Influence of Fifth-Wheel Location on Truck Ride Quality

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A combined analytical and experimental study is described that investigates the influence of the fifth-wheel position of a tractor-semitrailer on the vertical and longitudinal cab accelerations. The analytical model used was a relatively simple six-degree-of-freedom linear frequency domain model subject to statistical road-roughness inputs. The experimental study consisted of instrumenting a cab-over-engine tractor and loaded flatbed semitrailer combination with a three-axis accelerometer package in the cab and then measuring the resulting accelerations experienced on an Interstate highway at several speeds and fifth-wheel locations. Both the analytical and experimental studies show that the ride quality deteriorates as the fifth wheel is moved forward. It is also shown that the increase in root-mean-square acceleration primarily results from the development of a resonant condition of out-of-phase tractor-semitrailer pitch in the 4-Hz region.

The levels of ride quality that exist in a tractor cab have an influence on the long-term health as well as the short-term proficiency of drivers. At sufficiently high levels of exposure to vibration over a period of hours, fatigue and decreased proficiency can result. In this paper the influence of fifth-wheel position on tractor-cab ride is determined. In particular, the ride quality achieved in a tractor is predicted analytically and measured experimentally as the fifth wheel is moved forward to allow greater load capacity within existing regulations and/or to meet tractor-semitrailer length restrictions. For specific truck configurations, the fifth-wheel location can be moved by as much as 2.5 ft to meet federal axle-load regulations. A nominal fifth-wheel location for a truck when not fully loaded is 0.5 ft in front of the rear axle.

Although a number of studies (1-7) have been concerned with truck ride quality and, specifically, with methods for improvement, very little work has been published in which the influence of fifth-wheel position on ride quality is discussed. In 1965, LeFevre (1) conducted an analytical study that showed that jerk (rate of change of acceleration with respect to time) at the driver's seat in the vertical direction increases by 550 percent as the fifth wheel is moved forward 20 in from the rear axle. The model used in this study represented the influence of the trailer on the tractor fifth wheel as a constant force and considered a single wheel excited by a sinusoidal disturbance. In a more complete 1969 study, Walther and others (2) conducted computer simulation studies with a seven-degree-of-freedom model that included heave and pitch motions of the tractor and the semitrailer plus heave motion of tractor front and rear axles and the semitrailer axle. These studies showed that the natural frequencies associated with tractor-semitrailer heave and pitch (2-4 Hz) tend to increase, as does the acceleration level associated with these modes, when the fifth wheel is moved forward.

While the literature cited provides useful background information, none determines ride quality for the case in which a tractor-semitrailer is being driven down a highway subject to typical (random) highway disturbances, and none assesses ride quality in terms of standard ride-quality measures. In the study described in this paper, ride-quality measures are defined, and both vehicle and roadway models are formulated. The model used to evaluate the influence of fifth-wheel location on ride quality is a linear model; thus, the analytical results are valid primarily with respect to the general trends predicted rather than the detailed prediction of local motions and accelerations. Such detailed prediction requires more sophisticated nonlinear models. Finally, the measurements (and their trends) obtained from a tractor-semitrailer instrumented with accelerometers are compared with the analytical predictions.

DEFINITION OF RIDE QUALITY

Ride quality, in this study, is defined by two indices: (a) total root-mean-square (rms) acceleration at the driver's seat and (b) specifications of the International Standards Organization (ISO). The criteria for International Standard 2631:1978, shown in Figure 1, define ride quality in terms of human sensitivity to accelerations as a function of frequency and amplitude. The sensitivity is expressed in terms of the length of time a person can be subjected to an acceleration of a given amplitude and frequency in a specified direction and still be comfortable. For a vehicle's ride to be compared with the specification, it is necessary to obtain the rms of the vehicle accelerations for a specific direction in third-octave frequency bands and compare it with the specification at the band's center frequency. If the rms value exceeds the criteria for any third-octave band, then the ride fails to meet the ride-comfort criterion for that exposure time. Only the magnitude of the ISO curves (and not the shape) change as the exposure time is changed. (Exposure time in this paper is used only as a convenient relative measure. There is currently a great deal of controversy over the relation between exposure time and passenger comfort.)

Although the ISO standard allows for a quantitative evaluation of ride quality, it is often useful in studying trends to use total rms accelerations. Once these trends have been established, the ISO criteria can then be used to see which specific frequencies are causing the changes to occur.

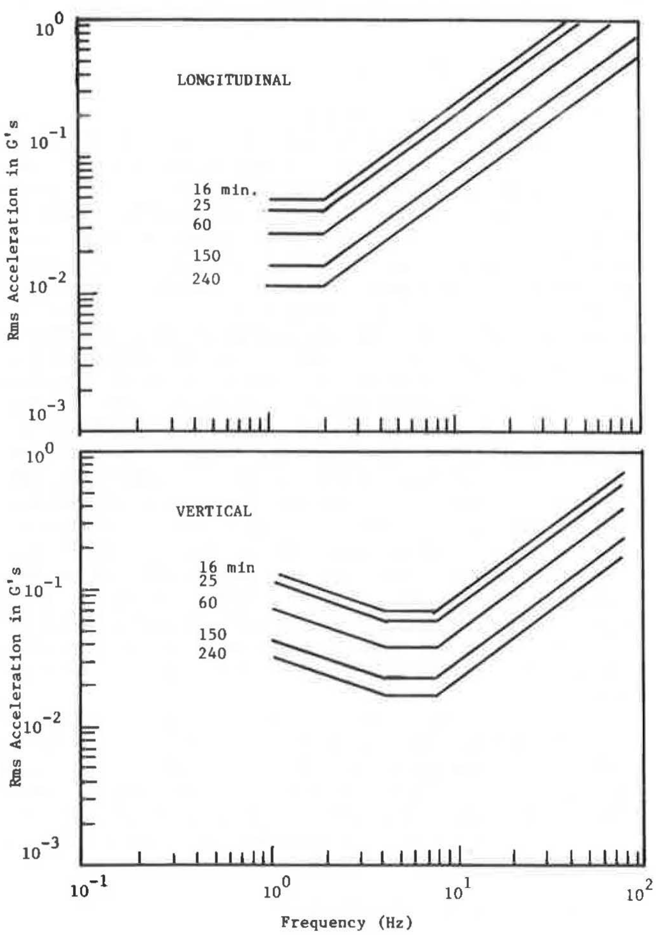
DESCRIPTION OF VEHICLE MODEL

A number of ride-quality models for tractor-semitrailers have been described in the literature. They range from a 3-degree-of-freedom model used by Ellis (3) to an 11-degree-of-freedom model including nonlinearities such as spring friction, shock-absorber characteristics, and wheel lift used by Harwood and Crosby (4). The model developed in the current study is linear; a complete description and derivation are given by Hedrick and others (8). It has been adapted from Walther's 7-degree-of-freedom model, which represents a compromise between the 3-degree-of-freedom Ellis model and the more advanced model of Harwood and Crosby.

The primary object of the vehicle model formulation has been to achieve a simple, yet sufficiently realistic, model to predict the general ride-quality trends in terms of accelerations at the driver's location as the fifth-wheel location is changed. To predict detailed vehicle motions, a model that includes the nonlinear suspension stiffness and damping effects, tractor and semitrailer frame bending, cab-mount-suspension characteristics, and the seat suspension would be required.

The model used in this study is illustrated in Figure 2. This model has six degrees of freedom: tractor heave and pitch, semitrailer heave, and the heave motion of the tractor front axle, tractor rear tandem axle, and semitrailer tandem axle. As shown in the figure, roll, lateral, and other yaw degrees of freedom are neglected, and only heave and pitch are considered. Semitrailer pitch is not a degree of freedom since, when the fifth wheel is represented as a pinned joint, semitrailer pitch motion may be represented directly in terms of tractor heave and pitch and semitrailer heave. In the model, the semitrailer and tractor bodies are assumed to be rigid (no frame bending). The axles on the truck are lumped into three equivalent axles (the front tractor axle, the rear tractor axle, and the semitrailer axle); thus, the detailed effects of tandem axles are neglected. The suspension elements between the axles and the body are represented as an equivalent linear spring and damper, and the tires on each axle are represented by a single equivalent linear spring. The disturbance inputs to the model are represented in terms of roadway displacements (U_1, U_2, and U_3, respectively), at each axle.

Figure 1. Longitudinal and vertical ISO specification for reduced comfort criteria.



A set of linear constant-coefficient differential equations can be derived directly from Figure 2 to describe the dynamic ride-quality model. These equations may be summarized in the matrix form:

$$\bar{M}\ddot{\underline{y}} + \bar{D}\dot{\underline{y}} + \bar{K}\underline{y} = \bar{F}\underline{U} \tag{1}$$

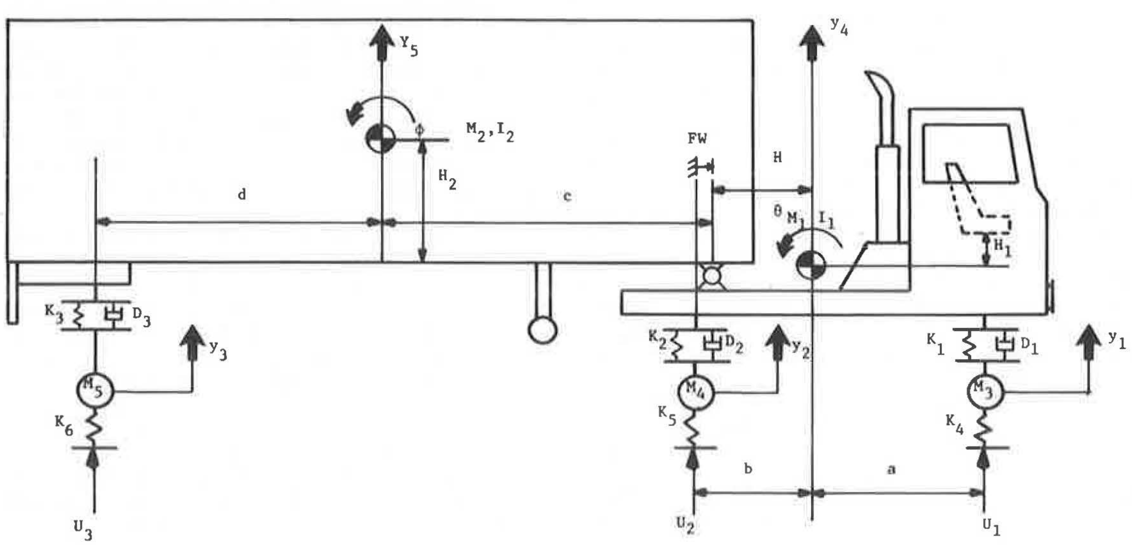
where $\underline{\bar{M}}$, $\underline{\bar{D}}$, and $\underline{\bar{K}}$ are 6x6 system mass, damping, and stiffness matrices, respectively, and $\underline{\bar{F}}$ is a 6x3 input matrix (all of which are given in the ride-quality equations of motion below). In addition, $\underline{y}$ is a 6x1 vector containing the variables associated with each independent degree of freedom where

- y_1 = front axle displacement in the vertical direction (heave),
- y_2 = tractor tandem displacement in the vertical direction (heave),
- y_3 = semitrailer tandem displacement in the vertical direction (heave),
- y_4 = tractor displacement in the vertical direction (heave),
- y_5 = semitrailer displacement in the vertical direction (heave),
- y_6 = tractor angular displacement (pitch, θ), and
- y = dy/dt.

$\underline{U}$ is a 3x1 vector of the systems inputs where

- U_1 = displacement input to the front tires,
- U_2 = displacement input to the tractor tandem tires, and

Figure 2. Dynamic ride-quality model.



U_3 = displacement input to the semitrailer tandem tires.

The equations of the model are linear; however, the input vector ($\underline{U}$) contains time-delayed terms because the second input is essentially equal to the first input but delayed by a time τ_1, and the third input is equivalent to the first input delayed by a time τ_2. The delay times depend on the velocity of the truck. If the input to the front wheel is $U_1(t)$, the input to the tractor driving wheels is

$$U_2(t) = U_1(t - \tau_1) \qquad (2)$$

where τ_1 = (a + b)/truck velocity and the input to the trailer wheels is

$$U_3(t) = U_1(t - \tau_2) \qquad (3)$$

where τ_2 = (a + H + c + d)/truck velocity and

a = distance from the front axle to the tractor's center of gravity (c.g.),
b = distance from the tractor's c.g. to the tractor tandem's centerline,
c = distance from the semitrailer's kingpin to the semitrailer's c.g.,
d = distance from the semitrailer's c.g. to the semitrailer tandem's centerline, and
H = distance from the fifth wheel to the tractor's c.g.

By using the model equations summarized above, given a description of the roadway in terms of $U_1(t)$, the response of the tractor cab can be computed directly. A complete description of the frequency domain model and numerical methods used in this study is contained in Hedrick and others (8).

Ride-Quality Equations of Motion

The linear equations of motion of the tractor-semitrailer model shown in Figure 2 were obtained by using Lagrange's equations. The result is a set of six linear constant-coefficient differential equations of the form $\underline{M}\,\ddot{\underline{y}} + \underline{D}\,\dot{\underline{y}} + \underline{K}\,\underline{y} = \underline{F}\,\underline{U}$, where $\underline{M}$, $\underline{D}$, and $\underline{K}$ are matrices of the forms shown in Figures 3, 4, and 5, where

K_1 = front suspension spring stiffness constant,
K_2 = tractor tandem's suspension spring stiffness constant,
K_3 = trailer tandem's suspension spring stiffness constant,

Figure 3. $\underline{M}$ is a 6x6 system mass matrix.

$$\begin{bmatrix} M_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & M_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & M_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & M_4 + \frac{I_2}{c^2} & -\frac{I_2}{c^2} & -\frac{HI_2}{c^2} \\ 0 & 0 & 0 & -\frac{I_2}{c^2} & M_5 + \frac{I_2}{c^2} & \frac{HI_2}{c^2} \\ 0 & 0 & 0 & -\frac{I_2 H}{c^2} & \frac{I_2 H}{c^2} & I_1 + \frac{H^2}{c^2} I_2 \end{bmatrix}$$

Figure 4. $\underline{D}$ is a 6x6 system damping matrix.

$$\begin{bmatrix} D_1 & 0 & 0 & -D_1 & 0 & -aD_1 \\ 0 & D_2 & 0 & -D_2 & 0 & bD_2 \\ 0 & 0 & D_3 & \frac{d}{c} D_3 & -(1+\frac{d}{c})D_3 & -\frac{d}{c} HD_3 \\ -D_1 & -D_2 & \frac{d}{c} D_3 & D_1+D_2+(\frac{d}{c})^2 D_3 & -\frac{d}{c}(1+\frac{d}{c})D_3 & aD_1-bD_2 -(\frac{d}{c})^2 HD_3 \\ 0 & 0 & -(1+\frac{d}{c})D_3 & -\frac{d}{c}(1+\frac{d}{c})D_3 & (1+\frac{d}{c})^2 D_3 & H\frac{d}{c}(1+\frac{d}{c})D_3 \\ -aD_1 & bD_2 & -\frac{d}{c} HD_3 & aD_1 - bD_2 -(\frac{d}{c})^2 HD_3 & \frac{d}{c}(1+\frac{d}{c})HD_3 & a^2D_1 + b^2D_2 + (\frac{d}{c})^2 H^2D_3 \end{bmatrix}$$

Figure 5. $\underline{K}$ is a 6x6 system stiffness maxtrix.

$$\begin{bmatrix} K_1+K_4 & 0 & 0 & -K_1 & 0 & -aK_1 \\ 0 & K_2+K_5 & 0 & -K_2 & 0 & bK_2 \\ 0 & 0 & K_6+K_3 & \frac{d}{c} K_3 & -(1+\frac{d}{c})K_3 & -\frac{d}{c} H K_3 \\ -K_1 & -K_2 & \frac{d}{c} K_3 & K_1+K_2+(\frac{d}{c})^2 K_3 & -\frac{d}{c}(1+\frac{d}{c})K_3 & -aK_1-bK_2 -(\frac{d}{c})^2 H K_3 \\ 0 & 0 & -(1+\frac{d}{c})K_3 & -\frac{d}{c}(1+\frac{d}{c})K_3 & (1+\frac{d}{c})^2 K_3 & H\frac{d}{c}(1+\frac{d}{c})K_3 \\ -aK_1 & bK_2 & -\frac{d}{c} H K_3 & aK_1 - bK_2 -(\frac{d}{c})^2 H K_3 & \frac{d}{c}(1+\frac{d}{c})HK_3 & a^2K_1 + b^2K_2 + (\frac{d}{c})^2 H^2K_3 \end{bmatrix}$$

K_4 = front tire stiffness constant,
K_5 = tractor tandem's tires stiffness constant,
K_6 = semitrailer tandem's tires stiffness constant,
D_1 = front suspension damping constant,
D_2 = tractor tandem's suspension damping constant,
D_3 = semitrailer tandem's suspension damping constant,
FW = distance fifth wheel is ahead of the tractor tandem's centerline,
H_1 = distance of the driver's seat above the tractor's c.g., and
H_2 = distance of the trailer's c.g. above the bottom of the semitrailer.

Road Input Model

The roadway has been modeled as a statistically stationary random process. Statistical roadway irregularities have traditionally been characterized in terms of mean square spectral densities. A great many road survey data have been processed for highways and runways (9) that indicate that a reasonable analytic representation is given by the spatial spectral density function

$$\phi(\Omega) = A/\Omega^2 \qquad (4)$$

where A = a statistical roughness parameter and Ω = spatial frequency (rad/ft). In this study, the value of A has been chosen as 5×10^{-6} ft, which corresponds to a very good highway. For a vehicle traveling at a constant forward speed V, the spatial density given by Equation 4 can be converted into a temporal spectral density

$$\omega = V\Omega \qquad (5)$$

$$S(\omega)d\omega = \phi(\Omega)d\Omega \qquad (6)$$

that, when applied to Equation 4, yields

$$S(\Omega) = AV/\omega^2 \qquad (7)$$

where

A = velocity of vehicle(ft/s),
ω = temporal frequency (rad/s), and
S = temporal spectral density [$(ft^2/rad)/s$].

RIDE-QUALITY PARAMETRIC ANALYSES

A series of parametric studies has been conducted by using the model described in the previous section. The influence of fifth-wheel location on the tractor-semitrailer natural frequencies and modes of vibration has been determined; vehicle acceleration levels in response to periodic and random roadway disturbances have also been determined. These parametric studies have been conducted for a tractor-semitrailer combination with the parameters summarized (10) below. These parameters represent the values corresponding to the model of Figure 6; thus, for example, the spring stiffness K_1 represents the combined stiffness of the tractor's left and right front suspension springs. Here the tractor was a 4x6 cab-over-engine White (sleeper type), and the trailer was a 40-ft Fruehauf van. D_1, D_2, and D_3 (the damping constants) were calculated by using damping ratios obtained from Walther and others (2).

Parameter	Value	Parameter	Value
a	3 ft	M_1	288 slugs
b	8.83 ft	M_2	1748.45 slugs
c	15.0 ft	M_3	40.37 slugs
d	17.58 ft	M_4	136.65 slugs
K_1	28 800 lbf/ft	M_5	93.17 slugs
K_2	249 000 lbf/ft	I_1	4447.83 $ft\cdot lbf\cdot s^2$
K_3	336 000 lbf/ft	I_2	171 250 $ft\cdot lbf\cdot s^2$
K_4	103 416 lbf/ft	H_1	1.7 ft
K_5	413 664 lbf/ft	D_1	1845 lbf·s/ft
K_6	413 664 lbf/ft	D_2	509 lbf·s/ft
		D_3	485 lbf·s/ft

Natural Frequency and Mode Shapes

The natural frequencies and mode shapes of the tractor-semitrailer unit have been determined for four fifth-wheel positions. The six natural frequencies are summarized in Table 1. The lowest-mode natural frequency decreases from 1.8 to 1.6 Hz, and the mode shape changes from a dominant tractor pitch to tractor heave as the fifth-wheel position is moved forward. The second and third natural frequencies both increase as the fifth wheel is moved forward; the second-mode shape corresponds to strong tractor pitch and semitrailer heave, and the third corresponds to the tractor and semitrailer pitching out of phase. Modes 4-6 correspond to axle motions and are essentially unaffected by fifth-wheel location. In summary, the primary influence of fifth-wheel movement forward is to accentuate tractor heave of the first mode and to increase the natural frequencies of the second and third modes.

Response to a Sinusoidal Disturbance

The steady-state response of the tractor-semitrailer to a sinusoidal disturbance has been computed [$U_1(t) = U_0 \sin\omega t$ with $U_2(t) = U_1(t - \tau_1)$ and $U_3(t) = U_1(t - \tau_2)$] and displayed in Figure 6 as a function of the disturbance frequency for two fifth-wheel positions. The plots illustrate the vertical amplitudes that occur at a rigid point in the cab corresponding to the seat location. Figure 6 shows that, as the fifth wheel is moved forward, the two resonant peaks in the 2- to 4-Hz range increase in frequency and in amplitude. These peaks correspond to the second and third modes of vibration discussed above. The resonance at 12 Hz also increases in amplitude as the fifth-wheel position is moved forward. The series of successive peaks and valleys in the response above 12 Hz corresponds to the phasing of the input between the axles. In summary, as the fifth wheel is moved forward, the amplitudes of the acceleration resonant peaks in the 2- to 12-Hz range increase, yielding a less comfortable ride.

RESPONSE TO A ROADWAY IN TERMS OF RANDOM ROUGHNESS

The tractor-semitrailer response to a roadway disturbance characterized in terms of its random roughness as represented in Equation 7 has been determined for three fifth-wheel positions for a vehicle traveling at 55 mph. The vertical accelerations have been computed at a rigid point in the cab corresponding to the seat. The computed acceleration spectral densities are displayed in Figure 7. These data show that the primary contributions to acceleration occur in the 2- to

Figure 6. Cab vertical response to a sinusoidal roadway disturbance for two fifth-wheel positions at 55 mph.

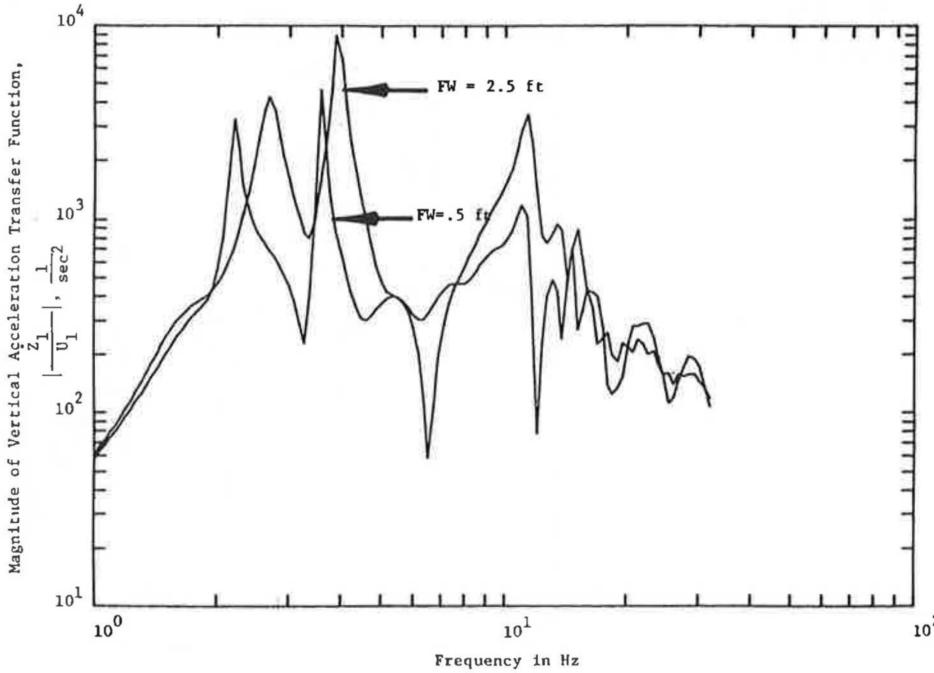


Table 1. Variations in resonant frequencies as a function of fifth-wheel position.

	Natural Frequency (Hz)			
	Fifth-Wheel Location[a]			
Mode Shape	0.5 ft Behind	0.5 ft Ahead	1.5 ft Ahead	2.5 ft Ahead
First	1.8	1.8	1.7	1.6
Second	2.0	2.2	2.4	2.7
Third	3.5	3.6	3.7	3.9
Fourth	8.7	8.7	8.7	8.7
Fifth	11.3	11.4	11.4	11.4
Sixth	14.6	14.6	14.6	14.6

[a]Location in relation to tractor tandem centerline.

12-Hz range and that, as the fifth wheel is moved forward from 0.5 to 2.5 ft, the magnitudes of the resonant peaks increase.

The total rms vertical and longitudinal accelerations at a rigid point corresponding to the driver's seat have been computed and are summarized in Figure 8. These data show that, as the fifth wheel is moved forward from 0.5 ft to 1.5 ft in front of the tractor rear axle, the rms vertical acceleration increases by a factor of 1.65 and the longitudinal acceleration by a factor of 1.5.

To provide a comparison of the ride quality predicted by the computer simulation with the standards set by ISO, the vertical seat-acceleration levels have been computed in accordance with ISO specifications. These levels and the ISO 25-min reduced comfort boundary are displayed in Figure 9. For all cases, the 25-min boundary is exceeded in the 2- to 5-Hz range because of the strong vehicle resonances in these frequency ranges. Thus, even though a relatively smooth roadway has been assumed, i.e., a roughness coefficient of $A = 5 \times 10^{-6}$ ft, the general level of ride quality predicted is relatively poor. These data also show that the vertical acceleration levels at 2-5 Hz exceed the 25-min boundary by an increasing degree as the fifth wheel is moved forward.

In summary, all the data presented have shown that vehicle acceleration in response to random road roughness increases as the fifth wheel is moved forward.

Experimental Measurement of Ride Quality

The acceleration levels in the cab of a tractor-semitrailer unit have been measured experimentally for a unit traveling on an Interstate highway. These measurements have been performed by means of an instrumentation package supplied by the Transportation Systems Center of the U.S. Department of Transportation on equipment supplied and operated by the National Highway Traffic Safety Administration Research Test Center in Maryland.

Test Equipment

For the tests, a 1975 International Harvester Transtar II with a loaded Fontaine flatbed semitrailer (19 110 lbf over the kingpin and 23 160 lbf over the rear tandem) was used (see Figure 10). [The full technical specifications for this experimental vehicle are available from the authors.]

The following instrumentation was employed. Three dc accelerometers were orthogonally oriented on a rigid aluminum housing that was securely screwed to a console located between the driver's and passenger's seats, as shown in Figure 11. (It is standard procedure not to include the seat dynamics in ride-quality measurements for convenience and because not all of the driver is isolated by the seat from the cab, e.g., hands on the steering wheel and feet on the floor.) The output sensitivity of the accelerometers was adjusted to make use of their full dynamic range and then filtered in the following manner. The longitudinal direction was low-passed filtered (Butterworth) with a break frequency of 30 Hz (the analytical models have shown no useful information above 30 Hz). The vertical direction was band-passed filtered (0.03-30 Hz) to remove the dc bias introduced by gravity. The signals were then simultaneously recorded on a seven-channel portable FM magnetic tape recorder.

Figure 7. Vertical mean square acceleration spectral density at the driver's seat for three fifth-wheel positions at 55 mph.

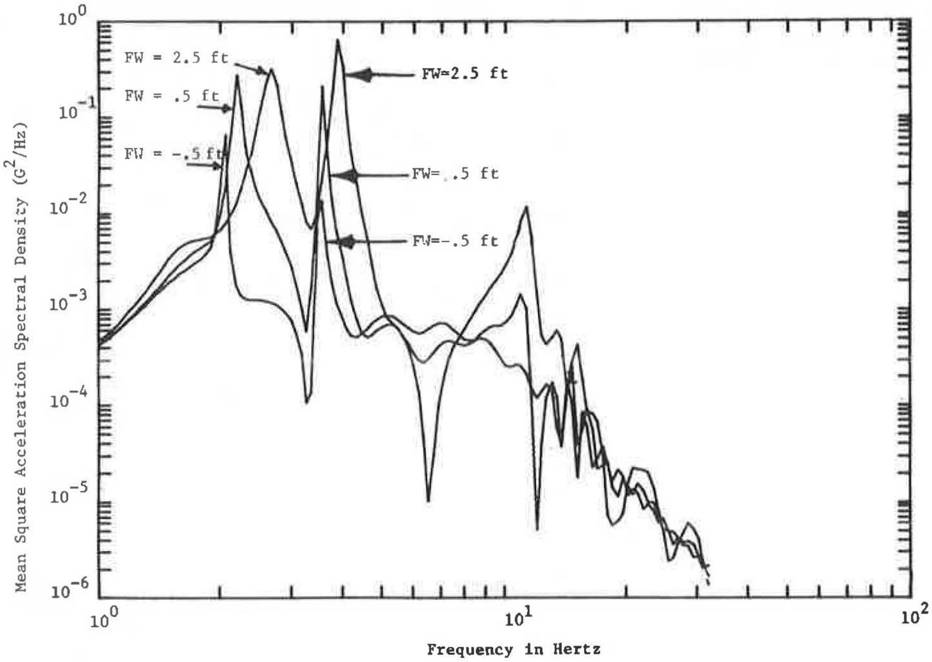
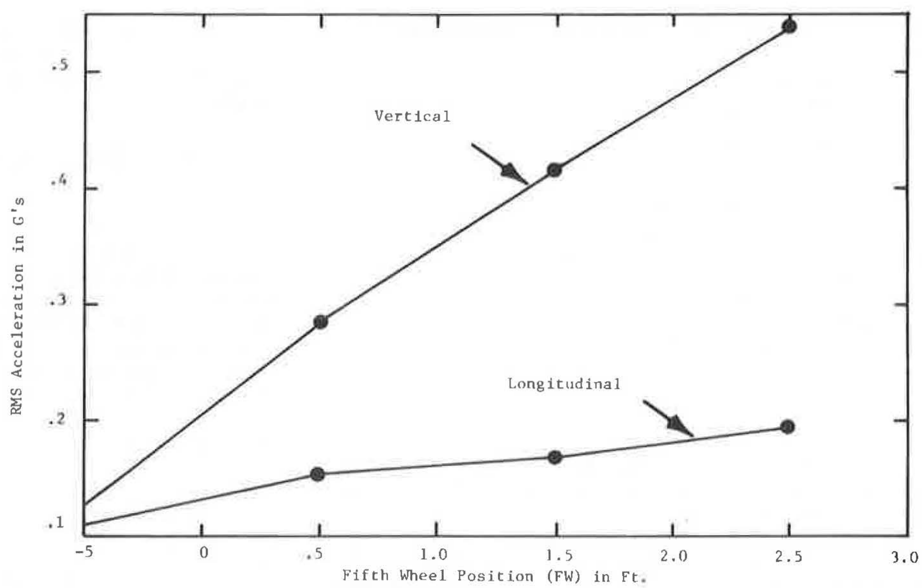


Figure 8. Fifth-wheel position versus rms acceleration at 55 mph.



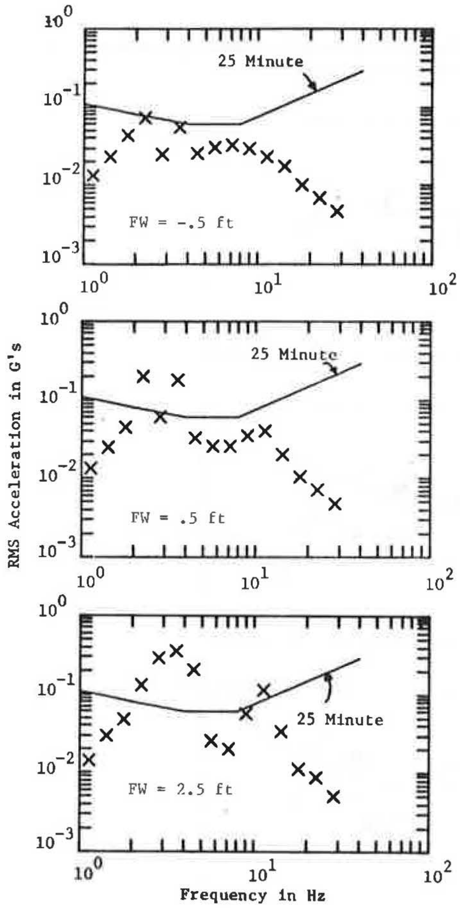
The tests were run on I-495 in Maryland between exits 28 and 37. The highway is poured concrete that has expansion joints (tar strips) every 40 ft; there are numerous overpasses. The first test made was a control test. A 5-min recording of the vehicle standing still with the engine running at a speed that corresponded to the truck traveling at 55 mph was performed to identify frequency components resulting from engine vibrations. Then six runs were made: tests at three different fifth-wheel positions (6 in behind and 3 and 15 in ahead of the tractor tandem's centerline) and at two different speeds (45 and 55 mph). All the runs at the same speed were made by traveling in the same direction while the test driver tried to maintain constant speed in the same lane. The fifth-wheel positions were changed in succession from the extreme rearward position to the extreme forward position. Approximately 20 min of data were collected on each run.

Data Analysis and Postprocessing

Each channel of data was individually processed in the following manner. The output of the tape recorder was band-passed filtered (0.06-40 Hz) to remove any dc bias and to further reduce noise that could have been introduced into the system. An analog computer was used as an amplifier to provide the signal and the proper level for the analog-digital converters. The signal was then sampled by using analog-digital converters at 102 Hz, giving 1024 points for a 10-s run. The data samples were then numerically analyzed on a digital computer by using a fast Fourier transform (FFT) Cooley-Tukey algorithm (11).

Twenty FFTs were computed from the 10-s data samples and averaged together in the frequency domain to give a consistent estimate of the mean square spectral density and to reduce the random error of this estimate to 22 percent. When 70 such 10-s samples were averaged, which reduced the random error of this estimate to 12 percent, no significant diffferences were found in the output mean square spectral density plots in comparison with the plots based on 20 sample runs. In addition, a test for stationarity was performed by comparing the mean square spectral density plots at one time in a run with those at another. No significant differences were found. The mean square spectral density plots, the total rms acceleration, and the third-octave band integration of the data were then performed on a digital computer.

Figure 9. Cab vertical ISO response at 55 mph versus 25-min reduced comfort criterion.



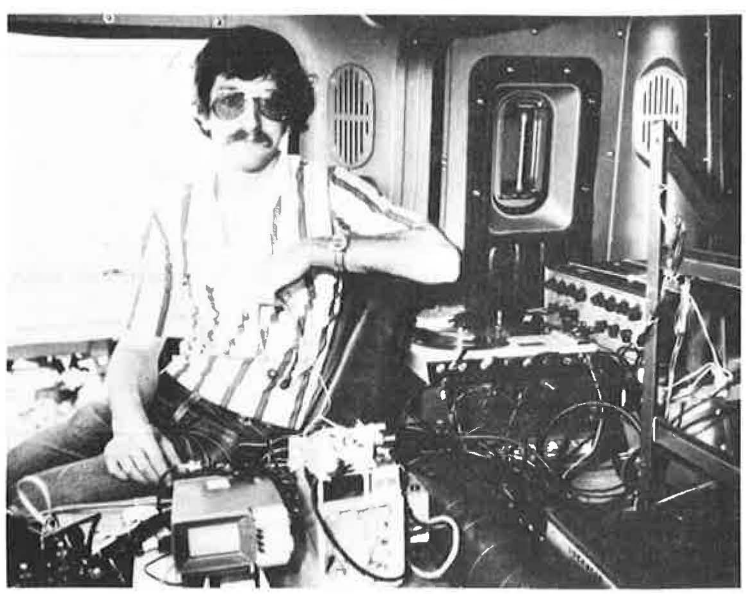
EXPERIMENTAL RESULTS

The vertical acceleration spectral densities computed from the data are shown in Figure 12 for the different fifth-wheel positions. Significant signal content in these data is associated with the 2- to 4-Hz region. As the fifth wheel is moved from 6 in behind to 15 in ahead of the tractor tandem's centerline, the dominant peak shifts upward in frequency approximately 1 Hz in the vertical direction and also increases in magnitude. The peaks at other frequencies are not strongly influenced by fifth-wheel positions. The higher-frequency peaks at approximately 15 and 30 Hz are a result of engine-induced vibrations.

The acceleration spectral density data have been processed in third-octave bands as specified by ISO and compared with the ISO 25-min reduced comfort boundary, as shown in Figure 13. In the vertical direction the criterion is violated when the fifth wheel is in the far-forward position (15 in ahead) in the 3- to 4-Hz region and is satisfied for all other fifth-wheel positions.

Figure 10. Tractor-semitrailer test vehicle.

Figure 11. Installed data acquisition and processing equipment.



The total rms accelerations in both the vertical and the longitudinal directions (at two truck speeds) are plotted against fifth-wheel position in Figure 14. In both the vertical and longitudinal directions, the worst ride is obtained with the fifth wheel in the far-forward position. In the vertical direction this increase in ride discomfort is more pronounced. The total vertical rms acceleration is higher at the lower speed because it is necessary to drive in the rough curb lane at 45 mph.

In summary, the ride quality generally degrades as the fifth wheel is moved forward. At 55 mph the total rms acceleration in the vertical direction increases by a factor of 1.5, whereas in the longitudinal direction the increase is 1.25 as the fifth wheel is moved from 0.5 ft behind to 1.25 ft forward of the axle centerline.

Figure 12. Vertical mean-square acceleration spectral densities for three fifth-wheel positions at 55 mph.

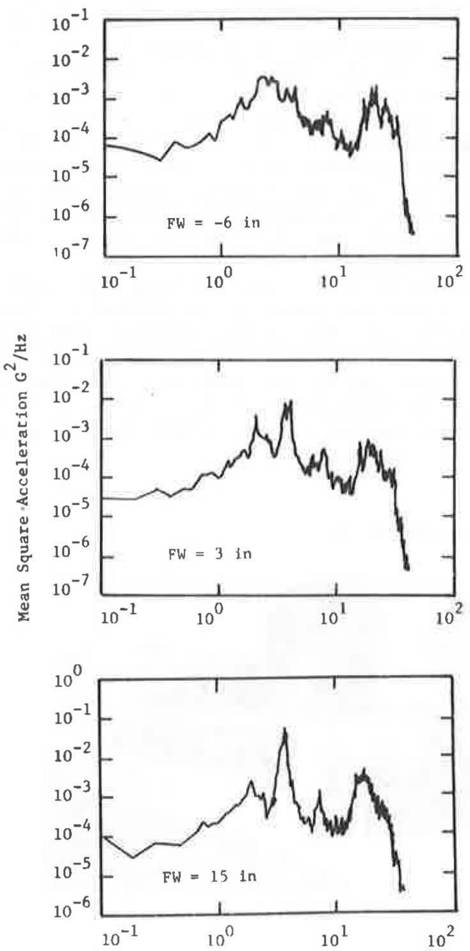


Figure 13. Vertical ISO response for three fifth-wheel positions at 55 mph versus the 25-min reduced comfort criterion.

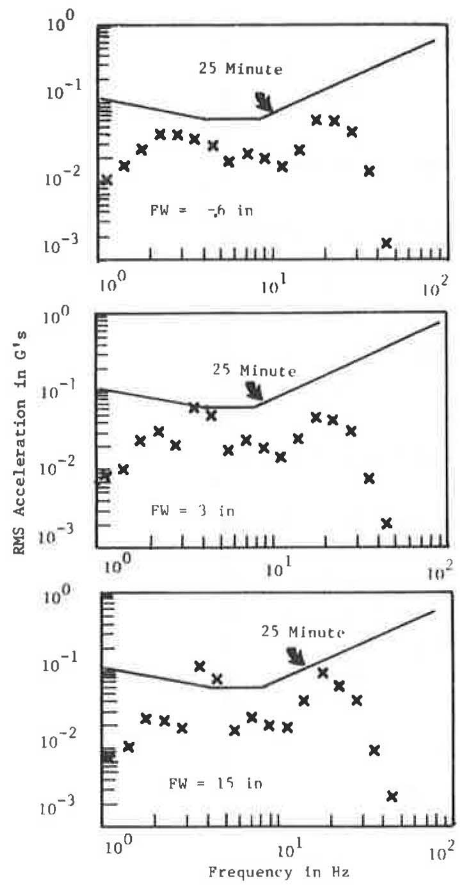
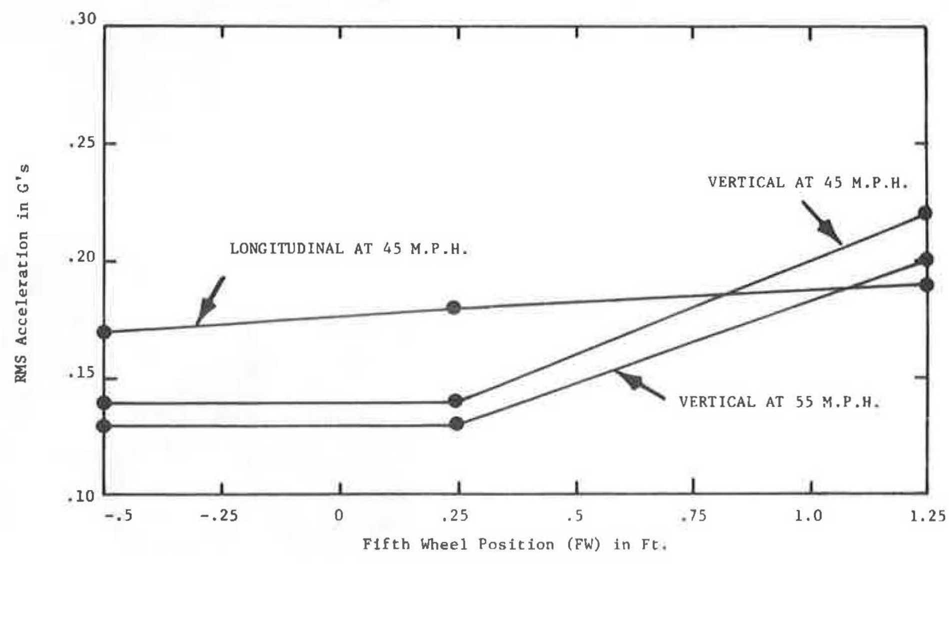


Figure 14. Fifth-wheel postion versus rms acceleration at two speeds.



SUMMARY

The objective of this paper was to determine the influence of fifth-wheel placement on ride quality. A six-degree-of-freedom linear model was formulated and the following analyses were performed:

1. Natural frequencies and mode shapes--(a) Natural frequencies associated with the tractor and semitrailer modes shift up in frequency as the fifth wheel is moved forward and (b) the amplitude of tractor pitching increases as the fifth wheel is moved forward.
2. Frequency response--The large resonant peaks at 2 and 4 Hz shift up in frequency and increase in magnitude as the fifth wheel is moved forward.
3. Random road response--(a) The total rms acceleration at a rigid point in the tractor representing the driver's location in both the vertical and longitudinal directions increases as the fifth wheel is moved forward and (b) the increase in rms accelerations results from increased output at frequencies in the 2- to 4-Hz range.

In addition to the analytical studies performed, an experimental investigation yielded the following conclusions: (a) The rms accelerations at the driver's location (not including the dynamics of the seat) in both the vertical and longitudinal direction increase as the fifth wheel is moved forward and (b) the analytical and experimental rms acceleration trends as a function of fifth-wheel placement agree closely.

In summary, the analytical and experimental data show that ride quality deteriorates as the fifth wheel is moved ahead of the tractor tandem's centerline. The increase in rms acceleration primarily results from the development of a resonant condition of out-of-phase tractor-semitrailer pitch in the 4-Hz region.

The comparison between the analytical and experimental results indicates good agreement when rms values are compared; however, it should be noted that the analytical model is valid only up to approximately 7 Hz. To model accurately the higher-frequency phenomena, features such as tractor-frame bending, cab-mount suspension, and wheel out of round would need to be included.

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Ride-Quality Models for Diverse Transportation Systems

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This research was undertaken to develop comfort (ride-quality) models for six specific vehicles and to refine an existing composite ride-quality model. The vehicles were an automated guideway transit vehicle, a short-haul intercity rail vehicle, an urban rapid rail vehicle, a luxury-type charter bus, a compact car, and a subcompact automobile. Experiments on most vehicles were conducted in two phases: model development and model validation. In both phases, physical variables were measured for a series of ride segments, and each segment was rated for comfort level by a group of paid subjects. The important determinants of comfort for most vehicles were roll, pitch, and vertical acceleration. These variables are highly intercorrelated; all load on the same principal component of the motion-correlation matrix. Two composite ride-quality models are presented; one has four variables (roll, pitch, and vertical and longitudinal acceleration) and one has two variables (vertical acceleration and roll). The two-variable comfort model is sufficient for most uses.