Role of Road Roughness in Vehicle Ride

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This paper describes the gross mechanics of motor vehicle ride vibrations to acquaint the highway engineer with the role played by road roughness. The acceleration spectrum observed on a typical passenger car is compared against available measures of human vibration tolerance to illustrate the frequency range of general interest. The characteristics of road roughness are presented in terms of both the elevation spectral density and the equivalent acceleration excitation to the vehicle. The mechanisms of ride isolation achieved through the suspension systems of motor vehicles are illustrated by attenuation of the response gain at high frequency. Examples are given to show effects of mass, suspension, and tire properties. Additional attenuation effects derive from wheelbase filtering and tire envelopment. In the case of commercial trucks, the compromises that occur in the ride isolation as a result of suspension friction and low frequency structural resonances are illustrated.

To the highway engineer, roughness is an important measure of roadway performance because of its known influence on user acceptability (1) and its potential link with highway safety. Anyone who rides in a highway vehicle is aware of road roughness as the major source of the ride vibrations experienced. Concurrently, the roughness may degrade the handling qualities of the vehicle (2,3) and contribute to wear and tear on the vehicle.

This paper examines the relation between road roughness and vehicle ride to illustrate the mechanisms involved and to reveal those aspects of road roughness that play the major role in determining the public judgment of road quality. In order to appreciate the significance of the ride motions observed, it is helpful to first begin with a brief introduction to the subject of human sensitivity to vibration as it relates to the perception of vehicle ride.

HUMAN PERCEPTION OF VIBRATION

Vehicle motion due to pavement roughness fits the general category of broad-band random vibrations. That is, the vibrations are not characterized by discrete frequencies but instead extend over a broad band of frequencies. Because they are random, the vibrations cannot be described exactly as a function of time but instead must be described statistically in terms of average properties.

The most basic statistical measures of random motion are the mean (average) value and the mean-square value. In the context of vehicle ride motions, mean values are zero and thus not useful numerics. However, the mean-square values, which are also the variances, are reasonable measures. In the automotive and trucking industry, the basic measure of ride has traditionally been the root-mean-square (RMS) accelerations measured at various points in the passenger compartment.

In other fields of transportation, particularly those of military aircraft and ground vehicles, the vibrations of the passenger compartment can be more severe than in ordinary passenger cars--to the point that the occupants can be impaired in their normal tasks and even risk injury in some cases. In early research to evaluate human tolerance to vibration in these environments, tests would frequently involve subjects seated or standing while exposed to sine-wave vibrations from laboratory shaker facilities. Such tests have shown that people are sensitive not only to the amplitude of the excitation but also to the frequency and the direction. The maximum sensitivity is usually found to be between 1 and 10 Hz, in which range various internal organs vibrate in resonance with the excitation. Because these tests involve sine waves and vehicle vibrations are broad band, the research has logically expanded to examine the influence of multiple components of sine waves (4) and the actual broad-band environments of transport vehicles (5,6).

In order to quantify the frequency content of the acceleration, the statistical power spectral density (PSD) function is employed. PSD is the partial derivative of the mean-square value of acceleration with respect to frequency and has units g^2/Hz. (The name power spectral density is based on its earliest applications in the electronics and communications field, when it was used to characterize voltage signals and had the units volts^2/Hz. Although volts^2 is proportional to power, a PSD is, in general, not a measure of power; clearly, an acceleration PSD, which has units of g^2/Hz, is not physically a measure of power.) Figure 1 shows the PSD of a measured vertical acceleration in a passenger car, plotted on linear axes, in which case the area under the entire curve is equal to the mean-square value. The area encompassed by any frequency band is the portion of the overall mean-square acceleration due only to that band.

A variety of ride quality standards have been proposed that, in one way or another, weight the acceleration PSD based on test results from the sine wave laboratory tests (7). But these various ride quality standards are largely incompatible with each other, probably because (a) they reflect different ideas about the subjective judgment of a good ride and (b) they are based on tests that are not always well documented and were designed and conducted for different applications. In order to provide a common measure for human vibration levels, the International Standards Organization (ISO) developed vibration standards that define an interaction among frequency, amplitude, direction (vertical or lateral and fore-aft vibrations), time of exposure, and human response level (i.e., noticeable reduced comfort, impaired efficiency, dangerous) (8). The standard provides boundaries of amplitude versus frequency that apply for a given direction, time of exposure, and response level. Figure 2 shows some of the boundaries. To apply the standard, the RMS value of the measured acceleration must be determined in third octave bands (either by using band pass filters or by taking the areas under the PSD in third octave frequency bands), and the resulting values are compared with the boundary in each band. The vertical bars in Figure 2 are taken from the PSD

Figure 1. Typical vertical acceleration PSD of a point on floor of passenger car.

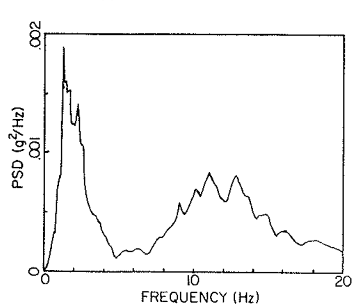


Figure 2. Example use of ISO vibration standard for PSD shown in Figure 1.

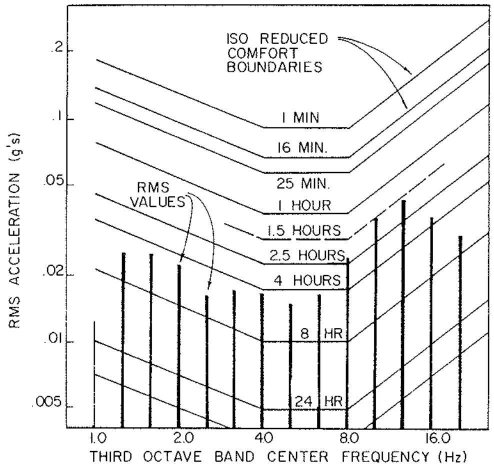


Figure 3. Typical road elevation PSD.

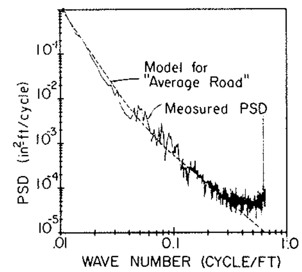
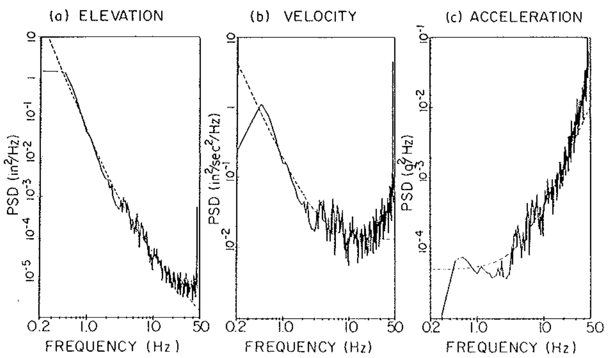


Figure 4. Elevation, velocity, and acceleration PSDs of road roughness input to vehicle traveling at 50 mph on real and model roads from Figure 3.



shown in Figure 1 and, when compared with the reduced comfort boundaries, show that the ISO reduced comfort criterion is satisfied for exposure times less than 1.5 h. But for longer exposure times, the boundary is violated for the third octave band centered at 10 Hz. The ISO standard can also be applied by multiplying the acceleration PSD by a weighting function and then using the resulting weighted RMS acceleration level.

The ISO criteria, along with a few other ride quality weighting functions [notably the absorbed power criterion (9) and the urban tracked air-cushion vehicle (UTACV) specification (10)] have been used at times by the research and automotive communities to objectively compare rides of different vehicles under different operating conditions. But ultimately, none of these criteria have been validated for highway applications by comparisons with subjective ratings of ride, and recent efforts to do so for passenger cars (11) have found that the simple total RMS acceleration in the passenger compartment correlates as well with the subjective ratings as does any one of the ride quality numerics.

Whether ride is quantified by RMS acceleration or by a more complex, frequency-weighted numeric, the PSD of the acceleration serves to characterize the vibrations of a vehicle as induced by road roughness.

NATURE OF ROAD ROUGHNESS

Road roughness encompasses everything from the tooth-jarring potholes that result from localized pavement failures to the ever-present random deviations that reflect the practical limits of precision to which a road surface can be constructed and maintained. Although potholes are the more spectacular examples in the public eye, random roughness is the more pervasive problem for the highway engineer, whose objective is to maintain the best possible highway network with the limited funds available.

Road roughness may be described by the elevation profile obtained along the wheel tracks over which vehicles pass. Such profiles fit the general category of broad-band random signals, and hence are well suited for the statistical description offered by the PSD function. Figure 3 shows such a PSD, with units elevation²/wave number (wave number is 1/wavelength), for a section of road 1 mile long.

The PSD of any particular road section is unique. However, when a number of PSDs of different sections are compared, a general similarity in their shape is evident. This leads to the notion of an average road, defined by a model PSD. A number of PSDs measured for European roads (12) led to the formulation of the average road model (13) shown in Figure 3, along with the measured PSD. Various types of road models have been in use for years (14) as a means to represent roads, in general, for analyzing vehicle ride behavior. In contrast to actual road profiles, such models eliminate effects of features that might be present in a single road but are not generally representative.

When traversed by a vehicle, the profile is perceived as an elevation that changes with time, where time and longitudinal distance are related by the speed of the vehicle. The time-varying elevation can also be characterized by a PSD that has units elevation²/frequency, where frequency and wave number are also related by speed. Figure 4a shows the PSD for the same section of road as used in Figure 3, when traversed at 50 mph.

As the elevation is perceived to be changing with time, it also has a velocity (proportional to slope) and acceleration (proportional to the derivative of slope), which also have PSDs as shown in Figures 4b and 4c for the same road section. Since velocity is the derivative of position, the velocity PSD is related to the elevation PSD by the scale factor $(2\pi f)^2$, where f is frequency in hertz. And, likewise, the acceleration PSD is related to the velocity PSD by the same scale factor. Although the elevation PSD is seen to be largest at low frequencies (long wavelengths) and to fall off drastically at higher frequencies, the scaling between elevation and velocity PSDs means that the velocity PSD is more uniform across the frequency range shown, and the acceleration PSD increases tremendously with frequency. The concept of the road as an acceleration input to the vehicle is important to understand because its ultimate effect--vehicle ride vibrations--are invariably quantified as accelerations.

The speed at which a vehicle travels affects the pavement acceleration PSD in two ways. First,

Figure 5. Effect of speed on perceived road input and vertical acceleration of car floor output.

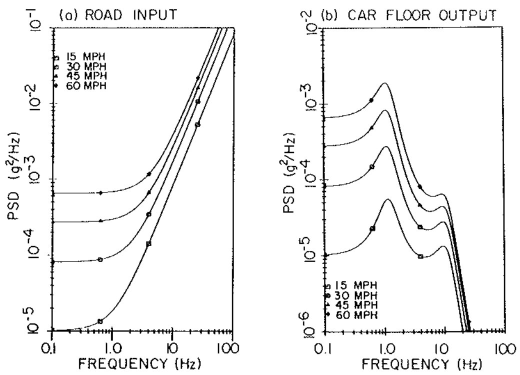


Figure 6. Simple suspended mass system.

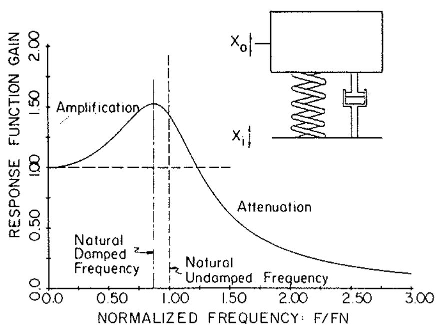
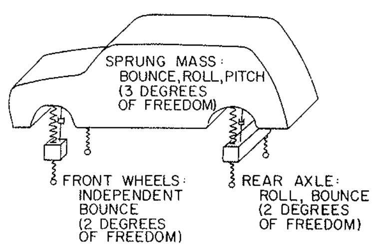


Figure 7. Ride model of passenger car with 7 df.

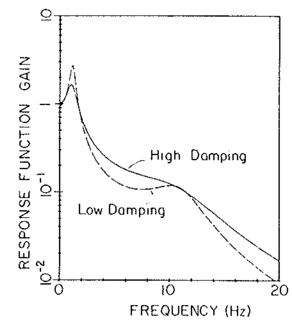


physical features occur at different wavelengths. A 44-ft wavelength is seen as a 2-Hz frequency at 60 mph and as a 1-Hz frequency at 30 mph. Second, the elevation changes faster at higher speeds. A slope of 1:88 is seen as a vertical velocity of 1 ft/s at 60 mph, but at 30 mph it is seen as a vertical velocity of 0.5 ft/s. Figure 5a illustrates the effect of speed on the acceleration PSD of the average road model, and Figure 5b shows the resulting effect on the vertical acceleration PSD in the passenger compartment. The RMS acceleration increases with speed, and more of the overall acceleration derives from the lower frequencies at higher speeds.

VEHICLE RESPONSE TO ROAD ROUGHNESS

The motor vehicle can be thought of as a multi-degree-of-freedom dynamic system, one function of which is to isolate the rider from the roughness imperfections of the roadway. The isolation is obtained by supporting the passenger compartment on a soft suspension system. Conceptually, at the most

Figure 8. Automobile response function between road and floor movement.



basic level, isolation of the vehicle ride capitalizes on the transmissibility benefits of the simple suspended mass system shown in Figure 6. For such a system, the ratio of the amplitude of the motion of the mass (x_o) to the motion of the input (x_i) varies with frequency. At very low frequency the mass will move with the input and provide no isolation. At a certain frequency (termed the natural frequency) the mass resonates and, in fact, amplifies the input motion. Then, at higher frequencies, the mass does not respond to the input and attenuation (or isolation) is obtained. When ride is important to the automotive designer, the objective is to achieve a low natural frequency so that most of the objectionable input occurs on the attenuating portion of the transmissibility curve.

The main application of a frequency response plot to vehicle ride is the way it relates the input PSD of the road to the output PSD of the body acceleration. The relation is a simple one:

$$PSD_{out} = PSD_{in} \times Gain^2 \quad (1)$$

Thus, the frequency response function acts to weight the input PSD.

Passenger Cars

The basic ride model of a passenger car (shown in Figure 7) has suspension points at each of the four wheels, and the wheels or axles also have mass. For simultaneous vertical inputs at all wheels, this model has the vertical transmissibility characteristics to the sprung mass center of gravity shown in Figure 8. The vehicle designer attempts to keep the body bounce resonant frequency as low as possible; however, 1 Hz is about the lowest that can be achieved because of constraints on the softness of the suspension imposed by the available range of suspension travel. Sport cars usually have higher natural frequencies and more damping due to considerations about their performance in handling. Figure 8 shows that reduction in the suspension damping improves isolation near 5 Hz and at higher frequencies but at the expense of a greater amplitude at body resonance.

Although Figure 8 would suggest that good ride isolation is obtained near the axle resonance at 10 Hz, the much greater road acceleration input at this frequency (see Figure 4c) can cause the transmissibility characteristics in this range to be critical. As seen in Figure 2, the vibrations at the axle resonance may often be as objectionable (or more so) than the vibrations at the body resonance. The vehicle designer attempts to minimize the transmissibility at axle resonance by holding down the mass of the unsprung components (e.g., driveline and brakes) and by using tires that have a low radial

Figure 9. Effect of tire radial stiffness on automobile response function.

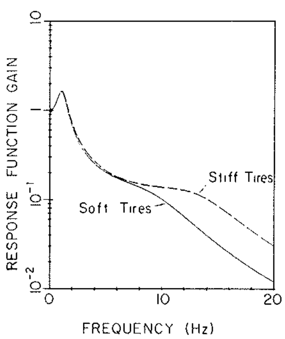
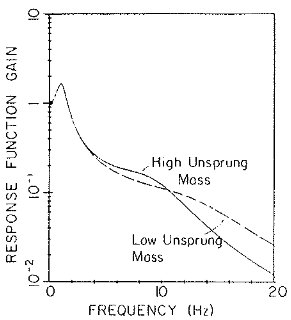


Figure 10. Effect of unsprung mass (axle weight) on automobile response function.



stiffness. Figures 9 and 10 illustrate the effects of these parameters.

Above axle resonance, the transmissibility attenuates even further because the axle can no longer follow the road inputs; roughness features are simply absorbed by deflection of the tire.

In actuality, the vehicle, as it travels down a road, does not experience equivalent inputs at all wheels simultaneously, as shown above. Instead, the rear wheels of the vehicle are excited by the same inputs as that of the front wheels, but delayed in time by an amount determined by the wheelbase and speed.

This type of excitation introduces a filtering effect to the frequency response function of the vehicle. The phenomenon can be easily visualized for a car that has uncoupled bounce and pitch modes. Vehicle bouncing is clearly excited when two axles move up and down together, and vehicle pitching is excited when the axles move completely out of phase. Figure 11 shows that a wavelength that is equal to the wheelbase can only excite the bounce motions, but a wavelength that is twice the wheelbase can only excite pitch motions. The effective wave number response function for bounce, which is the ratio of the average of the pavement elevation at the two axles to the elevation at the front axle, is shown in Figure 12.

When the frequency response function of a vehicle is found for a profile input, the wheelbase filtering effect is included, as shown in Figure 13. Although the vehicle response (discussed earlier) derives from dynamic (i.e., time-based) resonances and is purely a function of temporal frequency (Hz), the wheelbase filtering is purely a geometric property--characterized as a function of spatial frequency (wave number). Hence, the overall vehicle response function, as shown in the figure, would change with speed.

Figure 11. Relation of wheelbase and wavelength to bounce and pitch.

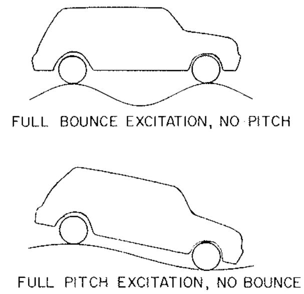


Figure 12. Effective wheelbase filtering for bounce excitation.

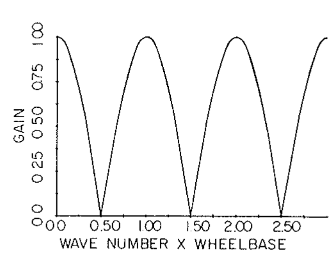


Figure 13. Automobile response function with wheelbase filtering.

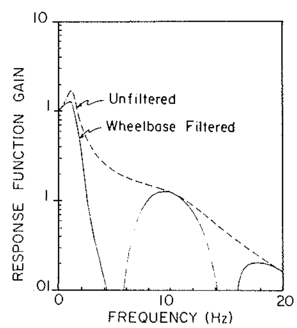
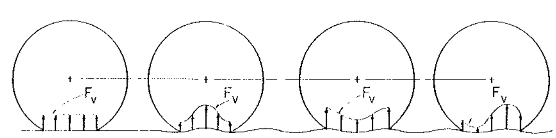


Figure 14. Illustration of tire envelopment phenomena at certain wavelengths.



A second source of geometric filtering is introduced into the vehicle response because the tires contact the pavement over an area rather than at a single point. At certain wavelengths, the increased vertical forces produced at one section of the contact patch between tire and road are completely cancelled by decreased forces from another section, with the net result that no variation in force occurs as the tire rolls over the profile. Figure 14 illustrates this phenomenon. For other wavelengths, there is imperfect cancellation and the tire experiences some force change, but not as much as would be expected from the linear spring representation shown in Figure 7. This behavior, called tire enveloping, results in the filtering effect shown in Figure 15 for a typical passenger-car tire (13). Again, the tire-enveloping effect will depend on the vehicle speed. Since the effect is only significant for short wavelengths, its effect on ride is only important at lower speeds.

In addition to these mechanisms that determine vertical response, similar analyses are necessary to determine motion in the pitch and roll degrees of freedom. Response characteristics similar to those shown in Figure 8 would be observed, although the resonances may be at different frequencies.

The potential for ride motions in all three of these directions exists with modern passenger cars. Although large luxury cars are easily designed to minimize pitch motions, this problem becomes more acute with smaller cars, which have shorter wheelbases. Likewise, the excitation of roll (or lateral) motions depends on the vehicle and type of road. Most vehicle ride studies represent the proprietary work of the vehicle manufacturers. In the limited published studies (11) that relate subjective ride ratings to vehicle accelerations, both vertical and lateral accelerations have been found to correlate with the ratings. Since the vertical and lateral accelerations are also highly correlated, it is not possible to identify which is the most significant, although it was noted that the higher correlation was usually found with the acceleration that had the higher amplitude at the location of the accelerometers.

Figure 15. Typical tire enveloping function.

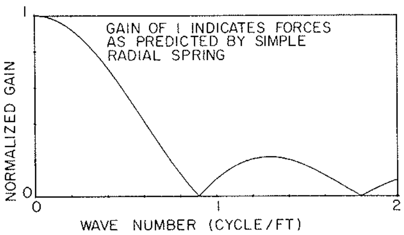


Figure 16. Ride model of tractor-trailer combination.

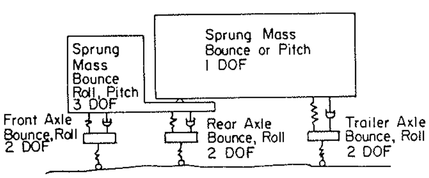


Figure 17. Typical impedance characteristics of commercial trucks.

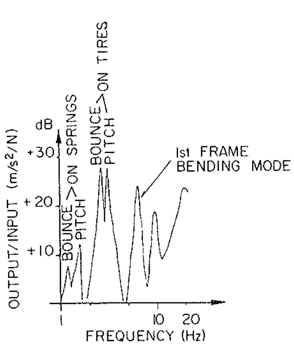
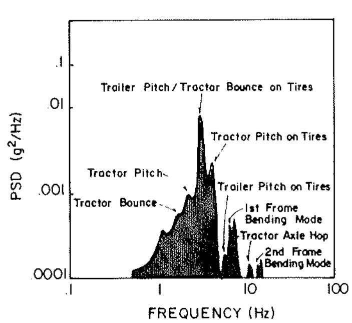


Figure 18. Typical PSD of vertical cab acceleration for tractor-trailer.



Commercial Vehicles

Heavy trucks and tractor-trailers rely basically on the same physical concept for ride isolation as the passenger car, although different results are achieved. The differences arise from factors related to the suspension system, structural differences, and rider location. Furthermore, the articulation of a tractor-trailer gives it additional degrees of freedom, as illustrated in the model of Figure 16. Instead of just bounce, pitch, and roll motions, the trailer is also free to move independently, which adds other resonances to the system.

The suspensions commonly used on trucks are proportionately much stiffer than on passenger cars. In part, this is necessary to limit height changes with loading. In addition, multiple leaf springs that have a high level of coulomb friction are commonly used. These deflect readily on large bumps, which yields a natural frequency near 1 Hz, as shown in Figure 17 for a straight truck that has a tank-type vocational body (15). But they may be effectively rigid on smaller bumps with the result that the tires may act as the primary suspension on smooth roads. Hence, pronounced vehicle resonances may also be observed in the figure in the vicinity of 3 Hz.

The longer wheelbases of trucks and trailers and the absence of a rigid body structure extending over the length of the vehicle allows chassis bending resonances to occur at low frequencies. First-order bending modes typically occur at 5-10 Hz on trucks and tractors. To allow for the load transport function, the riders must also be located at the vehicle extremities, far from the center of mass, where ride vibrations due to pitching and frame bending are more severe.

These factors, all combined, result in commercial vehicle ride motions that are more severe in all directions and richer in resonances in the mid-frequency range.

The additional modes of a tractor-trailer are indicated on the on-road acceleration spectrum of Figure 18 measured on a tractor-trailer (16). Here again, the characteristic resonant modes on the tires occur in the vicinity of 3 Hz and structural bending modes near 6-7 Hz.

As with passenger cars, roll or lateral modes may also exist, and the same wheelbase filtering and tire enveloping effects influence the final result. The preponderance of research on truck ride behavior in the United States focuses on the vertical and pitch modes with very little attention to the roll-lateral behavior. Hence, one may consider the behavior represented in Figures 17 and 18 indicative of the major response modes of commercial vehicles to road roughness. Although they, in general, cover the same frequency range as is important to the passenger car, road roughness has a more serious influence on rider comfort in commercial vehicles because of their greater responsiveness and the concentration of vibration energy in the 3-8 Hz frequency range.

SUMMARY AND CONCLUSIONS

The perceived ride of highway vehicles depends not only on the road characteristics and vehicle response but also on the rider's sensitivity to vibration. Viewed as an acceleration input to the moving vehicle, road roughness, on the average, tends to present a uniform excitation at lower frequencies and an increasing amplitude at higher frequencies. Highway vehicles, both cars and trucks, are designed to isolate the rider from road inputs above the body resonant frequency by the vibration attenuation properties of the suspension system. Vehicle designers attempt to keep the body resonant frequency as low as possible to maximize the attenuation. On passenger cars, body resonance as low as 1 Hz is often achieved, but the high-friction, leaf-spring suspensions common on commercial vehicles often result in a 3-Hz resonance.

Although the suspension serves to attenuate inputs at higher frequency, much of the benefit is offset by the growth in road roughness acceleration amplitude at higher frequencies. Secondary resonances of the axles or chassis compromise the attenuation achieved so that significant vibrations on the vehicle body may be evident out to 20 Hz.

The highway engineer, who is responsible for maintenance of road condition, has a direct influence on what is normally the major source of excitation for vehicle vibrations that involve body resonance through the axle resonance. Vehicle ride in the frequency range of approximately 0.5-20 Hz is dominated by the physics of these resonances with current vehicle designs. Vehicle manufacturers can alter, but not eliminate, their effects. Hence, the interest of the highway engineer should focus on roughness in that frequency range that then corresponds to a wave number range appropriate for the prevailing traffic speed.

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