Simple Computer Models for Predicting Ride Quality and Pavement Loading for Heavy Trucks

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Increasing pavement damage caused by the increasing number of heavy trucks on today's highways has promoted concern about the dynamic pavement loads and the ride quality of trucks. So far, these concerns have been analyzed using only experimental studies and complex computer programs. This paper presents three possible simple truck models—a quarter-truck, a half-single-unit truck, and a half-tractor semitrailer—that can be used on personal computers to predict ride quality and pavement loading. Numerical values for the model parameters are suggested for possible standardization. Sample results are presented in the form of vertical acceleration frequency responses and root mean square vertical acceleration for ride quality and tire force frequency responses and dynamic impact factors for pavement loading. The quarter-truck model overestimated both ride quality and pavement loading when compared to the half-single-unit truck model.

In recent years, the percentage of trucks in the highway traffic stream has increased significantly—up by 30 percent on some highways. As a result of improving brake and engine technology, longer and wider trucks are being constructed to carry heavier loads. In addition to affecting cornering and braking performance, increasing truck size and weight dramatically increases dynamic pavement loading. The resulting increase in pavement roughness and wear has made ride comfort a major concern for truck drivers covering long distances.

Various aspects of truck dynamics are being examined in several current research studies (*1*). Of three types of research methods—analytical, experimental, and computer simulation—only the last two find wide application in those studies. Analytical methods are practically useless in dealing with problems of the complexity associated with mathematical models of heavy trucks. Experimental methods offer the most valuable results; however, they are usually very costly. Moreover, the experimental methods are limited by safety requirements. Probably the most successful approach has been to conduct a limited number of field tests to provide actual truck performance data to validate computer simulation programs. These computer simulation programs are then used to extrapolate the experimental results over the range of test conditions where experimentation would be too dangerous or too expensive.

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Several truck simulation programs have been developed in recent years (*2–4*). In most cases these programs, such as the Phase 4 program (*2*) developed at the University of Michigan Transportation Research Institute, are products of long-term efforts. Although relatively accurate, these programs are very complex and require detailed input and long execution times even for simple problems.

The objectives of this paper are twofold. First, relatively simple mathematical models of truck dynamics are presented. The applicability of the proposed models is limited to those problems involving two-dimensional dynamics. Examples of such problems are dynamic pavement loading and ride quality. Second, possible numerical parameters for the various models are suggested to represent typical trucks. Acceptance of standard truck models similar to those developed for passenger cars (*5*) would allow for comparison of computer sim-

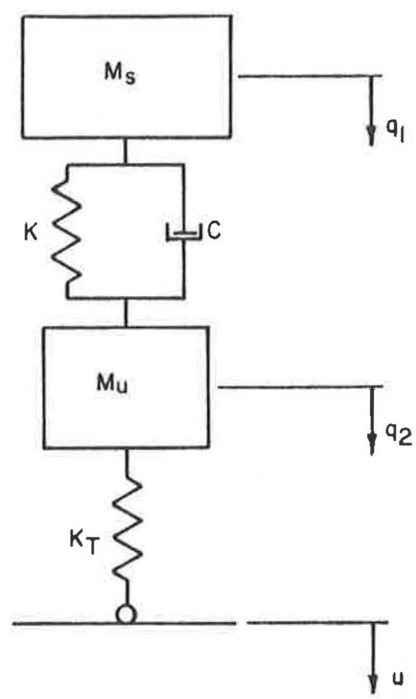

FIGURE 1 Quarter-truck model.

TABLE 1 QUARTER-TRUCK MODEL PARAMETERS

Parameters:

Symbol	Description
M_m	One quarter vehicle sprung mass
M_u	Unsprung mass corresponding to one wheel
K	Suspension spring constant
C	Suspension damping constant
K_t	Tire spring constant

Numerical Values Used:

Symbol	Single Unit Truck Front Axle		Single Unit Truck Rear Axle	
M_m	13.975	$lbsec^2/in$	22.9036	$lbsec^2/in$
M_u	1.553	$lbsec^2/in$	2.976	$lbsec^2/in$
K	1132.	lb/in	6500.	lb/in
C	15.	lbsec/in	15.	lbsec/in
K_t	4500.	lb/in	5000.	lb/in

ulation results conducted by different research groups. Sample simulation results are presented to demonstrate the use of these simple truck models and the numerical parameters selected.

MATHEMATICAL MODELS

To develop three mathematical truck models, the following was assumed:

- Constant vehicle velocity,
- No vehicle body or axle roll,
- Rigid vehicle bodies,
- Linear suspension and tire characteristics,
- Point tire to road contact, and
- Small pitch angles.

Quarter-Truck Model

The first model, a quarter-truck, is shown in Figure 1. The parameters are defined in Table 1; the state equations are presented in Table 2. The parameter values for this simple model can be derived in a variety of ways. One possible approach is to use front axle parameters for ride comfort studies and rear axle parameters for pavement loading studies. The quarter-truck model represents only the heave mode of the vehicle.

Transfer functions, listed in Table 3, can be developed easily in the frequency domain for both the tire force and the

TABLE 2 QUARTER-TRUCK STATE EQUATIONS

$$\dot{q}_1 = q_3 \qquad \dot{q}_2 = q_4$$

$$\dot{q}_3 = \{-q_1(K)+q_2(K)-q_3(C)+q_4(C)\}/M_s$$

$$\dot{q}_4 = \{q_1(K)-q_2(K_t+K)+q_3(C)-q_4(C)+u(K_t)\}/M_u$$

where:

q_1 = Vertical displacement of sprung mass

q_2 = Vertical displacement of unsprung mass

q_3 = Vertical velocity of sprung mass

q_4 = Vertical velocity of unsprung mass

u = Vertical displacement of road under wheel

vertical acceleration of the sprung mass. Considerable time can be saved by using the transfer functions instead of the simulation routines to calculate the frequency responses.

Half-Single-Unit Truck Model

The second model, a half-single-unit truck with single axles, is shown in Figure 2. The parameters are defined in Table 4; the state equations are presented in Table 5. This model includes both front and rear axles, resulting in both a pitch and a heave mode of the vehicle body being incorporated in the model.

Although this model is considerably more complicated than the quarter-truck model, the transfer function method could be used to determine specific frequency responses. Computer simulations can be used to determine frequency responses for any combination of the state variables and inputs. In this study, computer simulations are used to determine the half-single-unit truck frequency responses.

Half-Tractor Semitrailer Model

The third model, a half-tractor semitrailer, is shown in Figure 3. The parameters are defined in Table 6; the state equations are presented in Table 7. This model expands the half-single-unit truck model to include double axles and a semitrailer. The fifth wheel connecting the tractor to the semitrailer is modeled with a stiff spring and damper. This makes the fifth wheel appear nearly rigid without complicating the state equations. As with the half-single-unit truck, the pitch angles have been assumed small to make the mathematical model linear.

The complexity of this model makes developing transfer functions in the frequency domain a formidable task. Computer simulations are used to determine all frequency responses for this model.

NUMERICAL PARAMETER VALUES

Because truck sizes and loads vary greatly, it is much more difficult to select representative parameter values for trucks than for passenger cars. The numerical data used in this paper represent a fully loaded, single-unit, single-rear-axle truck and a fully loaded, 18-wheel tractor semitrailer with the payload evenly distributed (*6*). These values could be used with half-truck models to study typical loaded trucks.

Because the load often is unevenly distributed, selecting parameter values for the quarter-truck model is even more difficult. Two possible approaches are presented in this paper—one for ride comfort and one for pavement loading. In both cases the numerical values are based on the single-unit-truck parameter values. The first approach uses the front axle suspension parameters and half of the actual unsprung mass sup-

TABLE 3 QUARTER-TRUCK TRANSFER FUNCTIONS

Transfer Functions:

Vertical acceleration of sprung mass (a_1):

$$\frac{a_1(s)}{u(s)} = \frac{K_t s^2(C_s + K)}{(M_u s^2+Cs+k+K_t)\,(M_s s^2+Cs+K+K_t) - (Cs+K)^2}$$

Vertical tire force (F_t):

$$\frac{F_t(s)}{u(s)} = \frac{K_t^2\,(M_s s^2+C_s+K)}{(M_s s^2+Cs+K)\,(M_u s^2+Cs+k+K_t) - (Cs+k)^2} - K_t$$

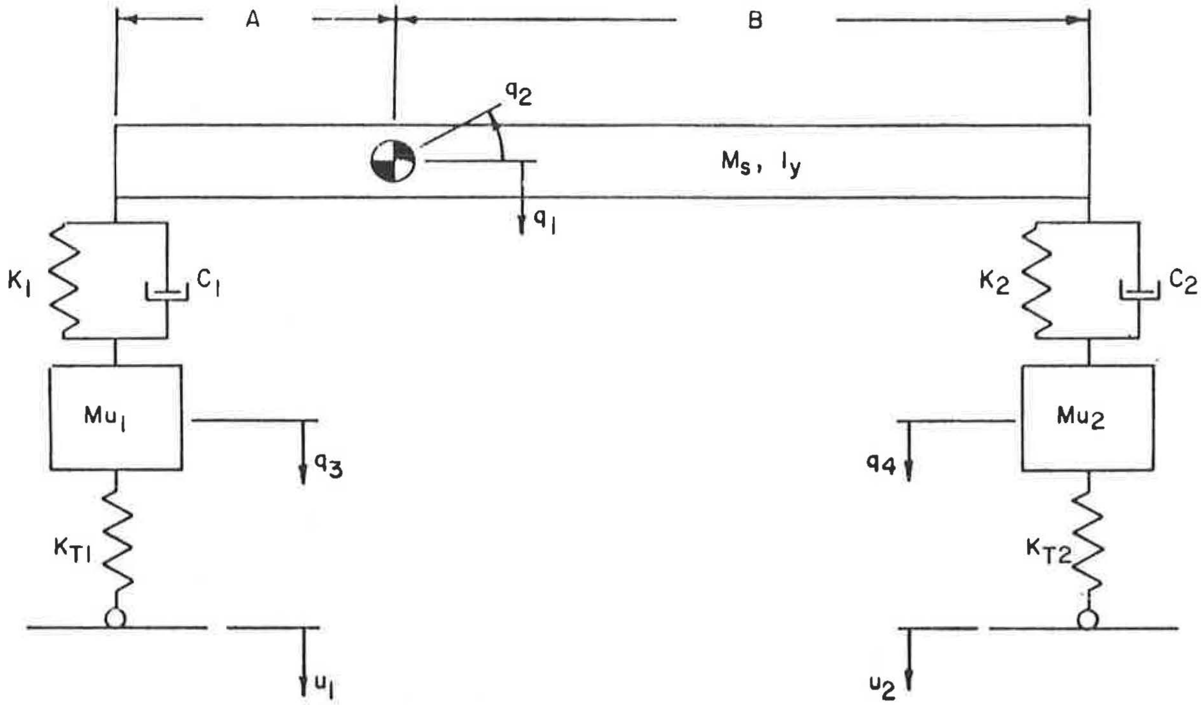

FIGURE 2 Half-single-unit truck model.

TABLE 4 HALF-SINGLE-UNIT TRUCK MODEL PARAMETERS

Parameters:

Symbol	Description	Numerical Value
M_s	One half vehicle sprung mass	36.8789 lbsec^2/in
I_y	One half sprung mass pitch moment	410876.4 lbsec^{2}in
M_{u1}	One half front axle unsprung mass	1.5528 lbsec^2/in
M_{u2}	One half rear axle unsprung mass	2.9762 lbsec^{2}in
K_1	Front suspension spring constant	1132. lb/in
K_2	Rear suspension spring constant	6500. lb/in
C_1	Front suspension damping constant	15. lbsec/in
C_2	Rear suspension damping constant	15. lbsec/in
K_{t1}	Front tire spring constant	4500. lb/in
K_{t2}	Rear tire spring constant	5000. lb/in
A	Horizontal distance from front axle to sprung mass center of gravity	149.052 in
B	Horizontal distance from rear axle to sprung mass center of gravity	90.948 in

TABLE 5 HALF-SINGLE-UNIT TRUCK STATE EQUATIONS

$\dot{q}_1 = q_5 \qquad \dot{q}_2 = q_6 \qquad \dot{q}_3 = q_7 \qquad \dot{q}_4 = q_8$

$$\dot{q}_5 = \{-q_1(K_1+K_2)+q_2(K_2B-K_1A)+q_3(K_1)+q_4(K_2)-q_5(C_1+C_2)+q_6(C_2B-C_1A) +q_7(C_1)+q_8(C_2)\}/M_s$$

$$\dot{q}_6 = \{-q_1(K_1A+K_2B)+q_2(K_2B^2-K_1A^2)+q_3(K_1A)+q_4(K_2B)-q_5(C_1A+C_2B) +q_6(C_2B^2-C_1A^2)+q_7(C_1A)+q_8(C_2B)\}/I_y$$

$$\dot{q}_7 = \{q_1(K_1)+q_2(K_1A)-q_3(K_1+K_{t1})+q_5(C_1)+q_6(C_1A)-q_7(C_1)+u_1(K_{t1})\}/M_{u1}$$

$$\dot{q}_8 = \{q_1(K_2)-q_2(K_2B)-q_4(K_2+K_{t2})+q_5(C_2)-q_6(C_2B)-q_8(C_2)+u_2(K_{t2})\}/M_{u2}$$

where:

- q_1 = Vertical displacement of sprung mass
- q_2 = Pitch angular displacement of sprung mass
- q_3 = Vertical displacement of front unsprung mass
- q_4 = Vertical displacement of rear unsprung mass
- q_5 = Vertical velocity of sprung mass
- q_6 = Pitch angular velocity of sprung mass
- q_7 = Vertical velocity of front unsprung mass
- q_8 = Vertical velocity of rear unsprung mass
- u_1 = Vertical displacement of road under front wheel
- u_2 = Vertical displacement of road under rear wheel

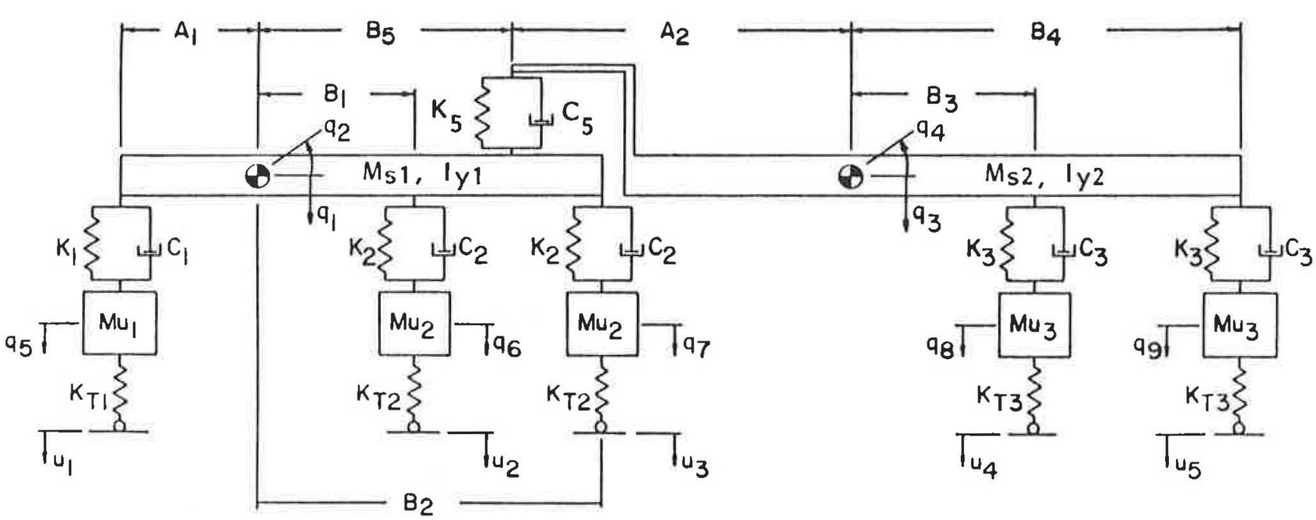

FIGURE 3 Half-tractor trailer model.

TABLE 6 HALF-TRACTOR SEMITRAILER MODEL PARAMETERS

Parameters:

Symbol	Description	Numerical Value	
M_{m1}	One half tractor sprung mass	10.401	$lbsec^2/in$
I_{y1}	One half tractor sprung mass pitch moment	200490.	$lbsec^2in$
M_{u1}	One half front axle unsprung mass	1.5528	$lbsec^2/in$
M_{u2}	One half tractor rear tandem axle unsprung mass (per axle)	2.9762	$lbsec^2in$
K_1	Tractor front suspension spring constant	1132.	lb/in
K_2	Tractor rear suspension spring constant	7200.	lb/in
C_1	Tractor front suspension damping constant	15.	lbsec/in
C_2	Tractor rear suspension damping constant	15.	lbsec/in
K_{t1}	Tractor front tire spring constant	4500.	lb/in
K_{t2}	Tractor rear tire spring constant	9000.	lb/in
A_1	Horizontal distance from front axle to tractor sprung mass center of gravity	60.108	in
B_1	Horizontal distance from leading rear axle to tractor sprung mass center of gravity	126.342	in
B_2	Horizontal distance from fifth wheel to tractor sprung mass center of gravity	177.442	in
B_5	Horizontal distance from fifth wheel to tractor sprung mass center of gravity	118.662	in
M_{s2}	One half trailer sprung mass	81.731	$lbsec^2/in$
I_{y2}	One half trailer spung mass pitch moment	90575.5	$lbsec^2in$
M_{u3}	One half trailer tandem axle unsprung mass (per axle)	1.941	$lbsec^2/in$
K_3	Trailer suspension spring constant	7500.	lb/in
C_3	Trailer suspension damping constant	15	lbsec/in
K_{t3}	Trailer tire spring constant	10000	lb/in
A_2	Horizontal distance from fifth wheel to trailer sprung mass center of gravity	235.581	in
B_3	Horizontal distance from leading rear axle to trailer sprung mass center of gravity	220.419	in
B_4	Horizontal distance from trailing rear axle to trailer sprung mass center of gravity	268.4	in
C_5	Fifth wheel damping constant	1000.	lbsec/in
K_5	Fifth wheel spring constant	100000.	lb/in

TABLE 7 HALF-TRACTOR SEMITRAILER STATE EQUATIONS

$\dot{q}_1 = q_{10}$ $\dot{q}_2 = q_{11}$ $\dot{q}_3 = q_{12}$ $\dot{q}_4 = q_{13}$ $\dot{q}_5 = q_{14}$

$\dot{q}_6 = q_{15}$ $\dot{q}_7 = q_{16}$ $\dot{q}_8 = q_{17}$ $\dot{q}_9 = q_{18}$

$$\dot{q}_{10} = \{-q_1(K_1+2K_2+K_5)+q_2(K_5B_5-K_1A_1+K_2(B_1+B_2))+q_3(K_5)+q_4(K_5A_2)$$
$$+q_5(K_1)+q_6(K_2)+q_7(K_2)-q_{10}(C_1+2C_2+C_5)+q_{11}(C_5B_5-C_1A_1+C_2(B_1+B_2))$$
$$+q_{12}(C_5)+q_{13}(C_5A_2)+q_{14}(C_1)+q_{15}(C_2)+q_{16}(C_2)\}/M_{s1}$$

$$\dot{q}_{11} = -\{q_1(K_1A_1-K_2(B_1+B_2)-K_5B_5)+q_2(K_1A_1^2+K_2(B_1^2+B_2^2)+K_5B_5^2)$$
$$+q_3(K_5B_5)+q_4(K_5B_5A_2)-q_5(K_1A_1)+q_6(K_2B_1)-q_7(K_2B_2)$$
$$+q_{10}(C_1A_1-C_2(B_1+B_2)-C_5B_5)+q_{11}(A_1C_1+C_2(B_1^2+B_2^2)+C_5B_5^2)$$
$$+q_{12}(C_5B_5)+q_{13}(C_5B_5A_2)-q_{14}(C_1A_1)+q_{15}(C_2B_1)+q_{16}(C_2B_2)\}/I_{y1}$$

$$\dot{q}_{12} = \{q_1(K_5)-q_2(K_5B_5)-q_3(K_5+2K_3)+q_4(K_3(B_3+B_4)-K_5A_5)+q_8(K_3)$$
$$+q_9(K_3)+q_{10}(C_5)-q_{11}(C_5B_5)-q_{12}(C_5+2C_3)+q_{13}(C_3(B_3+B_4)-C_5A_2)$$
$$+q_{17}(C_3)+q_{18}(C_3)\}/M_{s2}$$

$$\dot{q}_{13} = -\{-q_1(K_5A_2)+q_2(K_5A_2B_5)+q_3(K_5A_2-K_3(B_3+B_4))+q_4(K_5A_2^2$$
$$+K_3(B_3^2+B_4^2))+q_8(K_3B_3)+q_9(K_3B_4)-q_{10}(C_5A_2)+q_{11}(C_5A_2B_5)+q_{12}(C_5A_2$$
$$-C_3(B_3+B_4))+q_{13}(C_5A_2^2+C_3(B_3^2+B_4^2))+q_{17}(C_3B_3)+q_{18}(C_3B_4)\}/I_{y2}$$

$$\dot{q}_{14} = \{q_1K_1+q_2(K_1A_1)-q_5(K_1+K_{t1})+q_{10}C_1+q_{11}(C_1A_1)-q_{14}C_1+u_1K_{t1}\}/M_{u1}$$

$$\dot{q}_{15} = \{q_1K_2-q_2(K_2B_1)-q_6(K_2+K_{t2})+q_{10}C_2-q_{11}(C_2B_1)-q_{15}C_2+u_2K_{t2}\}/M_{u2}$$

$$\dot{q}_{16} = \{q_1K_2-q_2(K_2B_2)-q_7(K_2+K_{t2})+q_{10}C_2-q_{11}(C_2B_2)-q_{16}C_2+u_3K_{t2}\}/M_{u2}$$

$$\dot{q}_{17} = \{q_3K_3-q_4(K_3B_3)-q_8(K_3+K_{t3})+q_{12}C_3-q_{13}(C_3B_3)-q_{17}C_3+u_4K_{t3}\}/M_{u3}$$

where:

q_1 = Vertical displacement of tractor sprung mass

q_2 = Pitch angular displacement of tractor sprung mass

q_3 = Vertical displacement of trailer sprung mass

q_4 = Pitch angular displacement of trailer sprung mass

q_5 = Vertical displacement of tractor front unsprung mass

q_6 = Vertical displacement of tractor leading tandem axle

q_7 = Vertical displacement of tractor trailing tandem axle

q_8 = Vertical displacement of trailer leading tandem axle

(*continued on next page*)

TABLE 7 (*continued*)

q_9 - Vertical displacement of trailer trailing tandem axle

q_{10} - Vertical velocity of tractor sprung mass

q_{11} - Pitch angular velocity of tractor sprung mass

q_{12} - Vertical velocity of trailer sprung mass

q_{13} - Pitch angular velocity of trailer sprung mass

q_{14} - Vertical velocity of tractor front unsprung mass

q_{15} - Vertical velocity of tractor leading tandem axle

q_{16} - Vertical velocity of tractor trailing tandem axle

q_{17} - Vertical velocity of trailer leading tandem axle

q_{18} - Vertical velocity of trailer trailer tandem axle

u_1 - Vertical road displacement under tractor front wheel

u_2 - Vertical road displacement under tractor leading rear wheel

u_3 - Vertical road displacement under tractor trailing rear wheel

u_4 - Vertical road displacement under trailer leading wheel

u_5 - Vertical road displacement under trailer trailing wheel

ported by the front axle. The second approach uses the rear axle suspension parameters and half of the actual unsprung mass supported by the rear axle.

COMPUTER SIMULATIONS

A Fortran simulation routine was written for an IBM XT or compatible personal computer. To perform the different tasks involved in digital simulation, the program was divided into several subroutines. An integration subroutine performed the numerical integration of the state equations. For this study a constant time step, fourth-order, Runga-Kutta algorithm was used (*7*). State equation subroutines were created for each model so that the desired model could be selected when the program was compiled. An input subroutine defined the different road profiles. All subroutines were controlled by a main program that allowed the user to specify the simulation start time, end time, output interval, and time step.

As with any digital computer simulation, the integration time step must be selected carefully. A large time step can cause the results to be inaccurate and often unstable. A small time step causes the simulation program to use excessive computer time. The time step selection involves a compromise between speed and accuracy. The time step also depends on the vehicle parameters and the type of input used. Considerable care must be taken when selecting the time step for a fixed time step integration algorithm. A variable time step can achieve good results with a minimum of computer time.

APPLICATIONS

The primary application of these three truck models is to predict both ride comfort and dynamic pavement loading. Ride comfort is often determined from the vertical acceleration of the sprung mass. Using the models, acceleration frequency response and root mean square (RMS) acceleration can be calculated. Tire force frequency responses and dynamic impact factors (DIFs) can be determined to predict pavement loading.

The truck models were tested using two types of road profiles. A sinusoidal road profile was used to determine frequency responses. Actual road profiles were used to calculate RMS acceleration and DIF.

The sinusoidal road profile used in this study is defined by

$$U_i(t) = (.1 \text{ inch}) \sin (2 \eta f (t - t_{di})) \quad (1)$$

where

U_i = road elevation under wheel i (in.),
f = frequency (Hz),
t = time (sec), and
t_{di} = time delay between axles (sec).

Simulations were run using frequencies between 0 and 25 Hz and a vehicle velocity of 60 ft/sec. The resulting steady-state amplitudes were plotted as a function of frequency to obtain a frequency response plot. Transfer functions were used for the quarter-truck model.

The road profile used in this study is shown in Figure 4.

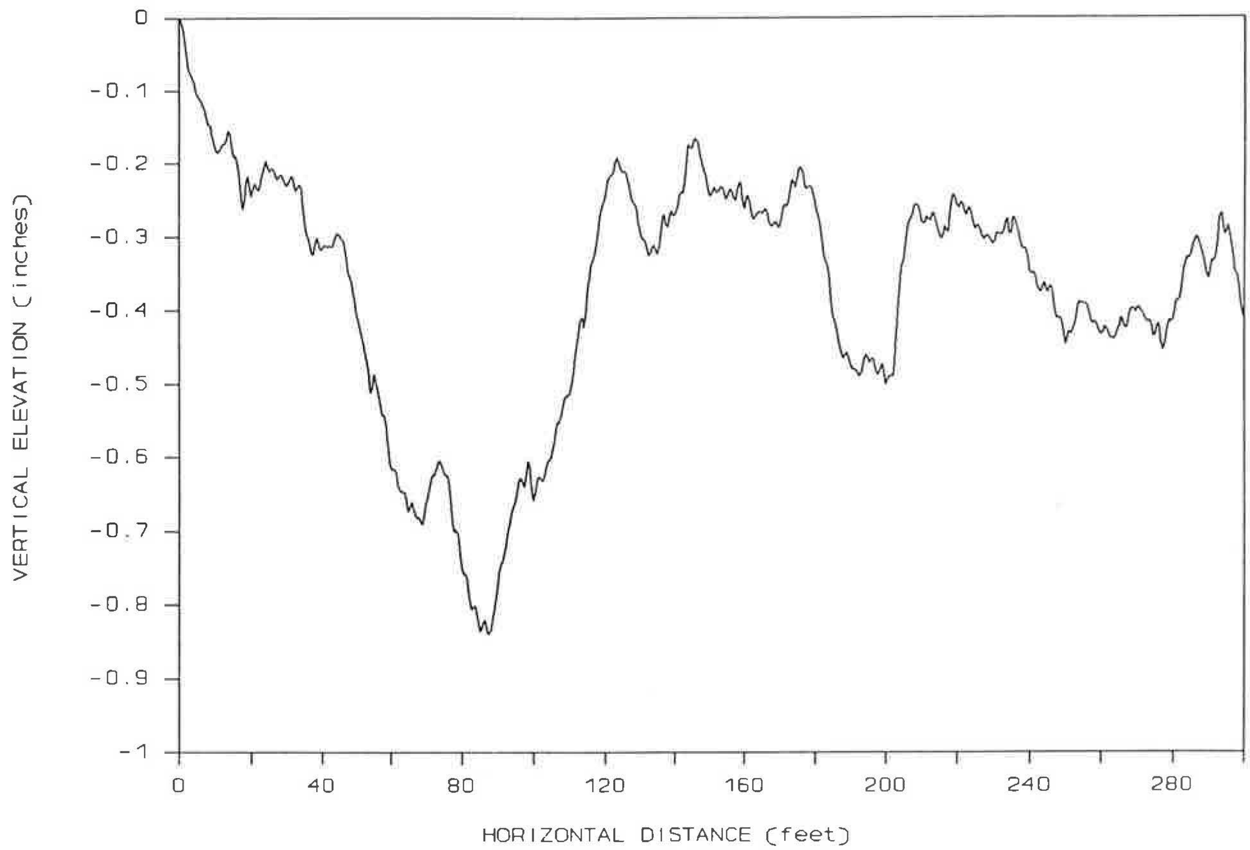

FIGURE 4 Road profile.

This profile is from a medium-roughness road having a quarter-car roughness index of 105 in./mile. The RMS acceleration is calculated from Equation 2. The DIF is calculated from Equation 3.

$$\mathrm{RMS} = \left[\left(\frac{1}{N}\sum_{i=1}^{N} a_i^2\right)\right]^{1/2} \tag{2}$$

where

RMS = root mean square acceleration,
N = total number of data points, and
a_i = acceleration at ith time step.

$$DIF = \left(\{[\sum_{i=1}^{N} (F_i - F)^2]/[(N - 1) * F^2]\}\right)^{1/2} \tag{3}$$

where

DIF = dynamic impact factor,
F_i = tire force at ith time step,
N = total number of data points, and
F = mean tire force.

RESULTS

Ride Comfort

The sprung mass, vertical acceleration frequency responses for each model are shown in Figures 5 through 7. Comparison of Figures 5 and 6 shows the vertical acceleration frequency response of the quarter-truck model using the front axle parameters to be similar to frequency response of the half-single-unit truck model. The resonant peaks predicted by the quarter-truck model occur at similar frequencies but have different amplitudes than those of the half-single-unit truck model.

The RMS vertical accelerations for each model, calculated using the actual road profile, came to 13.35 for the quarter-truck (front axle); 9.01, half-single-unit truck; and 22.36, half-tractor semitrailer. The lower the RMS, the smoother the ride. As might be expected from the frequency responses, the quarter-truck model predicts a higher RMS acceleration.

Pavement Loading

The vertical tire force frequency responses for each axle of each model are shown in Figures 8 through 11. The quarter-truck model using only rear-axle parameters predicts resonant peaks near 2 and 10 Hz. The half-single-unit truck shows similar resonant peaks, but the 2-Hz peak has a larger amplitude than that of the quarter-truck model.

Table 8 lists the DIF for each model determined from the actual road profile. The lower the DIF, the less pavement loading. The quarter-truck model prediction is considerably higher than the DIF predicted by the half-single-unit truck model. Thus, the simpler model overestimates pavement

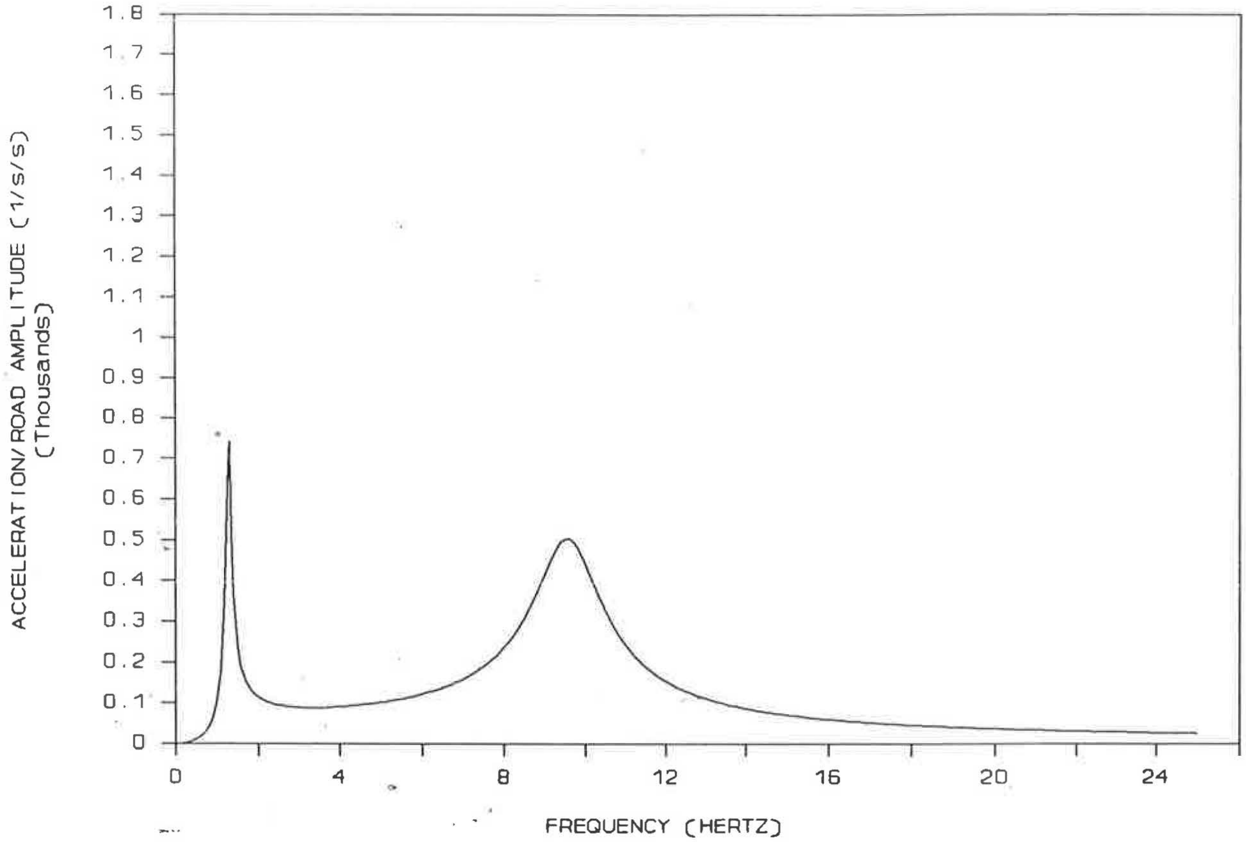

FIGURE 5 Quarter-truck model sprung mass vertical acceleration frequency response using single-unit truck front axle parameters.

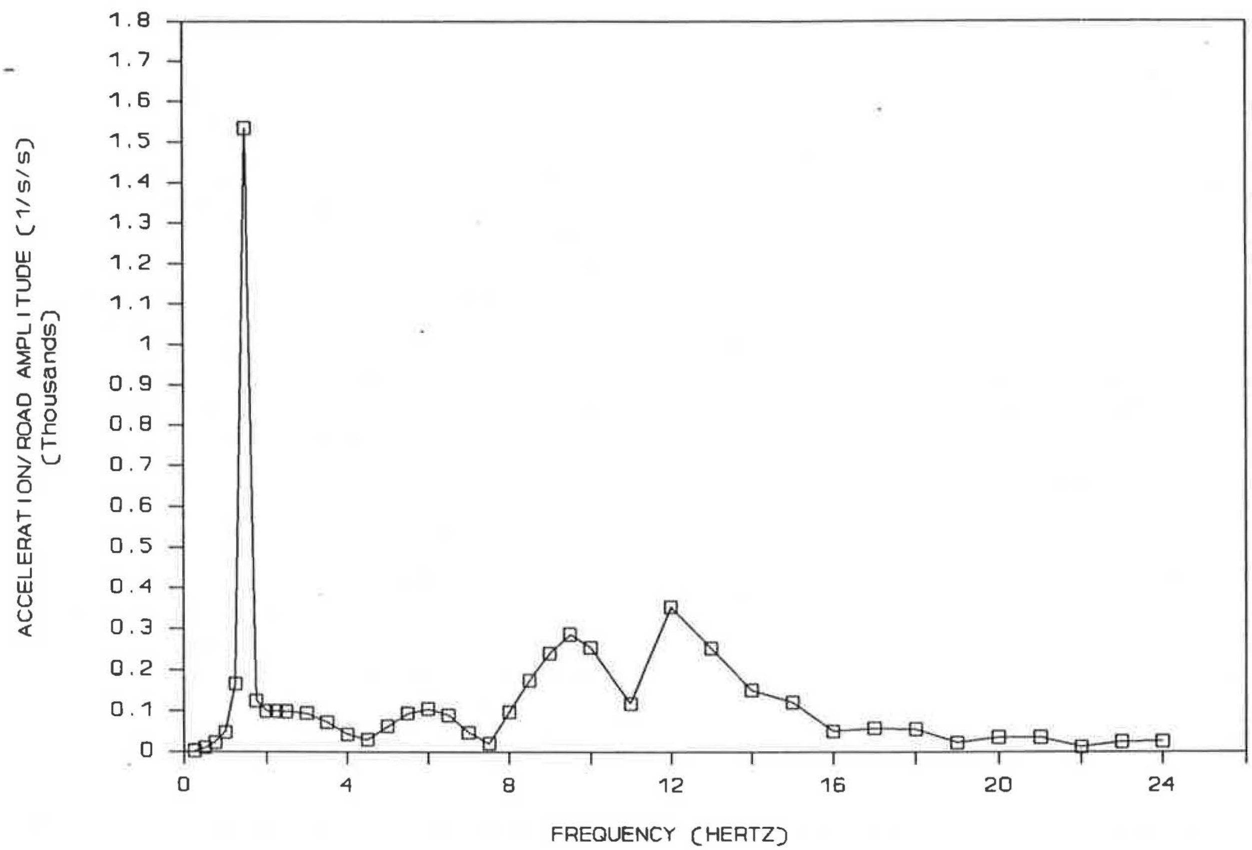

FIGURE 6 Half-single-unit truck model sprung mass vertical acceleration frequency response.

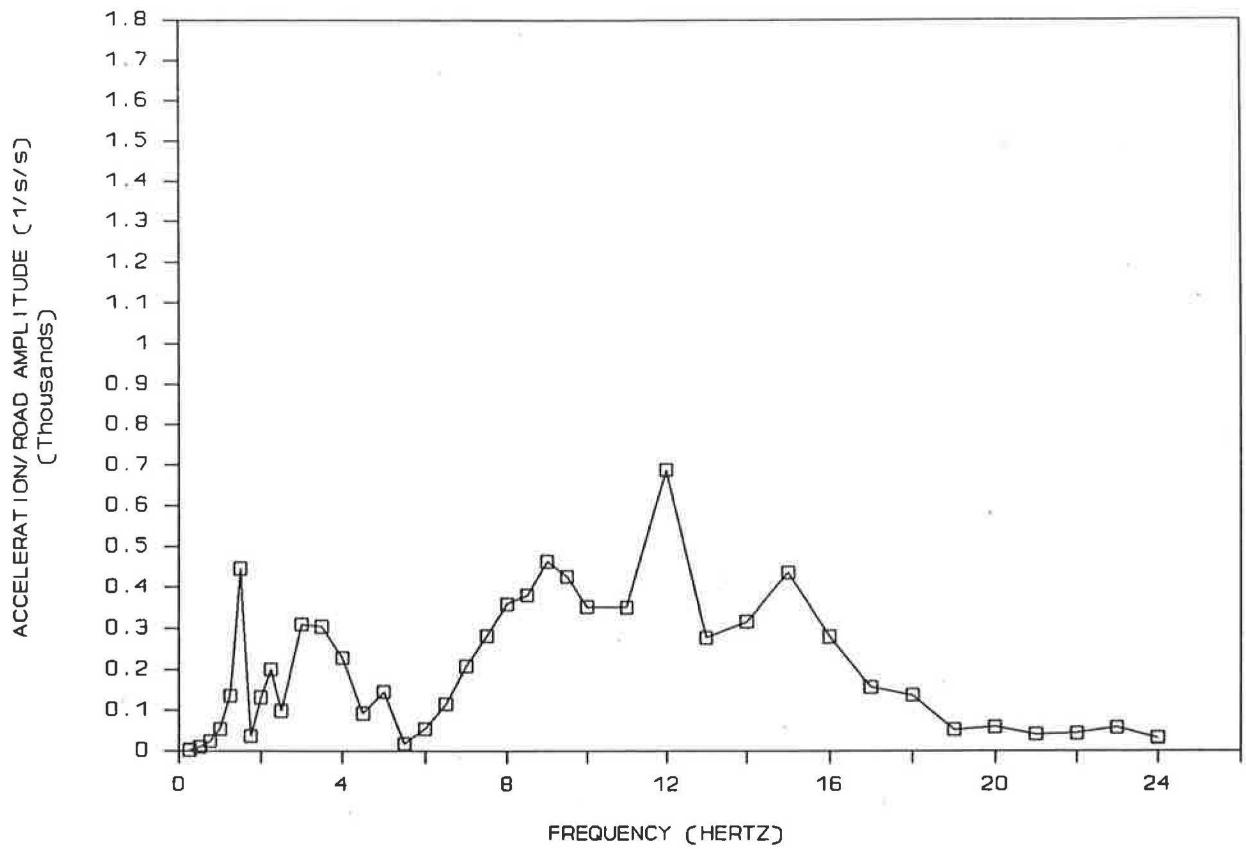

FIGURE 7 Half-tractor semitrailer model tractor sprung mass vertical acceleration frequency response.

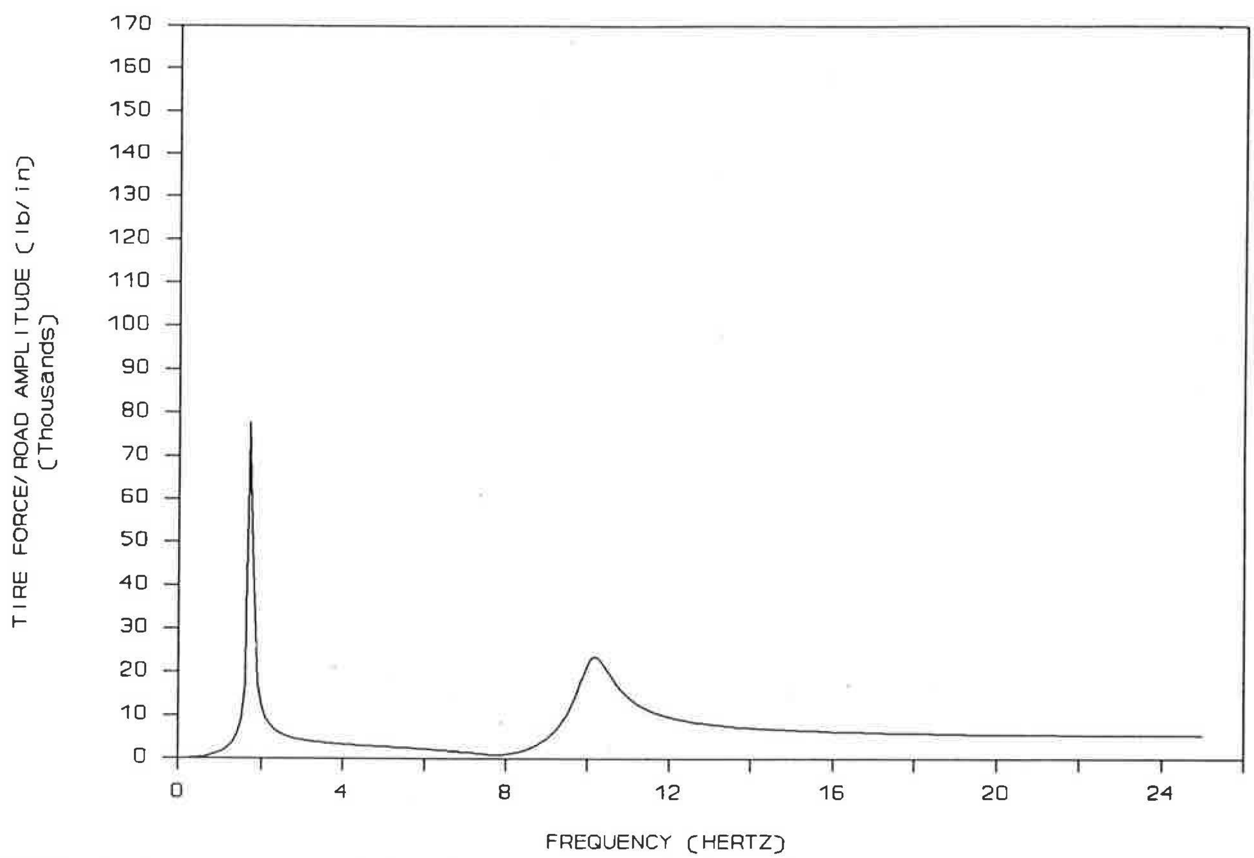

FIGURE 8 Quarter-truck model tire force frequency response using single-unit rear axle parameters.

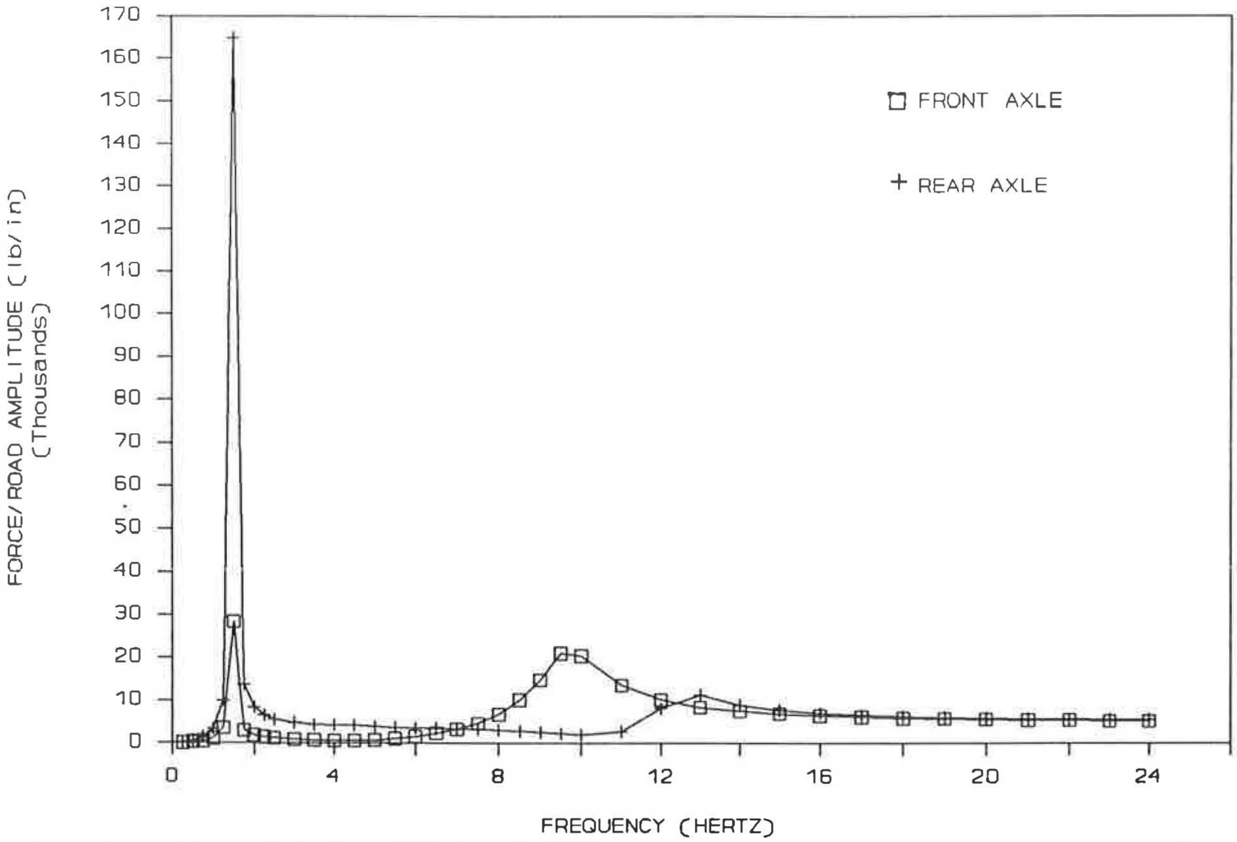

FIGURE 9 Half-single-unit truck model tire force frequency response.

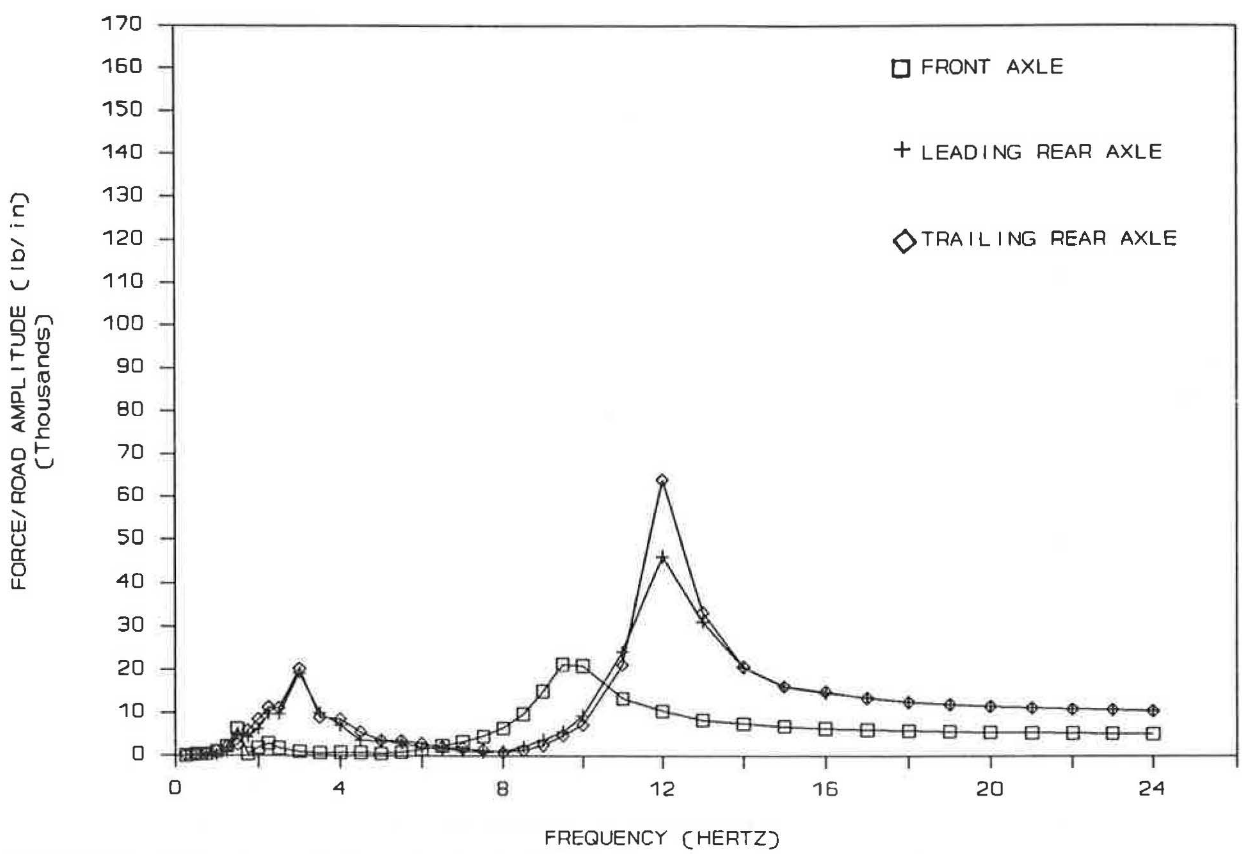

FIGURE 10 Half-tractor semitrailer model tractor tire force frequency responses.

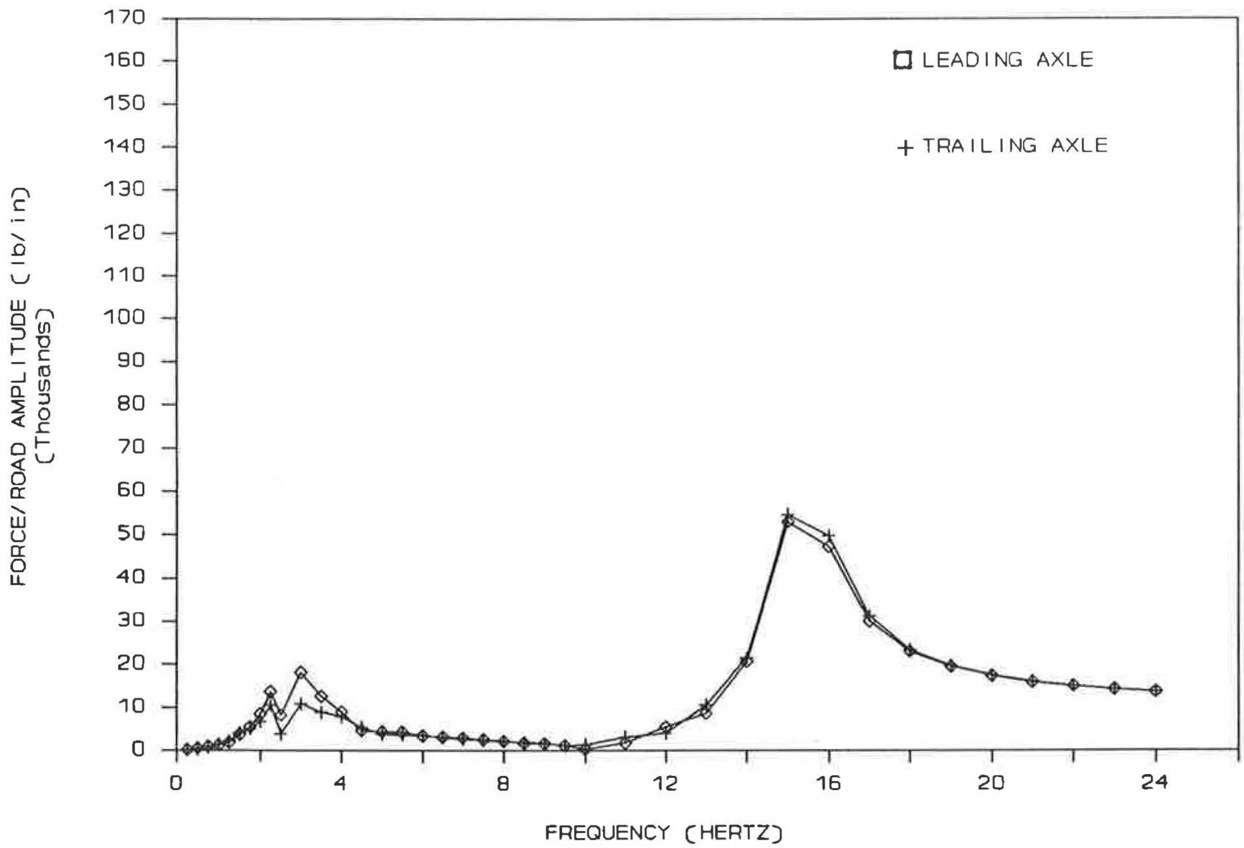

FIGURE 11 Half-tractor semitrailer model trailer tire force frequency responses.

TABLE 8 DYNAMIC IMPACT FACTORS FOR ACTUAL ROAD PROFILE

Axle Number	Quarter-Truck (Rear Axle)	Half-Single Unit Truck	Half-Tractor Semitrailer
1	.142	.030	.029
2	---	.071	.042
3	---	---	.039
4	---	---	.036
5	---	---	.035

--- Not applicable

loading, and a more complex half-truck model has to be used if more accurate results are needed.

CONCLUSIONS

Large complicated simulation programs should not be necessary for most studies concerning the vertical dynamics of heavy trucks. Simple two-dimensional truck models can be used with personal computers to predict ride quality and pavement loading. The quarter-truck model can be used as an initial estimate by selecting the parameters properly. For more accurate results that include pitching motion, the half-single-unit truck and the half-tractor semitrailer models can be used. With standardized parameter values, uniform simulations could be performed by different research groups to allow for comparison of the results of different studies.

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