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TOFC Lading Response Analyses for Several Track Profile and Hunting Conditions

The MITRE Corporation
McLean, Virginia

APRIL 1980

TECHNICAL REPORT

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George Kachadourian

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Metrek Division
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16. Abstract
The computer program FRATE is a non-linear, time domain digital computer program developed under Federal Railroad Administration sponsorship for the purpose of studying freight car response dynamics. The trailer on flatcar (TOFC) simulation contained in FRATE was expanded, for the purposes of the analyses of this report, to include a compliant lading representation. The compliant lading consisted of two spring mounted masses in each trailer with vertical, lateral and roll degrees of freedom. Analyses were performed to obtain the response of the TOFC vehicle and compliant lading to several track profiles and body hunting conditions. The analysis results characterize the response of a standard TOFC configuration to typical service conditions. Undesirable response conditions are noted and recommendations are made for improvements.

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Freight Car Dynamic Response, Lading, TOFC, Rock and Roll, Hunting Response

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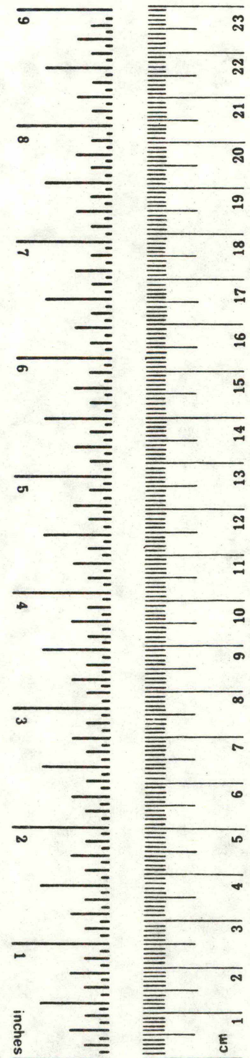
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

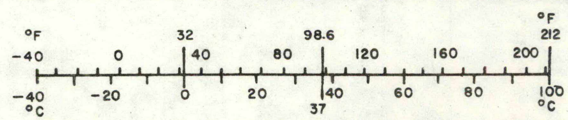
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
	MASS (weight)			
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
	VOLUME			
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



ABSTRACT

The computer program FRATE is a non-linear, time domain digital computer program developed under Federal Railroad Administration sponsorship for the purpose of studying freight car response dynamics. The trailer on flatcar (TOFC) simulation contained in FRATE was expanded, for the purposes of the analyses of this report, to include a compliant lading representation. The compliant lading consisted of two spring mounted masses in each trailer with vertical, lateral and roll degrees of freedom. Analyses were performed to obtain the response of the TOFC vehicle and compliant lading to several track profiles and body hunting conditions. The analysis results characterize the response of a standard TOFC configuration to typical service conditions. Undesirable response conditions are noted and recommendations are made for improvements.

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1.0 INTRODUCTION

The computer program FRATE has been developed for the purpose of studying the dynamic response characteristics of freight cars. The general objective of the studies is to define the response environments of and in the freight car and then to show how reduction of the response environments can be achieved. This report describes an initial phase of analyses performed with FRATE modified to incorporate compliant lading in each trailer. Three service conditions are simulated in the analyses: (1) the periodic cross level irregularities characteristic of staggered joint bolted rail, (2) vertical track irregularities caused by hard spots in the road such as with crossings, bridge encounters, culverts and switch blocks, and (3) simulated hunting conditions. The analyses were performed with two objectives. One was to demonstrate an application of FRATE including a lading model. The second objective was to quantify the responses of a nominal TOFC configuration to typical and extreme service conditions.

The need for the analysis capabilities provided by FRATE is based on the assumption that freight car motion causes wear on the car, the track and the road bed, causes damage to commodities carried and can result in derailment. There are two factors supportive of the analytic studies which provide justification for their performance. The first is that the toal annual costs for maintenance of rolling stock, track and roadbed and for lading damage settlements has reached the 10 billion dollar level (1977 statistics). Improvement in ride quality will reduce damage to lading and reduce wear on both the freight car and track roadbed. Because of the large numbers involved, a small improvement in ride quality can save a large amount of money. The second point is that innovations are continually being tried and incorporated into freight cars as part of the regular program of maintenance and replacement. Analytical studies which aid in the development of these innovations can be done more quickly and at less cost than development testing saving time and money and resulting in a better final product.

2.0 FREIGHT CAR MODEL

2.1 Model Description

The freight car configuration of this analysis is the trailer on flatcar (TOFC) shown in Figure 1. The flatcar is an 89 foot TTAX railcar with ASF ride control trucks using D-5 spring groups, friction snubbing and standard side bearings. There are two Trailmobile van trailers. The weight of the empty car body is 48,250 lbs. Each empty trailer weighs 12,187 lbs. and each is loaded with 49,920 lbs. of cartoned palletized lading. Including truck weights of 8620 pounds each, the gross weight at the rails is 189,704 lbs.

There are two basic differences between the configuration of this study and that of Reference (1)*: The Reference (1) configuration contains a platform and a van trailer and includes all of the lading as a rigid and integral part of the trailer bodies; the configuration of this report contains two van trailers and includes a flexible lading representation. A schematic of the lumped-mass spring-connected model is shown in Figure 2. Table I indicates the number of degrees of freedom for each lumped mass. There are a total of eleven lumped masses, including four representing spring mounted lading masses, and a total of 43 degrees of freedom. Car body flexibility is included through a modal superposition method using normal modes obtained in a separate analysis of the empty carbody.

Problem solution is in the time domain, using numerical integration. This method allows the inclusion of some of the non-linear characteristics of freight cars. Specifically, there are no small angle assumptions made; separation is permitted at the wheel-rail interface and at the trailer tire-carbody deck interface; the

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* The list of references is located on page B-1.

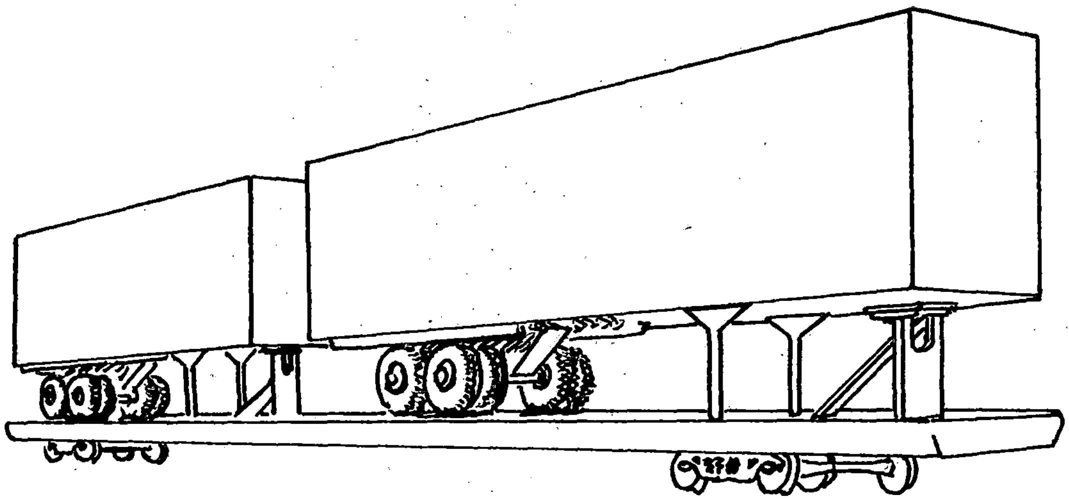


FIGURE 1
TRAILER ON FLATCAR (TOFC) CONFIGURATION

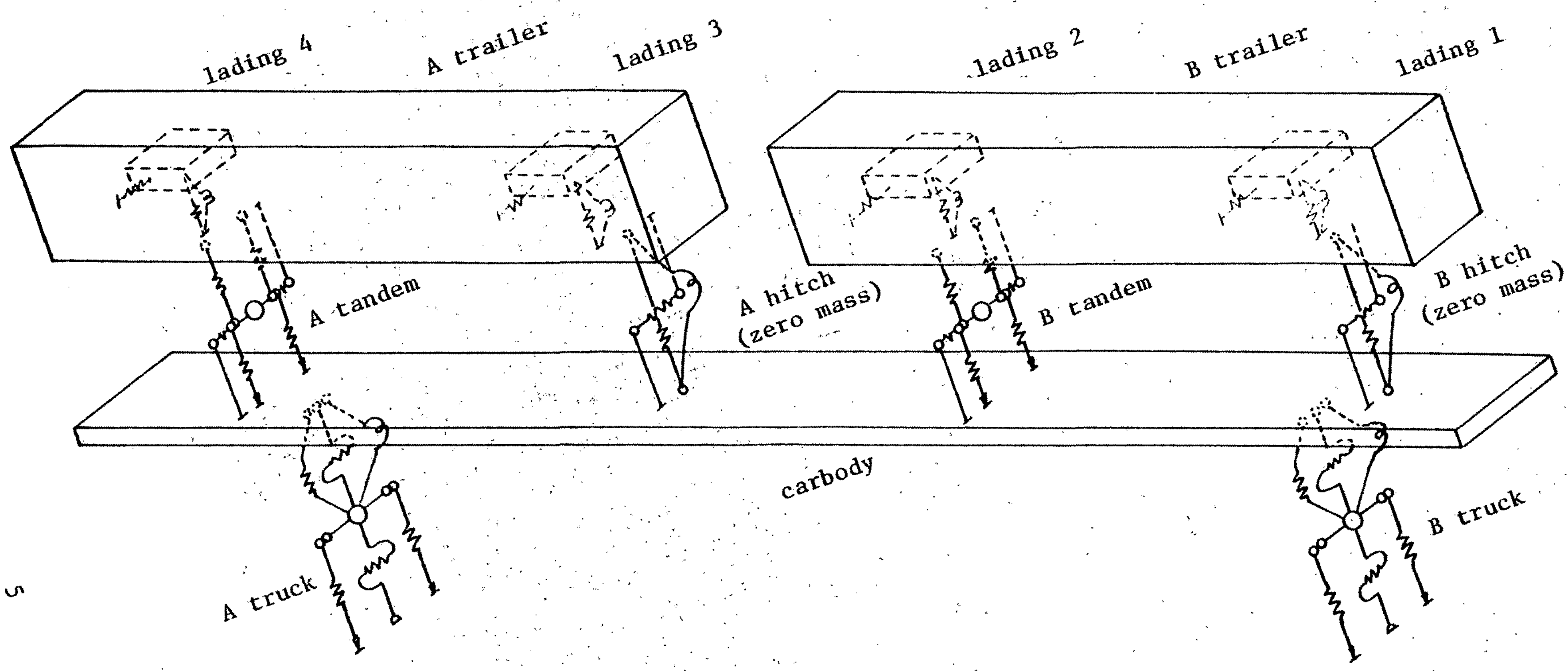


FIGURE 2
MATH MODEL SCHEMATIC OF TOFC WITH FLEXIBLE LADING

TABLE I

MODEL DEGREES OF FREEDOM

LUMPED MASS	DEGREES OF FREEDOM INCLUDED				
	LATERAL X	VERTICAL Z	PITCH θ	ROLL φ	YAW α
M(1) B Truck	√	√	—	√	—
M(2) A Truck	√	√	—	√	—
M(3) Carbody	√	√	√	√	√
M(4) B Tandem	√	√	—	√	—
M(5) B Trailer	√	√	√	√	√
M(6) A Tandem	√	√	—	√	—
M(7) A Trailer	√	√	√	√	√
MLAD(1) - MLAD(4)	√	√	—	√	—

Notes: The first four normal modes of the flexible carbody are also included.

There are a total of 43 degrees of freedom.

non-linear roll spring rate of the rail car is simulated with a bilinear spring; the difference in lateral spring rates at the wheel-rail interface, with and without flange contact, is simulated with a bilinear spring and the friction snubbers in the railcar trucks were modeled as coulomb dampers.

A detailed description of the computer program and the TOFC model, including numerical values for all model parameters, is contained in Appendix A.

2.2 Resonant Frequencies of the Model

In the analyses performed, maximum response was anticipated where the excitation frequency fell on one of the resonant frequencies of the freight car. The excitation frequencies are discussed in Sections 3.1, 4.1 and 5.1. It was consequently helpful to determine beforehand the resonant frequencies of the TOFC being analyzed. This was done in two ways: (1) by noting the frequency of the decaying oscillation after a sinusoidal input motion at the wheel-rail interface was abruptly stopped and (2) by varying the frequency of a sinusoidal input motion to find the frequency of maximum response. The resonant frequencies thus obtained are shown in Table II.

The frequencies of resonance shown in Table II are nominal values. The frequencies generally will vary as much as $\pm$ 0.10 Hertz depending on the amplitude of motion of the freight car. The frequency variation can be even greater for certain resonances if the side bearing gap is varied or if the wheel gage clearance is varied.

TABLE II

TOFC RESONANT FREQUENCIES

	FREQUENCY (Hertz)	DESCRIPTION OF RESPONSE MOTION
1	.56	Low center roll
2	.90	Carbody yaw, trailer roll and yaw
3	1.59	Carbody yaw, trailer high center roll
4	1.70	Carbody vertical translation and body bending
5	1.76	High center roll
6	2.2	Carbody pitch
7	2.9	Trailer roll
8	4.5	Trailer pitch and carbody bending
9	6.1	Trailer yaw
10	9.0	Carbody second bending
11	10.4	Carbody torsion

3.0 ROCK AND ROLL ANALYSIS

3.1 Track Profile Simulation

Track rails will have reduced stiffness at the joints. This loss in stiffness is minimal with welded rail, thus is generally associated with bolted joint rail. The result of the reduced joint stiffness is for the rail to acquire a cuspate deformation at the joint. The cusp will be in the downward direction, as a result of vertical loading on the rail, but can also have an outward deformation resulting from lateral loading of the wheel flange on the rails.

Most of the bolted joint rail in use in the United States is 39 foot rail, laid with staggered joints. For the purposes of this analysis it was assumed that the track profile could be adequately represented by a rectified sine wave with wave length equal to twice the rail length. The expression for input motion at the wheel-rail interface and attendant assumptions are shown below:

<u>Rectified Sine Input Motion</u>

ZI(I) = AMP (I) x FIN x ABS (SIN (TH + PH (I)))

where ZI(I) = input motion at location (I), inches.

(I) = 1,6

1 = B truck, left side, vertical

2 = B truck, lateral

3 = B truck, right side, vertical

4 = A truck, left side, vertical

5 = A truck, lateral

6 = A truck, left side, vertical

AMP(I) = amplitude scale factor, input data

FIN = amplitude of input motion vector, may be constant or variable with frequency depending on input data, inches

TH = angle of input motion vector, radians

PH(I) = phase angle of input motion at location (I) relative to other input locations, radians.

Assumptions

1. Rail is staggered and 39 feet in length.
2. Truck center distance is 66 feet.
3. The railcar truck is assumed to follow the rectified sine profile without the "flattening" of the cusp afforded by two wheel sets.
4. The phase angles between the input locations were determined as shown in Figure 3.
5. The alignment offset was assumed zero for all but two cases.

Speed-Frequency Relationship

The relationship between rail length, train speed and frequency is defined by the following expression:

$$f = V/\lambda$$

where f = frequency in Hertz

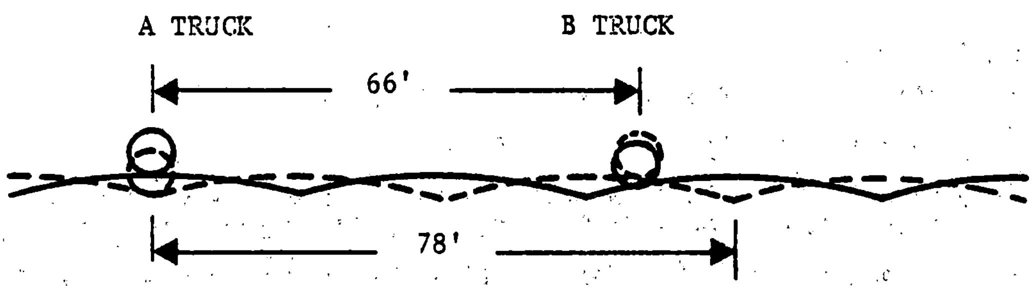
V = velocity in feet per second

λ = rail length in feet

Thus with 39 foot rail length (λ) and a train speed of 26.591 mph the effective motion felt by the freight car is at a frequency of 1.0 Hertz.

With a rectified sine representation the frequency of the rectified sine (frs) equals half the excitation frequency

$$frs = f/2 = V/(2\lambda)$$



INPUT LOCATION	INPUT PHASE ANGLE (degrees)
I = 1	0
I = 3	90
I = 4	304.6 (360 x 66/78)
I = 6	34.6

FIGURE 3
RECTIFIED SINE SIMULATION OF STAGGERED BOLTED TRACK

Cross Level Differences

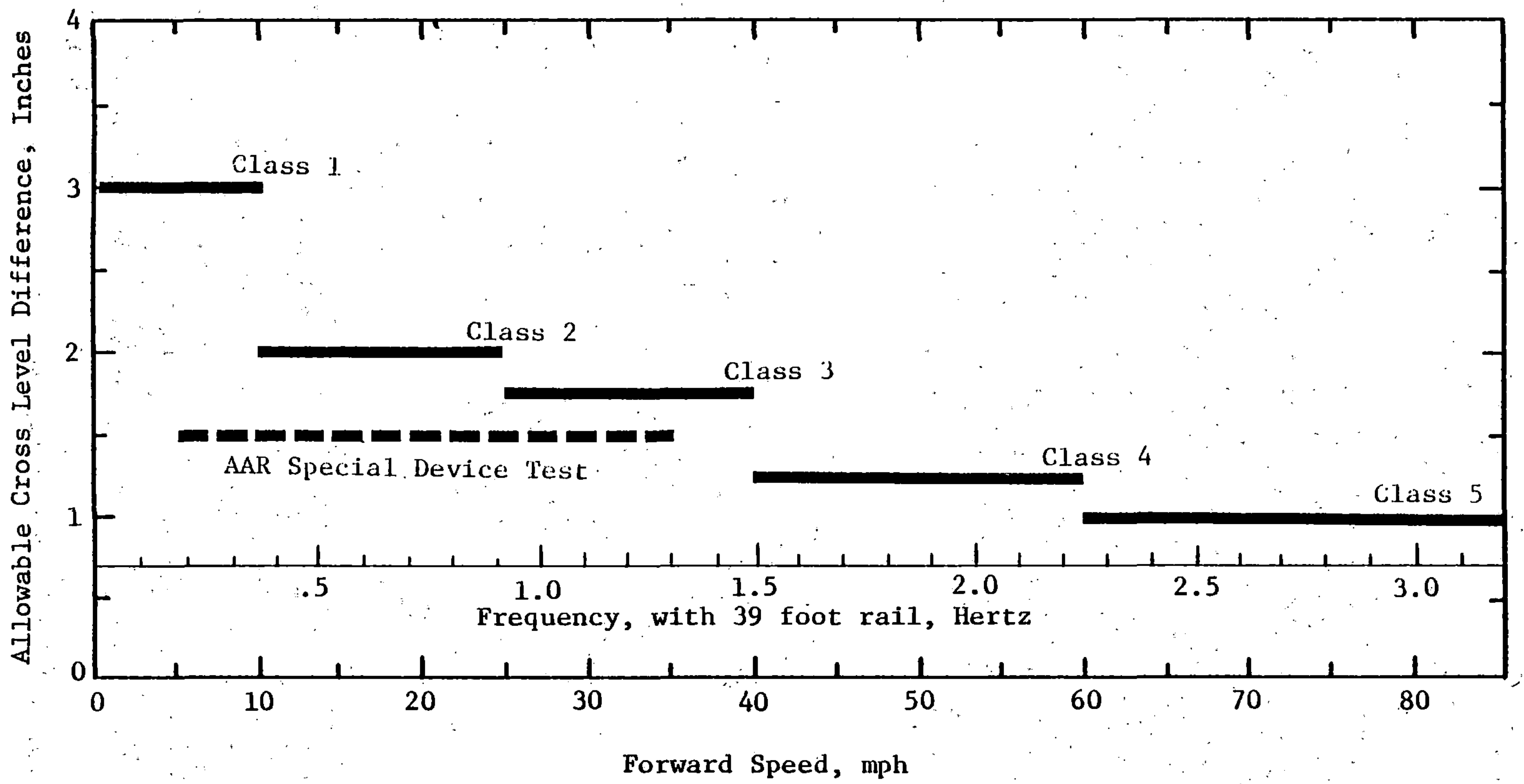
The amplitude of the rectified sine was made equal to one half the allowable cross level difference of tangent track defined in the Track Safety Standards of Reference 2. Figure 4 shows these allowable cross level differences according to class of track and train speed.

Analyses were performed for class 2 and 3 track and for the AAR special device test specifications. The AAR Special Devices Test calls for the rail to "be shimmed opposite 20 consecutive joints to 3/4 inches higher than the general elevation..." Since this will result in cross level variations of $\pm$ 3/4 inches, the maximum cross level difference between any two points will be 1.50 inches.

3.2 Description of Rock and Roll Analysis

Testing results and over-the-road experience has shown that the rock and roll response of a freight car over staggered bolted rail is very speed dependent. That is, there is a critical speed for any given freight car where its rock and roll response will reach a maximum. Experience has also shown that, because of the non-linear properties of the freight car, the largest response amplitudes will be reached with the train speed decreasing through the critical speed. This phenomenon was duplicated in the analysis of this report, for cross level amplitudes above a threshold limit. In view of this most of the analyses performed for the rock and roll case were with deceleration through the critical speed. All the data presented is for this type of run.

It was also found in the analysis that the starting frequency (or speed) and the rate of change had an effect on the maximum response. After several exploratory runs, start and rate of change values which had minimum effect were selected and used throughout



from Code of Federal Regulations, Title 49 and the AAR Specification
for Testing Special Devices to Control Stability in Freight Cars

FIGURE 4
ALLOWABLE CROSS LEVEL DIFFERENCES

for the comparative study; i.e., starting at a higher speed and decelerating at a slower rate would not cause change in the results.

The response parameters which were used to evaluate results were:

1. The amplitude of the roll motions of the trailers and carbody
2. The amplitude and acceleration of lading motion
3. The wheel-rail loadings including L/V values and incidence of wheel lift

The bulk of the analyses was performed for critical speeds corresponding to the first (low center) roll resonance with 39 foot staggered rail. An analysis was also performed with 33 foot rail since this would be a worst case condition for the 66 foot truck spacing of the 89 foot flatcar and since there is some 33 foot rail in use today.

Three check cases were also performed for critical speeds corresponding to the first yaw resonance. One case was with cross level offset only; the second case was with alignment offset only, and the third case was with combined cross level and alignment offset.

3.3 Rock and Roll Analysis Results

The results of the rock and roll response analyses performed in the two van trailer TOFC configuration are summarized in Figure 5 and Tables IIIa and IIIb.

The performance of the TOFC under the AAR Special Devices Test was well within the acceptability limits defined by those specifications. Specifically, the maximum total angle of roll was 2.96 degrees compared to an allowable 6.0 degrees; the lateral acceleration at the center of gravity was about 0.15 g compared to an allowable 0.35 and there was no wheel lift compared to an allowable 0.5 inches.

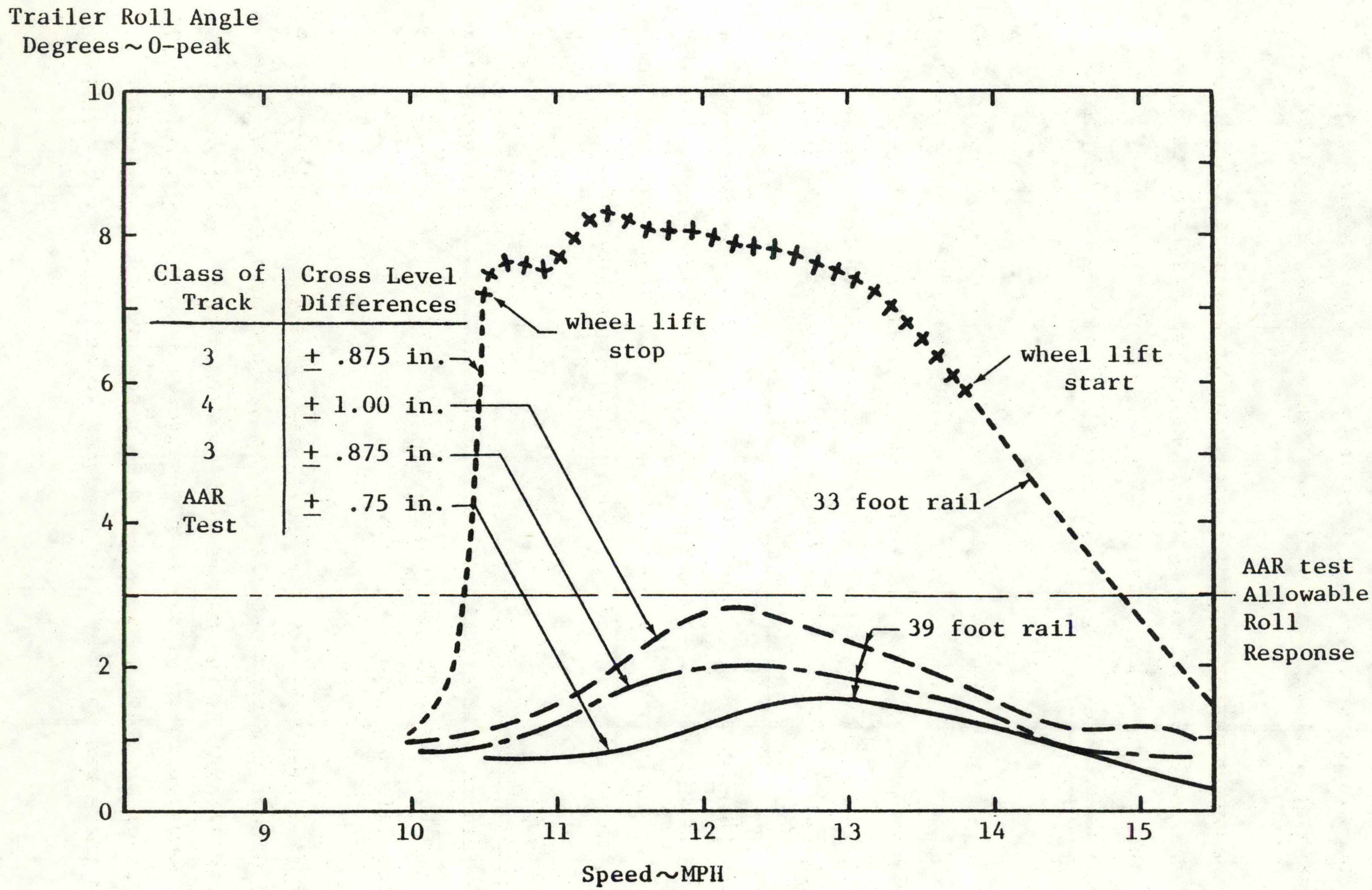


FIGURE 5
SUMMARY RESULTS ROCK AND ROLL ANALYSES RESPONSE OF TOFC TO STAGGERED BOLTED RAIL WITH CLASS LIMIT CROSS LEVEL OFFSET

TABLE IIIa

SUMMARY OF ROCK AND ROLL ANALYSIS RESULTS

Analysis Conditions	Trailer Roll (Degrees)	Carbody Roll (Degrees)	Lading g Load (g's)	Lading Displacement (inches)	Wheel Lift	L/V_o V_o=47426	L/V
Special Device Test Crosslevel = 0.75"	1.48	1.04	.18	3.59	No	.21	.15
Class 3 Track Crosslevel = .875" 39 ft. rail	1.93	1.51	.182	4.59	No	.18	.13
Class 2 Track Crosslevel = 1.00" 39 ft. rail	2.76	2.10	.232	6.88	No	.23	.18
Class 3 Track Crosslevel = .875" 33 ft. rail	8.36	6.86	.75	19.2	11 Cycles	.61	.30

NOTE:
- In the calculation of L/V, the lateral force at the wheel flange divided by the vertical wheel force, V_o is the static vertical force and V is the dynamic vertical wheel force at the time of max lateral force.
- Crosslevels are at point of maximum.
- All values in Tables IIIa and IIIb are single amplitude.

TABLE IIIb

SUMMARY OF ROCK AND ROLL ANALYSIS RESULTS RESPONSE

Analysis Conditions	Trailer Roll Angle (Degrees)	Carbody Roll Angle (Degrees)	Lading Accel. (g's)	Lading Displacement (inches)	Wheel Lift	L/V_o V_o=47426	L/V
Sine Input Cross level = ± .875 in. Alignment = 0.0 Max. Resp. at 20.58 mph (.774 Hz)	.86	.20	.34	1.48	no	.31	.26
Sine Input Cross level = 0.0 Alignment = ± .875 in. Max. Resp. at 22.87 mph (.86 Hz)	2.42	.53	.835	5.49	no	.58	.44
Sine Input(1) Cross level = ± .875 in. Alignment = ± .4375 in. Max. Resp. at 22.34 mph (.840 Hz)	2.16	.48	.807	4.74	no	.57	.44

(1) Sine track input with both cross level and alignment offset, phased so that when rail is at low point it is also at its maximum outward alignment deviation.

The responses with maximum cross level differences of 1.75 and 2.00 inches for class 3 and 2 track respectively were also within acceptable limits for rock and roll performance. This good performance of the 89 foot flat car is laid to the fact that the 66 foot truck spacing results in being relatively insensitive to the 39 foot wave length of standard track. This is borne out by the very severe response obtained with class 3 track with 33 foot rail. For this case total roll angle of 16.72 degrees is predicted for the trailer motion and 13.72 degrees for the carbody. Wheel lift starts after four rail lengths have been traversed and wheel lift occurs on each side for 11 cycles of motions in a total distance of about 15 rail lengths.

When the TOFC freight car is traveling on staggered joint rail at speeds in the 20-30 mph range there will be a tendency for the vehicle to respond in its first yaw resonance. In the TOFC of this study the resonant frequency was at about .90 Hertz and the critical speed was between 22 and 24 mph. The results of the three cases analyzed are shown in Table IIIb. In the first case the track was assumed to have the maximum allowable cross level difference for class 3 track (1.75 inches) and no alignment deviation. The response conditions are seen to be significantly less than for the roll resonance case over the same class track.

The second case of yaw resonance analyzed was for the effects of 1.75 inch alignment deviation. The results show the lading acceleration to be higher than for any of the roll cases. Also, even though there was no wheel lift the lateral/vertical wheel load ratio (L/V) was higher than for any other case in this study. The (L/V_o) of .58 although higher than would be desired is below any derailment threshold level.

The third case of combined cross level and alignment had responses which were very similar to case 2, alignment offset only, even though the alignment offset for case 3 was half that used for case 2.

4.0 HUNTING MOTION ANALYSIS

The analyses performed in this study are to predict the responses of the freight car assuming a hunting condition exists. The purpose of the analysis is to quantify the freight car responses under various hunting conditions and to gain some insight on the behavior of the freight car. An extension of the analyses would be to find configurational modifications which would reduce freight car response.

4.1 Hunting Motion Simulation

The motion of the freight car truck in a hunting condition is basically more or less sinusoidal lateral movement between the limits of gage clearance. At speeds below the critical hunting speed the lateral movements will be relatively small and sinusoidal. At or close to the critical speed the amplitudes may be large enough for the wheel flanges to make contact with the rails, and even hold contact for an instant of time, in each half cycle of motion. Figure 6 shows the hypothetical truck motion in these two conditions of hunting.

In order to accurately reproduce the hunting motion of the trucks the lateral stiffness between the truck and rails, K(2) and K(8), were set up as bilinear springs. For small deflections the springs were made to represent the no-flange-contact rolling condition. For deflections greater than the gage clearance the spring stiffness was made representative of flange-contact conditions. With this arrangement the input parameters of input motion and gage clearance can be set so as to duplicate truck motions representative of either of the conditions shown in Figure 6.

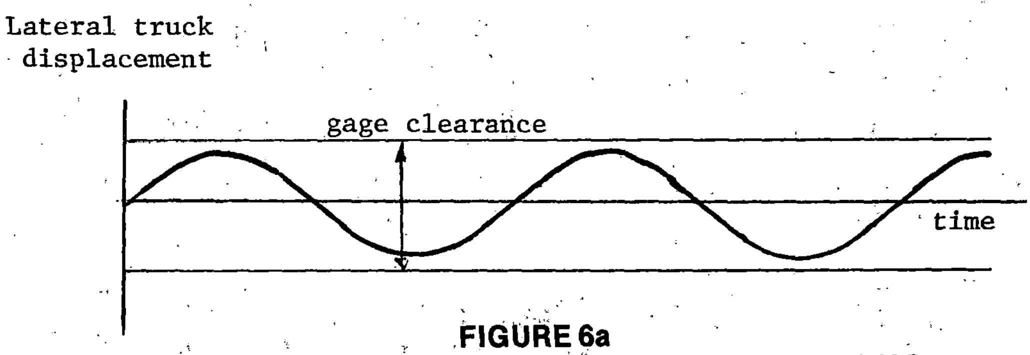


FIGURE 6a
TRUCK MOTION HYPOTHESIZED FOR MILD HUNTING

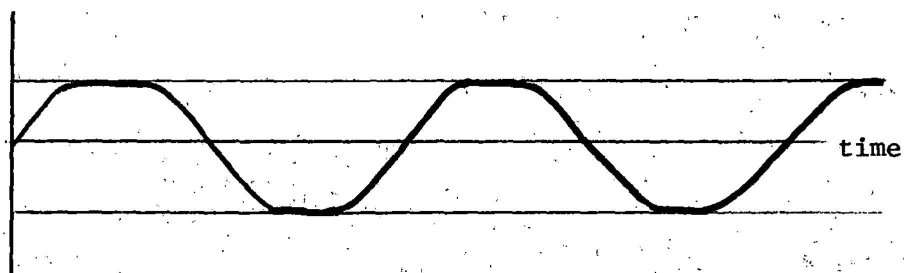


FIGURE 6b
TRUCK MOTION HYPOTHESIZED FOR SEVERE HUNTING

4.2 Description of Hunting Simulation Analysis

In the study of hunting of freight car trucks, largely empirical relationships have been established for the damping of the hunting motions and the oscillatory frequency relative to track speed. (See Reynolds[3] and Wickens[4].) A set of hypothesized predictive curves are presented here in Figures 7a and 7b relating the frequency and damping of the oscillating hunting motion to train speed. The frequency curves are from Reynolds using the expression:

$$f = \frac{V}{2\pi \times 8.58} \quad \text{for new 33 inch wheels}$$

$$f = \frac{V}{2\pi \times 6.20} \quad \text{for worn 33 inch wheels}$$

where f is in Hertz

V is in feet per second.

The damping curves are hypothesized results of the empirical formulas presented in References 3 and 4. The hunting frequency line and the hunting damping curve, in Figures 7a and 7b are to be used in conjunction with each other. That is, for any given speed the frequency line will indicate what the frequency of the hunting motion will be and the damping curve gives a qualitative measure of damping. For example, if a car with new wheels (Figure 7a) encounters an alignment irregularity at 40 mph, the frequency of the resulting lateral motion in the truck will be at about 1.0 Hertz and the motion will damp out very quickly. If the same irregularity is encountered at 80 mph, the frequency will be 2.0 Hertz or slightly higher and the motion will be essentially undamped and will continue at an amplitude bounded by gage clearance until the speed is reduced. The speed at which the damping curve reaches zero is known as the critical speed and the associated frequency is referred to in this report as the critical hunting frequency.

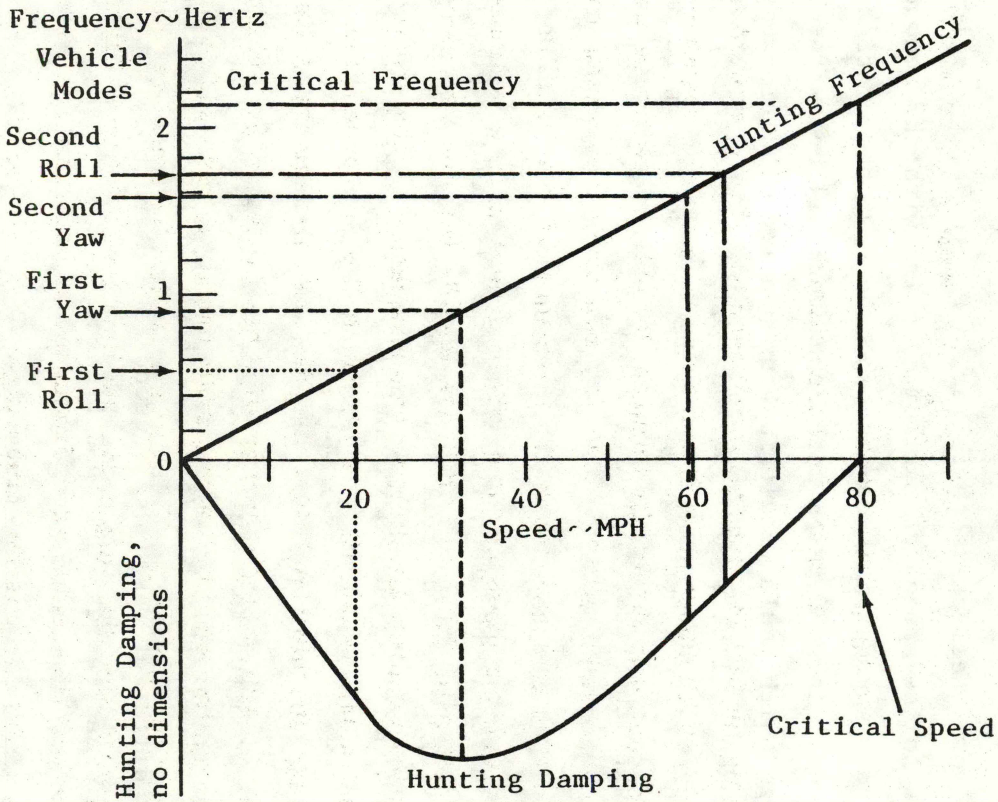


FIGURE 7a
NEW WHEEL HUNTING CHARACTERISTICS

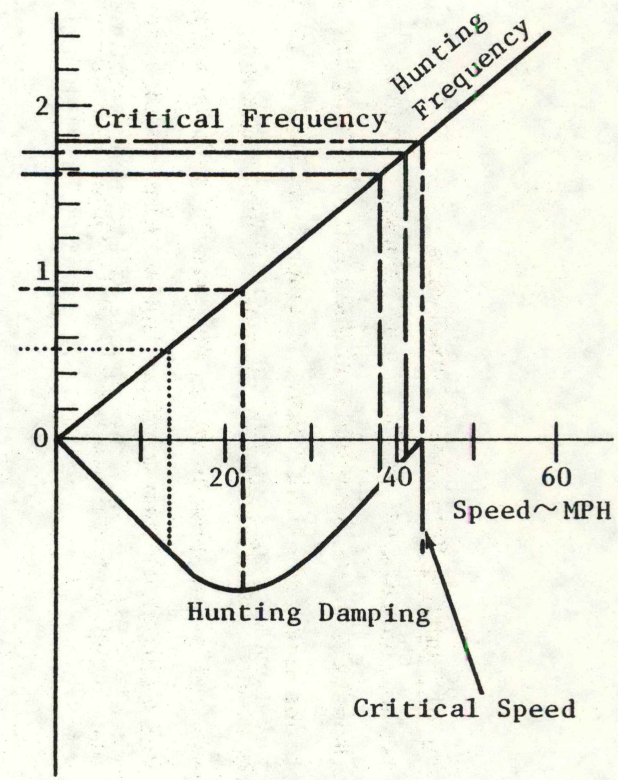


FIGURE 7b
WORN WHEEL HUNTING CHARACTERISTICS

Body hunting occurs when the hunting frequency is close to a body frequency. In the case of the TOFC configuration of this report there are four body frequencies which may participate, .56, .90, 1.59 and 1.70 Hertz. From Figures 7a and 7b body motions will couple with truck hunting motions in the four speed ranges of 18-21, 22-34, 37-59 and 40-63.

With new wheels, Figure 7a, the speeds where body motions will couple are in the range where the hunting damping is high and body hunting will not occur.

With the worn wheel characteristics in Figure 7b, the critical speed is close to the speeds of maximum coupling with the second roll and second yaw modes. It can be expected that the critical hunting speed will in some cases match and even fall below the 35-40 mph range which would result in body hunting conditions involving either the second roll or second yaw body modes. However, it is not likely that the critical hunting speed could get so low as to develop body hunting with the first roll or first yaw mode, i.e., below 20 mph.

The conclusion to be drawn from this exercise is that body hunting involving the second roll and second yaw modes can be expected under service conditions but body hunting involving the first roll and first yaw body modes are very unlikely.

It should be noted that this discussion specifically applies to the freight car of this analysis. Another freight car will have significantly different resonant frequencies and the body hunting-truck hunting speed ranges will be different.

The analyses performed used the assmption of a gage clearance and the input of lateral sinusoidal motions at the wheel-rail interface with frequency varied to cover the speed range of 10 to 60 miles per hour. The bulk of the analyses was made with gage clearance assumed to be 1.20 inches with check cases of 1.6 and 2.0 inches. These values overlap the average allowable gage clearances shown in Table IV.

Runs were made with the input motions at either end in-phase and out-of-phase so that both roll and yaw body hunting conditions were obtained.

4.3 Hunting Simulation Analysis Results

results of the hunting analyses are summarized in Figures 8 through 17 and in Table V. It should be noted that the results presented are not a prediction of whether or not hunting will occur, but are a prediction of the magnitude of response motions within the freight car with the assumption that a hunting condition exists. Accordingly, each plot of Figures 8 through 11 shows the magnitude of various responses with a notation as to the general probability of hunting occurrence.

Body hunting with the first yaw mode (frequency about 0.9 Hertz) and with the second roll mode (frequency about 1.7 Hertz) are seen to result in the largest responses for both acceleration loads and L/V values. Since the first yaw mode body hunting depends on a critical hunting speed below 30 mph, its likelihood of occurrence is very low. The second roll body hunting condition will occur if the critical hunting speed falls below 55 mph. This represents the most likely worst case hunting condition for the TOFC configuration analyzed.

TABLE IV

GAGE CLEARANCE AS A FUNCTION OF TRACK CLASS

Reference 2, Code of Federal Regulations,
Title 49, Transportation

Class of Track	Allowable Gage Clearances (in)			Max Speed MPH
	Minimum	Maximum	Average	
1	.6875	3.0625	1.875	10
2	.6875	2.8125	2.750	25
3	.6875	2.8125	1.750	40
4	.6875	2.5625	1.625	60
5	.6875	2.3125	1.500	80
6	.6875	2.0625	1.375	110

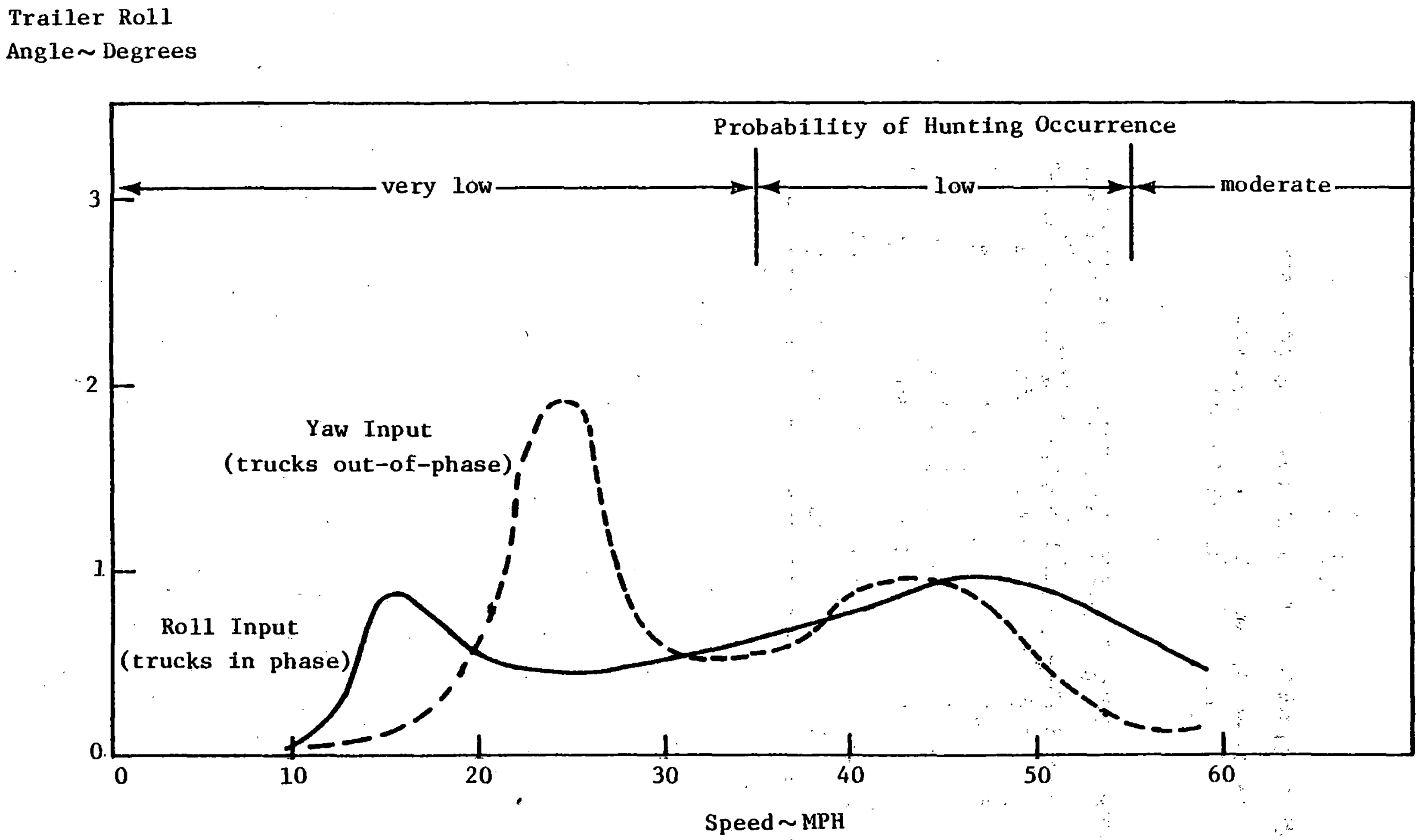


FIGURE 8
MAXIMUM TRAILER ROLL ANGLE RESPONSE WITH ASSUMED HUNTING CONDITION OF LATERAL SINUSOIDAL MOTION

input amplitude = 1.20 inches peak to peak

speed = 2πf * 6.2 * .68, mph

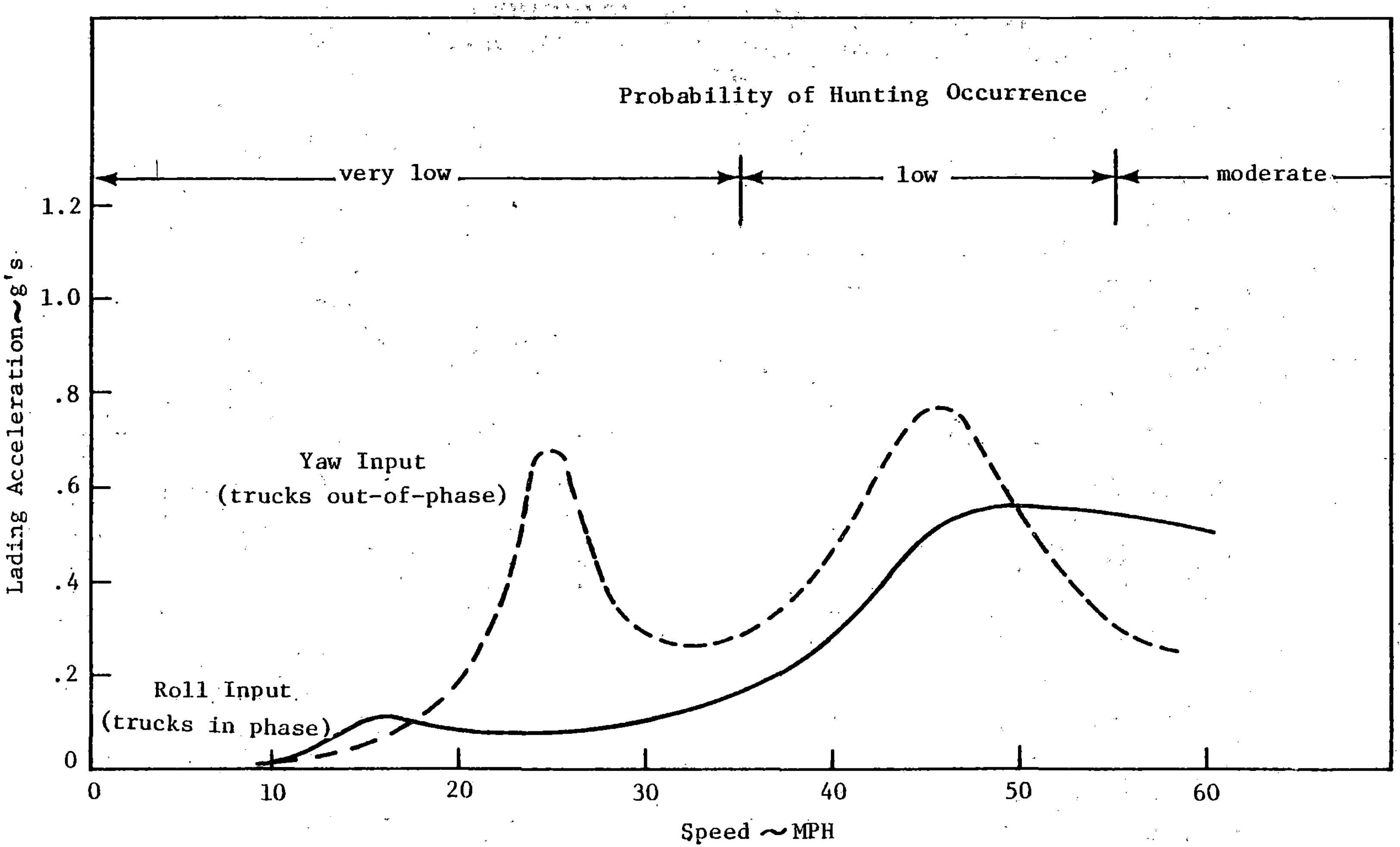


FIGURE 9
LADING ACCELERATION RESPONSE WITH ASSUMED HUNTING CONDITION OF LATERAL SINUSOIDAL MOTION

input amplitude = 1.20 inches, peak to peak

speed = $2\pi f * 6.2 * .68$ mph

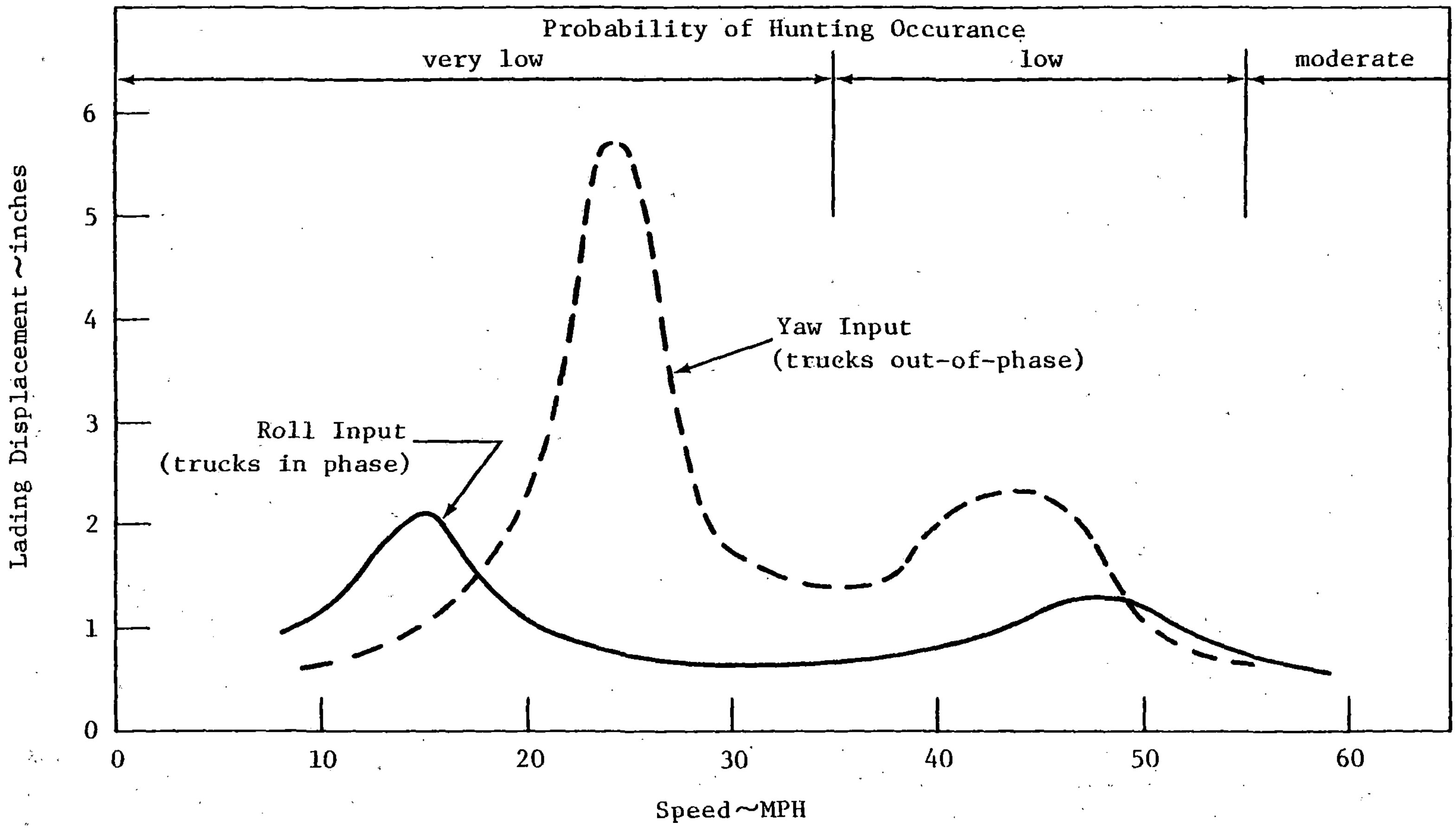


FIGURE 10
LADING DISPLACEMENT WITH ASSUMED HUNTING CONDITION OF LATERAL SINUSOIDAL MOTION

input amplitude = 1.20 inches, peak to peak

speed = $2\pi f * 6.2 * .68$ mph

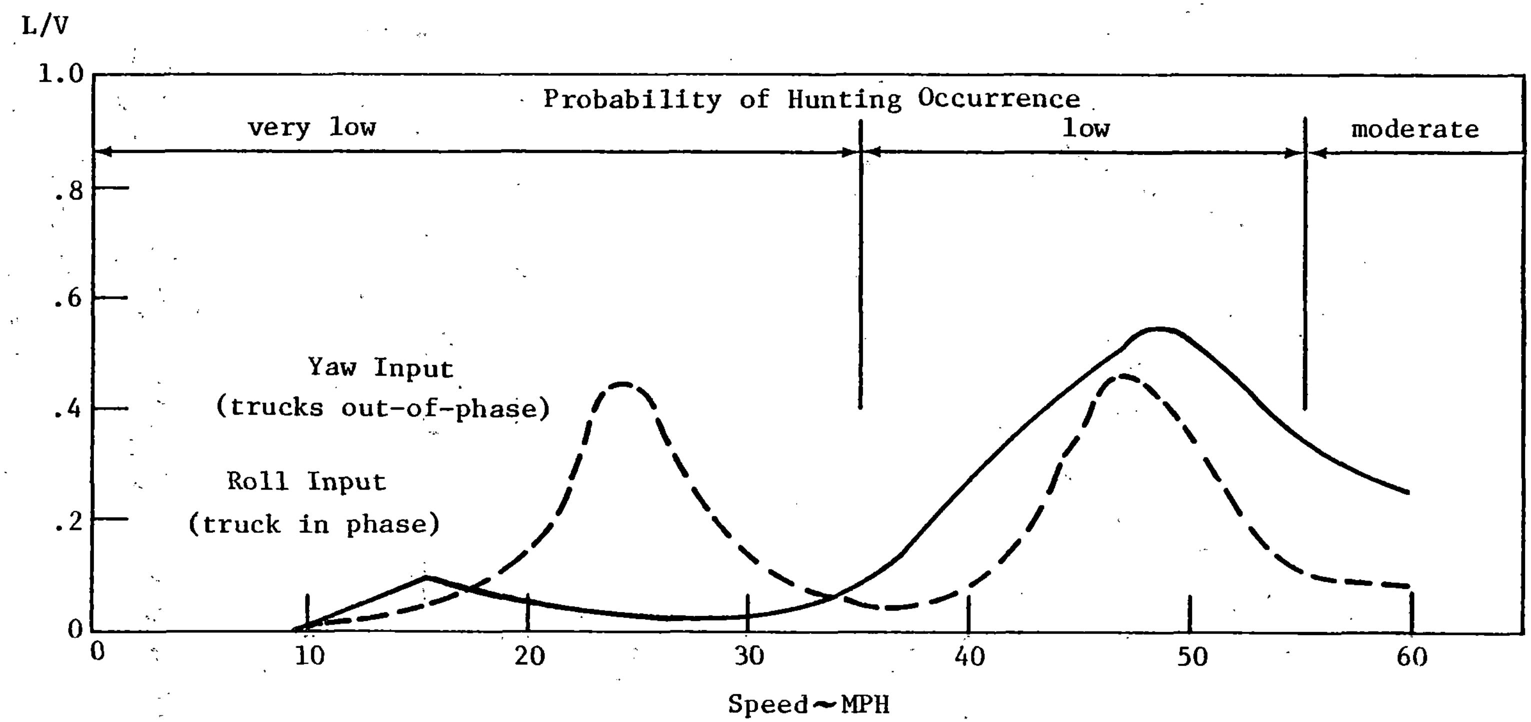


FIGURE 11
LATERAL/VERTICAL WHEEL-RAIL FORCE RATIO WITH ASSUMED HUNTING CONDITION OF LATERAL SINUSOIDAL MOTION

input amplitude = 1.20 inches peak to peak

speed = $2\pi f * 6.2 * .68$ mph

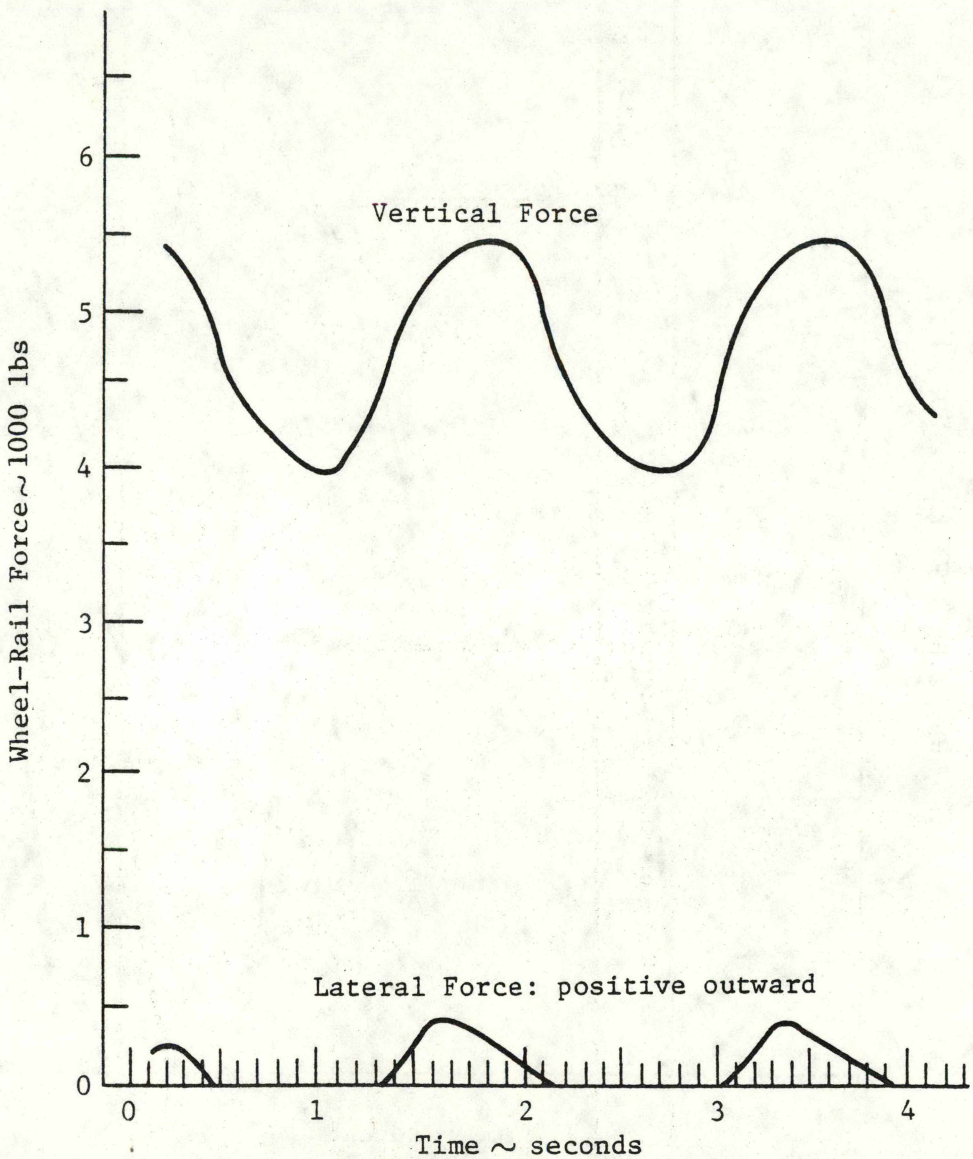


FIGURE 12
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITION TUNED TO FIRST ROLL MODE

f = .56 Hertz

input amplitude = 1.20 inches, peak to peak

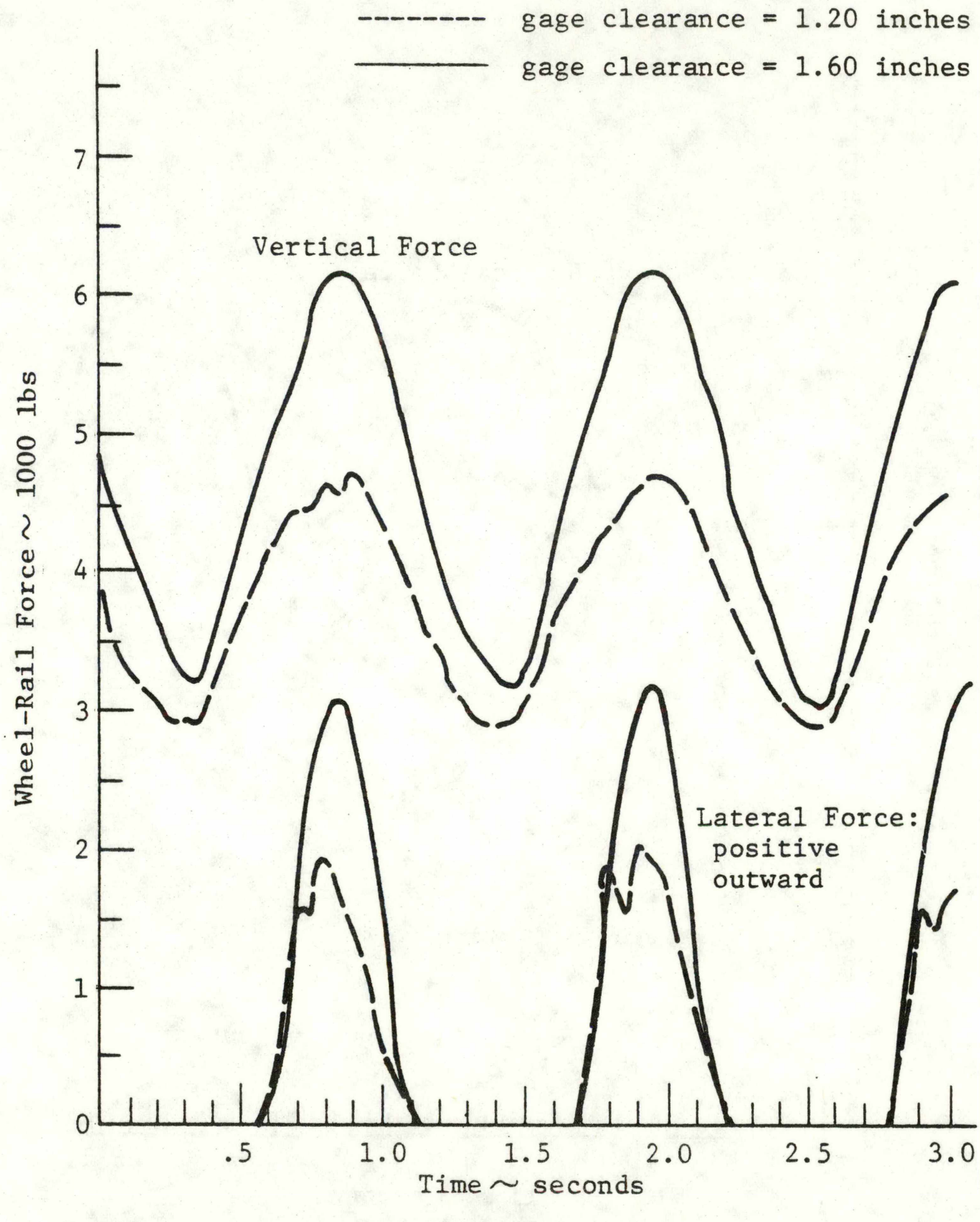


FIGURE 13
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITION TUNED TO FIRST YAW MODE

f = .90 Hertz

input amplitude = gage clearance

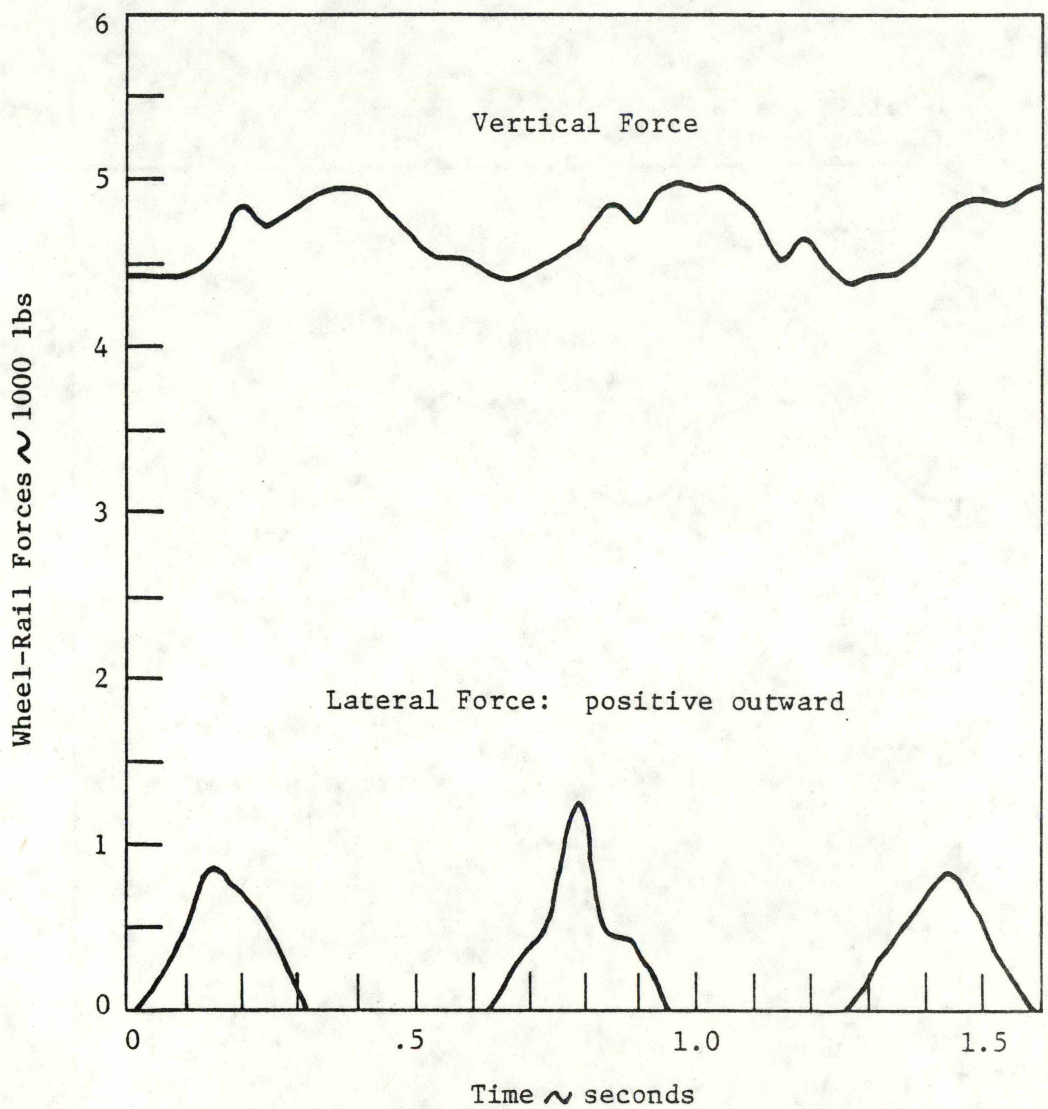


FIGURE 14
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITIONS TUNED TO SECOND YAW MODE

f = 1.59 Hertz

input amplitude = 1.20 inches, peak to peak

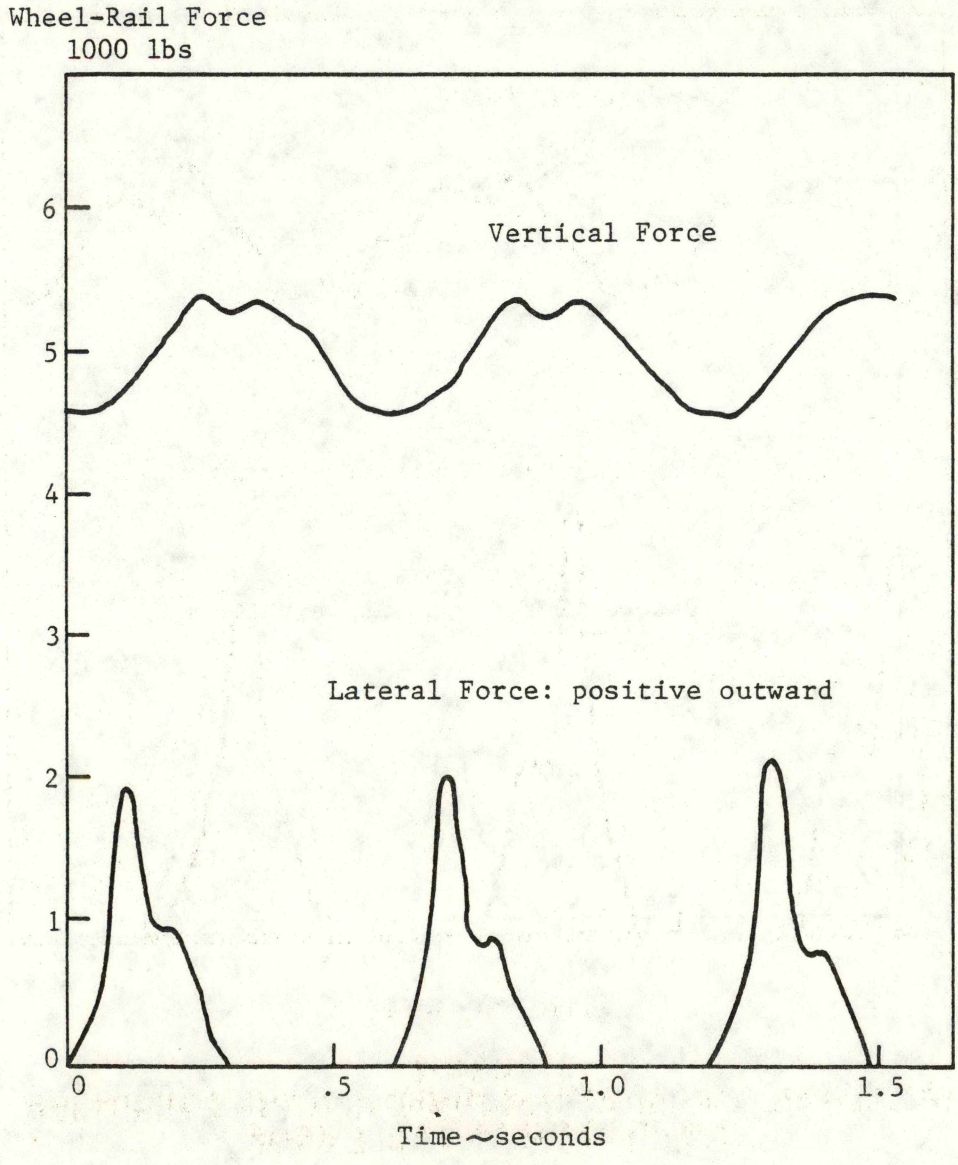


FIGURE 15
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITIONS TUNED TO SECOND ROLL MODE

f = 1.68 Hertz

input amplitude = 1.20 inches

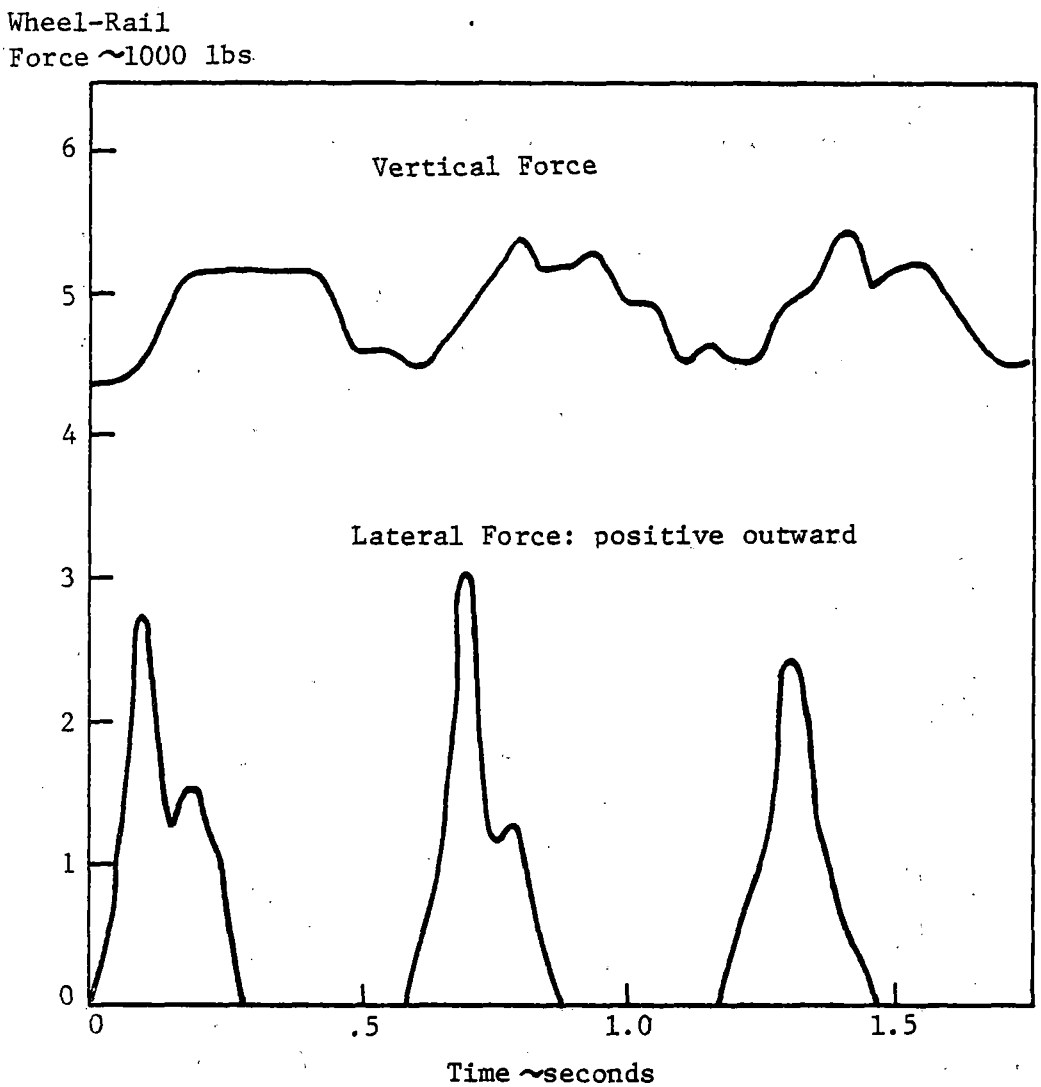


FIGURE 16
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITIONS TUNED TO SECOND ROLL MODE

f = 1.70 Hertz

input amplitude = 1.60 inches, peak to peak

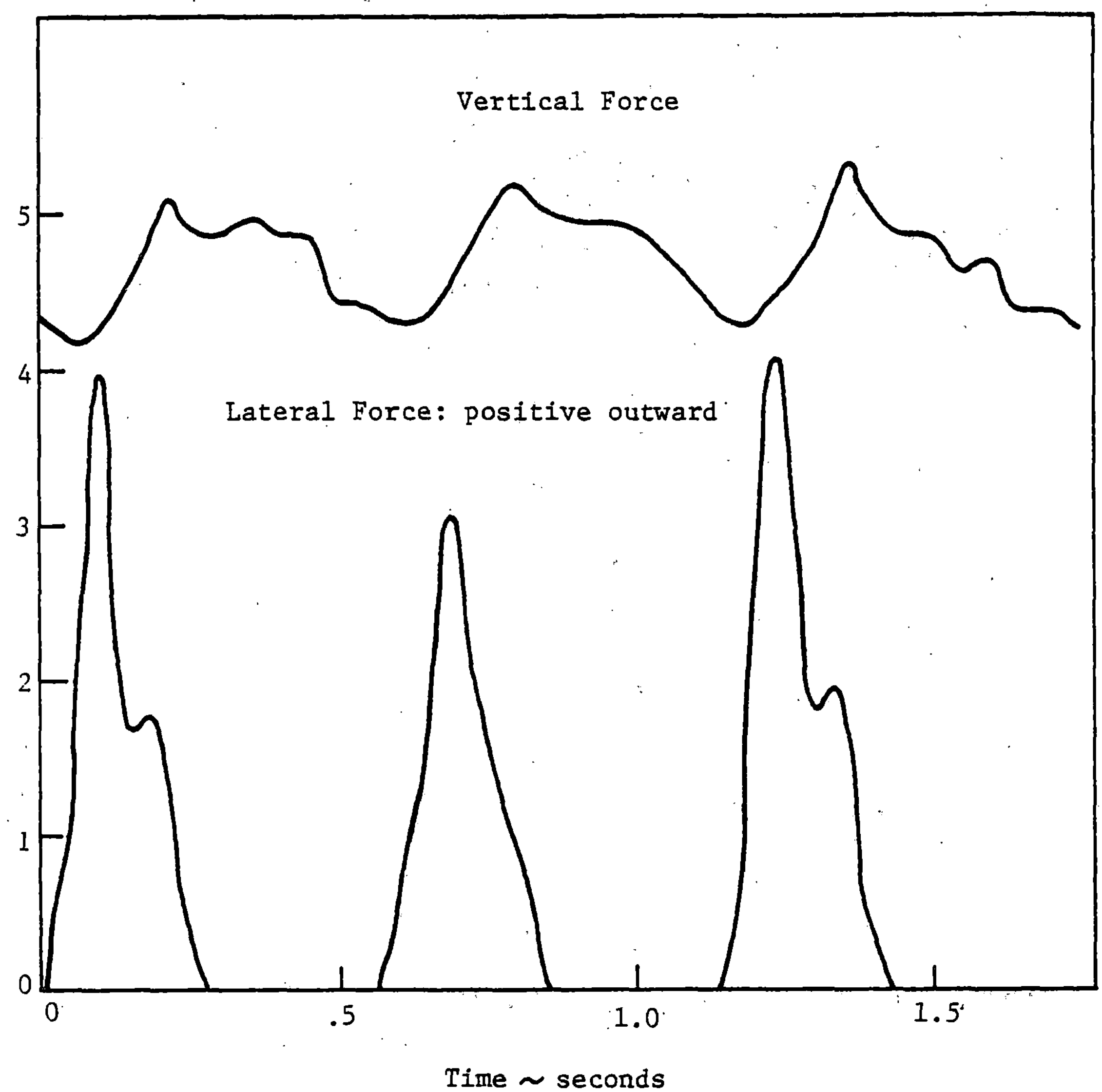


FIGURE 17
WHEEL-RAIL FORCES WITH ASSUMED HUNTING CONDITIONS TUNED TO SECOND ROLL MODE

f = 1.70 Hertz

input amplitude = 2.00 inches, peak to peak

Two additional cases were analyzed to find the responses to second roll mode body hunting for gage clearances of 1.6 and 2.0 inches. Gage clearance of 1.6 inches is about average and 2.0 is about 80 percent of maximum allowable for 60 mph (class 4) track. The results of these two analyses are shown in Table V. The lading acceleration values and the L/V values indicate that lading damage, and derailment, are very likely to occur.

Figures 12 through 17 are representative plots of vertical and lateral wheel-rail forces for each of the four types of body hunting analyzed. It is of interest to note the phasing between the vertical and lateral wheel-rail forces. For the first roll and first yaw modes the forces are in phase so that the instantaneous L/V is less than the maximum lateral force divided by the static vertical force In contrast, for the second yaw and roll body hunting conditions the phasing has shifted so that the instantaneous L/V is equal to L/V_o, as can be seen in Table V. It is likely that at slightly higher frequencies (speeds) the phasing will shift to the point where L/V is greater than L/V_o.

TABLE V

SUMMARY OF HUNTING MOTION RESPONSE ANALYSIS

ANALYSIS CONDITIONS	ANALYSIS RESULTS						
	Trailer Roll (degrees)	Max. Lateral Acceleration (g's)	Max. Lading Acceleration (g's)	Max. Lading Deflection (inches)	Wheel Lift	L/Vo Vo=47426	L/V
1st Roll Mode f = .56 Hertz input motion ± .6"	.81	.069	.088	2.00	No	.09	.08
1st Yaw Mode f = .89 Hertz input = ± .60"	1.89	.712	.628	5.71	No	.53	.45
2nd Yaw Mode f = 1.59 Hertz input = ± .60"	.94	.428	.655	2.23	No	.28	.26
2nd Roll Mode f = 1.68 Hertz input = ± .60"	.968	.180	.493	1.18	No	.40	.44
2nd Roll Mode f = 1.72 Hertz input = ± .80"	1.42	.343	.777	1.81	No	.68	.68
2nd Roll Mode f = 1.72 Hertz input = ± 1.00"	1.83	.467	1.068	2.36	No	.90	.90

Note: All response values in this table are 0-peak values.

5.0 VERTICAL PULSE ANALYSIS

5.1 Track Irregularity Simulation

There are irregularities in the track profile other than the periodic variations discussed in Section 3.0. These irregularities can be divided into two types: (1) those generally random variations that arise from the construction of the track and the uneven settling of the road bed, and (2) elevated hard spots that are found at road crossings, switch blocks and similar road bed and track structure deviations.

It was assumed, for the purposes of this analysis, that a vertical hump on both rails would provide a representative indicator of TOFC response to general track irregularities. The hump was assumed to have a (1-cos) shape. The effect of the irregularity on the trailing truck was delayed by the ratio of truck spacing and train speed.

$$\text{time delay} = \frac{\text{truck spacing}}{\text{train speed}}$$

The hump length was included as a variable. Thus, the length of the hump and the train speed could be tuned for maximum response of the TOFC being studied.

5.2 Description of Vertical Pulse Analysis

The analysis of this section consisted simply of assuming a pulse height representative of allowable track profile variations and of varying pulse length and train speed to get maximum response. It was found that maximum response was obtained when

$$PD = 0.8 \times T$$

where PD = pulse duration in seconds

T = TOFC resonant frequency period in seconds and when

$$V = \frac{L * f}{N} \qquad \text{for resonances with vertical motions for}$$

$N = 1, 2, 3,$ etc.

and $$V = \frac{L * f}{N + .5} \qquad \text{for resonances with pitch motions for}$$

$N = 0, 1, 2, 3,$ etc.

where

V = train speed, ft. per second

L = truck spacing, feet

f = freight car resonant frequency, Hertz

There are four resonant frequencies which were expected to be responsive to vertical track irregularities. These are the vertical and pitch modes listed in Table II. Table VI lists these frequencies again along with the train speeds and pulse lengths which should result in maximum responses.

5.3 Vertical Pulse Analysis Results

Maximum response to track irregularities was expected to occur in the 60 to 80 mph speed range. This is in the class 5 track speed range which allows elevation deviations of 1.0 inch. Analyses were run for the four speed and pulse length conditions shown in Table VI with pulse height of 1.00 inch. A summary of results is presented in Table VII.

One apparent trend in the data is for the acceleration response and wheel loads to increase with decreases in pulse length. This trend is explainable because the "frequency" of the pulse is inversely proportional to its duration and the acceleration is proportional to the frequency squared. That is:

TABLE VI

PULSE DEFINITION FOR
VERTICAL TRACK IRREGULARITY ANALYSIS

FREQUENCY (Hertz)	MOTION	FACTOR	V (mph)	PL (feet)
1.7	Vertical	N = 1	76.5	52.8
2.2	Pitch	N+.5 = 1.5	66.0	35.2
4.5	Vertical	N = 3	67.5	17.6
9.0	Pitch	N+.5 = 5.5	73.6	9.6

The values in this table are problem input conditions which were expected to result in maximum responses of the vehicle. They were obtained with the following formulae:

$$V = \frac{L * f}{\text{Factor}} * \frac{60}{88}$$

$$PD = \frac{.8}{f}$$

$$PL = PD * V * \frac{88}{60}$$

where:

V = speed expected to result in maximum responses, mph

f = freight car resonant frequency, Hertz

Factor = N or N+.5, integer

PD = pulse duration, seconds

PL = pulse length, feet

TABLE VII

SUMMARY OF VERTICAL PULSE ANALYSIS RESULTS

TRACK CONDITION		MAXIMUM RESPONSE				
SPEED MPH	PULSE LENGTH FEET	CARBODY (g's)	TRAILER AT LADING (g's)	LADING cg (g's)	WHEEL RAIL VERT LOAD (lbs) Max/Min	RESPONSE FREQ. (Hertz)
76.5	52.8	.318	.420	.427	57,900 / 33,584	1.74
66.0	35.2	.567	.647	1.57	57,400 / 28,700	1.74
67.5	17.6	1.23	1.11	1.48	61,800 / 27,192	?
73.6	9.6	3.43	1.02	2.05	77,600 / 18,624	9.0

NOTE: Wheel-rail load is the combined vertical load on the two wheels on one side of one truck.

Response "g" values are 0-peak.

$$f \sim 1/\tau$$

$$\text{acceleration} \sim f^2$$

Using these relationships the effective g's of each pulse duration were calculated and used as the denominator in determining the ratio of output to input. The results are given in Table VIII and show that there is little amplification and the high loadings are due to the "sharpness" of the track pulse.

The conclusion is that in general, for a given track elevation deviation, the freight car acceleration responses will be higher for shorter (longitudinal) deviations and faster speeds.

Returning to Table VII we note that the predominant frequency of response in case 1 is the 1.74 Hertz first bending resonance, as expected, and the frequency in Case 4 is the second carbody bending resonance, again as expected. In Case 2 the 1.74 Hertz was predominant and in Case 3 there were a number of frequencies present but no one predominated. Of significance, neither the 2.2 nor the 4.5 Hertz resonance was found in any of the cases, leading to the conclusion that these two resonances are not excited by vertical input at the trucks.

Also note that the vertical load at the wheel-rail interface increases as the pulse duration is decreased.

TABLE VIII

PULSE WIDTH TO PULSE ACCELERATION RELATIONSHIP TO
(1-COS) PULSE, 1 INCH HIGH

CASE	PULSE WIDTH (FEET)	PULSE MAX. ACCEL. (g's)	MAX. CARBODY RESPONSE (g's)	Q
1	52.8	.30	.318	1.06
2	35.2	.50	.567	1.13
3	17.6	2.07	1.23	.59
4	9.6	8.29	3.43	.41

6.0 SUMMARY AND RECOMMENDATIONS

The FRATE computer program was used to study the dynamic response of a TOFC with compliant lading to service conditions. Three basic types of conditions were studied: (1) the response to vertical track irregularities; (2) the rock and roll response to bolted rail and (3) the yaw and roll carbody motions in response to hypothesized hunting conditions. We were able to show what particular conditions caused the maximum responses as well as quantifying the responses. The responses included accelerations, displacements and forces. A summary comparison of responses is shown in Table IX, which compares worst cases of probable occurrence. Results from the 33 foot bolted track analysis and the vertical pulse analysis with a 9.6 foot pulse length at 73.6 mph were not included in the comparison because of their low probability of occurrence.

The response to the 39 foot staggered joint track resulted in the mildest responses of the three track conditions studied. For class 2 track with ± 1.00 inches cross level variation, the worst case rock and roll studied, carbody accelerations were ± .18g, lading accelerations were ± .23g, carbody roll angles were ± 2.8 degrees and L/V was .18. The mildness of this rock and roll response is laid to the fact that with 66 foot truck spacing the TOFC is relatively insensitive to the 39 foot wave length of standard track.

As one would consequently expect the TOFC configuration does respond much more readily when run over 33 foot staggered rail. However, since 33 foot rail is in very limited use it is not a problem that needs action other than to avoid running TOFC over track that has 33 foot rail.

The yaw response of TOFC over staggered joint rail in the 20-25 mph speed range resulted in larger responses than the rock and roll

TABLE IX

SUMMARY OF PREDICTED RESPONSES

RESPONSE MEASURE	RESPONSE VALUE			
	39 Foot Jointed Track		Hunting	Vertical Pulse
	First Roll	First Yaw		
Input Conditions				
Amplitude - inches	1.00	.875	.80	2.00
Speed - mph	12.3	22.9	45.7	67.5
Frequency - Hertz	.46	.86	1.72	5.6
Carbody Acceleration - g's	.18	.52	.95	1.23
Carbody roll angle - degrees	2.10	.53	.54	--
Trailer roll angle - degrees	2.76	2.42	1.42	--
Lading Acceleration - g's	.236	.835	.78	1.48
Lading Deflection - inches	6.88	5.49	1.80	1.46
Wheel-rail Forces				
Max. Vertical lb	74000.0	62400.0	54000.0	61800.0
Min. Vertical lb	9250.0	32500.0	40000.0	27200.0
Max. Lateral lb	11100.0	28700.0	32400.0	0
L/V	.18	.46	.68	0

NOTE: The input amplitudes and all response values are single amplitude; i.e., $\pm$, except for the vertical pulse input which is + only.

response. The accelerations of .53g on carbody and .84g on lading are high enough to contribute to lading damage. The relatively high lateral wheel-rail forces and high L/V ratios will result in accelerated wear on both track and wheel.

Responses in the hunting simulations also resulted in relatively high accelerations of the carbody ($\pm$ .95g) of the lading ($\pm$ .78g) high lateral wheel-rail forces and high L/V ratios. If a TOFC vehicle has a body hunting critical speed in or near the train operating speed range the hunting motion can be expected to occur frequently with resulting accelerated wear on the track, car and lading.

The vertical pulse analysis performed resulted in larger vertical acclerations than any other conditions studied. The g levels predicted could lead to lading damage. Since wheel lift did not occur and since the minimum vertical force was about 57 percent of static load the vertical track irregularity studied should not, by itself, cause derailment.

One of the conclusions to be drawn from this work is that the 89 foot flat car with its 66 foot truck spacing has good rock and roll behavior over standard 39 foot staggered joint bolted rail. However, this characteristic is offset by the tendency of the TOFC to respond in its first yaw mode when traversing staggered joint rail in the 20-25 mph speed range. The response motion consists primarily of yaw of the carbody and roll of the trailers.

Improvement to roll response characteristics is not necessary. Improvement to the yaw response characteristics cannot be accomplished by changing the truck primary suspension but can be best effected by modifying the trailers in some way. For example, the A trailer was positioned with its tandem directly above the A truck and had 50% greater roll response than the B truck which was positioned with its hitch directly above the B truck. Consequently, a 33 percent reduction in response could probably be achieved, for this condition, if the trailer were mounted tandem to tandem on the flatcar.

Improvement in ride quality for the vertical track anomalies is best achieved through a softening of the suspension system. A step in the right direction is to change from the D5 to the softer, longer stroke D7 spring set. Also reduced friction snubber would improve the vertical ride quality. There is some optimal value of friction, or hydraulic snubber which is enough for rock and roll control and which does not cause a harsh vertical suspension. An ideal solution would be a snubber between the body bolster and truck bolster which would be active with roll motions but inactive in vertical motions by virtue of its location. Such a device would have the added advantage of being in effect for small roll angles of the carbody where existing snubbing devices are all active only after side bearing contact is made.

Body hunting with the second yaw or second roll body mode are the hunting conditions most likely to occur. When they do occur they also will have high wheel-rail forces and lading acceleration response. These two hunting conditions are in the 1.6 to 1.7 Hertz frequency range and will occur in the 35 to 65 mph speed range. Whether hunting occurs or not depends primarily on the design and condition of the truck and wear conditions of the wheels. However, some control of severity of response in hunting can be had by modifying the car body and trailers. The motions of the car body consist of lateral translation in the roll case and twist and yaw in the yaw case. The trailer motions are roll in both cases; in the roll case they roll together, in the yaw case they roll in opposition to each other.

Since trailer roll is the strongest motion in either of these hunting conditions, it will probably be possible to reduce the body response in hunting by some changes to the trailer. The work of this report did not investigate the effectiveness of any changes. The following are consequently suggestions of three changes to be studied which may reduce hunting loads. (1) Dissimilar trailers will probably reduce loads. (2) Auxiliary damping devices between trailer body and flatcar body that are in effect for relative roll motions between trailer and car body (3) The addition of a roll damper in the trailer suspension system.

The flexible lading did have a measurable effect on the vertical response. This conclusion is based on the comparisons of g loading at opposite ends of the spring supporting the spring mounted lading. This ratio varied from about 1.2-2.0, the highest value caused by a 73.6 mph encounter of a hump in the track 2.0 inches high and 9.6 feet long. However, in the other cases studied the carbody and trailer response motions were all below 2.0 Hertz and there were only small differences between the responses of rigid and compliant lading. This is to be expected since the lowest lading resonant frequency was 5 Hertz. The compliant lading model was useful in providing more accurate lading response numbers, but it is not apparent that there was any effect on the overall response of the total vehicle. This of course can change for another lading configuration; one for example which has a larger lumped mass on a low frequency support.

In conclusion, this work has provided the response characteristic of a basic TOFC configuration which can be used as a datum for the investigation and evaluation of proposed modifications.

Appendix A

The Computer Program FRATE with TOFC Model

FRATE is a digital computer program for the analysis of railcar dynamic response which was developed under the sponsorship of the Federal Railroad Administration (FRA). The name FRATE is an acronym for Freight Car Response Analysis and Test Evaluation. The computer program is written in Fortran for Control Data Corporation (CDC) computers with solution in the time domain by numerical integration methods. A detailed description of the basic program and its use can be found in Reference 1. Validation of the FRATE/TOFC simulation is presented in Reference 5.

Time domain solution of the Newtonian equations of motion is used in FRATE in order to be able to include certain nonlinear properties of the railcar. The number of nonlinearities included have gradually increased in the development and application usage of the program. The ones included in the version of FRATE used in this report are listed below:

1. There are no small angle assumptions.
2. Wheel lift is programmed for the freight car wheels and for the trailer wheels.
3. The springs representing the lateral stiffness of the track and wheels, K(2) and K(8), are bilinear to simulate flange contact and no flange contact rolling conditions. The gage difference between rail and wheel set can be varied.
4. The springs representing the roll stiffness of the freight car truck suspension, K(6) and K(12) are bilinear to simulate the roll stiffness with and without side bearing contact. The side bearing gap can be varied.
5. Coulomb damping is included in the freight car truck model to simulate friction snubbers.

The TOFC model used in this study contains six basic modifications to the previous work. First the model was expanded to include four spring supported lading mass representations. This is discussed later in this appendix. Second the platform trailer in the original model was replaced with a van trailer resulting in a TOFC with two identical fully loaded van trailers. Third, the coulomb damper equations in roll were modified to be acting full stroke where it had been half stroke. Fourth, the lateral spring at the wheel rail interface was modified to a bilinear spring: for small amplitudes the spring was made representative of wheel rolling with no flange contact, for amplitudes greater than the defined gage clearance the spring was made representative of flange contact conditions. Fifth, the original FRATE assumed zero height for the freight car wheels and truck. In the FRATE of this study this assumption was dropped and the equations of motion were accordingly rewritten to include wheel radius (HAXL) and effective truck height (HTRK). Sixth and final, input forcing function options were enlarged to include rectified sine and a 1-cos shape pulse with phased applications to the front and back truck based on track speed and truck spacing.

The general analysis procedure which was followed was to impose time-deflection functions at the wheel rail interfaces and to output time histories of resulting response forces, accelerations and displacements. The input motions were simulation of various track profile geometries found in service. The output responses gave indication of the effects of these profiles on freight car and lading dynamic environment.

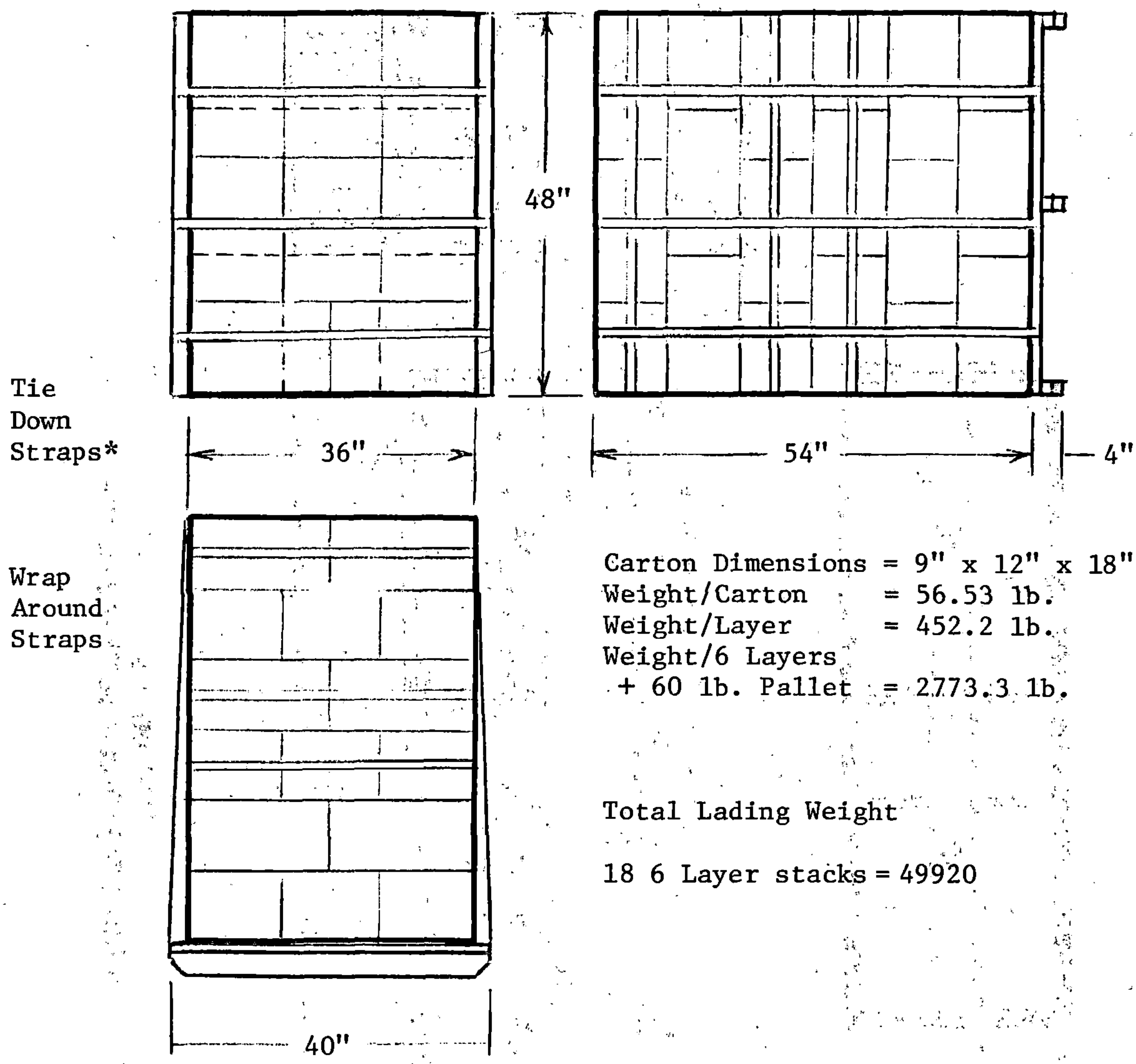
In this appendix a development of the lading model is presented followed by a tabulation of mass, inertia, stiffness, damping and dimensional values used in the TOFC configurations of this study.

Lading Model

The TOFC configuration of this report assumed two equally loaded and configured highway van trailers. The lading was assumed to be the carton on pallet configurations shown in Figure A1 and loaded in the trailer as shown in Figure A2.

The lading arrangement of Figure A1 characteristically will have three resonant frequencies of interest, two lateral at 90° to each other and one vertical. There are other resonances but their frequencies are high enough to be out of the range of interest. Estimates for these resonances have been obtained based on unpublished data from tests performed on a TOFC configuration at the Rail Dynamics Laboratory (RDL), Pueblo, Colorado, and cartoned lading tests performed by Rutgers University, Piscataway, New Jersey, under contract to FRA. In both tests the lading was in steel cans three inches in diameter and 4 3/8 inches high. In both tests the cartons were two cans high. The RDL test cartons were roughly 12 x 18 inches and carried 48 cans. The Rutgers test cartons were roughly 12 x 9 inches and carried 24 cans each. The contents of the RDL cans had a density of roughly 60 lbs./cu. ft. The Rutger content density ranged from 32.1 to 62.4 lb./cu.ft. The Rutgers vibration testing was along the vertical axis only.

It has been generally understood that in a stack of can filled cartons the controlling "springs" are the layers of corrugated board which are the tops and bottoms of the cartons and which are sandwiched by the double layers of cans. And further, that these layers of corrugated board are crushed by the cans in circular footprints and as a result become stiffer. The Rutgers testing verified these understandings. The Rutgers tests showed that the crush up and resulting stiffening was dependent on the density of lading, the g loading and the number of load cycles imposed. The



Carton Dimensions = 9" x 12" x 18"
Weight/Carton = 56.53 lb.
Weight/Layer = 452.2 lb.
Weight/6 Layers
+ 60 lb. Pallet = 2773.3 lb.

Total Lading Weight

18 6 Layer stacks = 49920

*Straps are 1" or 3/4" filament tape.
Seventh Layer also has wrap-around.

FIGURE A-1
CARTON STACK ON PALLET CONFIGURATIONS, TOFC TESTS, RAIL DYNAMICS LABORATORY, 1976

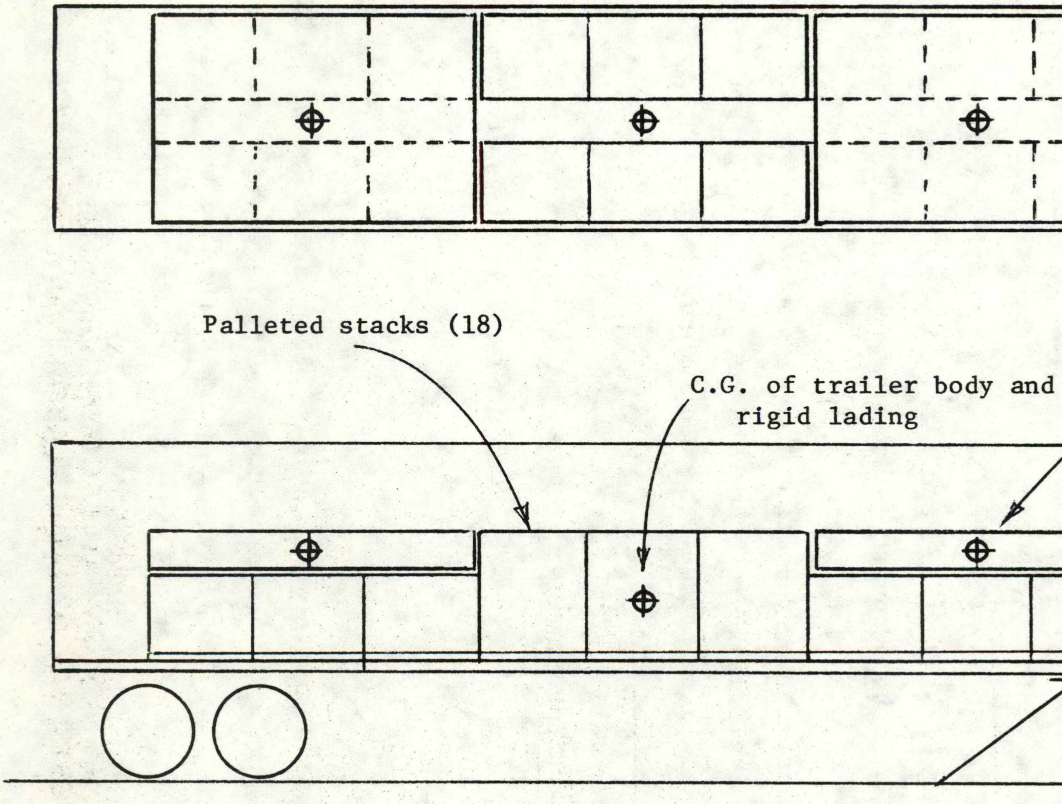
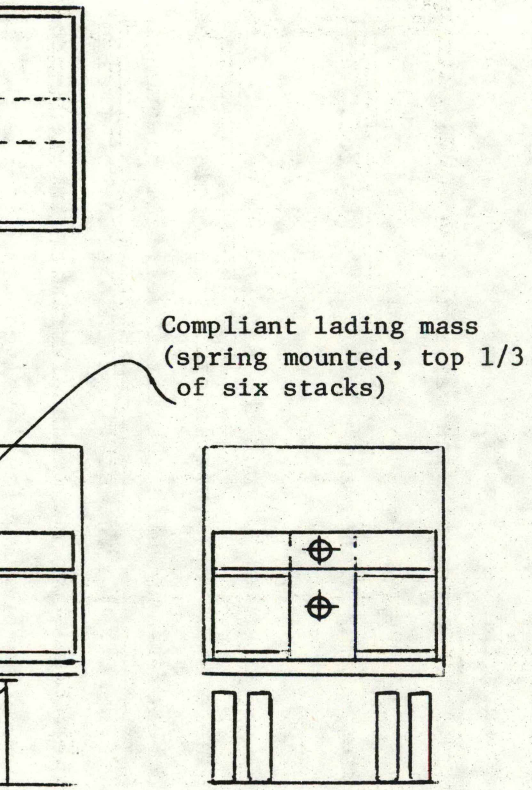


FIGURE A-2
LADING IN TRAILER CONFIGURATION


Compliant lading mass
(spring mounted, top 1/3
of six stacks)

testing also showed that the resonant frequency of a 6 high stack of cartons was essentially constant with variations of lading density. This was explained by the heavier lading causing more crush up and a corresponding higher stiffness of the carton.

The conclusions drawn from study of the RDL and Rutgers test data are summarized in Table AI. The first lateral frequencies are seen to fall in the 3-5 Hertz range while the first vertical frequencies are between 8 and 12 Hertz. In the start of this study of freight car dynamic response there were two phenomena of primary interest--rock and roll and hunting. The frequencies involved were in the 0.5 to 3.0 Hertz range. The transient response due to track irregularities extend this frequency range to about 10 Hertz. Consequently, a model of lading which includes the first vertical and first lateral resonances of a lading stack should be adequate insofar as frequency range is concerned. This could be done with a single mass supported by vertical and lateral spring/dampers. The suspended mass would be sized to some equivalence-at-resonance basis. The springs and dampers would be sized to produce the desired frequencies and amplifications at resonance.

It was felt that the roll degree of freedom should also be included in the lading model. This would be the rough equivalent of the second lateral stack resonance which was estimated to be at 15.0 Hertz.

If one pallet stack were modeled in this way a measure of lading response could be had. If the effect of lading response on the response of the trailer and flatcar is wanted, then it would be best if all lading pallet-stacks were modeled. As a compromise, the six pallet-stacks in the front of each trailer and the six pallet-stacks in the back of each trailer were lumped into one sprung lading

TABLE AI

RESONANT FREQUENCIES AND AMPLIFICATION FACTORS
OF STACKED CARTON CANNED LADING

	Lateral Resonance		Vertical Resonance	
	Hertz	Q	Hertz	Q
6 Layer Stack	4.5-5.4	4.0	9.4-12.	8.5
7 Layer Stack	2.4-3.4	6.5	8.1-11.	15.

Notes: 1. The data in this table are from testing of canned lading in corrugated cartons. The cans were three inches in diameter and 4 3/8 inches high

2. Resonant frequencies appeared to be independent of lading density

3. Data scatter was attributed to variations of loading history on individual test cartons and the stiffening effect of cartons material crushing.

mass each. (That is two sprung lading masses in each trailer.) The top 1/3 of each six stack model was separated as the sprung mass with its center of gravity assumed to be at is geometric center.

The springs and dampers were sized so that the uncoupled natural frequencies and the amplification factors were as follows:

Lateral Resonance:	f = 5.0 Hertz	Q = 3.3
Vertical Resonance:	f = 9.5 Hertz	Q = 7.9
Roll Resonance:	f = 15.0 Hertz	Q = 8.3

Freight Car Model

The flatcar and trailer models are the same as presented in Reference 5 with the changes discussed above. The model has 11 lumped masses including the four spring mounted lading masses. Car body flexibility is included with a modal superposition method using four carbody normal modes, bringing the total degrees of freedom to 43. The TOFC/Lading configuration is shown schematically in Figures A3, A4 and A5. Figure A3 defines the notation used for masses, inertias and degrees of freedom; Figure A4 defines geometry notation; and Figure A5 defines spring-damper.

Mass and inertia values used are shown in Table AII. Spring and damper values are shown in Tables AIII and AIV. Dimensional values are given in Table AV.

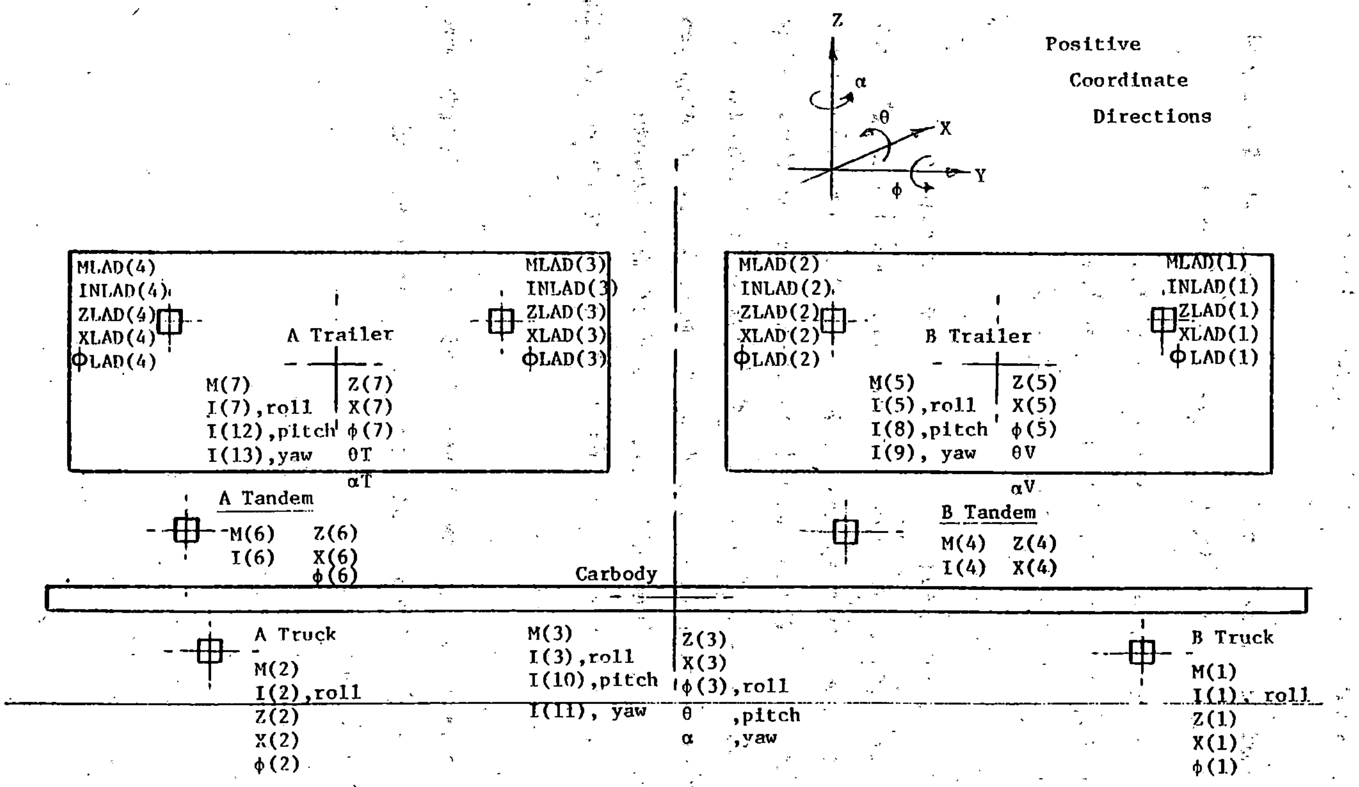


FIGURE A-3
FRATE/TOFC MASS, INERTION AND DEGREE OF FREEDOM NOTATION

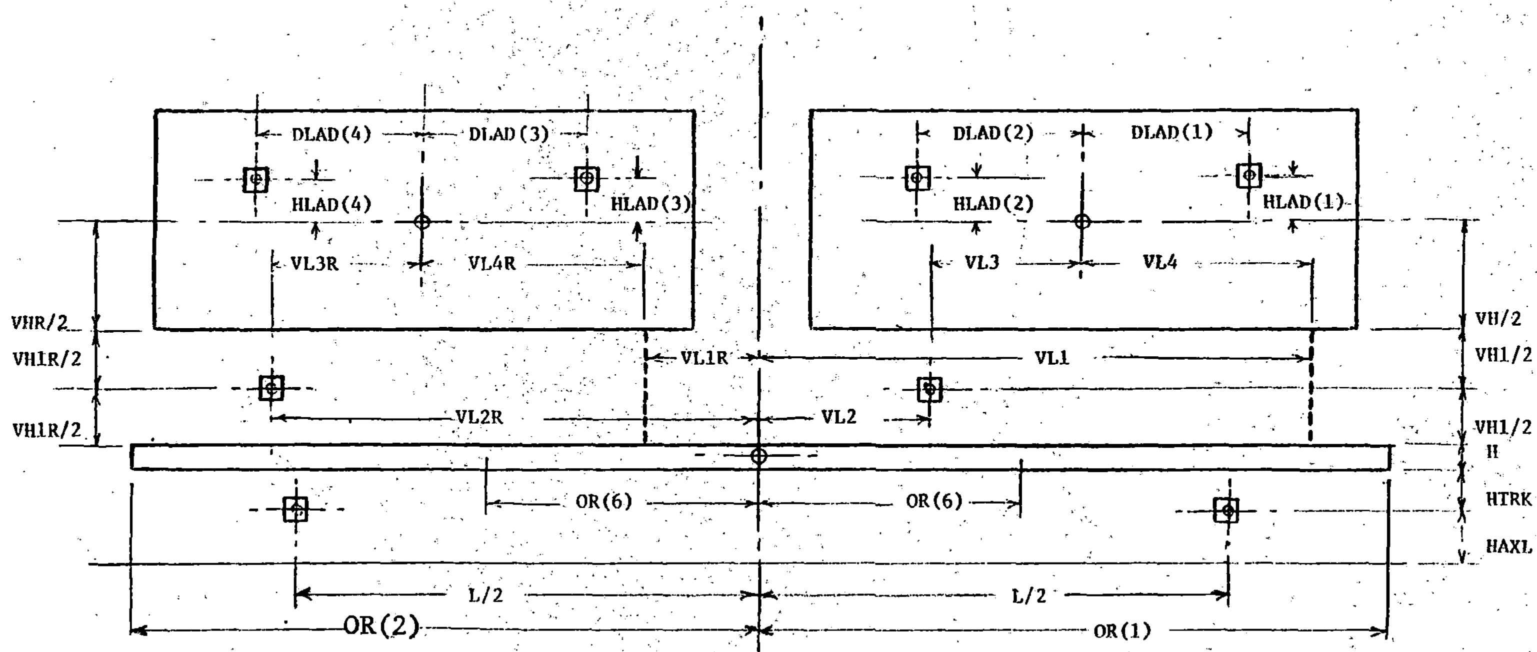


FIGURE A-4
FRATE/TOFC GEOMETRY NOTATION

Notes for Figure A-4

Horizontal Geometry Notation for FRATE/TOFC is as follows:

R(1)	=	wheel gage distance, B truck
R(2)	=	distance between spring nest centroids, B truck
R(3)	=	wheel gage distance, A truck
R(4)	=	distance between spring nest centroids, A truck
R(5)	=	effective width of tandem tread of B end trailer
R(6)	=	effective width between tandem spring centroids of B end trailer
R(7)	=	effective width of tandem trend of A end trailer
R(8)	=	effective width between tandem spring centroids of A end trailer
RLAD(I)	=	x distance of lading mass I cg from trailer center line

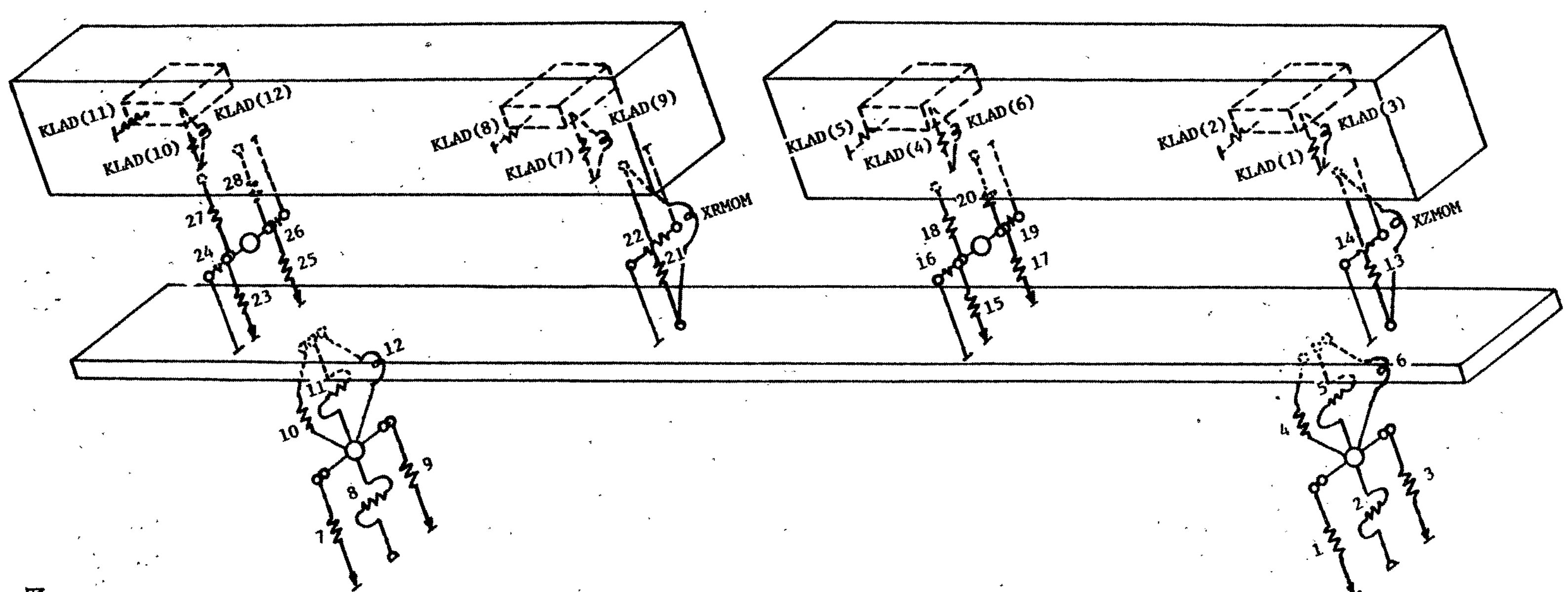


FIGURE A-5
FRATE/TOFC SPRING-DAMPER NOTATION

TABLE AII

TOFC MASS PROPERTIES WITH TWO IDENTICAL VAN TRAILERS
EACH HAVING FLEXIBLY MOUNTED LADING MASSES

Symbol	Description	Value (mass units)	Weight Units
M(1), M(2)	B Truck Mass, A Truck Mass	22.33	8619.
M(3)	Carbody Mass (Empty)	125.00	48250.
M(4), M(6)	Tandem Masses, B and A Trailers	8.179	3157.
M(5), M(7)	Trailer Body Masses, B and A[(1)]	119.6	46166
MLAD(1)-MLAD(4)	Flexibly mounted lading masses (4)	16.56	6392
I(1), I(2)	Truck Roll Inertias, B and A	.02208E6	8.523E6
I(3)	Carbody Roll Inertia	.1085E6	41.88E6
I(10)	Carbody Pitch Inertia	15.00E6	5790.E6
I(11)	Carbody Yaw Inertia	15.00E6	5790.E6
I(4), I(6)	Tandem Roll Inertia, Trailers B and A	.0200E6	7.72E6
I(5), I(7)	Trailer Body Roll Inertias, B and A[(1)]	.213E6	82.22E6
I(8), I(12)	Trailer Body Pitch Inertias, B and A[(1)]	1.926E6	7434.E6
I(9), I(13)	Trailer Body, Yaw Inertia, B and A[(1)]	2.085E6	8048.E6
INLAD(1)-INLAD(4)	Roll Inertia of Sprung lading masses	.01337E6	5.161E6

(1) Trailer body mass and inertia values include that portion of lading assumed to be a rigid, integral part of the trailer.

Units: M(I) = lb. sec.^{2}/in. or lb.
I(I) = lb. in. sec.^{2}, or lb. in.^{2}

TABLE AIII

TOFC SPRING AND DAMPER VALUES, FLATCAR TRUCKS

Spring/Damper Number	Spring Constant (lb/in)	Viscous Damping (lb/sec/in)	Damping Ratio Estimate (C/C_c)†	Structure or Element Represented
K(1),K(3),K(7),K(9)	.91E5	300.	.020	Side frame, wheels and track, vertical. Two per truck
K(2), K(8)	.10E5	333.	.209	Side frame, wheels and track, lateral, without frange contact
KFC2, KFC8	.95E5	333.	.022	Side frame, wheels and track, lateral, with flange contact
K(4), K(10)	.48E5	140.	.018	Truck spring-vertical
KS4, KS10	.24E6	0	0	Structure local to truck friction snubbers
KCP6, KCP12	.20E8	.2DE6	.063	Truck roll before side bearing contact
K(6), K(12)*	.618E8	.30E6	.030	Truck roll after side bearing contact
KS6, KS12*	.34E9	0	0	Structure local to truck friction snubbers
K(5), K(11)	.16E5	200.	.079	Truck lateral
MFS4 MFS10	--	6000 (lb)		Snubber vertical friction load
MFS6 MFS12	--	.237E6(in.lb.)		Snubber roll friction moment

*Angular springs and dampers - units are in.lb./radian and in.lb./radian/sec.
†C/C_c is estimated by assuming $C_c = K/\pi f$ and $f = 2.0$ Hertz.

TABLE AIV

TOFC SPRING AND DAMPER VALUES, HIGHWAY TRAILERS

NUMBER	SPRING CONSTANT K lb/in	VISCOUS DAMPING CONSTANT C lb/sec/in	DAMPING RATIO C/C_c †	REPRESENTING
13,21	.225E6	1000.	.028	Trailer hitch vertical
14,22	.150E5	200.	.084	Trailer hitch lateral
15,17,23,25	.225E5	300.	.092	Tandem tires, vertical, 2 per tandem
16,24	.180E5	200.	.070	Tandem tires, lateral 1 per tandem
18,20,26,28	.5276E5	775.	.092	Tandem springs vertical, 2 per tandem
19,27	.180E5	200.	.070	Tandem springs lateral, 1 per tandem
XZMOM* XRMOM	.30E8	.10E6	.021	Trailer hitch, roll moment

* Angular spring and dampers, units are inch pounds per radian and inch pound seconds per radian.

† See Table AIII.

TABLE AV

TOFC DIMENSION VALUES

(Units are inches except as noted)

R(1) = R(3) = 58.00
R(2) = R(4) = 79.00
R(5) = R(7) = 62.25
R(6) = R(8) = 43.50

L = 792.0	
VL1 = 469.0	VL1R = -85.0
VL2 = 148.0	VL2R = -411.0
VL3 = 129.4	VL3R = 129.4
VL4 = 191.7	VL4R = 191.7

H = 16.0		
HTRK = 9.0	VH = 49.6	VHR = 49.6
HAXL = 16.5	VH1 = 47.0	VH1R = 47.0

OR(1) = 536.0	HL = 30.27
OR(2) = -536.0	
OR(3) = -39.0	
OR(6) = 224.0	
OR(7) = 226.0	
OR(8) = 254.0	
OR(9) = 245.0	
OR(10) = 235.0	

GAPB = .01 radians	DFC2 = .60 (DFC = ½ gage clearance)
GAPA = .01 radians	DFC8 = .60

DLAD(I) = 156.8, -131.2, 156.8, -131.2
RLAD(I) = 0., 0., 0., 0.
HL(I) = 30.27 30.27, 30.27, 30.27
BX = 48
BY = -72
BZ = 13.5

Note: GAP angle is based on side bearing clearance at 25 inches from carbody centerline and assumes equal gap on each side. (.01 radians = .25 inch gap.)

APPENDIX B

LIST OF REFERENCES

1. Kachadourian, G., N. E. Sussman and J. R. Anderes, FRATE Volume 1: User's Manual, NTIS Report No. FRA/ORD-78/59, September 1978.

2. Code of Federal Regulations, Title 49 Transportation, Part 213, Track Safety Standards and Part 215, Railroad Freight Car Safety Standards.

3. Renolds, D. J., Hunting in Freight Cars, ASME Publication 74-RT-2.

4. Wickens, A. H., The Dynamics of Railway Vehicles on Straight Track, Proceedings of the Institution of Mechanical Engineers, Vol. 180, 1965-1966.

5. Kachadourian, G. and N. E. Sussman, Validation of FRATE, Freight Car Response Analysis and Test Evaluation, The MITRE Corporation, Report No. MTR-8007, December 1978.

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