

Axle Tramp Contributions to the Dynamic Wheel Loads of a Heavy Truck

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ABSTRACT

The road wear caused by heavy trucks depends on the magnitude and distribution of loads imposed on the roadway structure. For theoretical analyses, pitch-plane vehicle models are commonly used to predict dynamic wheel loads. A pitch-plane model treats the vehicle as two-dimensional, neglecting the axle tramp mode.

A three-dimensional truck model is used to evaluate the significance to the axle tramp mode to dynamic wheel loads. The model is calibrated to duplicate frequency responses of a five-axle tractor-semitrailer measured on a hydraulic road simulator. The drive axle of the vehicle has an axle tramp mode with a resonant frequency that is different than the axle hop mode, but a similar damping level. The tramp mode of the steer axle occurs at the same resonant frequency as the hop mode, with a much lower level of damping.

The roll component of a set of measured road profiles of typical highway pavements is investigated. The profiles are fed into a two- and three-dimensional model of the five-axle tractor-semitrailer to assess the relative significance of the axle tramp mode to dynamic wheel loads and pavement wear.

INTRODUCTION

Wheel loads of heavy trucks are major contributors to pavement stresses that ultimately damage the structure and reduce pavement life. Pavement stresses caused by trucks relate in part to the static load distribution among the axles, and in part to the dynamic components of those loads. On typical roads, truck dynamics increase the average pavement wear by 10 to 40 percent over that associated with static loading. (1, 2) At the most severely loaded locations, truck dynamics increase pavement wear by 50 to 400 percent. (3-5) Understanding the manner in which dynamics increase wheel loads provides a foundation for developing truck designs that cause less pavement wear, and pavement designs that are more resistant to damage.

Experimental measurements provide the most reliable understanding of the role of truck dynamics in road loading. However, the expense and time required to carry out large

testing programs prohibits the use of testing alone to investigate complicated trade-offs in truck and pavement design. The alternative to testing is computer simulation using valid models of the truck and the subsystems that influence dynamic behavior.

The modes of vibration relevant to truck ride and dynamic pavement loading are well known. (6) From the perspective of dynamic wheel loads that stress and fatigue pavements, roll dynamics are frequently ignored. Most simulation models in use for predicting dynamic pavement loading from trucks are two-dimensional models that allow movement of the vehicle in the pitch plane only. (7) In part, this is justified by the observation that ride motions in heavy trucks are largely vertical and fore-aft, arising from vertical and pitch dynamics, with little energy in the lateral or roll directions.

Roll motions of the vehicle body contribute to low-frequency load differences between the left and right sides of the vehicle, while roll vibrations of the axle, known as *axle tramp* (8), contribute to high-frequency dynamic loads under individual wheels. A study of the significance of roll motions on dynamic pavement loading requires a three-dimensional vehicle model that includes roll degrees of freedom of the vehicle body and each axle.

In a validation study of a three-dimensional truck model, Cole and Cebon (9) concluded that at highway speeds the roll mode of the truck body is not sufficiently excited to contribute to dynamic loading. The study also concluded that the roll modes of truck axles could be ignored if they had similar resonant frequencies and damping as the axle hop mode.

The study presented here investigates the dynamic behavior of the individual wheels on a truck with air-spring (drive) and four-leaf (trailer) tandem suspensions using a hydraulic road simulator. Measurements of the input-output relationships between various wheels are used to investigate the modes involved. Axle tramp (roll) vibrations appear at approximately 15 Hz in the air suspension and have dynamic levels comparable to axle hop (vertical) vibrations. Tramp

vibrations of the steer axle appear at approximately the same frequency as the hop resonance, but are much more significant.

An analytical model of an articulated tractor-semitrailer with vertical, pitch, and roll-plane degrees of freedom is used to duplicate the dynamic behavior of a truck measured on the hydraulic road simulator. The wavelength content of measured road profiles is investigated with the aim of assessing the relative levels of excitation provided to the roll modes of vibration. The three-dimensional truck model is then exercised using the measured road profiles as input to evaluate the relative significance of the axle tramp vibration mode as a contributing source of dynamic wheel loads excited by road roughness. Results are presented which illustrate the importance of the axle tramp mode to pavement wear induced by the vehicle.

VEHICLE MODELS

The motions of the various components in a vehicle can be predicted mechanistically by solving the ordinary differential equations that describe their dynamics and kinematics. The assumption is made that the truck consists of a system of rigid bodies upon which forces and moments act. The primary masses in the system are the bodies of tractor and the semitrailer. These are supported at each axle by suspension systems, and are appropriately designated as the *sprung masses*. The additional masses significant to dynamic wheel load performance are those concentrated at each axle arising from the mass of the axle, brakes, steering knuckle, wheels, and portions of the suspension linkage. These are denoted as *unsprung masses*.

The treatment of the various masses as rigid bodies ignores structural vibrations of the individual components. Unsprung masses generally have no structural vibration modes within the frequency range of interest. Trucks, tractors, and trailers usually have frame-bending vibration modes within this frequency range. While these may be significant to vibrations present on the body of the vehicle (the ride behavior), in general they have little influence on the loads experienced under the wheels.

The equations of motion for the simulation models used in this study were written using the AUTOSIM™ software package. AUTOSIM automatically generates computationally efficient **simulation** programs for mechanical systems composed of multiple rigid bodies. (10, 11) In AUTOSIM, the modeler describes the vehicle system in terms of the rigid bodies, the manner in which they are connected, compliant elements (linear or non-linear) within the system, and disturbance inputs. AUTOSIM formulates the equations of motion symbolically, and then writes a file containing the source code for a FORTRAN or C program that can numerically integrate the equations of motion. When the program is compiled and executed, it produces ready to plot data files of time histories of output variables of interest.

In this study, two simulation models of a five-axle tractor-semitrailer were used: a 9-degree-of-freedom, two-dimensional pitch-plane model, and a 24-degree-of-freedom three-dimensional full vehicle model. The pitch-plane model accepts a road profile for a single wheeltrack as a disturbance input. The full vehicle model accepts two profiles, one for the left side and one for the right.

The full vehicle model was used to draw conclusions about the significance of roll motions to pavement loading. The pitch-plane model was used primarily to assess the consequences of ignoring roll motions.

PITCH-PLANE VEHICLE MODEL

The pitch-plane model, illustrated in Figure 1, includes two sprung masses (the tractor and semitrailer) and five unsprung masses (one for each axle). The tractor sprung mass translates vertically and longitudinally, and rotates in pitch. The trailer is attached to the tractor at the hitch point with a pin joint. The pin joint allows the semitrailer to rotate in pitch relative to the tractor. This gives the sprung masses a total of four kinematical degrees of freedom. Each unsprung mass translates vertically relative to the sprung mass it supports. Suspensions are modeled as non-linear springs and linear viscous shock absorbers. In addition, they include load equalizers. The tires are modeled as linear springs and linear shock absorbers in parallel.

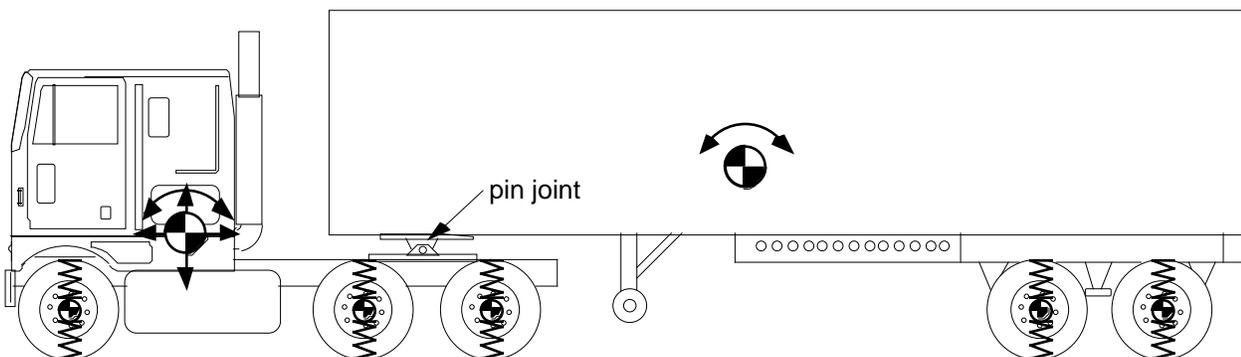


Figure 1. Tractor-semitrailer pitch-plane model.

Full vehicle model

The full vehicle model is similar to the above, but the tractor sprung mass has six kinematical degrees of freedom arising from translation in the vertical, lateral, and longitudinal directions, and rotation in yaw, pitch, and roll. The trailer is attached to the tractor at the hitch point with a ball joint. The ball joint allows the semitrailer to rotate in pitch, roll, and yaw relative to the tractor. A linear torsional spring is used to resist roll of the semitrailer relative to the tractor. This represents the torsional stiffness of the hitch.

Axle motions are defined relative to the sprung masses. Each axle has two translational and one rotational degree of freedom relative to the sprung mass it supports. The axles move vertically, and rotate in roll about a roll axis. A lateral degree of freedom is included to allow lateral movement of the axle mass as it rotates about a roll axis fixed in the sprung mass.

Suspensions are modeled as non-linear springs and linear viscous shock absorbers on the left and right side. The suspensions also include anti-roll bars modeled as linear torsional springs, and a linear spring that resists lateral axle motion. The tandem suspensions in the full model do not include load equalizers.

ROAD PROFILE INPUT

Both vehicle models accept road profiles as disturbance inputs. The road surface profile is described by a series of road elevation values spaced at fixed intervals along the road. Road profiles stored in this form can be measured or synthesized. The models assume the profile to be straight (constant slope) between points. The pitch-plane model requires a single wheeltrack profile, whereas the full vehicle model requires separate profiles for the left and right side.

Note that the pitch-plane model (with load equalization disabled) predicts the same wheel loads for a given road profile input as the full vehicle model when that profile is entered on both the left and right sides.

SUSPENSION SPRING MODEL

A key system that must be modeled properly in order to accurately predict dynamic load performance of heavy trucks is the suspension spring, particularly leaf springs. Truck leaf springs, as well as other suspension components, exhibit a great deal of friction in their operation which produces complex force-displacement characteristics.

Figure 2 shows the force-displacement characteristics for a typical truck leaf spring measured experimentally. (12) An analytical model for duplicating this behavior has been developed in previous research and is documented elsewhere. (13) This model is used for all suspension springs in the pitch-plane and full vehicle models. The behavior of the analytical model over the small displacements typical of ride motions is shown in Figure 3.

The model is based on representing the force properties of the system on a nominal stiffness plus a coulomb friction

force which is dependent on previous motions. The model also includes a parameter for describing the exponential rate at which the suspension force within a hysteresis loop approaches the outer boundary of the force-deflection envelope. This type of behavior is inherent not only to leaf springs, but to many other components of truck suspensions. Therefore, the model is useful for replicating friction effects from other components in truck suspensions, such as air springs.

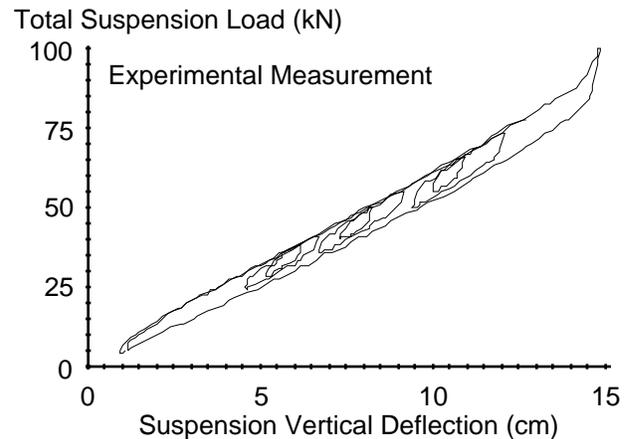


Figure 2. Measured vertical force-deflection behavior of a leaf spring.

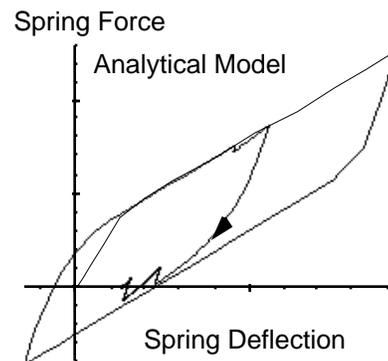


Figure 3. Simulated vertical force-deflection behavior of a leaf spring.

TANDEM SUSPENSION LOAD SHARING

Four-spring tandem suspensions incorporate compliant elements (such as leaf springs), along with linkages to equalize the load between the two axles. (14) A schematic is shown in Figure 4, with degrees of freedom for the suspension indicated with arrows. The axles may translate vertically, and the equalizer beam is free to rotate. There is no interaction between equalizers on opposite sides of the vehicle. Many tandem suspensions are functionally equivalent to the four-leaf shown, in which the load on both axles is equalized by a balance of forces in the springs, accomplished by an equalizer beam.

Including the inertial properties of the equalizer link would result in dynamical equations of motion that are “stiff” and require an order of magnitude more computation

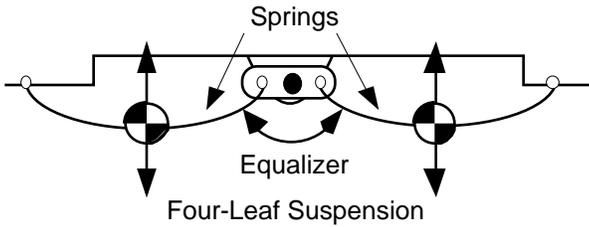


Figure 4. Tandem suspension load sharing.

to predict vehicle motions. However, the inertial effects of the link are insignificant in comparison to the loading forces. A quasi-static solution method can be used to account for the load-equalization and the frictional behavior of the link. This approach is used in the pitch-plane model. Thus, tandem suspensions in the model only include two degrees of freedom.

The springs in Figure 4 are shown as leaf-springs. However, the same mathematical model can be used to describe air-springs or torsion-bar springs, differing only in the parametric values necessary to describe their characteristics. (15)

TIRE MODEL

Tires are normally modeled as springs and dampers in parallel connecting the axles to the ground. The tire springs and dampers are simple linear elements while in contact with the ground. Should the tire leave the ground, the tire force goes to zero (the model does not allow the ground to pull back on the tire).

The full vehicle model also includes combined slip for lateral compliance and aligning moment behavior of the tires. These effects are not significant, however, for this study, because the vehicle under investigation is always run in a straight line.

VEHICLE PROPERTIES

The determination of parameters to describe the behavior of a vehicle requires measurements of force-displacement behavior under dynamic conditions. Ideally, this would be accomplished using on-road measurements, but this is complicated by the difficulty and expense of carrying out such tests.

Hydraulic road simulators used to study the ride and vibration behavior of heavy trucks provide an alternative method that can be adapted to this purpose. During the setup of a vehicle on many simulators, a *remote parameter characterization* (RPC) is often performed to determine transmissibility from road inputs at each wheel to vibrations at various points on the vehicle. (16, 17) The RPC process involves excitation at different wheel positions from which to determine the transmissibility to different parts of the vehicle.

RPC data for a loaded five-axle tractor-semitrailer were obtained from a truck manufacturer. The truck has an air-spring suspension on the drive axle and a four-leaf

suspension on the trailer axle. These data were used to calibrate dynamic properties for the pitch-plane and full vehicle models.

The relationship between the truck measurements and the pitch-plane simulation model are illustrated in Figure 5. To obtain these results the truck was characterized by parameter values measured in the laboratory with some variation of parameters within their normal range of uncertainty to obtain the best fit. Good agreement was obtained over the frequency range of 0-14 Hz encompassing the rigid body bounce and pitch resonances, centered around 3 Hz, and the axle-hop resonance near 13 Hz. Since the transmissibility is measured across a tandem suspension, it also validates the inter-axle load transfer across the equalizer system. Notable in this figure is the additional response above 14 Hz observed in the experimental measurements. This arises from the axle tramp mode, which is not be included in the pitch-plane model.

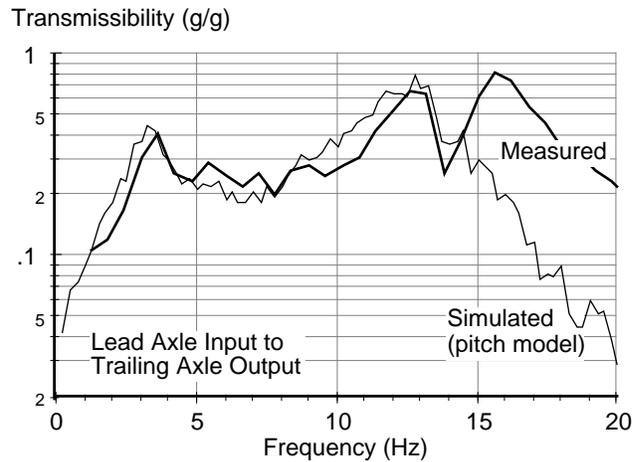


Figure 5. Comparison between measured and simulated behavior of a four-leaf tandem suspension.

The axle tramp mode of vibration is controlled by the roll moment of inertia of the axle resonating against the compliance of the tires and suspension system. Figure 6 shows the elements of the system. The roll moment of inertia is reflected in the figure by representing the unsprung mass of the axle as split between two point masses distributed laterally.

In the absence of auxiliary roll stiffness, the natural frequency for the axle tramp mode is:

$$f_t = \frac{1}{2} \sqrt{\frac{2K_t S_3^2 + 2K_t S_2^2 + 2K_s S_1^2}{mk^2}}$$

where

m = Unsprung mass

k = Radius of gyration = $\sqrt{\frac{I_{xx}}{m}}$

I_{xx} = Roll moment of inertia of the axle

For typical truck drive axles the natural frequency in tramp equals or exceeds that of axle hop. This is particularly true

with suspensions that have relatively soft springs such as the air spring. Considering only the tire springs, the axle hop natural frequency will be:

$$f_u = \frac{1}{2} \sqrt{\frac{4K_t}{m}}$$

The comparable expression for the tramp mode is:

$$f_t = \frac{1}{2} \sqrt{\frac{2K_t S_3^2 + 2K_t S_2^2}{mk^2}}$$

Then their ratio will be:

$$\frac{f_t}{f_u} = \sqrt{\frac{S_3^2 + S_2^2}{2k^2}}$$

The radius of gyration, k , for a drive axle is usually of the same order as S_2 . Typical values are 0.7 m for k , 0.75 m of S_2 , and 1.08 for S_3 . (18, 19) The resulting frequency ratio is 1.33. Note that the true ratio may be even higher if the contribution of the auxiliary roll stiffness of the suspension is significant. This is more apparent in the transmissibility plot for an air-spring suspension as shown in Figure 7. In this case the rigid body modes center around 2 Hz, with axle hop at 10 Hz. The axle tramp mode accounts for the peak in response at a frequency just below 14 Hz.

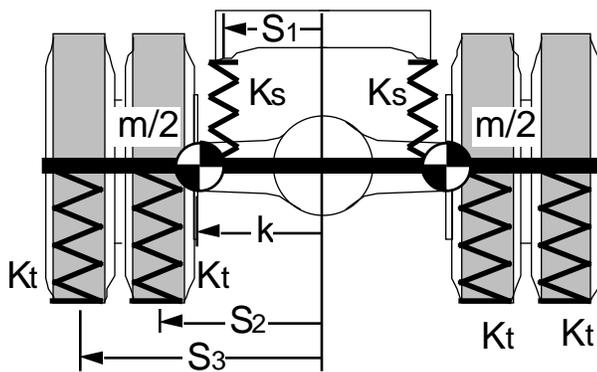


Figure 6. Simple roll model for a truck axle.

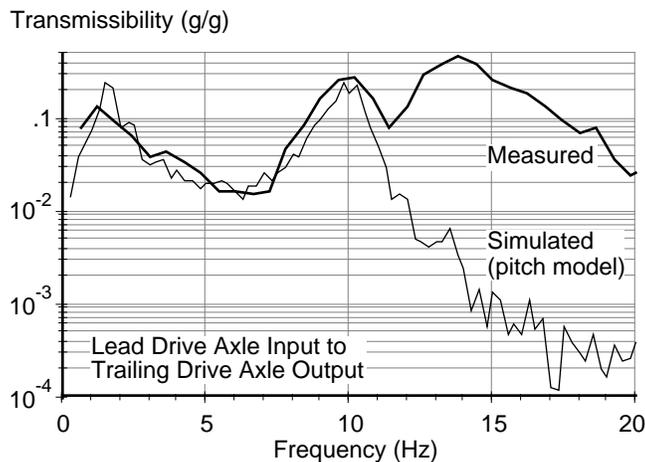


Figure 7. Comparison between measured and simulated behavior of an air-spring tandem suspension.

Unlike the pitch-plane model, the full model can duplicate the axle tramp mode. This is demonstrated by examining the response of the truck axle to inputs from the ground. Figure 8 shows a comparison of the transmissibility measured on the truck drive axle to those predicted by the pitch-plane and full vehicle models using essentially the same parameters. The difference in response of the pitch-plane and full vehicle models is due to the additional modes of vibration contained in the full model.

The curves in Figure 8 follow closely up through 10 Hz, which is the axle-hop mode present in both models. However, near 13 Hz the full model exhibits an increase in response attributable to tramp resonance. Note that the axle tramp mode occurs at a different resonance frequency than the axle hop mode, but has similar damping.

Figure 9 compares the measured and simulated transmissibility for one of the wheels of the steer axle. As was the case for the drive axle, the full model is able to duplicate the axle tramp mode fairly accurately, and the pitch-plane model is not. In the case of the steer axle, however, the tramp mode occurs at the same resonance

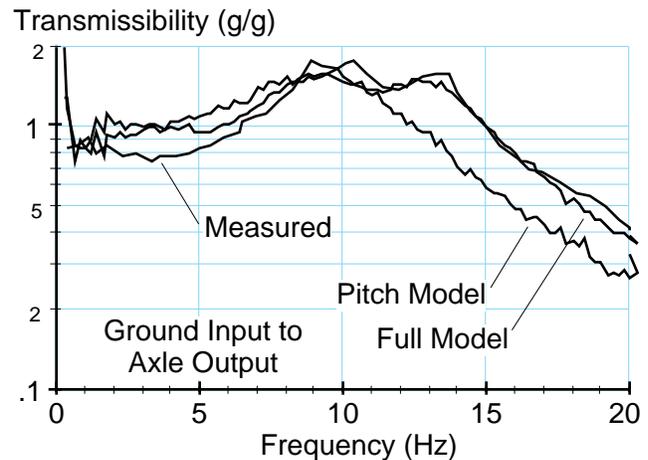


Figure 8. Comparison of simulated and measured transmissibility for a wheel on the air-spring suspension.

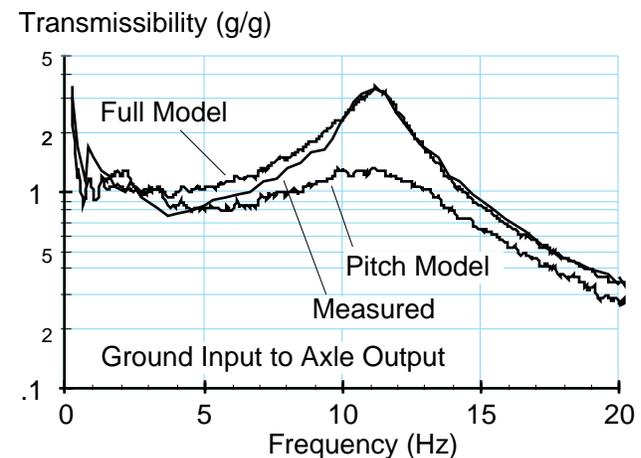


Figure 9. Comparison of simulated and measured transmissibility for a wheel on the steer axle suspension.

frequency as the hop mode, but is not damped nearly as well.

The full vehicle model predicts the transmissibility, and several other measured frequency response characteristics, reasonably well for the steer axle and air-sprung drive axle suspensions. However, the absence of the equalizer system in the full model prevents it from predicting the frequency response of the four-leaf trailer suspension accurately. Thus, the comparisons of computed dynamic wheel loads presented in this paper focus on the steer axle and drive axle suspensions only.

ROAD PROFILES

A set of measured road profiles was assembled to support statistical comparisons between dynamic loads predicted with and without considering roll motions. These measurements cover three pavement types: asphalt concrete (AC), Portland cement concrete (PCC), and asphalt overlay on PCC (composite). The measurements include roughness levels over the range of 0.7 to 3.7 m/km (45 to 235 in/mi), in terms of the International Roughness Index (IRI) for PCC and composite pavements; and 0.7 to 5.0 m/km (45 to 320 in/mi) for AC pavements.

The set of measurements selected for analysis cover the roughness ranges as uniformly as possible. Target roughness levels were defined within the overall ranges, and an attempt was made to represent them all equally for each pavement type. Whenever possible, the measurements included at least three pavement sections near each target roughness level, and no more than five. Although this approach does not represent a common distribution of roughness for each pavement type, it does represent a broad range of roughness uniformly. This enables the identification of trends that depend on roughness level. These measurements include 37 AC, 22 PCC, and 21 composite pavement sections.

All of the road profile measurements used in this study were made by a K. J. Law Profilometer. The measurements cover pavement sections that are approximately 800 m (1/2 mile) in length and include the left and right wheeltrack. The individual elevation values are measured at 25.4-mm (1-inch) distance intervals, averaged over a 300-mm interval (1-foot) and stored at 152.4-mm (6-inch) intervals. This is approximately equivalent to averaging the elevation values over the length of a tire contact patch, which reflects the envelopment properties of the tires. (20) The profiles are also high-pass filtered with a cutoff wavelength of about 90 m.

Since the profile data validly covers a range of wavelengths from 0.5 to 55 m, the analyses performed on them must not rely on wavelengths outside this band. This study is concerned with truck dynamic loading behavior in the frequency range from 1 to 25 Hz. Thus, the simulations must not be run outside the speed range of 12.5 to 33 m/s (27 to 74 mi/hr).

ROAD PROFILE CHARACTERISTICS

The importance of the axle tramp vibration mode to dynamic wheel loads depends on the extent to which it is excited by road roughness. The axle tramp mode is excited by the roll input to the wheels at the axle tramp resonant frequency. Although measured road profiles are usually represented as left and right elevation as a function of distance along the road, they can be represented as a combination of vertical and roll inputs to a vehicle. This is accomplished by taking the point-by-point average of the left and right profiles, known as the *average profile*, and the point-by-point difference of each profile, known as the *difference profile*. (21) The difference profile represents the roll input to the vehicle, and the average profile represents the vertical input.

In general, the left and right profile are approximately the same for very long wavelengths (the left and right profiles follow the same hills and valleys); and the profiles are uncorrelated for very short wavelengths (in the texture range). (22) Thus, the difference profile is expected to have a magnitude similar to the average profile for short wavelengths, and decrease to zero as the wavelengths grow very long. Figure 10 demonstrates this phenomenon.

Figure 10 shows power spectral density (PSD) functions for the average and difference profiles of an asphalt road of moderate roughness. The PSDs have equal magnitudes for wavelengths shorter than 1 m, but the PSD of the difference profile decreases steadily relative to the PSD of the average profile with increasing wavelengths. At wavelengths longer than 20 m (0.05 cycles/m), the difference profile is insignificant relative to the average profile (a factor of 10 or more lower). Note that the PSD functions are scaled to show the distribution of road slope variance as a function of wavenumber. The slope variance of roads typically exhibits a fairly uniform spectral content over the range of wavenumbers that affect vehicle ride and dynamic wheel loading, thereby making the PSD of road slope much easier to interpret than that of road elevation.

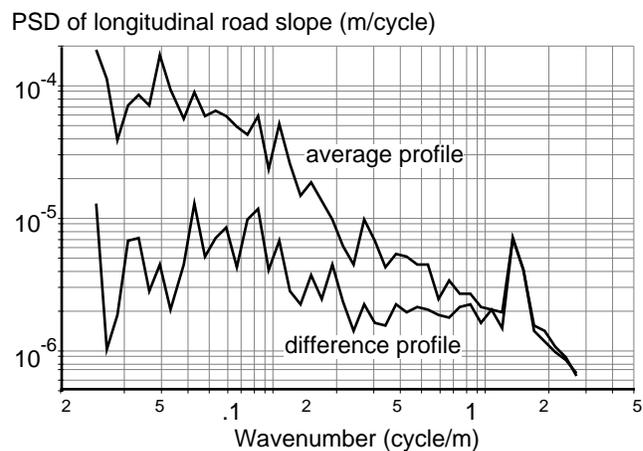


Figure 10. PSD of the vertical and roll components of a measured road profile.

A simple way to characterize the significance of the roll inputs relative to the vertical inputs a vehicle will experience on a road is to examine the ratio of the PSD functions. This *PSD ratio* is obtained by normalizing the

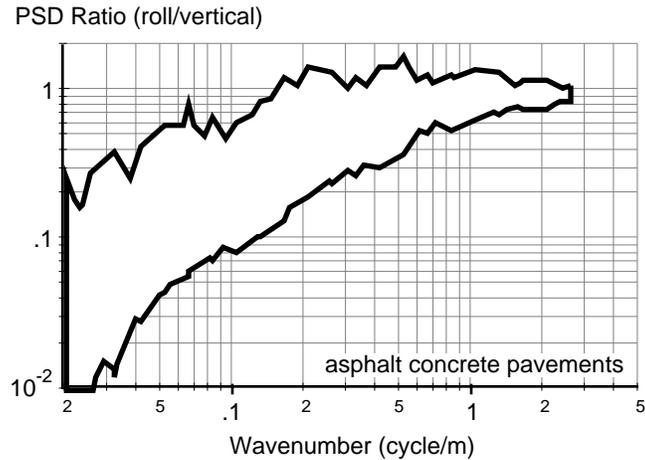


Figure 11. Aggregate PSD ratio functions for 37 AC pavement sections.

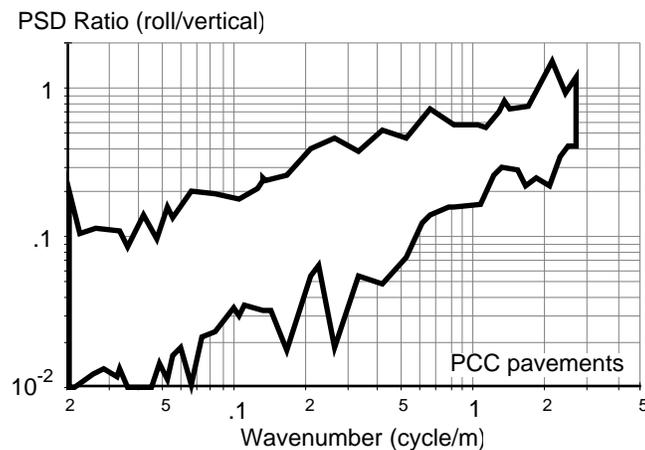


Figure 12. Aggregate PSD ratio functions for 22 PCC pavement sections.

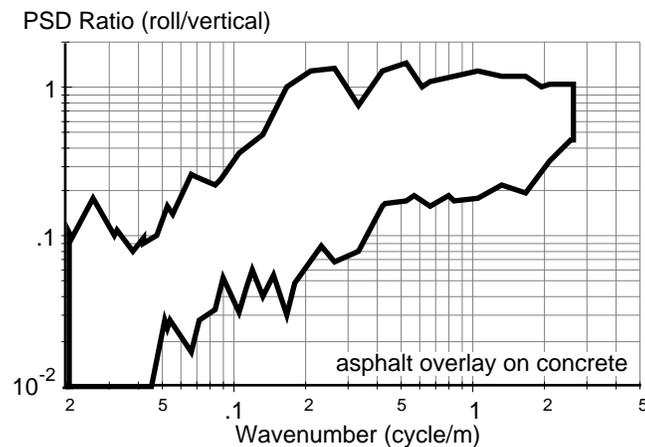


Figure 13. Aggregate PSD ratio functions for 21 composite pavement sections.

PSD coefficient of the difference profile by the PSD coefficient of the average profile at each center frequency. Figures 11, 12, and 13 show these functions for measurements of the AC, PCC, and composite pavement sections used in this study, respectively. The outlined areas represent the aggregate of the PSD ratio functions—that is, the region covered by all of the PSD ratio functions for each pavement surface type.

The aggregate PSD ratio functions for all three pavement types have values near one at 2 cycles/m. As wavenumber decreases, the AC pavements maintain a high aggregate PSD ratio. In contrast, the aggregate PSD ratio for the PCC pavements decreases rapidly with wavenumber.

In general, the AC pavements exhibit roughness with a more significant roll component over the entire range of wavenumbers shown. This is because AC pavement is surfaced with flexible material, whereas PCC pavement is surfaced with rigid slabs. Thus, AC pavement is less likely to maintain its shape across a lane when it is constructed and, in particular, as it wears. PCC pavement failures, on the other hand, often occur across a lane (faulting, cracking, etc.). These failures will contribute to the vertical component of roughness rather than roll, even early in the life of a PCC pavement.

Composite pavement has roughness properties similar to AC pavement when it is first overlaid, but takes on the roughness properties of the PCC underneath after some exposure to traffic and the environment. This is reflected in the aggregate PSD ratios (see Figure 13), which take on the frequency characteristics of both AC and PCC pavement.

SIGNIFICANCE TO AXLE TRAMP

The axle tramp vibration mode of the drive axle of the vehicle investigated in this paper resonates at a frequency of about 14 Hz. The significance of these road profile characteristics to dynamic wheel loads caused by axle tramp depends on the excitation level they provide at 14 Hz. At a typical highway speed of 96 km/hr (60 mi/hr), the content of road roughness with a wavelength of about 2 m is experienced by the vehicle at 14 Hz. This corresponds to a wavenumber of 0.5 cycles/m. Note that the AC pavements (see Figure 11) have an aggregate PSD ratio ranging from 0.4 to 1.5 at this wavenumber. The PCC pavements, on the other hand, have an aggregate PSD ratio ranging from 0.07 to 0.4 (see Figure 12). Thus, axle tramp vibrations are expected to impact dynamic wheel loading much more significantly on AC pavement than on PCC pavement.

DYNAMIC LOADS

The consequence of the axle tramp vibration mode is additional dynamic response that may contribute to elevated dynamic wheel loads. This section compares dynamic loads predicted by a pitch-plane and full vehicle model as a means of assessing the relative significance of the axle tramp mode to dynamic wheel loads and pavement wear. The vehicle examined in this section is the five-axle tractor-semitrailer

studied on the hydraulic road simulator. The properties used to describe the vehicle are the same as those that resulted in the calibration of the models demonstrated in Figures 5, 7, 8, and 9.

Both models of the vehicle were exercised over all of the measured road profiles investigated above. The full model was run using the left and right wheeltrack profiles as input simultaneously. The pitch-plane model was run twice over each measured road profile, once using the left wheeltrack profile as input, and once using the right. The predicted dynamic load histories output by the models were examined by comparing the overall level of dynamic loading and the pavement wear.

The results presented here focus on the air-sprung drive axle and the steer axle. Although the static load of the steer axle was much lower than the drive axle, it was included in the analysis because it was fitted with a conventional single tire. The smaller footprint area of the single tire results in pavement strains that are comparable to those caused by the drive axle. (4, 23) Thus, the dynamic component of wheel loads imposed by the steer axle of the vehicle are also relevant to pavement wear.

The additional dynamics present in the full model might be expected to cause the predicted dynamic wheel loads to be more severe than those of the pitch-plane model. However, this is not always the case. A large, short-duration bump that occurs on one side of a road is perceived by the pitch-plane model as a pure vertical input to the vehicle that excites both the axle hop mode and the body bounce and pitch modes. The full model, on the other hand, treats the same bump as a combination of vertical and roll inputs that are half as severe. Thus, if the axle tramp mode is well damped, the full model may not predict higher dynamic loads.

OVERALL LEVEL OF DYNAMIC LOADING

A well known measure of the dynamic variation in wheel loads for a specific combination of road roughness and speed is the dynamic load coefficient (DLC). The dynamic load coefficient is defined as the coefficient of variation of the dynamic load history of a wheel. (5) It is calculated by normalizing the standard deviation of dynamic wheel load by the overall mean wheel load.

The DLCs predicted by the full model were, on average, nearly equal to those predicted by the pitch-plane model. Figure 14 shows a comparison of the DLCs predicted for all four wheels of the drive axle, with a line of equality included for reference. The DLCs predicted by the full vehicle model range from 23 percent higher to 25 percent lower than those predicted by the pitch-plane model, and are, on average, 2 percent lower. The relative insignificance of the axle tramp mode of the drive axle is due to the fact that it is well-damped, and occurs at a relatively high resonance frequency (14 Hz), which is not heavily excited by road roughness at typical highway speeds.

The DLCs for the steer axle predicted by the full model average 8 percent higher than those predicted by the pitch-plane model (see Figure 15). This is because the steer axle of this vehicle has a poorly-damped tramp mode. On the asphalt roads, which provide greater excitation to the axle tramp mode, the DLCs average 10 percent higher, with worst-case values as high as 30 percent. For PCC the roads, on the other hand, the DLCs were only 4 percent higher.

PAVEMENT WEAR

If the distribution of wheel loads applied to the pavement surface by a fleet of truck traffic is truly random, then pavement wear caused by a wheel is directly related to its DLC. However, if the most severe dynamic loads are applied to the same pavement locations, then pavement wear is most directly related to the peak dynamic wheel loads.

The relative pavement wear caused by a truck wheel is often related exponentially to the dynamic load it applies to the pavement by raising it to the 4th power (though many other powers have been suggested). (24) Thus, the relative

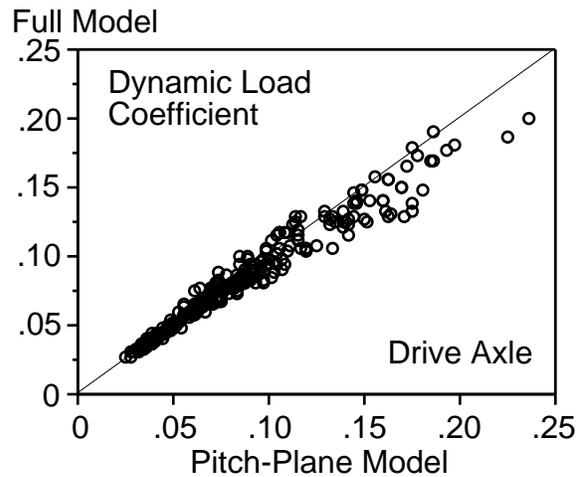


Figure 14. Comparisons of DLCs predicted by the pitch-plane and full vehicle models.

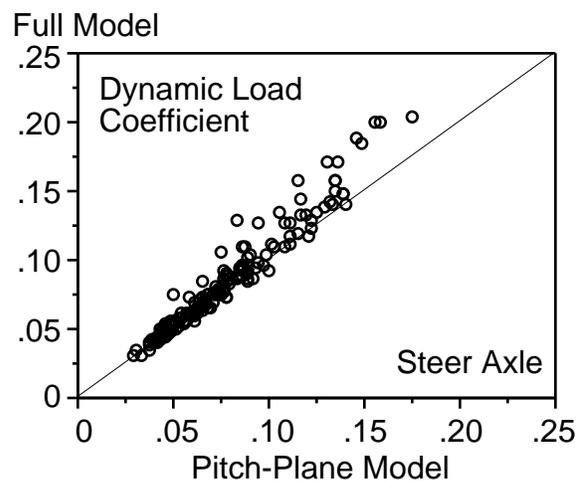


Figure 15. Comparisons of DLCs predicted by the pitch-plane and full vehicle models.

effect of dynamic wheel loads on pavement wear can be assessed by raising the dynamic load to the 4th power.

As a means of quantifying the effect of the most severe dynamic wheel loads predicted by the models, the 95th percentile weighted dynamic load was calculated. This is obtained by normalizing each dynamic load history by the average load, then raising the normalized instantaneous loads to the 4th power. Each load history is then cast into a frequency distribution, and the 95th percentile level is determined. This represents the relative level of wear sustained by the 5 percent of the pavement length that is subjected to the most severe wheel loads. Such a small portion of the overall pavement length was chosen because only 5 percent of the road surface area needs to fail before the road becomes unserviceable.

As expected, the 95th percentile level of relative wear predicted by the full model for the drive axle was, on average, nearly the same as that predicted by the pitch-plane model (within 1 percent).

For the steer axle, on the other hand, the full model predicts a 95th percentile level that is 4 percent high when averaged over all roads (see Figure 16), and 6 percent higher on the asphalt concrete roads. In the most extreme cases, the full model predicted a wear level 20 percent higher than the pitch-plane model.

These findings generally support those from a recent study of a three-dimensional vehicle model by Cole and Cebon. Their study concluded that the axle tramp mode can be ignored if it occurs at the same natural frequency and at a similar level of damping as the axle hop mode. (9) In this study the steer axle hop and tramp modes occur at the same frequency, but the damping is not at the same level, hence the larger contribution of axle tramp to dynamic loads and pavement wear. On steer axles the narrow separation of left and right suspensions necessary to allow wheel clearance in turning will tend to result in poorly damped tramp modes.

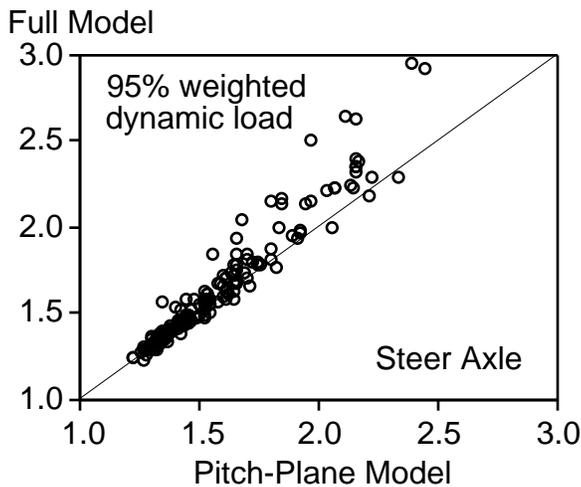


Figure 16. Comparisons of 95% weighted dynamic load predicted by the pitch-plane and full vehicle models.

CONCLUSIONS

In this study, a three-dimensional tractor-semitrailer model was developed and calibrated using measured frequency responses on a hydraulic road simulator.

The findings from the study can be summarized as follows:

- Frequency responses of a five-axle tractor-semitrailer measured on a hydraulic road simulator could be duplicated using a three-dimensional rigid-body model.
- On typical roads, the axle tramp mode will not exacerbate the level of dynamic loading if it is damped as well as the axle hop mode, even if it occurs at a different resonant frequency.
- On typical roads, the axle tramp mode will significantly increase dynamic loading and pavement wear if it is not as well damped as the axle hop mode.
- Truck front (steer) axles may be poorly damped in the tramp mode because of the narrow spacing of the suspensions.
- The contribution of the axle tramp mode to dynamic loads and pavement wear tends to be greater on asphalt concrete roads than on Portland cement concrete roads.

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