
Collector		Provides a less highly developed level of service at a lower speed for shorter distances by collecting traffic from local roads and connecting them with arterials.	30 mph or higher
Local		Consists of all roads not defined as arterials or collectors; primarily provides access to land with little or no through-movement.	20-30 mph

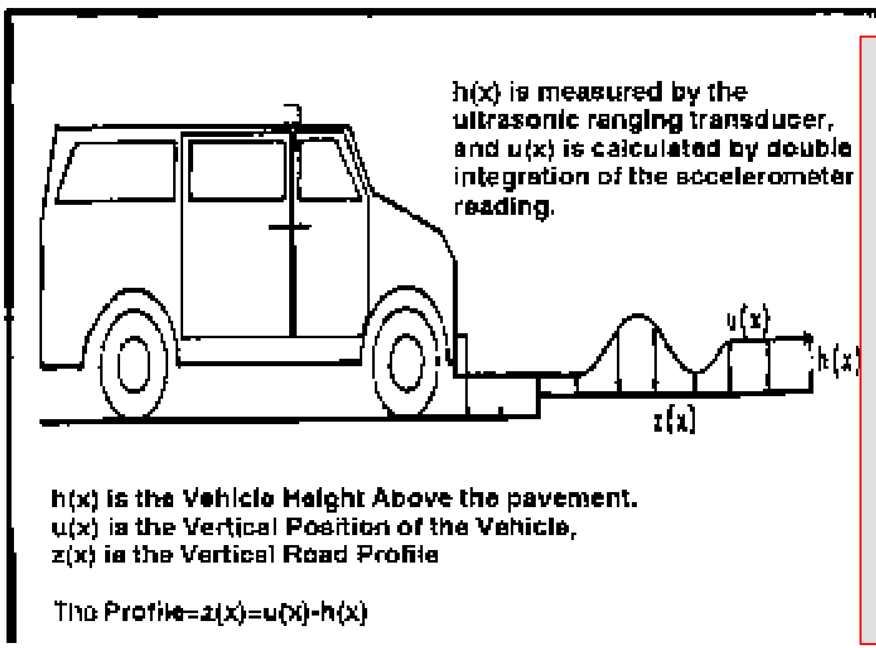
Road- Power Spectral Density (PSD)

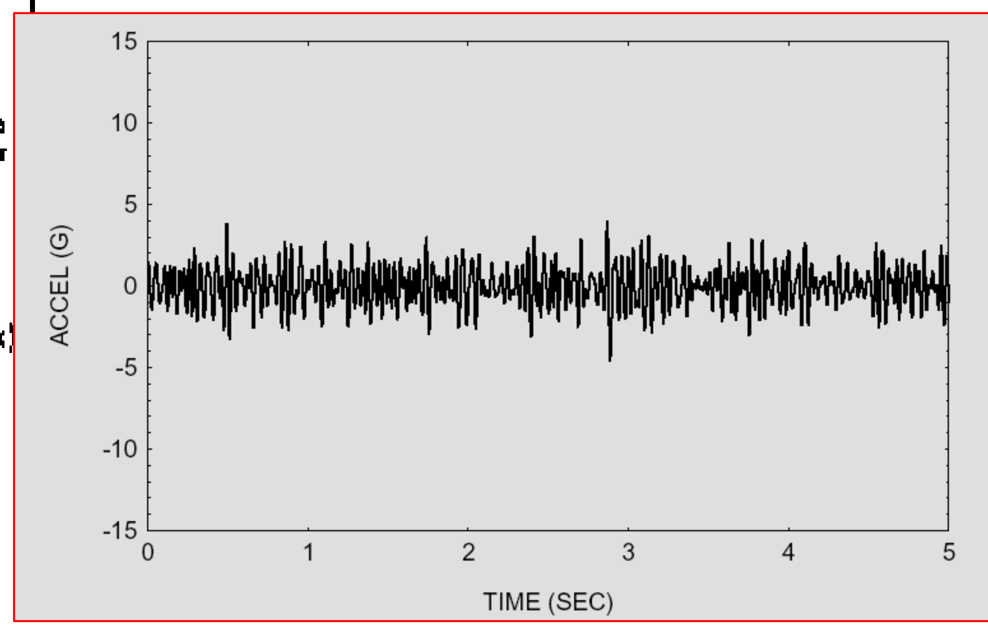
The distribution of signal power with spatial frequency domain

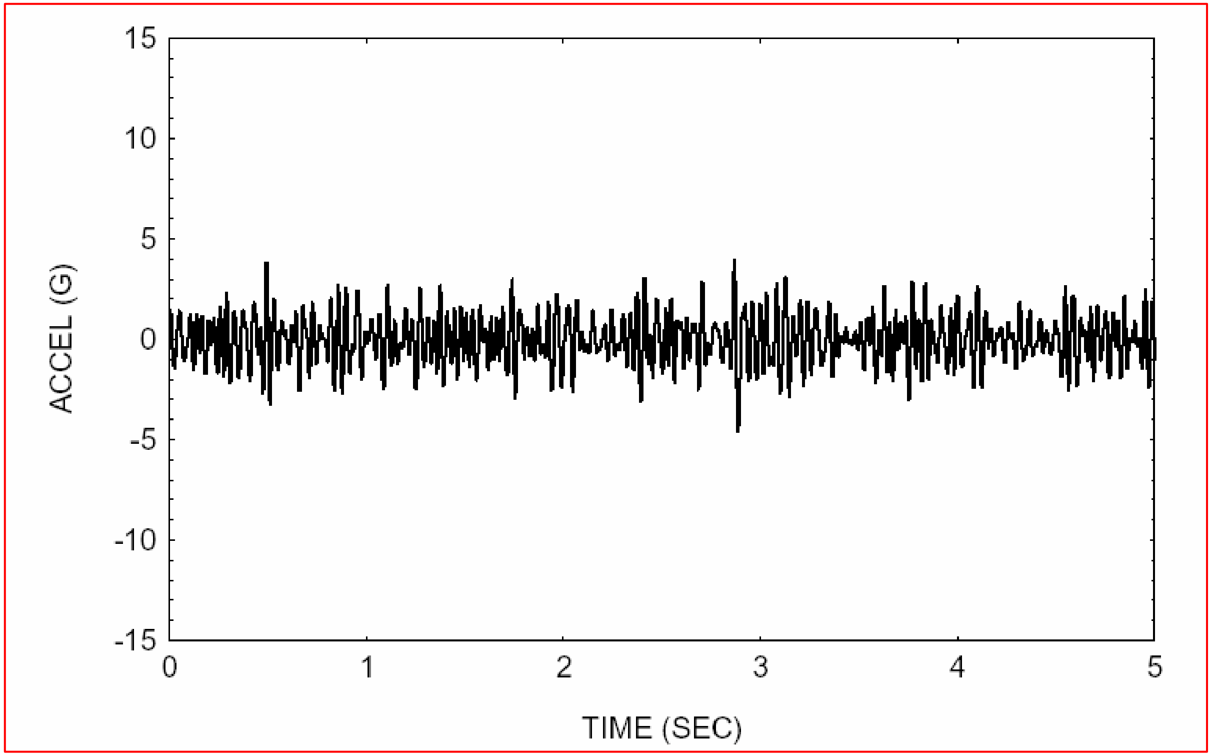
$$S(n\Omega_0) = \frac{z_n^2}{2\Delta\Omega} = \bar{z}_n^2 / \Delta\Omega \qquad \text{m}^2\text{ /cycles/m}$$

$$z = z_0 \sin \omega t : \omega - angular\ \ frequency\ rad / s$$

$$z = z_0 \sin \Omega x : \Omega - spatial\ \ frequency = \frac{2\pi}{l} rad(cycles)/m$$



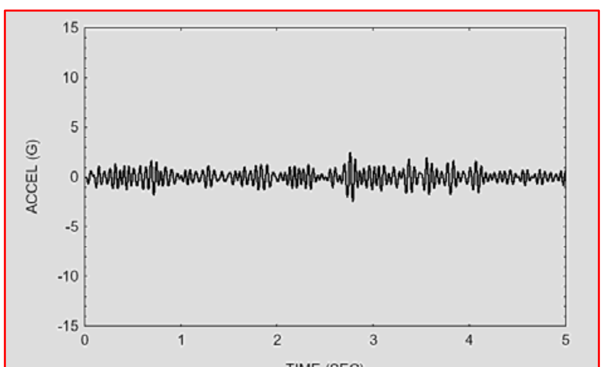




10 Hz and 40 Hz.

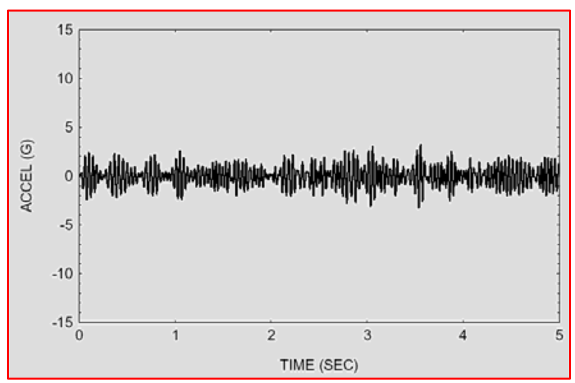
Overall Level = 1.49 GRMS

Band Pass Filtered



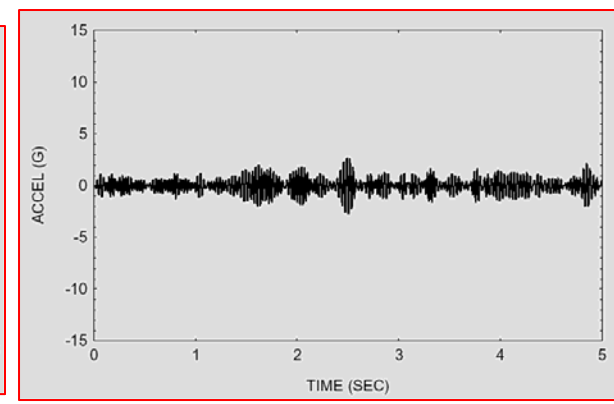
10 Hz and 20 Hz.

Overall Level = 0.68 GRMS



20 Hz and 30 Hz.

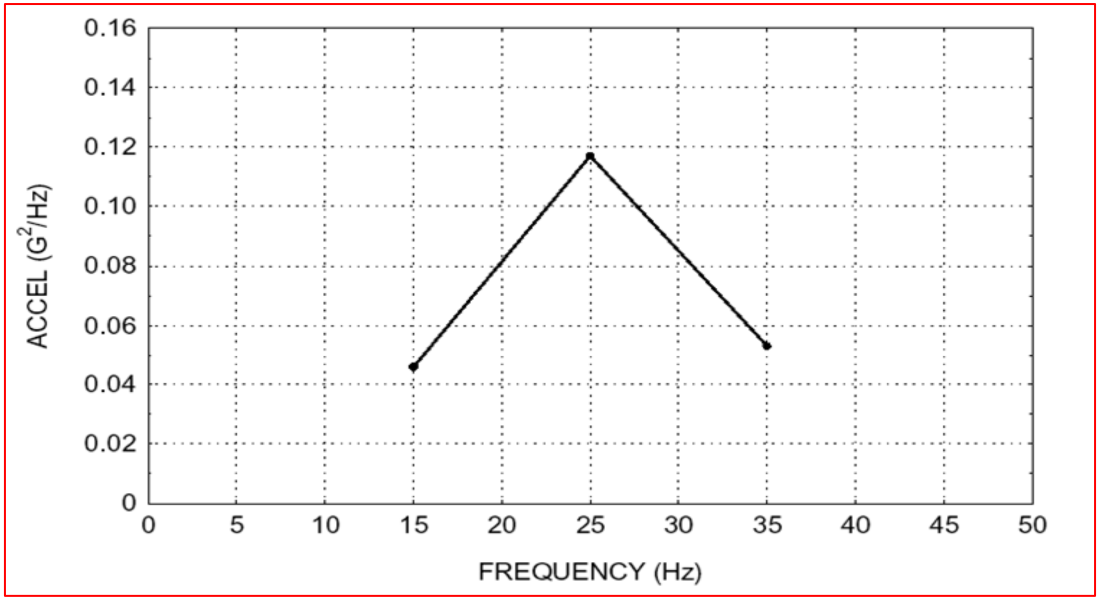
Overall Level = 1.08 GRMS



30 Hz and 40 Hz.

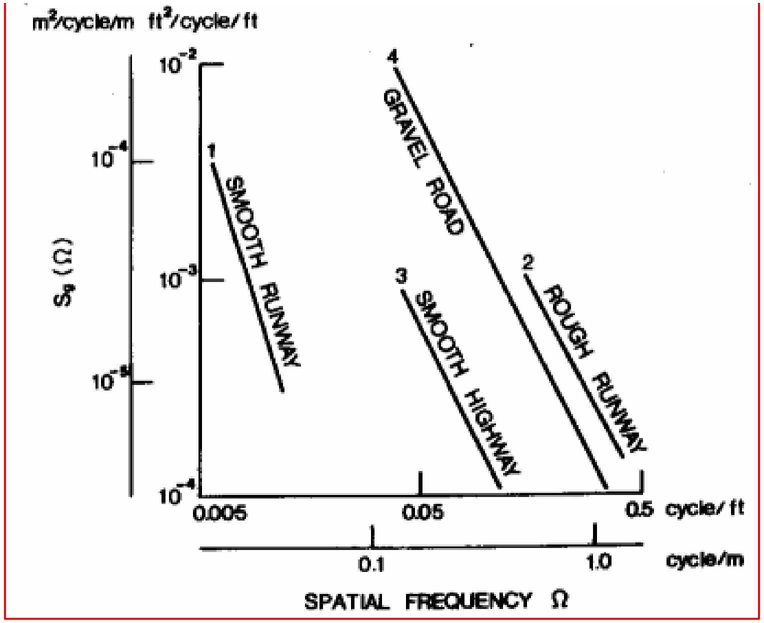
Overall Level = 0.73 GRMS

Table 2. Power Spectral Density Calculation					
Bandpass Filter	Band Center Frequency (Hz)	Overall Level (GRMS)	Overall Level ^2 (GRMS^2)	Bandwidth (Hz)	PSD (GRMS^2/Hz)
10 Hz to 20 Hz	15	0.68	0.46	10	0.046
20 Hz to 30 Hz	25	1.08	1.17	10	0.117
30 Hz to 40 Hz	35	0.73	0.53	10	0.053



Road- Power Spectral Density (PSD)

Road Class	Degree of Roughness $S_g(\Omega_0)$, 10^{-6} m^2/cycles/m	
	Range	Geometric Mean
A (Very Good)	<8	4
B (Good)	8–32	16
C (Average)	32–128	64
D (Poor)	128–512	256
E (Very Poor)	512–2048	1024
F	2048–8192	4096
G	8192–32,768	4096
H	>32,768	16384



Surface Irregularities

$S_g(f)$

Input

Vehicle System

Abs (H(f))

Output

Vehicle Vibrations

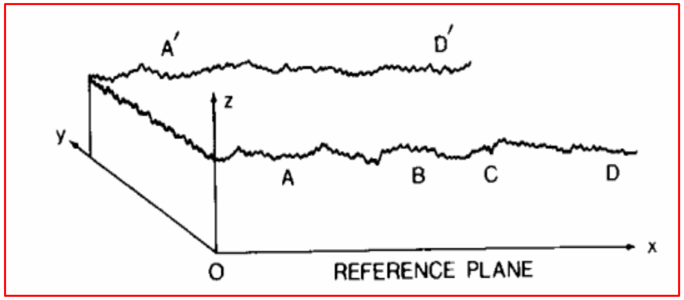
$S_v(f)$= square (Abs(h(f))) $S_g(f)$

$$|H(f)| = \left|(2\pi f)^2 \sqrt{\frac{1+(2\xi f/f_n)^2}{[1-(f/f_n)^2]^2+[2\xi f/f_n]^2}}\right|$$

$$z_v(t) = |H(f)|\, z_g(t)$$

$$\bar{z}_v^{\,2} = |H(f)|^2\, \bar{z}_g^{\,2}$$

$$S_v = |H(f)|^2\, S_g$$

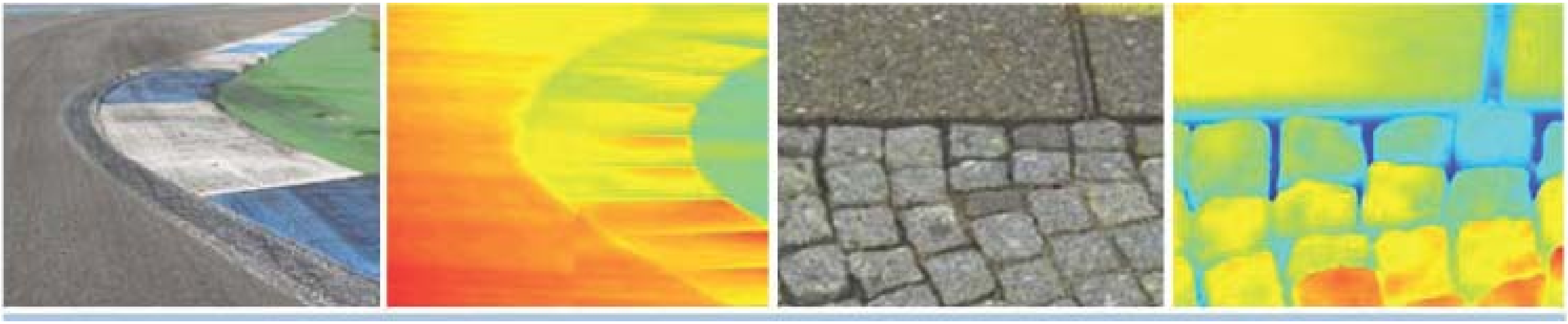


$$\bar{z}_n^{\,2} = \frac{1}{l_{wn}} \int_0^{lwn} [z_n \sin(\frac{2\pi x}{l_{wn}})]^2\, dx = \frac{z_n^2}{2}$$

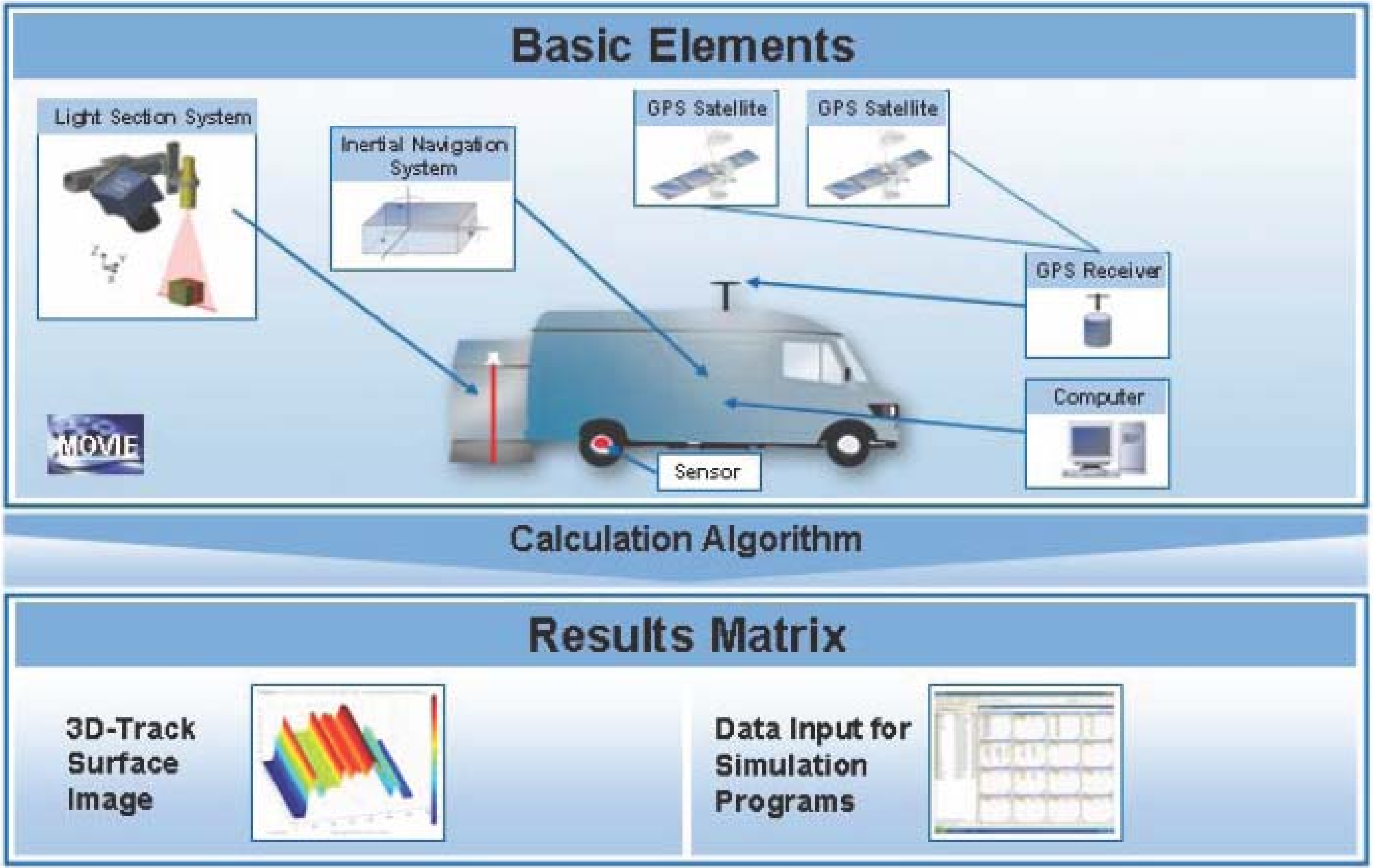
$$S(n\Omega_0) = \frac{z_n^2}{2\Delta\Omega} = \bar{z}_n^{\,2} / \Delta\Omega$$

3D -Road Models

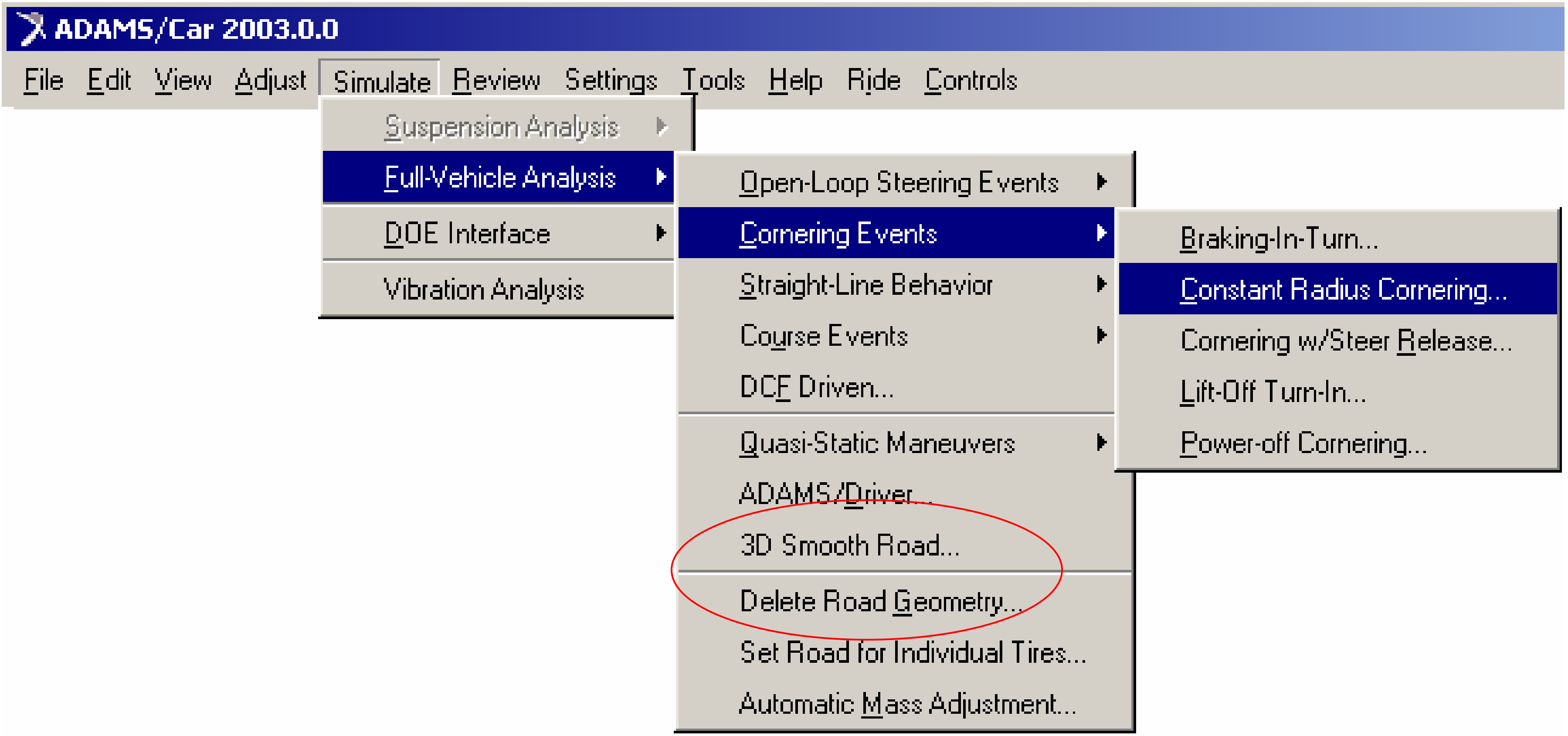
3D-Track
• 3D-Track enables high-resolution track measurements of any type of road surface (asphalt, rough roads, public roads, proving grounds, race tracks)
• 3D-Track can use for various application in product development or racing environments such as simulation
• 3D-Track opens up new possibilities of researching road influences with respect to ride, performance and noise
• "Unique" – it combines quick measurements with high-resolution images. The measurements can even be made on public roads in moving traffic



3D -Road Models

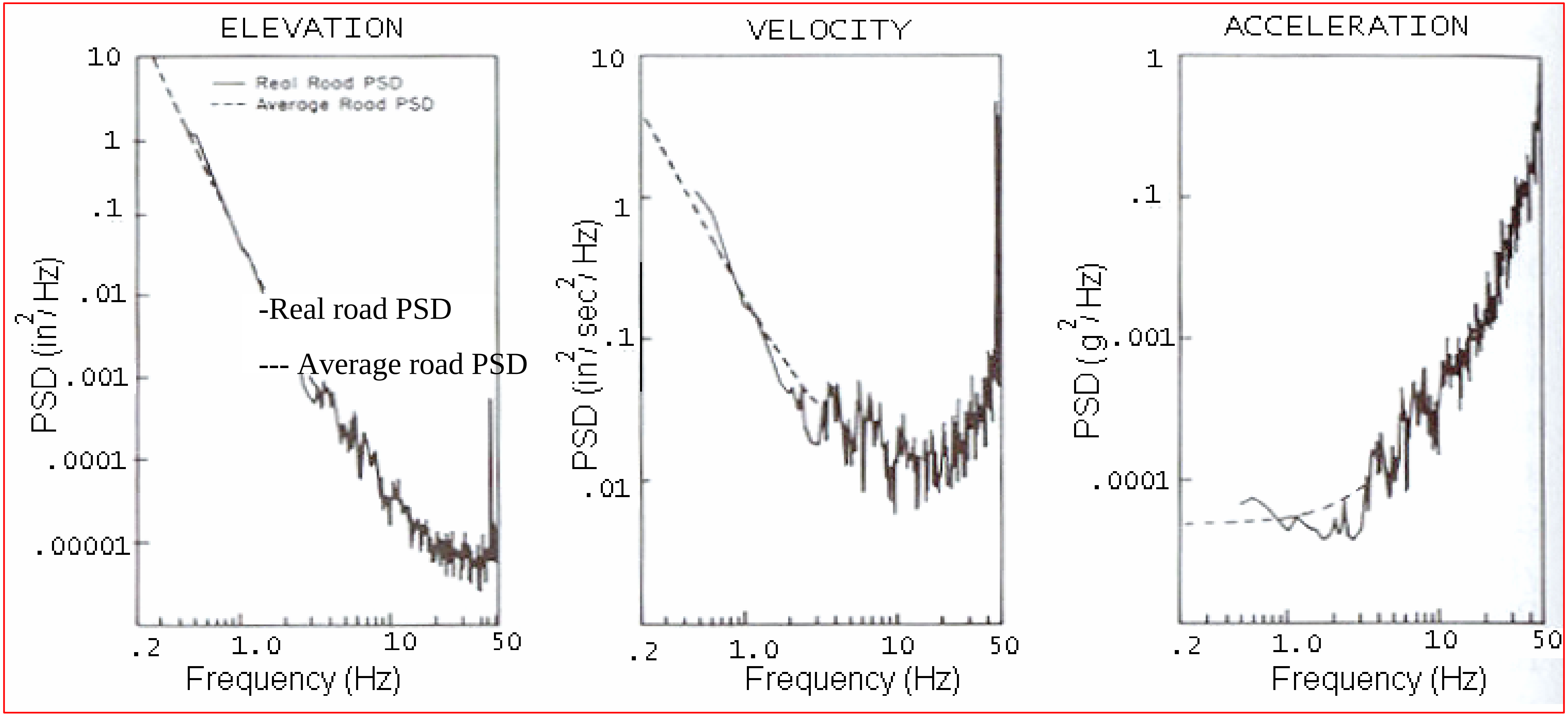


3D Road Models



Road Roughness as an Acceleration Input

- The roughness in a road is the deviation in elevation seen by a vehicle as it moves along the road
- The roughness acts as a vertical displacement input to the wheels, thus exciting ride vibrations
- Most common and meaningful measure of ride vibration is the acceleration produced
- Roughness should be viewed as an acceleration input at the wheels
- Assume a speed of travel, convert elevation profile to displacement,differentiate and get velocity, once again differentiate and get acceleration


ELEVATION
Real Road PSD
Average Road PSD
PSD (in^2/Hz)
10
1
.1
.01
.001
.0001
.00001
.2
1.0
10
50
Frequency (Hz)
-Real road PSD
--- Average road PSD
VELOCITY
PSD ($(in/sec)^2$/Hz)
10
1
.1
.01
Frequency (Hz)
ACCELERATION
PSD (g^2/Hz)
1
.1
.001
.0001
Frequency (Hz)

PSD and Roll Excitation

- The difference in elevation between the left and right road profile points represents a roll excitation input to the vehicle.
- The PSD for the roll displacement input to a vehicle is typically similar to that for the elevation as was shown earlier, although its amplitude is attenuated at wavenumbers below 0.02 to 0.03 cycle/foot.

Tire and Wheel Excitations

- Mass imbalance in Tire and Wheel assembly
- Dimensional variations in Tire
- Stiffness variations in Tire

- Mass imbalance in Tire and Wheel assembly

$F_i = (m\ r)\ \omega^2$

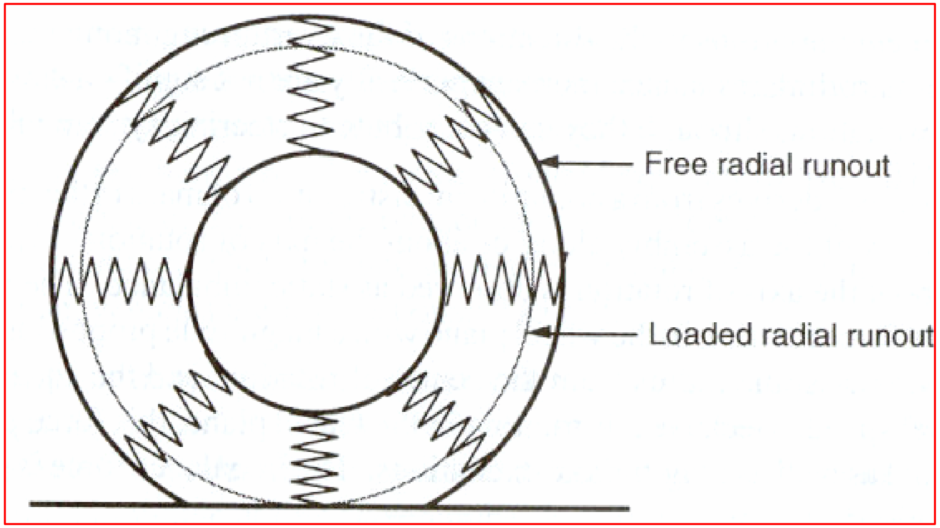
Where:

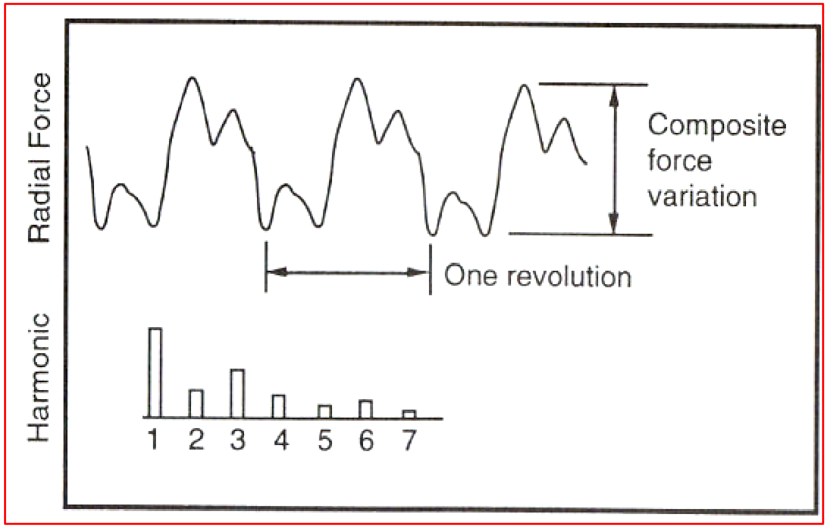
F_i = Imbalance force

m r = The imbalance magnitude (mass times radius)

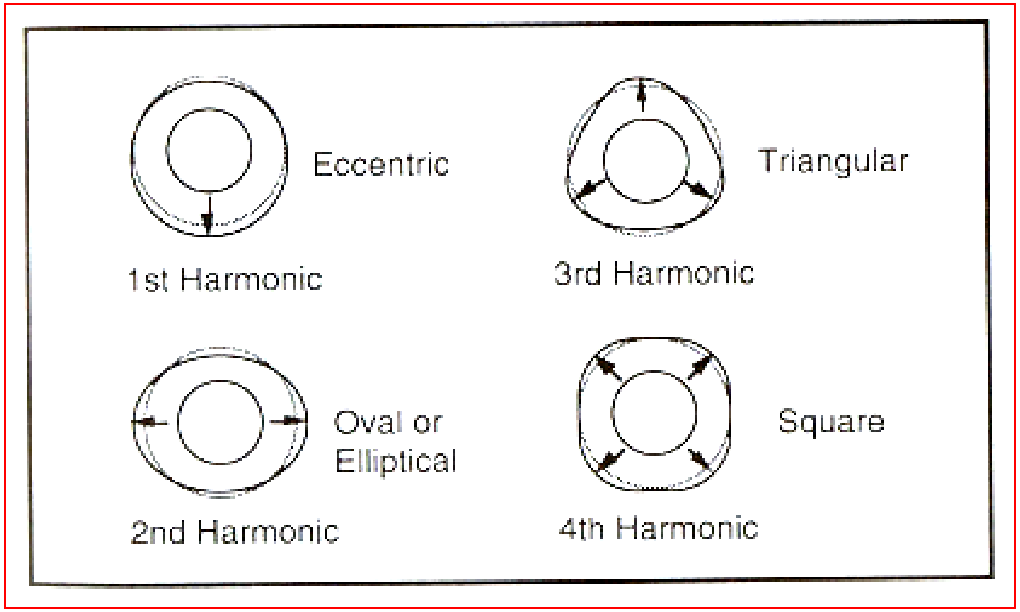
ω = The rotational speed (radians/second)

Dimensional Variations in Tyre

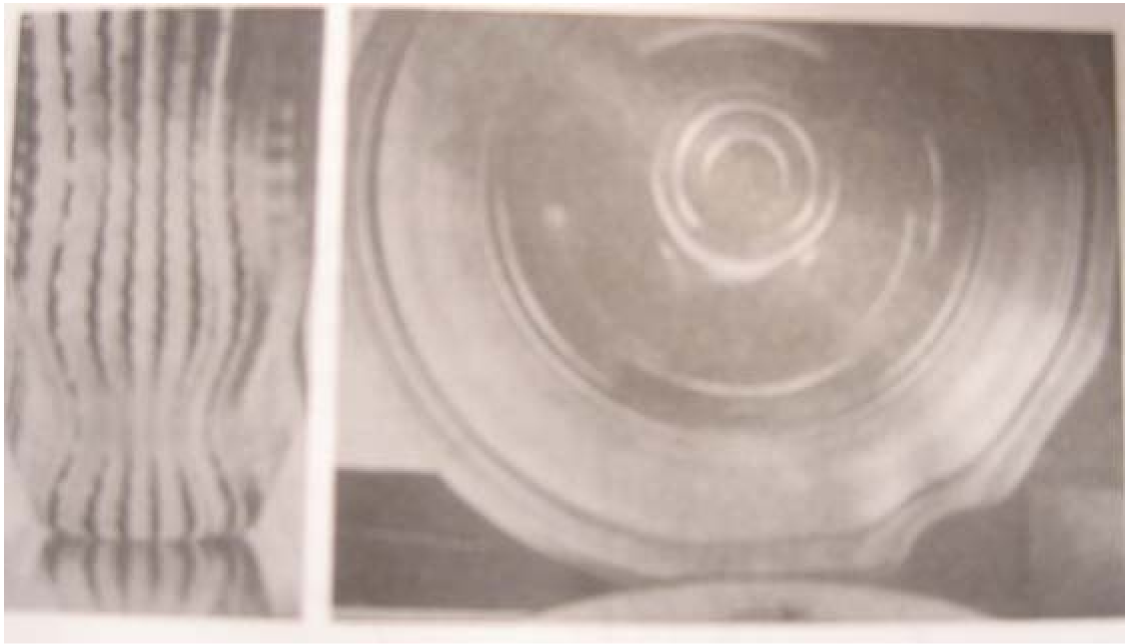




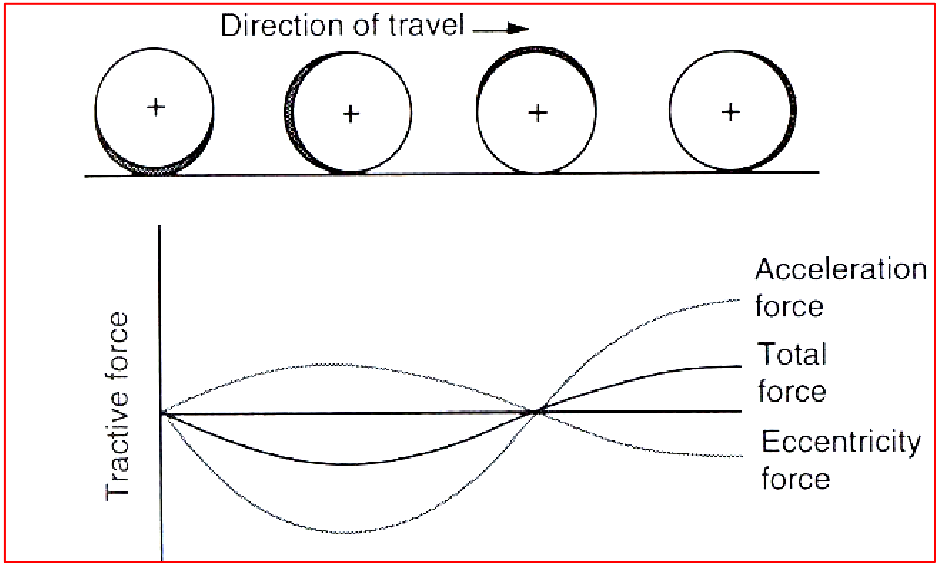
- The tire, being an elastic body analogous to an array of radial springs, may exhibit variations in stiffness about its circumference.



Standing Waves



Stiffness Variations in Tyre

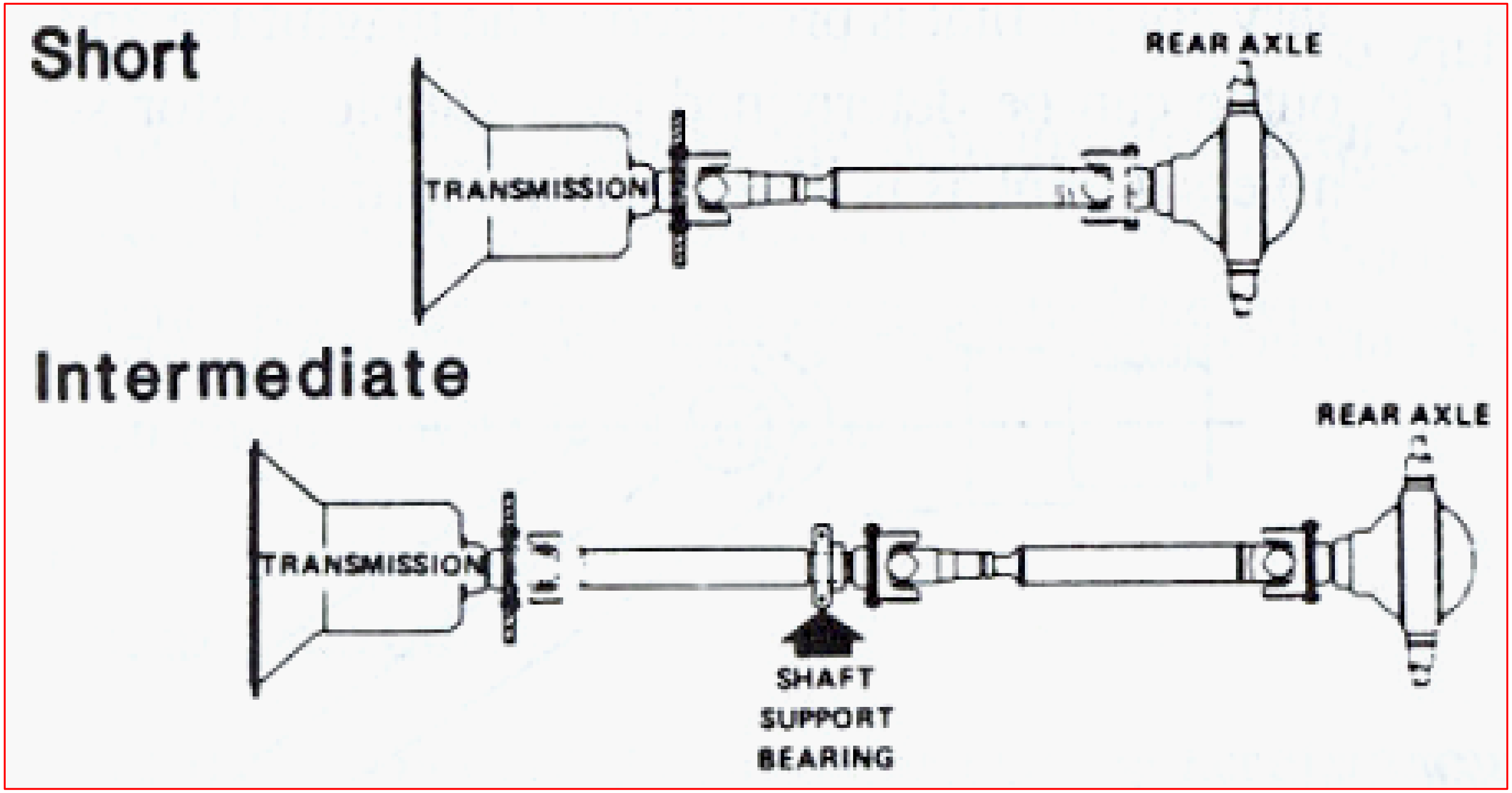


- Tractive force variations arise from dimensional and stiffness non-uniformities
- With eccentricity, even at low speed the axle must roll up and down the "hill" represented by the variation in radius of the wheel assembly. Thus a longitudinal force is involved and a tractive force variation is observed. Its magnitude will be dependent on the load carried and the amount of eccentricity; however, it is independent of speed.

Stiffness Variations in Tyre

- Lateral force variations may arise from nonuniformities in the tire, but be readily related to lateral runout effects in the wheel or hub components. They tend to be independent of speed, thus measurements of the force magnitudes at low speed are also valid for high speed.
- First-order lateral variations in the tires or wheels, or in the way in which they are mounted, will cause wobble. These will affect the dynamic balance of the assembly. The wobble in the wheel may contribute a minor lateral force variation, but may also result in radial and tractive force variations comparable to the effect of ovality because the wheel is elliptical in the vertical plane.
- Higher-order lateral variations are predominantly important in the tire only. Wheel variations are substantially absorbed by the tire. These sources could potentially cause steering vibrations, but have not been identified as the cause of ride problems.

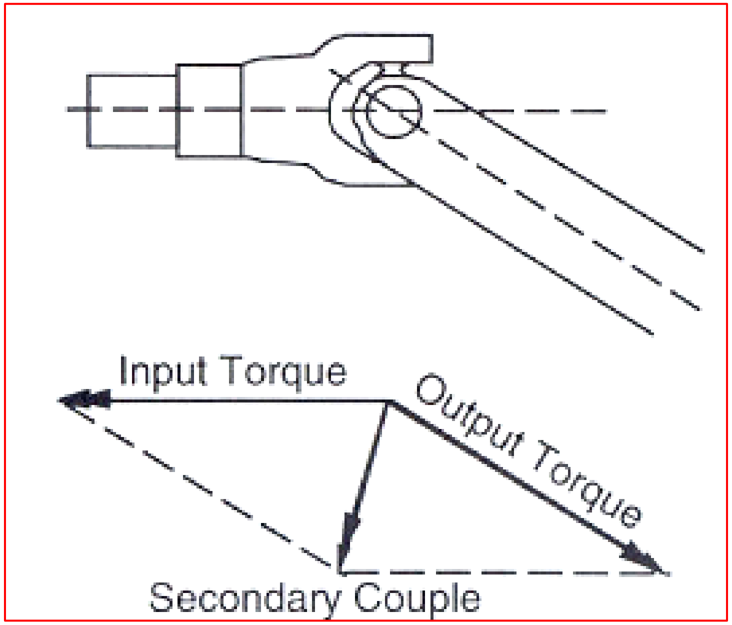
Drive Line Excitations



- Of these various components, the driveshaft with its spline and universal joints has the most potential for exciting ride vibrations.

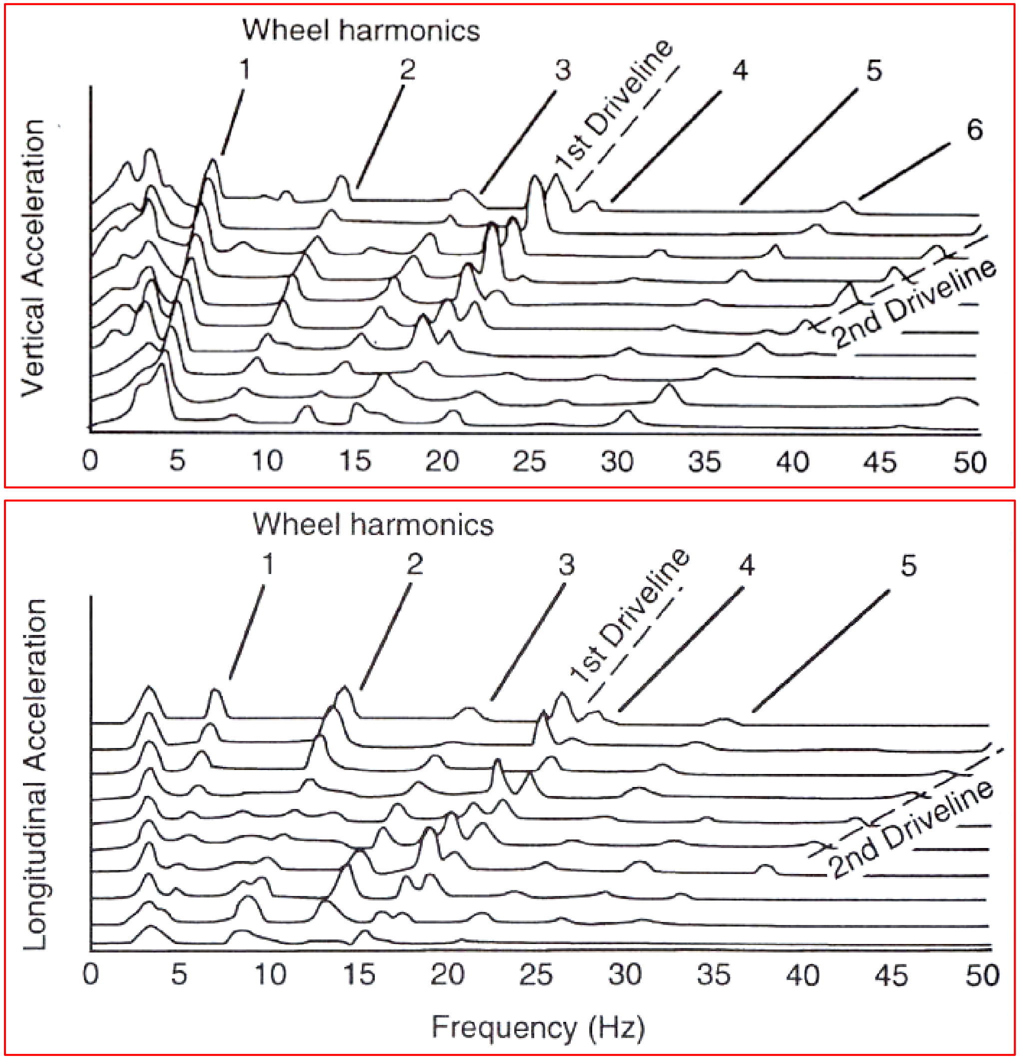
Driveline Excitations

- Excitations to the vehicle arise directly from two sources
  - Mass imbalance of the driveshaft hardware,
  - Secondary couples, or moments, imposed on the driveshaft due to angulation of the cross-type universal joints.
- Mass Imbalance – Imbalance of the drive shaft may result from the combination of any of the five following factors:
  - Asymmetry of the rotating parts
  - The shaft may be off-center on its supporting flange or end yoke
  - The shaft may not be straight
  - Running clearances may allow the shaft to run off center
  - The shaft is an elastic member and may deflect



- The magnitude of the secondary torque is proportional to the torque applied to the driveline and the angle of the universal joint.
- When the torque varies during rotation due to engine torque pulsations and/or nonconstant-velocity joints, the secondary couple will vary accordingly.
- The secondary couple reacts as forces at the support points of the driveline on the transmission, crossmembers supporting the driveline intermediated bearings, and at the rear axle. Hence, these forces vary with driveline rotation and impose excitation forces on the vehicle.
- Drive line basically introduces torsional vibrations

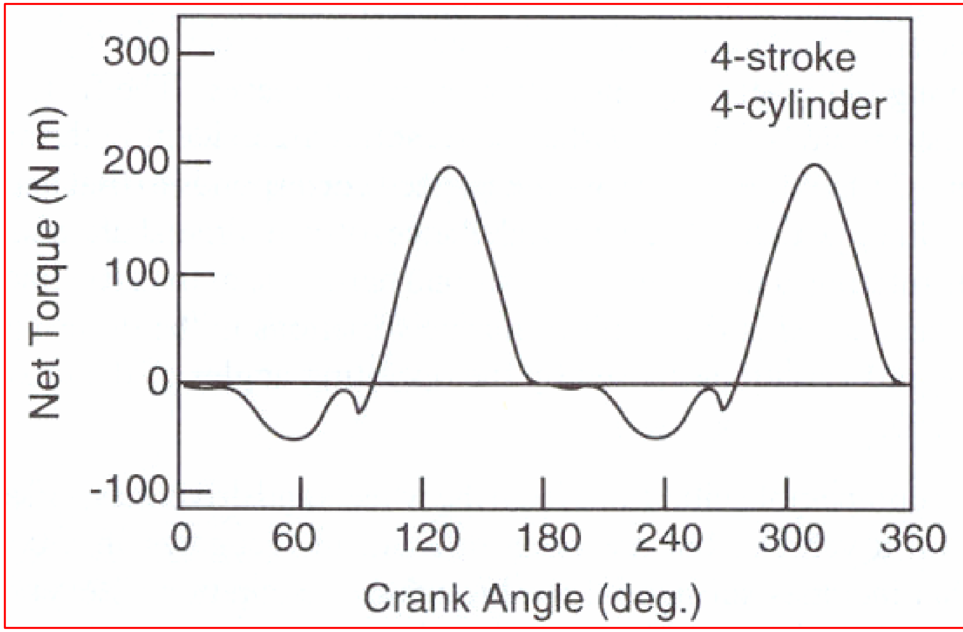
- Nature of the vibrations that may be produced as a result of driveline and tire/wheel nonuniformities.



- Figures show a map of the vibration spectra measured at different vehicle speeds. Excitation from tire/wheel inputs appear as ridges in the spectra moving to higher frequency as the speed increases.
- The first, second and higher harmonics of the tire/wheel assemblies are evident in the spectra.
- The ridge at 3.7 times the wheel rotational speed corresponds to first harmonic of the driveline, which is due to the imbalance of the driveshaft and other components rotating at this speed.
- The second harmonic of the driveline at 7.4 times the wheel speed is the result of torque variations in the driveshaft that arise from speed variations caused by operating angles of the cross-type universal joints.

Engine Excitations

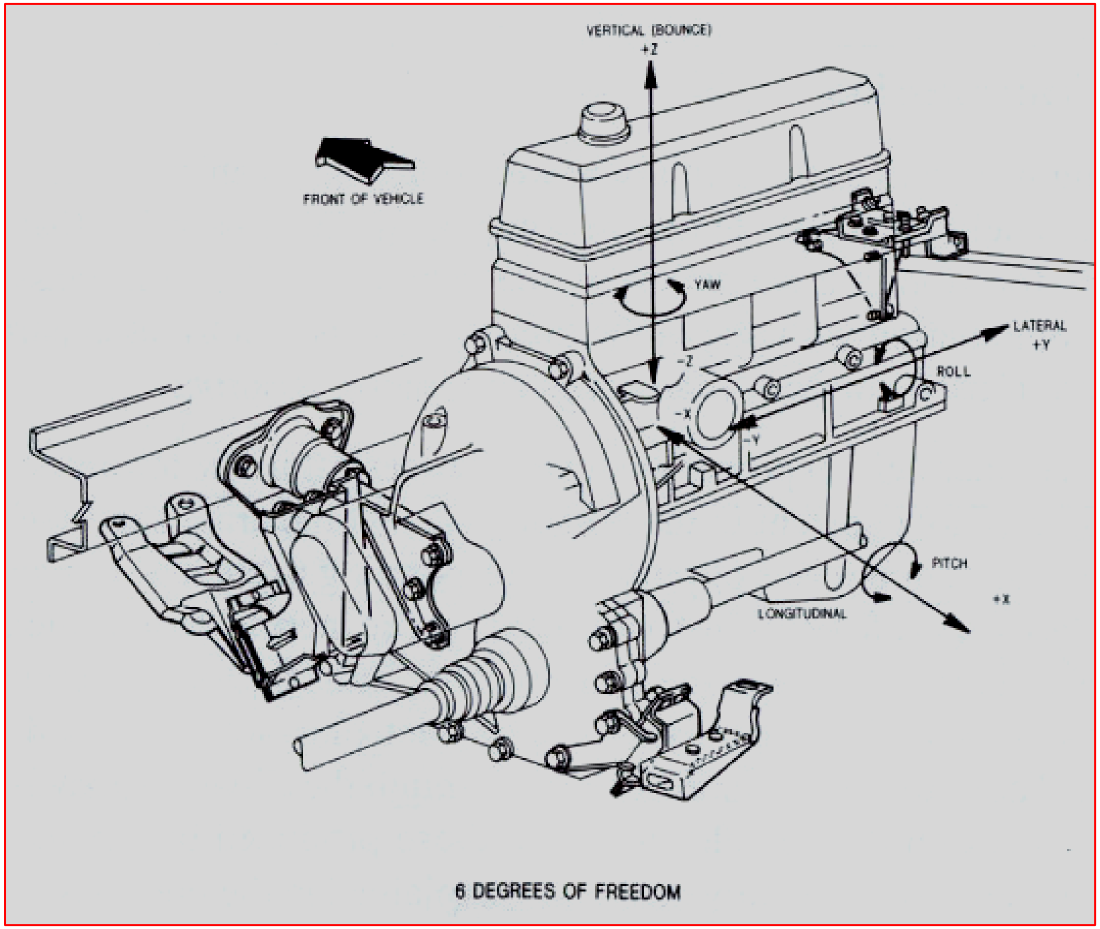
- The engine serves as the primary power source on a vehicle. The fact that it rotates and delivers torque to the driveline opens the possibility that it may be a source for vibration excitation on the vehicle



$$T_c \cong -\frac{W_c}{g}a(l-L)\frac{\omega^2}{2n^2}\sin 2\theta + T_p + T_W$$

Engine Excitations

- The system will vibrate in six directions – the three translational directions and three rotations around the translational axes.
- The figure also shows a three-point mount typical of those used with most transverse engines today.



- Of all the directions of motion, the most important to vibration is the engine roll direction which is excited by drive torque oscillations.
- Torque oscillations occur at the engine firing frequency as well as at sub-harmonics of that frequency due to cylinder-to-cylinder variations in the torque.
- A key to isolating these excitations from the vehicle body is to design a mounting system with a roll axis that aligns with the engine inertial roll axis, and provide a resonance about this axis at a frequency that is below the lowest firing frequency.

- By doing so, torque variations which occur above the resonant frequency are attenuated. In effect, the torque is absorbed in the inertial motion of the engine rather than being transmitted to the vehicle body.
- The engine inertial axis of four-cylinder engines will generally be inclined downward toward the transmission because of the contribution from the mass of the transmission.
- Thus the mounting system must be low at the transmission end and high at the front of the engine. On V-type engines (six- and eight-cylinder) the inertial axis is lower in the front permitting a mounting system more closely aligned with the crankshaft.

- The worst-case problem is isolation of idle speed torque variations for a four-cylinder engine with the transmission in drive, which may have a firing frequency of 20 Hz or below. Therefore, successful isolation requires a roll axis resonance of 10 Hz or below.
- Engines may produce forces and moments in directions other than roll as a result of the inherent imbalances in the reciprocating/rotating masses. These take the form of forces or couples at the engine rotational frequency or its second harmonic, and must be isolated in the same manner as for the roll mode

- For the more commonly used engine configurations the balance conditions are as follows:
  - Four-cylinder inline – Vertical force at twice engine rotational frequency; can be balanced with counter-rotating shafts.
  - Four-cylinder, opposed, flat – Various forces and moments at rotational frequency and twice rotational frequency depending on crank-shaft arrangement.
  - Six-cylinder inline – Inherently balanced in all directions.
  - Six-cylinder inline, two-cycle – Vertical couple generating yaw and pitch moments at the engine rotational frequency; can be balanced.
  - Six-cylinder, 60-degree V – Generates a counter-rotating couple at rotational frequency that can be balanced with counter-rotating shaft.
  - Six-cylinder, 90-degree V (uneven firing) – Generates yaw moment of twice rotational frequency; can be balanced with counter-rotating shaft
  - Six-cylinder, 90-degree V (even firing) - Generates yaw and pitching moments at crankshaft speed, which can be balanced. Also generates complex yaw and pitching moments at twice rotational speed which are difficult to balance.
  - Eight-cylinder inline – Inherently balanced in all directions
  - Eight-cylinder, 90-degree V – Couple at primary rotational speed; can be counter-balanced

- With proper design of the mounting system the mass of the engine-transmission combination can be utilized as a vibration absorber attenuating other vibrations to which the vehicle is prone.
- Most often it is used to control vertical shake vibrations arising from the wheel excitations. For this purpose the mounting system is designed to provide a vertical resonance frequency near that of the front wheel hop frequency (12-15 Hz), so that the engine can act as a vibration damper for this mode of vehicle vibration.

Vehicle Lumped Model

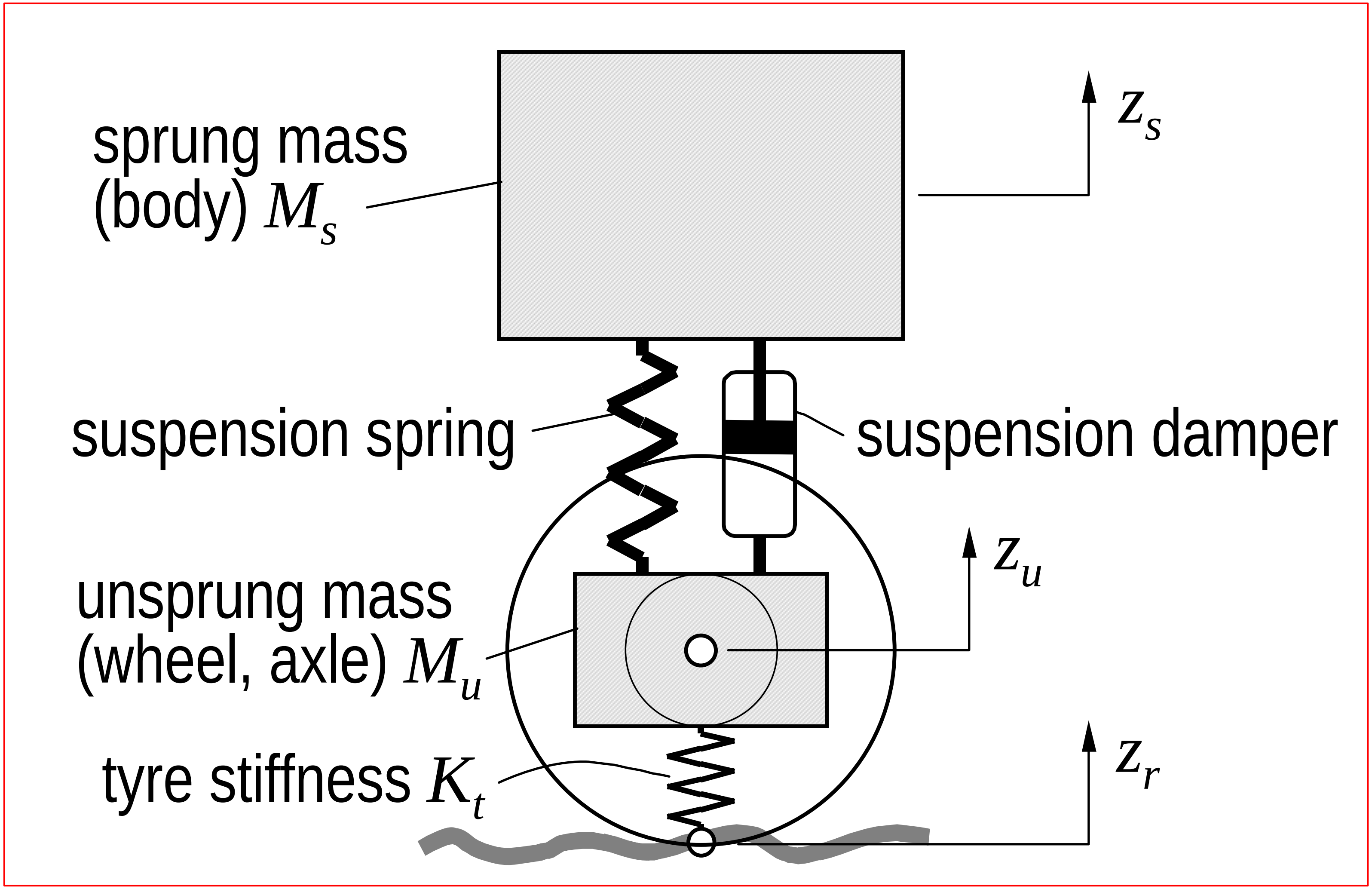


Fig: 9.1

Dynamic Behaviour

Road Roughness

Tire/Wheel

Driveline

Engine

→ Automobile → Vibrations

Displacement
Velocity
Acceleration

Ratio of O/P amplitude to input amplitude is Gain of the Dynamic system

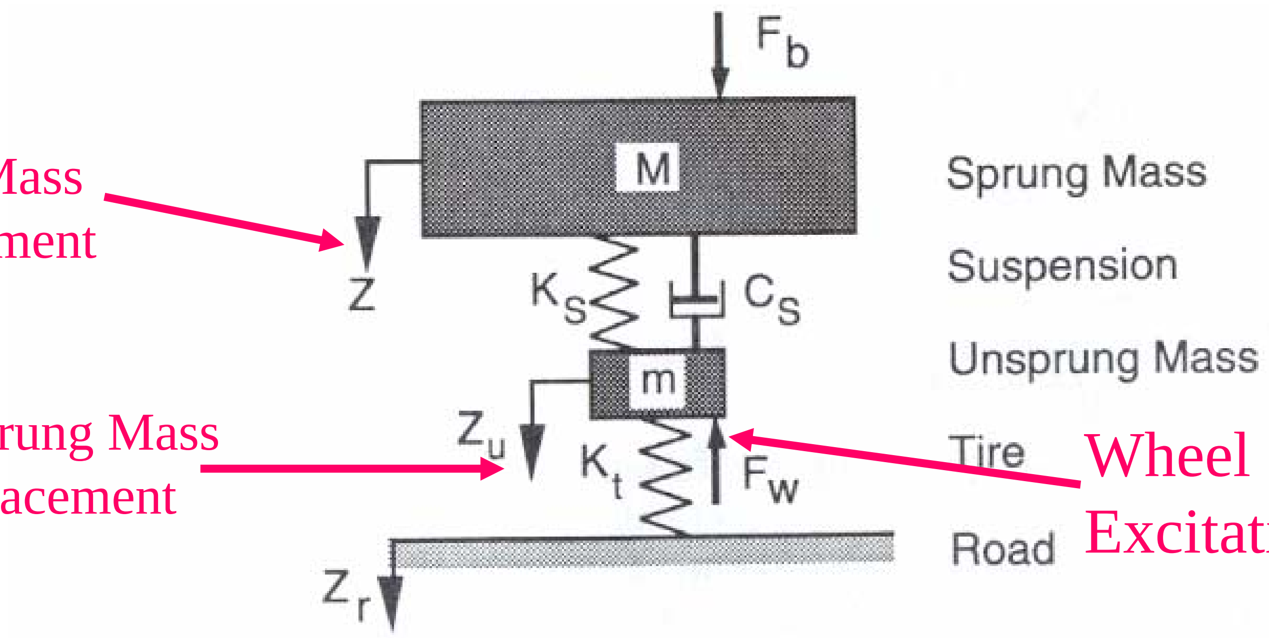
Transmissibility is often used to denote the Gain

Quarter Car Model

Engine & Transmission
Excitation force

Sprung Mass
Displacement

Unsprung Mass
Displacement



Wheel
Excitation Force

Road Displacement

Ride Rate

- The sprung mass resting on the suspension and tire springs is capable of motion in the vertical direction.
- The effective stiffness of the suspension and tire springs in series is called the "ride rate" determined as follows:

$$RR = \frac{K_s K_t}{K_s + K_t}$$

Where:

RR = Ride rate

K_s=Suspension stiffness

K_t=Suspension stiffness

- In the absence of damping, the bounce natural frequency at each corner of the vehicle can be determined from:

$$\omega_n = \sqrt{\frac{RR}{M}} \qquad \text{(radians/sec)}$$

or:

$$f_n = 0.159\sqrt{\frac{RR}{W/g}} \qquad \text{(cycles/sec)}$$

Where: M = Sprung mass

W = M g = Weight of the sprung mass

g = Acceleration of gravity

- When damping is present, as it is in the suspension, the resonance occurs at the "damped natural frequency," ω_d, given by:

$$\omega_d = \omega_n \sqrt{1 - \xi_s^{\,2}}$$

Where:

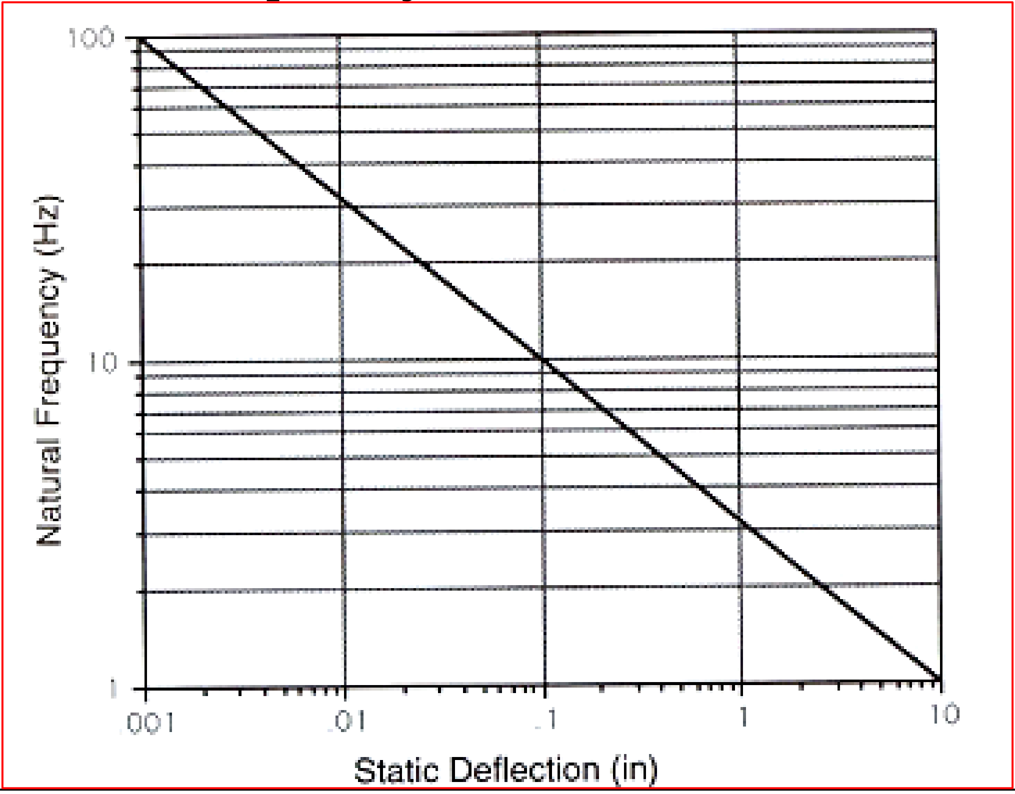
ξ_s = Damping ratio

$$\xi_s = C_s \sqrt{4 K_s M}$$

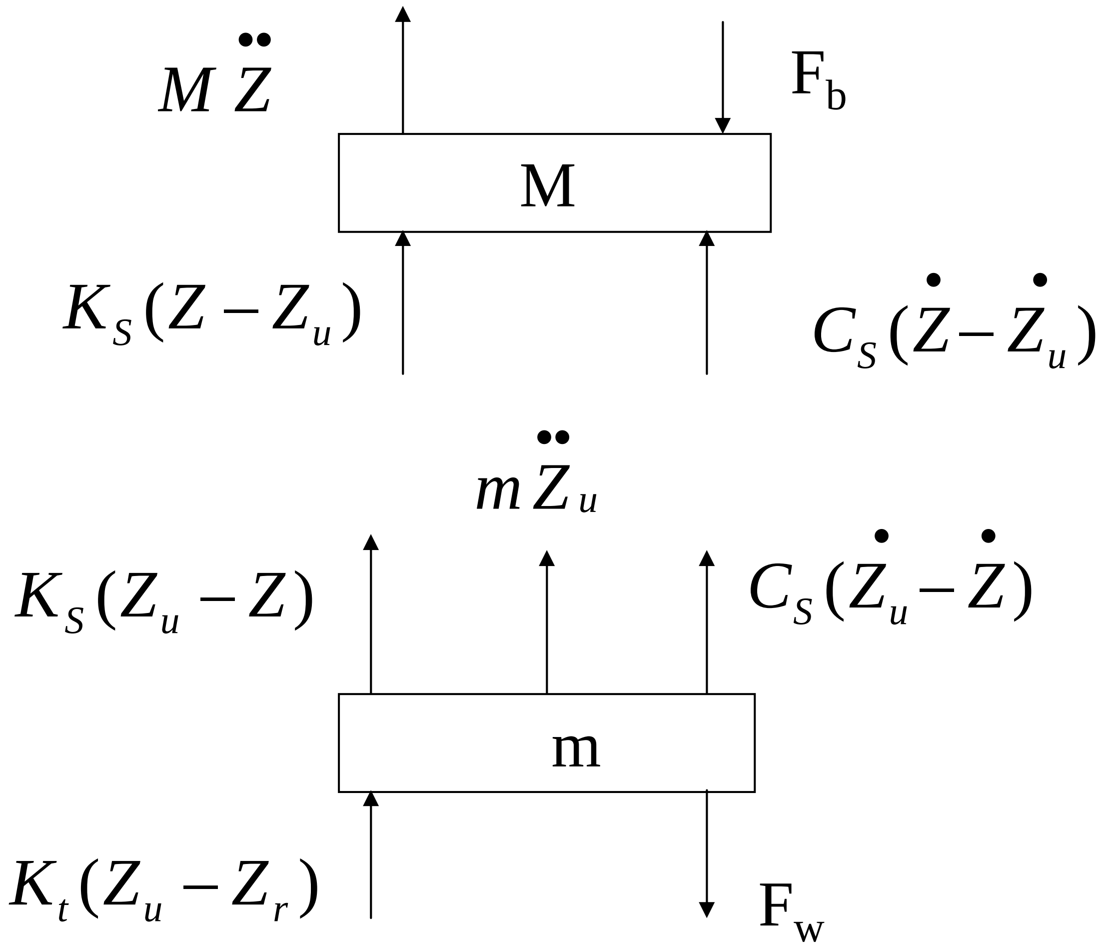
C_s= Suspension damping coefficient

- For good ride the suspension damping ratio on modern passenger cars usually falls between 0.2 and 0.4. Because of the way damping influences the resonant frequency in the equation above (i.e., under the square root sign), it is usually quite close to the natural frequency.
- With a damping ratio of 0.2, the damped natural frequency is 98%. Because there is so little difference the undamped natural frequency, ω_n, is commonly used to characterize the vehicle.

- The ratio of W/K_s represents the static deflection of the suspension due to the weight of the vehicle. Because the "static deflection" predominates in determining the natural frequency, it is a straightforward and simple parameter indicative of the lower bound on the isolation of a system. **Fig:** provides a nomograph relating the natural frequency to static deflection.



- The dynamic behavior for the complete quarter-car model in steady-state vibration can be obtained by writing Newton’s Second Law for the sprung and unsprung masses.



$$\frac{\ddot{Z}}{\ddot{Z}_r} = \frac{K_1K_2 + j[K_1C\omega]}{[x\omega^4 - (K_1 + K_2x + K_2)\omega^2 + K_1K_2] + j[K_1C\omega - (1+x)C\omega^3]}$$

$$\frac{\ddot{Z}}{F_w / M} = \frac{K_2w^2 + j\ [C\omega]}{[x\omega^4 - (K_1 + K_2x + K_2)\omega^2 + K_1K_2] + j[K_1C\omega - (1+x)C\omega^3]}$$

$$\frac{\ddot{Z}}{F_b / M} = \frac{[\mu\omega^4 - (K_1 + K_2)\omega^2] + j[C\omega^3]}{[x\omega^4 - (K_1 + K_2x + K_2)\omega^2 + K_1K_2] + j[K_1C\omega - (1+x)C\omega^3]}$$

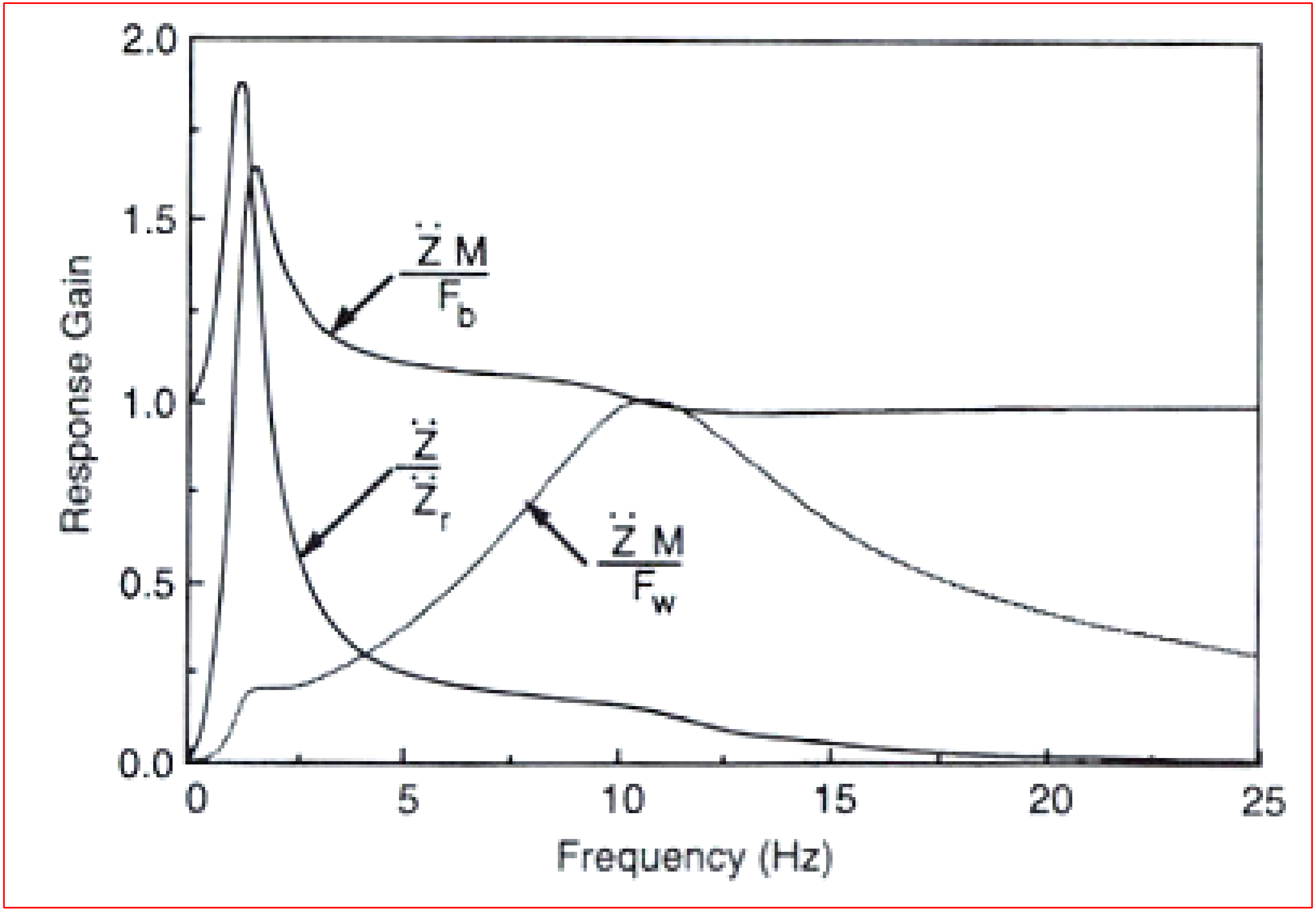
$$x = m / M$$

$$C = C_s / M$$

$$K_1 = K_t / M$$

$$K_2 = K_s / M$$

j = Complex operator


Response Gain
2.0
1.5
1.0
0.5
0.0
0
5
10
15
20
25
Frequency (Hz)
$\frac{\ddot{Z}M}{F_b}$
$\frac{\ddot{Z}}{\ddot{Z}_r}$
$\frac{\ddot{Z}M}{F_w}$

Surface Irregularities

$S_g(f)$

Input

Vehicle System

Abs (H(f))

Output

Vehicle Vibrations

$S_v(f)$= square (Abs(h(f))) $S_g(f)$

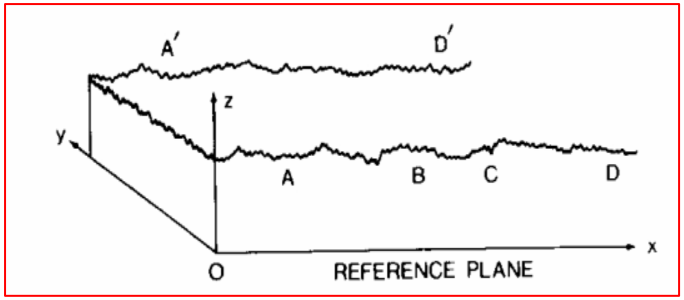
$$|H(f)| = \left| (2\pi f)^2 \sqrt{\frac{1+(2\xi f/f_n)^2}{[1-(f/f_n)^2]^2+[2\xi f/f_n]^2}} \right|$$

$$z_v(t) = |H(f)| z_g(t)$$

$$\bar{z}_v^2 = |H(f)|^2 \bar{z}_g^2$$

$$S_v = |H(f)|^2 S_g$$

$$G_{zs}(f) = |H_v(f)|^2 G_{zs}$$



$$\bar{z}_n^2 = \frac{1}{l_{wn}} \int_0^{lwn} [z_n \sin(\frac{2\pi x}{l_{wn}})]^2 dx = \frac{z_n^2}{2}$$

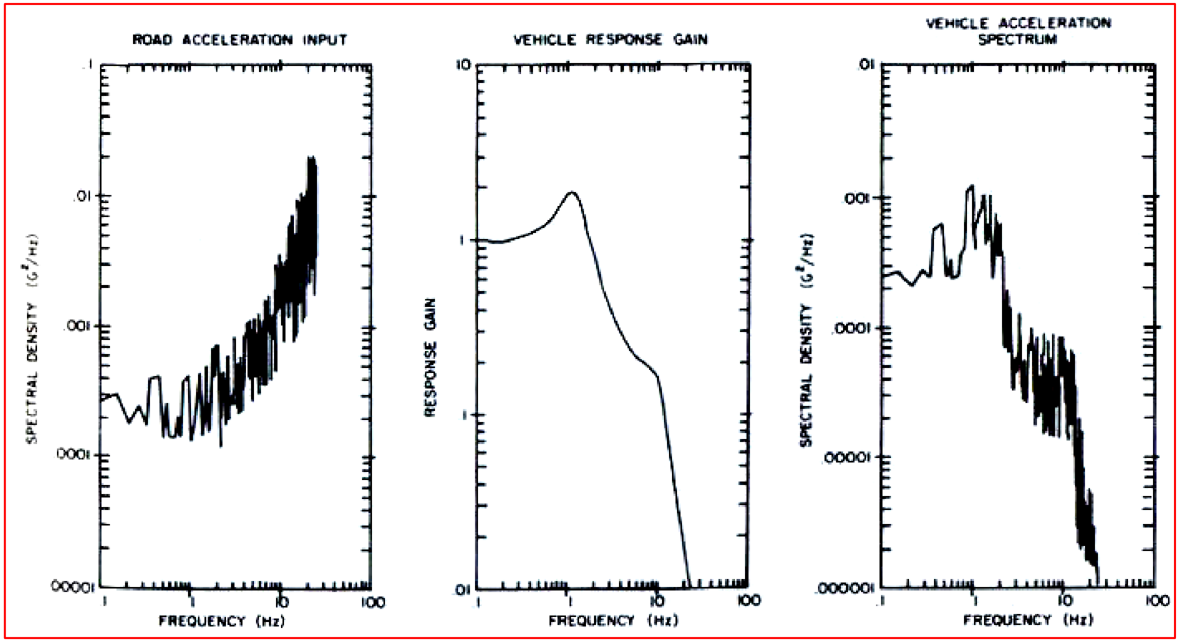
$$S(n\Omega_0) = \frac{z_n^2}{2\Delta\Omega} = \bar{z}_n^2 / \Delta\Omega$$

Response of Sprung Mass to Road Roughness

$$G_{zs}(f) = |H_v(f)|^2 \, G_{zs} \qquad H_v(f) = \text{Response gain for road input}$$

$$G_{zs}(f) = \text{Acceleration PSD on the sprung mass}$$

$$G_{zs} = \text{Acceleration PSD of the road input}$$



- The results obtained are illustrated in **Fig:** While the road represents an input of acceleration amplitude which grows with frequency, the isolation properties of the suspension system compensate by a decrease in the vehicle's response gain.

- The results obtained are illustrated in **Fig: 9.5.** While the road represents an input of acceleration amplitude which grows with frequency, the isolation properties of the suspension system compensate by a decrease in the vehicle's response gain.

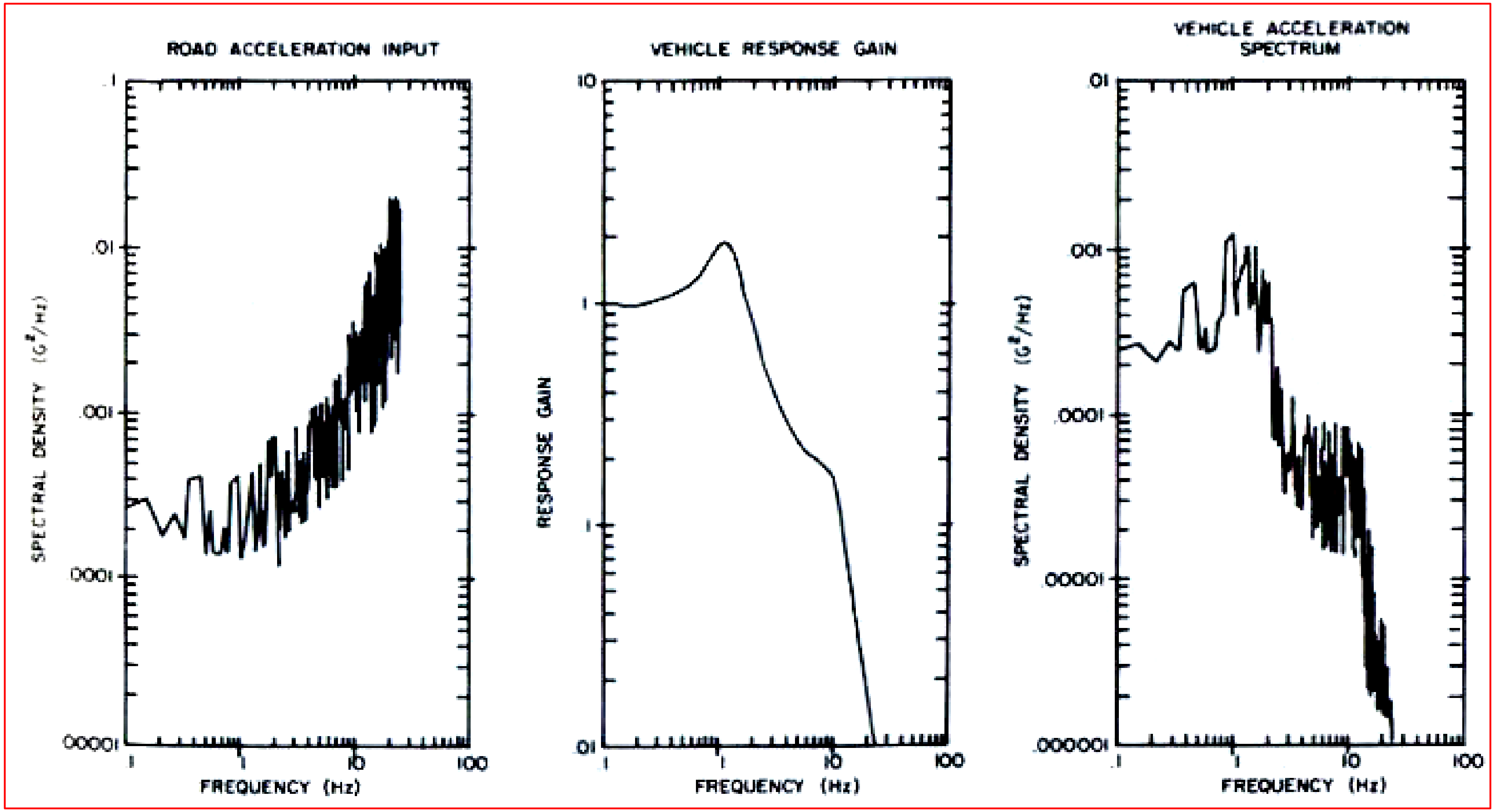


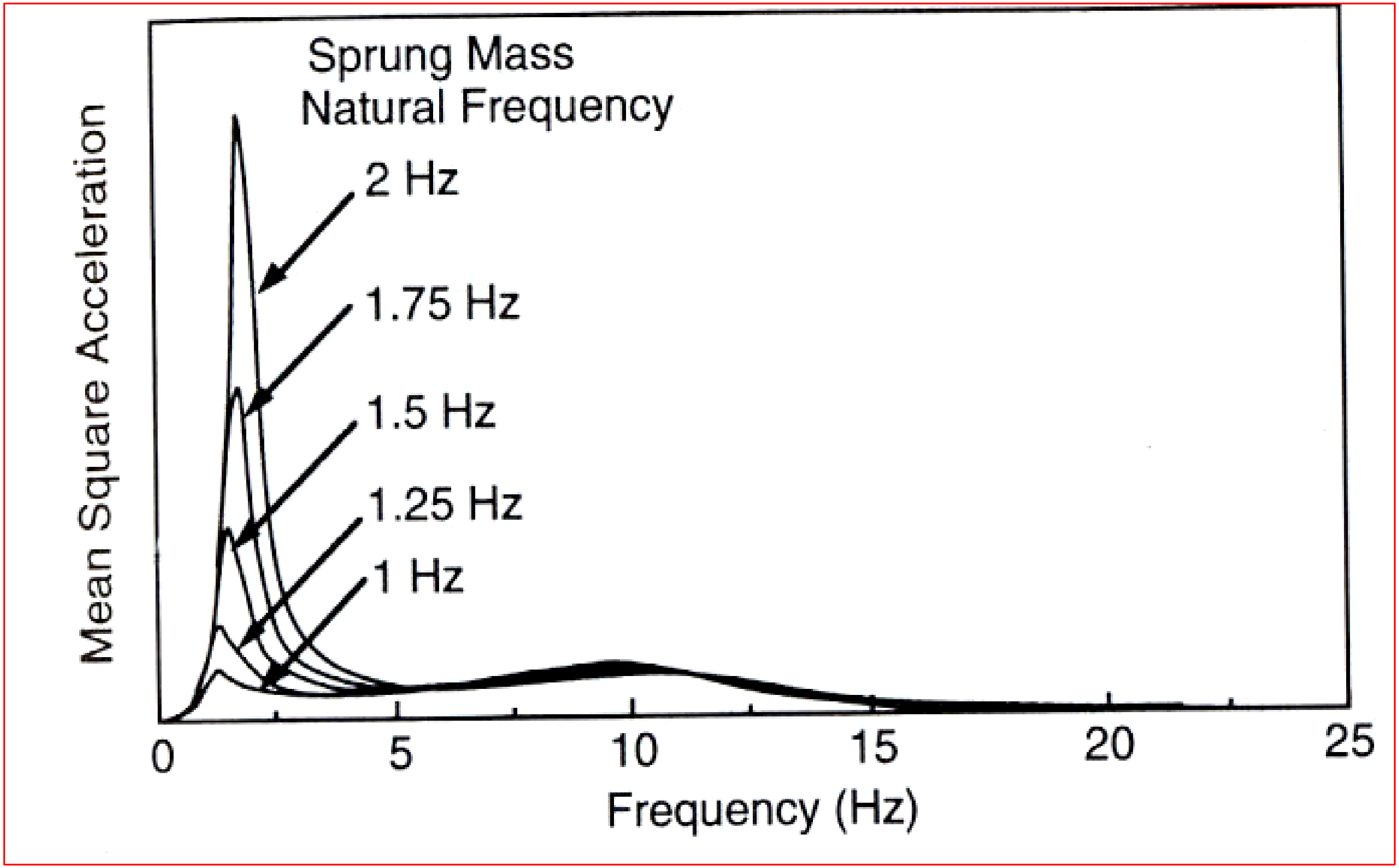
Fig: 9.5

- The net result is an acceleration spectrum on the vehicle with a high amplitude at the sprung-mass resonant frequency, moderate attenuation out through the resonant frequency of the wheel, and a rapid attenuation thereafter.
- Note that, even though the road input amplitude increases with frequency, the acceleration response on the vehicle is qualitatively similar to the vehicle's response gain.
- Thus the acceleration spectrum seen on a vehicle does provide some idea of the response gain of the system even when the exact properties of the road are not known.

Suspension Stiffness

- Because the suspension spring is in series with a relatively stiff tire spring, the suspension spring predominates in establishing the ride rate and, hence, the natural frequency of the system in the bounce (vertical) mode.
- Since road acceleration inputs increase in amplitude at higher frequencies, the best isolation is achieved by keeping the natural frequency as low as possible.
- For a vehicle with a given weight, it is therefore desirable to use the lowest practical suspension spring rate to minimize the natural frequency.

- The effect on accelerations transmitted to the sprung mass can be estimated analytically by approximating the road acceleration input as a function that increases with the square of the frequency.
- Then the mean-square acceleration can be calculated as a function of frequency.
- **Fig** shows the acceleration spectra thus calculated for a quarter-car model in which the suspension spring rate has been varied to achieve natural frequencies in the range of 1 to 2 Hz.
- Because it is plotted on a linear scale, the area under the curves indicates the relative level of mean-square acceleration over the frequency range shown.



Sprung Mass
Natural Frequency
2 Hz
1.75 Hz
1.5 Hz
1.25 Hz
1 Hz
Mean Square Acceleration
0
5
10
15
20
25
Frequency (Hz)

- The lowest acceleration occurs at the natural frequency of 1 Hz. At higher values of natural frequency (stiffer suspension springs), the acceleration peak in the 1 to 5 Hz range increases, reflecting a greater transmission of road acceleration inputs, and the mean square acceleration increases by several hundred percent.
- In addition, the stiffer springs elevate the natural frequency of the wheel hop mode near 10 Hz, allowing more acceleration transmission in the high-frequency range.
- While this analysis clearly shows the benefits of keeping the suspension soft for ride isolation, the practical limits of stroke that can be accommodated within a given vehicle size and suspension envelope constrain the natural frequency for most cars to a minimum in the 1 to 1.5 Hz range.
- Performance cars on which ride is sacrificed for the handling benefits of a stiff suspension will have natural frequencies up to 2 or 2.5 Hz.

Suspension Damping

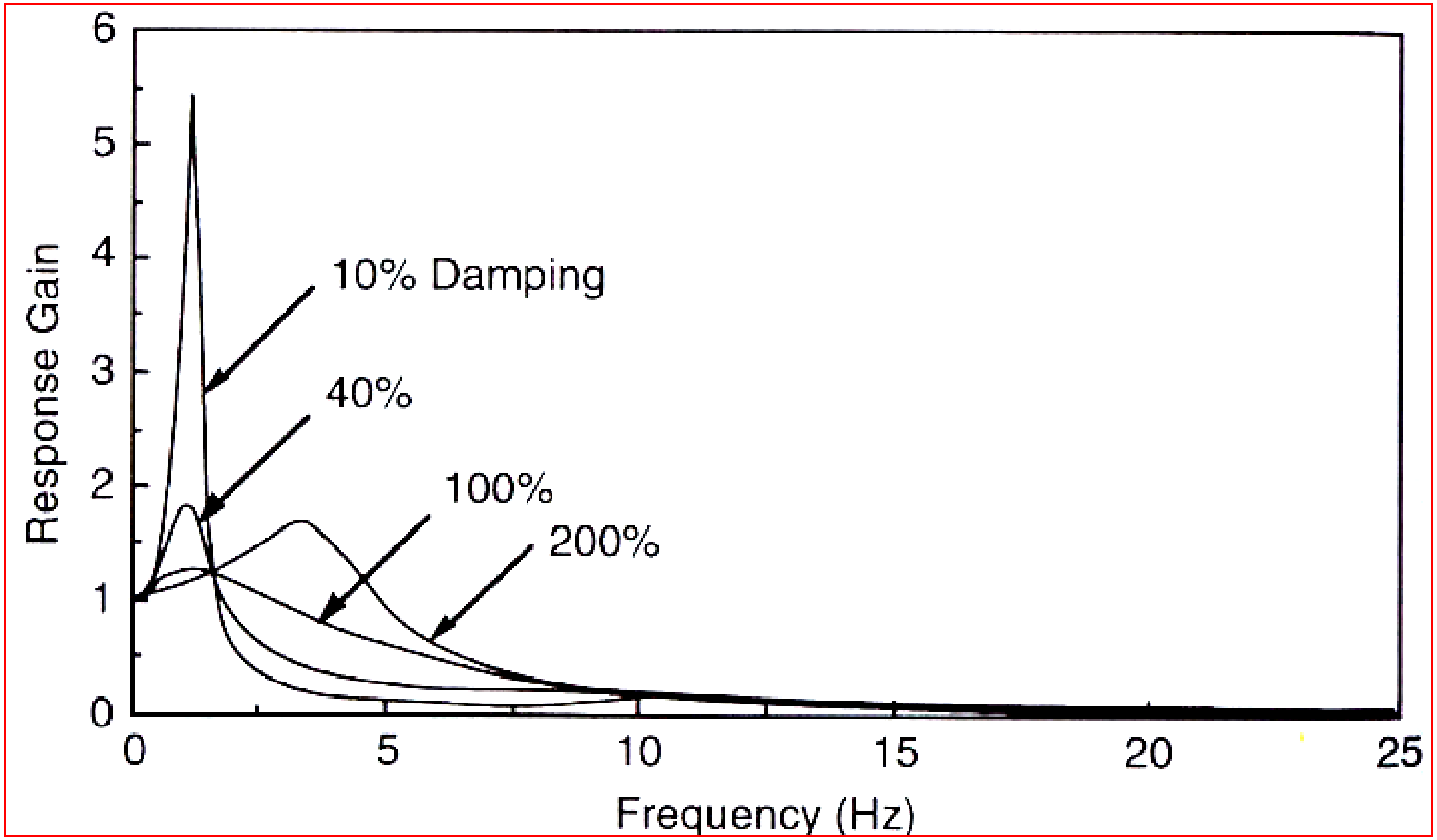
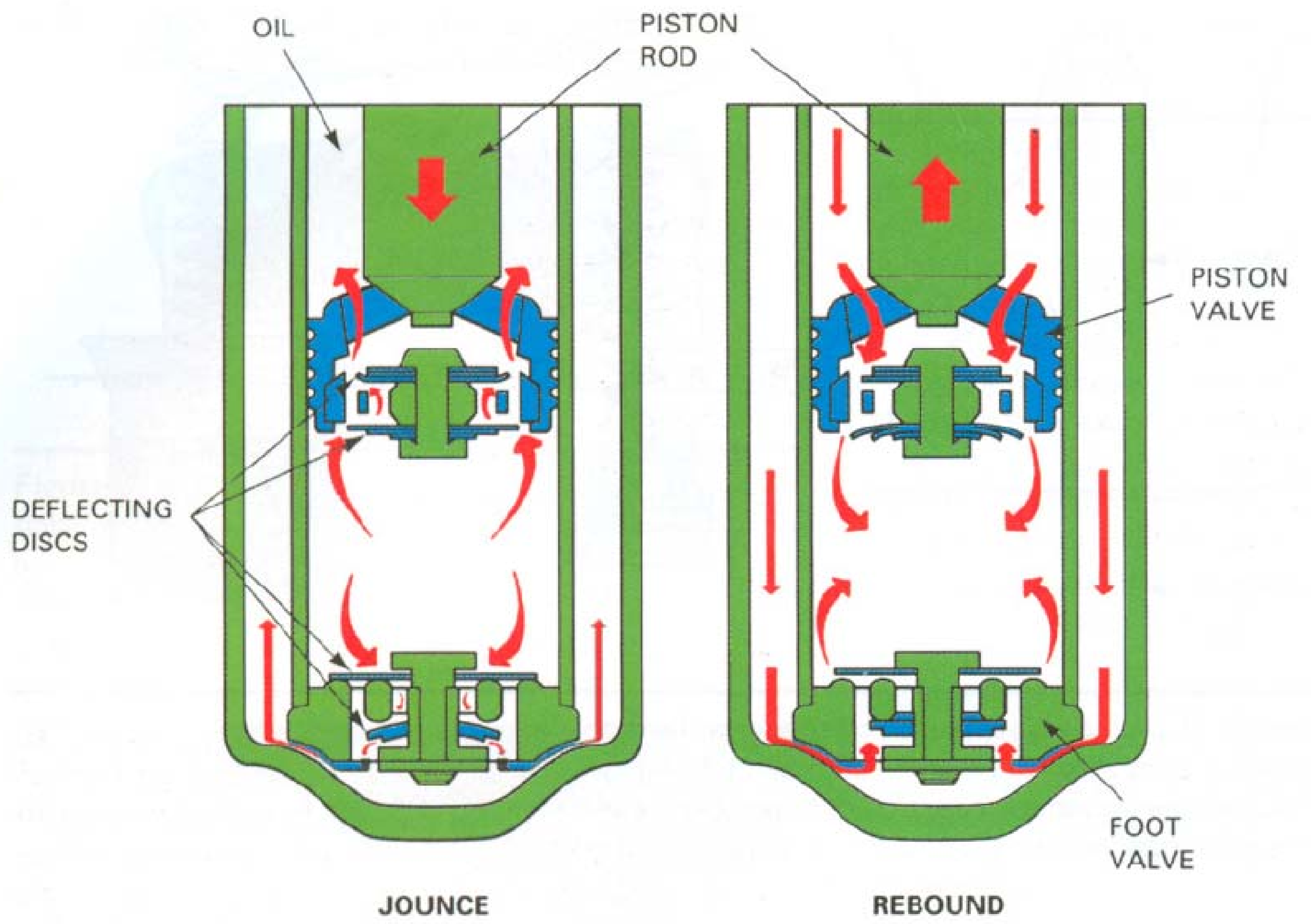


Fig: 9.7

- The 40% damping ratio curve is reasonably representative of most cars, recognizable by the fact that the amplification at the resonant frequency is in the range of 1.5 to 2.0.
- At 100% (critical) damping, the 1 Hz bounce motions of the sprung mass are well contributed, but with penalties in the isolation at higher frequencies.
- If damping is pushed beyond the critical, for example to 200%, the damper becomes so stiff that the suspension no longer moves and the vehicle bounces on its tires, resonating in the 3 to 4 Hz range.

- Shock absorbers must be tailored not only to achieve the desired ride characteristics, but also play a key role in keeping good tire-to-road contact essential for handling and safety
- First, the damping in suspension jounce (compression) and rebound (extension) directions is not equal. Damping in the jounce direction adds to the force transmitted to the sprung mass when a wheel encounters a bump, thus it is undesirable to have high damping in this direction.
- On the other hand, damping in the rebound direction is desirable to dissipate the energy stored in the spring from the encounter with the bump.
- Consequently, typical shock absorbers are dual-rate with approximately a three-to-one ratio between rebound and jounce damping


OIL
PISTON ROD
PISTON VALVE
DEFLECTING DISCS
FOOT VALVE
JOUNCE
REBOUND

Wheel Hop Resonance

- Wheel hop resonance frequency 10-12 Hz

$$f_a = 0.159\sqrt{(K_t + K_s)g / W_a}$$

Where:

f_a = Wheel hop resonant frequency (Hz)

K_t = Tire sping rate

K_s = Suspension spring rate

W_a = axle weight

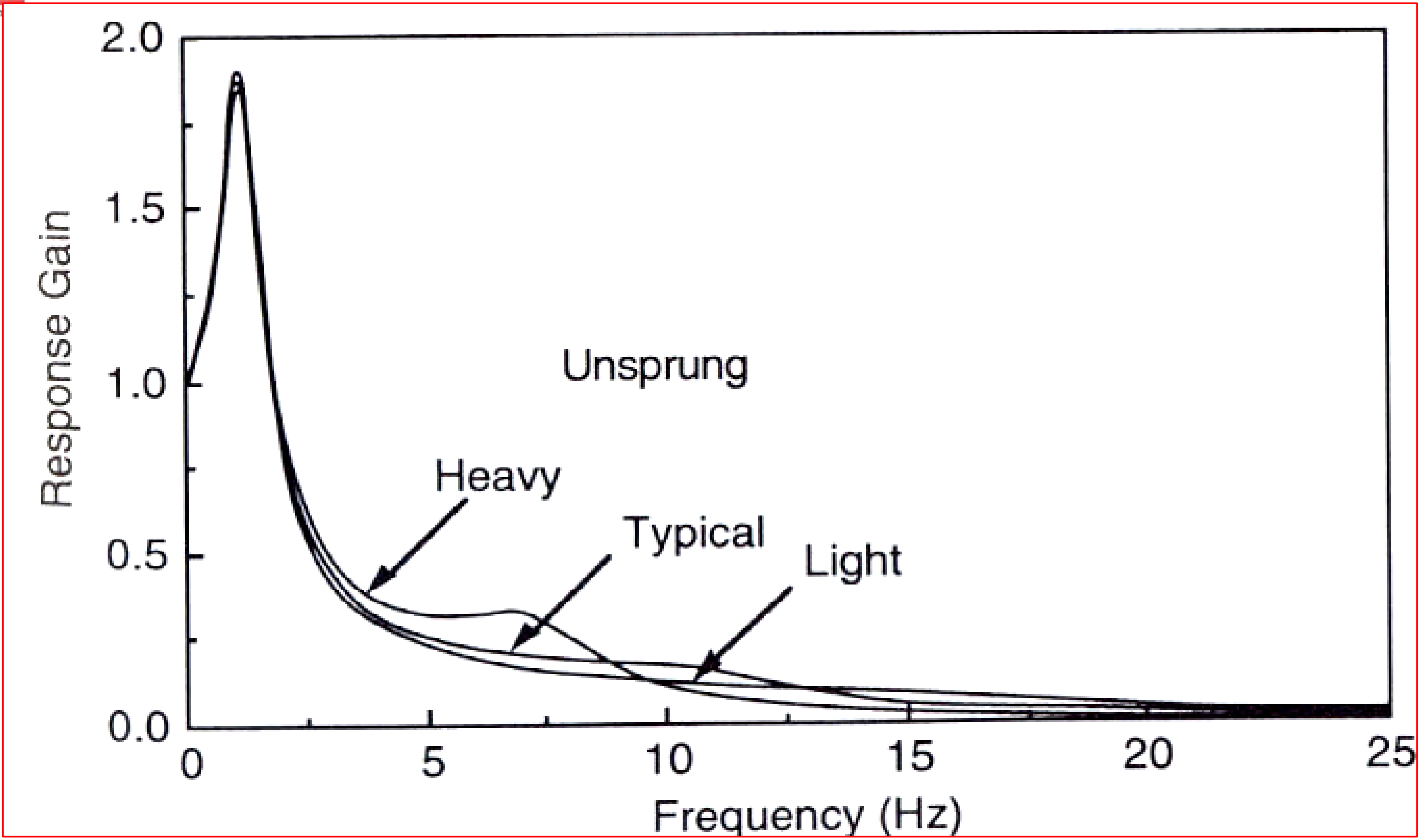


Fig: 10.8 Effect of unsprung mass on suspension isolation behavior

- Characteristically, the unsprung mass will correspond to a weight that is proportional to the gross axle weight rating (GAWR), which in turn is indicative of the loads normally carried by the axle.
- For nondriven axles, the weight, W_a, is typically about 10 percent of the GAWR, whereas for drive axles it will be about 15 percent of the GAWR.
- In as much as the tires and suspension springs are normally sized in proportion to the GAWR, and the resonant frequency depends on the ratio of the mass to the total spring rate of the tires and suspension springs, the resonant frequencies of most wheels, at least theoretically, would fall in a limited range.

Human Perception and Tolerance of Vibrations