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Response Analyses of a Boxcar with Compliant Lading for Several Track Profile and Hunting Conditions

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TECHNICAL REPORT

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ABSTRACT

The simulation in the FRATE computer program (Freight Car Response Analysis and Test Evaluation) was modified from a flexible TOFC (trailer on flatcar) to a rigid boxcar with compliant lading. Analyses were performed to obtain the response of boxcar elements and compliant lading to several track profile and body hunting conditions. Three types of track profile irregularities were simulated: (1) a single vertical irregularity on both rails, (2) a single vertical irregularity on one rail and (3) rectified sine representations of staggered joint bolted rail. Hunting conditions were simulated by imposing sinusoidal lateral motions at the wheel rail interface. Worst case conditions resulted in wheel-rail separations and in potentially damaging accelerations of the lading.

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METRIC CONVERSION FACTORS

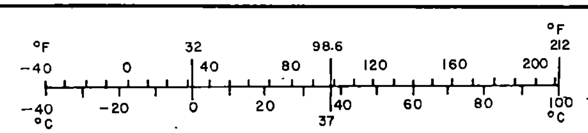
Approximate Conversions to Metric Measures

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 simulation in the FRATE computer program (Freight Car Response Analysis and Test Evaluation) was modified from a flexible TOFC (trailer on flatcar) to a rigid boxcar with compliant lading. Analyses were performed to obtain the response of boxcar elements and compliant lading to several track profile and body hunting conditions. Three types of track profile irregularities were simulated: (1) a single vertical irregularity on both rails, (2) a single vertical irregularity on one rail and (3) rectified sine representations of staggered joint bolted rail. Hunting conditions were simulated by imposing sinusoidal lateral motions at the wheel rail interface. Worst case conditions resulted in potentially damaging accelerations of the lading and wheel-rail separations.

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1.0 INTRODUCTION

Under the sponsorship of the Federal Railroad Administration (FRA) MITRE has performed a series of dynamic studies related to railroad freight cars. Previous work has concentrated on the establishment of a viable, validated analysis which could be used with confidence in calculating the dynamic response of a single freight car to track irregularities. The result of this effort is an analysis which is realized in a digital computer program; this program has been named FRATE for Freight Car Response Analysis and Test Evaluation. A users manual[1*] and a validation report[2] have been issued describing the basic features of this computer program.

At the time References 1 and 2 were issued, FRATE simulated a trailer-on-flatcar (TOFC) configuration. However, the program had the option to remove one or both trailers so that, with appropriate mass and dimensional parameter values, any freight car could be simulated. In order to be able to include the effects of lading dynamics FRATE was modified to include up to 4 spring supported lading masses. Using the new FRATE with lading program two sets of analyses have been run: one with a boxcar simulation which is the subject of this report and a second with a TOFC simulation which is reported in Reference 3. Other enhancements to FRATE have occurred over the past two years. Of most significance for this study are the various options available for exciting the vehicle at the wheel/rail interface and the output options.

The primary emphasis of this study is the response of the lading to various track irregularities of both transient and steady state nature. Responses of the vehicle are also noted and documented, especially those that indicate excessive acceleration and/or forces.

* References are located in Appendix D.

The lading consists of a portion which is capable of motion relative to the boxcar structure and a portion which is not.

The number of degrees of freedom of this simulation is 23, and reflects the concern that a proper balance be achieved between modeling complexity, accuracy of input data and computer costs. Overly complex simulation of the lading could not be warranted because of the relative inaccuracy of the data feeding the lading model and because of the potentially large computer costs incurred by time domain models of many degrees of freedom.

2.0 DESCRIPTION OF THE SIMULATION

2.1 Vehicle

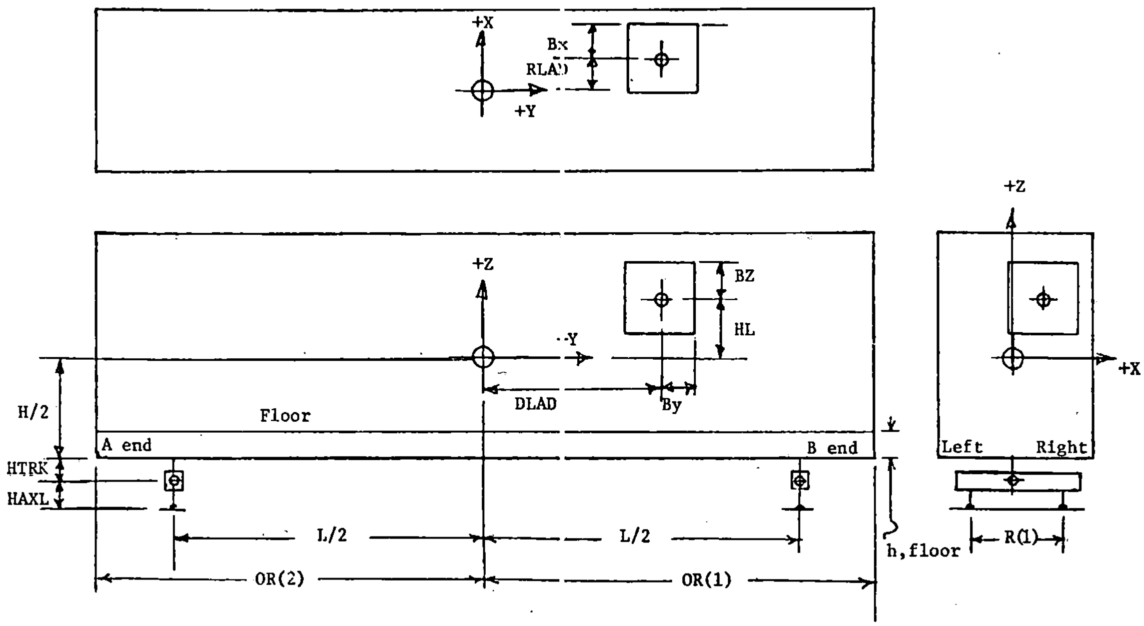
2.1.1 Boxcar

The boxcar is assumed to be a rigid body containing both rigid and compliant lading mounted on two trucks. Figure 1 gives the fundamental dimensions assumed for this study. The trucks are modeled as shown in Figure 2 which is taken from Reference 1. The combination of boxcar and lading is symmetric along the longitudinal and lateral axes. The center of gravity (c.g.) of the combined boxcar and lading is located midway between the trucks and is on the longitudinal center line.

The boxcar structure is assumed to be a six-sided hollow box with all but the bottom of equal thickness. Reference 4 provides these thicknesses, the interior dimensions and the c.g. location of the empty boxcar, and the associated moments of inertia.

The values of all boxcar and truck parameters required as input data for the simulation are summarized in Tables I and II. Model 1 has 112842 pounds of cargo which is roughly 72 percent of the maximum allowable load and was used as a typical loading condition. Model 2 has 38814 pounds of cargo, roughly 25 percent allowable and was used in the hunting simulation since service experience have shown light weight configurations more likely to be involved in body hunting conditions.

There are seven lumped masses in the model and a total of 23 degrees of freedom. The masses are listed in Table III along with the degrees of freedom permitted for each.



L = 474
R(1) = 58.0 (B truck)
R(3) = 58.0 (A truck)
OR(1) = 303
OR(2) = -303
H = 74.3 (Model 1)
H = 42.36 (Model 2)
HTRK = 9.0
HAXL = 16.5
GAPA = .01 radians
GAPA = .01 radians
Boxcar Inside Dimensions
  length = 606
  width = 114
  height = 125

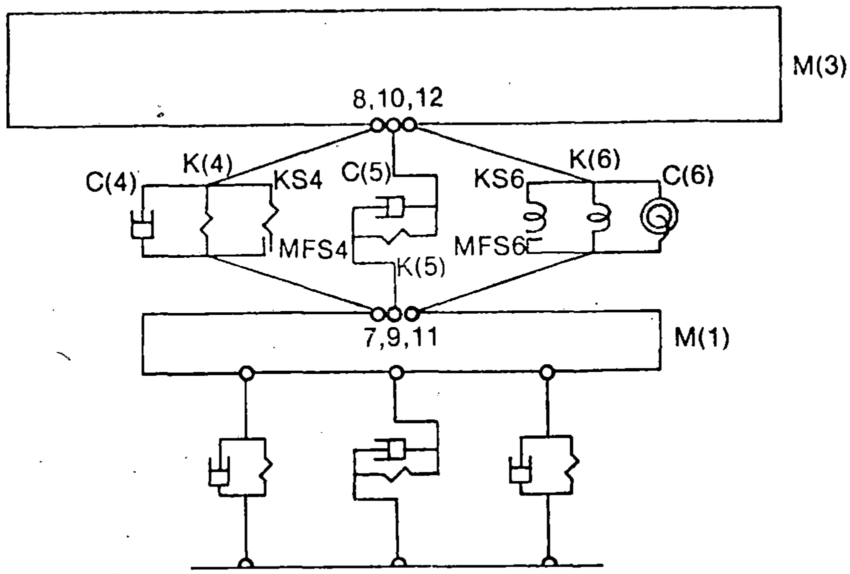
RLAD(I) = 0., 0., 0., 0.,
DLAD(I) = 262.4, 60.6, -60.6, -262.4
HL(I) = 42.8, 42.8, 42.8, 42.8, (Model 1)
HL(I) = 12.0, 12.0, 12.0, 12.0 (Model 2)
BX = 51.0
BY = 40.2
BZ = 15.5

h, floor = 43.8

Notes:

1. All dimensions are in inches except where noted otherwise.
2. The boxcar height, H, is a synthetic number equal to two times the distance from the center plate plane to the carbody + rigid lading c.g.

FIGURE 1
BOXCAR MODEL DIMENSIONS



Legend: K(4) = total vertical spring rate for B truck
C(4) = viscous damper in paralled with K(4)
MFS4 = vertical friction damping force
KS4 = spring in series with MFS4
K(6) = total, bilinear, roll moment spring rate for B truck
= KCP6 before side bearing contact
= K(6) after side bearing contact
C(6) = viscous bilinear damper in parallel with K(6)
= CCP6 before side bearing contact
= C(6) after side bearing contact
MFS6 = angular friction damping moment activated after side bearing contact
KS6 = roll moment spring constant in series with MFS6
M(1) = mass of B Truck
M(3) = carbody mass
K(5) = lateral stiffness of truck primary suspension
C(5) = viscous damper in parallel with K(5)

FIGURE 2
B TRUCK MATH MODEL SCHEMATIC, TYPICAL OF A TRUCK

TABLE I

BOXCAR MASS PROPERTIES

SYMBOL	MODEL 1	MODEL 2	ELEMENT	UNITS
M(1), M(2)	22.33	22.33	B Truck, A Truck Mass	lb. Sec^2/in
M(3)	350.2	199.3	Carbody Mass (including rigid lading)	lb. Sec^2/in
MLAD(1) -MLAD(4)	15.6	5.36	Complaint Lading Masses	lb. Sec^2/in
I(1), I(2)	.2208E5	.2208E5	B Truck, A Truck roll inertia	lb. in. Sec^2
I(3)	.821E6	.514E6	Carbody and Rigid Lading Roll Inertia	lb. in. Sec^2
I(10)	.111E8	.658E7	Carbody and Rigid Lading Pitch Inertia	lb. in. Sec^2
I(11)	.111E8	.656E7	Carbody and Rigid Lading Yaw Inertia	lb. in. Sec^2
INLAD(1) -INLAD(4)	.1477E5	.47E4	Complaint Lading Roll Inertia	lb. in. Sec^2

TABLE II

BOXCAR MODEL SPRING AND DAMPER VALUES

SPRING/DAMPER NUMBER	MODEL 1 K	MODEL 1 C	MODEL 2 K	MODEL 2 C	STRUCTURE OR ELEMENT REPRESENTED	UNITS
K(1), K(3), K(7), K(9)	.91E5		.91E5		Side frame, wheels and track	lb/in
C(1), C(3), C(7), C(9)		300.		300.	Vertical. Two per truck.	lb/in/sec
K(2), K(8)	.95E5		.35E6		Sideframe, wheels and track	lb/in
C(2), C(8)		333.		333.	Lateral. One per truck.	lb/in/sec
K(4), K(10)	.48E5		.48E5		Truck spring and viscous damper	lb/in
C(4), C(10)		140.		140.	Vertical. One per truck	lb/in/sec
KS4, KS10	.24E6		.24E6		Spring in series with friction damper	lb/in
MFS4, MFS10		6000.		3000.	Friction Damper	lb
K(5), K(11)	.16E5		.67E4		Truck. lateral. one per truck	lb/in
C(5), C(11)		400.		400.		lb/in/sec
KCP6, KCP12	.20E8		.20E8		Truck spring and viscous damper	in lb/rad
CCP6, CCP12		.20E6		.20E6	Roll without side barring contact	in lb/rad/sec
K(6), K(12)	.6185E8		.6185E8		Truck spring and viscous damper	in lb/rad
C(6), C(12)		.30E6		.30E6	Roll with side bearing contact	in lb/rad/sec
KS6, KS12	.34E9				Spring in series with friction damper	in lb/rad
MFS6, MFS12		.237E6		.1185E6	Truck roll friction damper	in lb
KLAD(1), (4), (7), (10)	.556E5	223.	.191E5		Lading Vertical Spring	lb/in
CLAD (1), (4), (7), (10)				76.8	Lading Vertical Damping	lb/in/sec
KLAD (2), (5), (8), (11)	.154E5		.529E4		Lading Lateral Spring	lb/in
CLAD (2), (5), (8), (11)		78.4		27.0	Lading Lateral Damping	lb/in/sec
KLAD (3), (6), (9), (12)	.131E9		.418E8		Lading Roll Spring	in lb/rad
CLAD (3), (6), (9), (12)		.167E6		.532E5	Lading Roll Damping	in lb/rad/sec

TABLE III

MODEL DEGREES OF FREEDOM

LUMPED MASS	DEGREES OF FREEDOM INCLUDED LATERAL (X)	VERTICAL (Z)	PITCH (θ)	ROLL (ϕ)	YAW (α)
M(1) B Truck	Yes	Yes	No	Yes	No
M(2) A Truck	Yes	Yes	No	Yes	No
M(3) Carbody	Yes	Yes	Yes	Yes	Yes
MLAD (1) - MLAD (4) (compliant lading)	Yes	Yes	No	Yes	No

(23 degrees of freedom total)

2.1.2 Lading

The lading model is based on test results[3] of pallet mounted stacks of canned goods in cardboard shipping cartons. It is assumed that 30 such pallet stacks are arranged in two parallel rows in the boxcar. Based on previous work the top 40 percent of specified stacks are considered to be compliant and dynamically responsive. Appendix A provides a complete summary of the development of the lading model, the car body/rigid lading moments of inertia and the total mass.

2.2 Excitation

Excitation, as defined in this report, is the input motion, vertical and/or lateral, imposed at the wheel-rail interface. These input motions are defined as either transient or steady state.

2.2.1 Transient Excitation

The transient excitations used here are representations of isolated track irregularities, which are caused by such things as road crossings, switch blocks, road bed soft spots and track deterioration. These irregularities can be vertical, up or down (hump or dip), can be lateral (alignment) and can be one rail or both rails.

A (1-COS) function was used to simulate the shape of the irregularity. The amplitude of the irregularity was taken from track class allowables given in Title 49 Transportation Code of Federal Regulations[5] and shown here in Table IV. A time delay was included between the applications of the transient to the leading truck and the trailing truck. This time delay depends on train speed and truck spacing. Train speed was also varied in conjunction with the length of the track irregularity to obtain maximum response. The 1-cos shape and equations are shown in Figure 3.

TABLE IV-a
ALLOWABLE TRACK DEVIATIONS: TITLE 49 (REF 5) TANGENT TRACK

Max. Allowables	Class of Track 1	2	3	4	5	6
Speed, MPH	10	25	40	60	80	110
Profile Deviation - 62'chord, inches	3	2 3/4	2 1/4	2	1 1/4	1/2
Cross Level differences 62' chord, inches	3	2	1 3/4	1 1/4	1	5/8
Gage - inches (Min.=56" all classes)	57 3/4	57 1/2	57 1/2	57 1/4	57	56 3/4
Alignment, inches	5	3	1 3/4	1 1/2	3/4	1/2

TABLE IV-b
GAGE CLEARANCE ESTIMATES (inches)

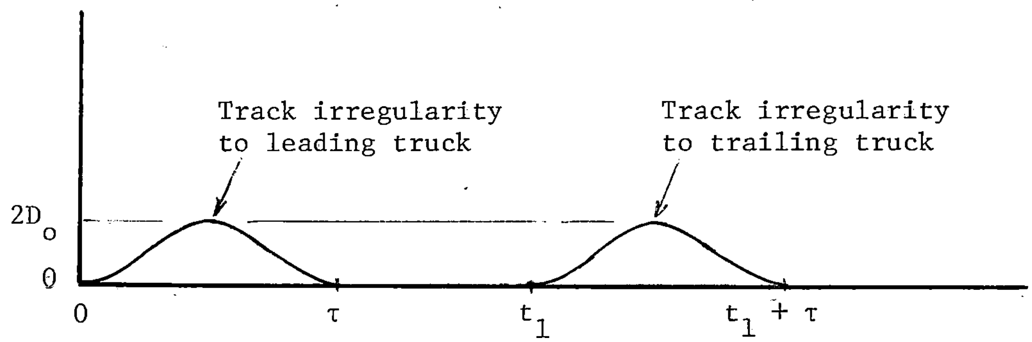
Condition	Class of Track 1	2	3	4	5	6
Min. (new wheel)	3/16	3/16	3/16	3/16	3/16	3/16
Max. (new wheel)	2 1/2	2 1/4	2 1/4	2	1 3/4	1 1/2
Max. (worn wheel)	2 7/8	2 5/8	2 5/8	2 3/8	2 1/8	1 7/8
Avg. (worn wheel)	1.53	1.41	1.41	1.28	1.16	1.03

The data in this table is based on track allowables as given here in Table IVa and wheel dimensions and allowables from Ref. 5 and AAR standard car dimensions as follows:

Wheel set flange point to flange point distance.

minimum allowable = 55 1/4 inches (new wheel)
= 54 7/8 inches (worn wheel)

maximum allowable = 55 13/16 inches (new wheel)



<u>Leading Truck</u>

$Zi = D_o \left(1 - \cos \frac{2\pi}{\tau}\right)$ if $t > 0$ and $< \tau$

$Zi = 0$ if $t \leq 0$ or $t \geq \tau$

<u>Trailing Truck</u>

$Zi = D_o \left(1 - \cos \frac{2\pi}{\tau}(t-t_1)\right)$ if $t > t_1$ and $< (t_1 + \tau)$

$Zi = 0$ if $t \leq t_1$ or $t \geq (t_1 + \tau)$

where:

Zi = input motion , inches

D_o = single amplitude of cosine function, inches

τ = duration of track irregularity, seconds

= (irregularity length)/(speed)

t_1 = time trailing truck reaches track irregularity, seconds

= (truck spacing)/(speed)

FIGURE 3
(1-cos) SHAPED TRACK IRREGULARITY

It was expected that the speed of maximum response would occur when the time delay between trucks is a factor of the vehicle period as follows:

for bounce or roll resonant frequencies

$$L/V = NT$$

$$\text{or } V = L \times f/N$$

and for pitch or yaw resonant frequencies

$$L/V = (N + .5)/T$$

$$\text{or } V = L \times f/(N + .5)$$

where V = speed, feet per second

L = period of vehicle resonant frequency, seconds

= 1/f

N = integer

= 1, 2, 3, etc. for bounce or roll resonances

= 0, 1, 2, etc. for pitch or yaw resonances

(Note: Distinction is not made here between natural and resonant frequencies. In a time domain analysis we are restricted to working with resonant frequencies. Uncoupled natural frequencies calculated to estimate resonant frequencies are therefore referred to as vehicle resonant frequencies.)

The length of the irregularity was also varied to obtain maximum response in the analysis vehicle. First estimates of irregularity length were based on single degree of freedom transient response theory that shows maximum response occurs when τ, the input function time duration, equals about 0.8 x T, the period of the single degree of freedom natural frequency. Based on this the following relationships were used for a first estimate on irregularity length:

$$PD = .8\ T$$

$$T = 1/f$$

$$PL = V * PD$$

$$PL = .8 * V/f$$

where

PD = duration, seconds

PL = track irregularity length, feet

T = period of vehicle resonant frequency, seconds

f = vehicle resonant frequency, Hertz

V = speed, feet per second

2.2.1.1 Vertical Coincident Pulse

For example, consider a coincident vertical pulse, that is a hump in the track where both rails have the same shape side by side. There are two boxcar frequencies which will be most responsive, the first bounce frequency at about 2.2 Hertz and the first pitch frequency at about 2.8 Hertz. The following first approximations were made:

First Bounce

$$f = 2.2 \text{ Hertz}$$

$$L = 474/12 = 39.5 \text{ feet}$$

if $N = 1$ is used:

$$V = L * f/1 = 86.9 \text{ fps} = 59.2 \text{ mph}$$

and $PL = .8 * V/f = 31.6$ feet

or if N = 2 is used:

V = 43.45 fps = 29.6 mph

and PL = 15.8 feet

use N = 1

First Pitch

f = 2.8 Hertz

L = 39.5 feet

for N = 0

V = L * f/.5 = 221.2 fps = 151 mph

PL = 63.2 feet

or for N = 1

V = L * f/1.5 = 73.7 fps = 50.3 mph

PL = 21.1 feet

use N = 1

2.2.1.2 Cross Level Pulse

The cross level pulse of this report is a simulation of an irregularity occurring in one rail. A 1-COS shape is used as in the vertical coincident pulse discussed above. The first approximation for worst case track speed and pulse length, was made in the same manner discussed in the previous section, using the resonant frequencies estimates for the first and second car body roll.

Low Center Boxcar Roll

f = 1.05 Hertz

V = 39.5 * 1.05 = 41.5 fps = 28.5 mph

PL = .8 * 41.5/1.05 = 31.6 feet

High Center Boxcar Roll

$$f = 1.79 \text{ Hertz}$$

$$V = 39.5 * 1.79 = 70.7 \text{ fps} = 48.2 \text{ mph}$$

$$PL = .8 * 70.7/1.79 = 31.6 \text{ feet}$$

2.2.2 Steady State Excitations Caused by Jointed Rail

The use of a rectified sine curve to simulate the vertical profile of jointed rail is well documented. This study incorporates a 39-foot jointed rail as a steady state excitation function. This excitation function is used to excite low center rock and roll at speeds of 15 to 20 mph. Also, simulations using it were performed to develop lading acceleration responses at high speeds.

Two types of simulations were performed. One was to assume a constant forward speed (constant input frequency) and the other to assume a constant decrease in forward speed to simulate braking. This latter simulation was performed because experimental and service experience has shown that rock and roll motions are generally greatest when decelerating through the critical speed. This phenomenon is evidence of a nonlinear softening spring.

Phasing, right rail to left rail, was maintained at 90° since it was assumed that the 39-foot jointed rail was staggered. Also, phasing, front truck to back truck, allowed for the wheel base of 39.5 feet. Therefore, at each of the four wheel rail interfaces a different value of vertical excitation was applied at each time interval, the values dependent on the phasing. Consequently, as one would expect, the car body response contained all modes of body motion with rock and roll predominating at the resonant speed.

Three deceleration values were used: 2.9, 1.33 and .53 mph/sec. which correspond to .055, .025 and .010 Hertz/sec. rate of change of the rectified sine wave with 39 foot rail. The 2.9 mph/sec. value, as shown in Figure 4, represents a design limit. The .025 and .010 Hertz/sec. rates were used as the typical operational range.

Constant frequency (or constant speed) runs were made over the range of 10 to 70 mph to develop car body and lading acceleration responses across the speed spectrum. The time duration of these runs was 9 seconds, which was a long enough time for responses to reach steady state conditions. Time history plots were made by allowing 3 seconds to elapse to let the starting transient die out, and plotting the responses for the remaining 6 seconds.

2.3 Simulation of Truck Hunting

The purpose of the hunting simulations performed for this report was to calculate the lading and boxcar responses to assumed hunting conditions. Since hunting usually occurs under lightly loaded cars, parameters for canned goods in corrugated shippers were developed for a partial load. The resulting configuration had about 24 percent of maximum load capability and was called Model #2. There are two kinds of car body motions which will generally couple with truck hunting motion to form the conditions known as body hunting. These are car body high center roll motions and car body yaw motions. The uncoupled natural frequencies for these two modes of motions were estimated as follows:

High Center Roll

$$f = \frac{1}{2\pi} \sqrt{\frac{K}{I}} = 2.46 \text{ Hertz}$$

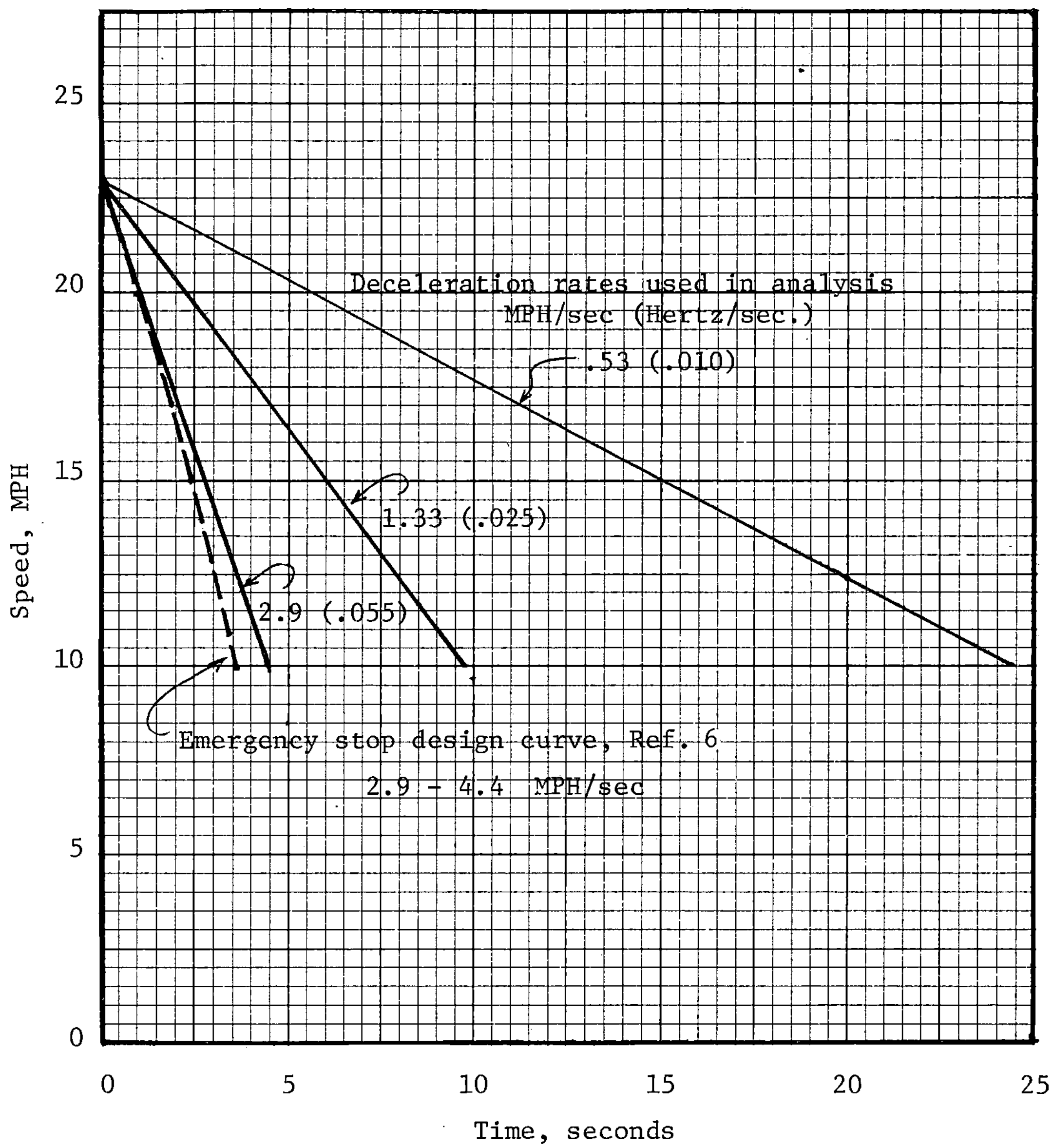


FIGURE 4
DECELERATION CURVES USED IN ROCK AND ROLL SIMULATION

where:

It is assumed that the car body is rotating about its center of gravity

$$K = 2 * (K(5) * (H/2 + HTRK)^2 + K(6))$$

$$= 1.358E8 \text{ in. lb./radian}$$

$$I = I(3) * \frac{M(3) + 4MLAD}{M(3)}$$

$$= .569E6 \text{ lb. in. sec.}^2$$

<u>Car Body Yaw</u>

$$f = \frac{1}{2\pi} \sqrt{\frac{K}{I}} = 1.62 \text{ Hertz}$$

$$\text{when } K = 2 * K(5) * (474/2)^2$$

$$= 7.5266E8 \text{ in. lb./radian}$$

$$I = I(11) * \frac{M(3) + 4\ MLAD}{M(3)}$$

$$= .726E7 \text{ lb.in. sec.}^2$$

The hunting condition is simulated by forcing lateral sinusoidal motions of the truck (or trucks). Since input motions in the analysis are imposed at the rail it is desirable to have K(2) and K(8) much stiffer than K(5) and K(11). (Where K(2) and K(8) are the lateral springs between ground and trucks Band A respectively and K(5) and K(11) are the lateral springs between the trucks and car body). Consequently K(2) and K(8) were increased from .95E5 to .35E6 lb./in. to become almost 22 times stiffer than K(5) and (K(11) which are .16E5 lb./in. Thus with motion imposed at the rail essentially all of the compliance is in K(5) and K(11) and the truck lateral motions are the same as the input motions.

The kinematic frequency of hunting can be approximated with the following empirical relationship from D. J. Reynolds.[7]

$$f_k = \frac{V}{2\pi \times 8.22} \text{ for new 33-inch wheels}$$

$$f_k = \frac{V}{2\pi \times 4.11} \text{ for worn 33-inch wheels}$$

Then the speed ranges were the hunting kinematic frequency and the car body yaw and high center roll frequencies coincide are as follows:

Car Body Yaw

$$\begin{aligned} f &= 1.62 \text{ Hertz} \\ V &= 2\pi f * 8.22 \\ &= 83.67 \text{ fps} \\ &= 57.0 \text{ mph (new wheels)} \\ \text{and } V &= 28.5 \text{ mph (worn wheels} \end{aligned}$$

Car Body High Center Roll

$$\begin{aligned} f &= 2.46 \\ V &= 127.05 \text{ fps} \\ &= 86.6 \text{ mph (new wheels)} \\ \text{and } V &= 43.3 \text{ mph (worn wheels)} \end{aligned}$$

A final factor to be accounted for is the critical hunting speed. For the car body yaw case, in order for body hunting to occur the critical hunting speed must be in the range of 55-65 mph for new wheels and 30-55 mph for worn wheels. For the high center roll case, the critical hunting speed must be in the range of 80-90 mph for new wheels and 40-80 mph for worn wheels. Consequently, we can conclude that body hunting will occur in either the yaw or high center roll mode but the high center roll condition will occur more frequently.

2.3.1 Amplitude and Duration of Hunting Motion

It is generally agreed, that at or near the critical hunting speed the truck will oscillate from flange contact on one rail to flange contact on the other rail in a roughly sinusoidal path (see References 7 and 8). Consequently, the input motions used in the analyses of this report were assumed to be related to gage clearance.

An estimate of the range of gage clearance which can be expected in service is shown in Table IV b. The average values for track classes 2, 3, 4 and 5 were increased by 25 percent to be representative of a worst case service condition and were used in the hunting simulation anslyses. The resulting values are as follows:

Class of Track	2 and 3	4	5
Total Gage Clearance, Inches	1.7	1.6	1.4
Maximum Speed for Track Class, mph	40	60	80

The typical hunting motions will consist of short sequences of several cycles of the lateral oscillatory motions at subcritical speeds. As the critical hunting speed is approached the oscillations will be sustained for a greater number of cycles in each burst and when the critical speed is reached the oscillations will become a sustained oscillation. For the purposes of this report our objective is to determine the magnitudes of the carbody motions and lading loads in a hunting condition. Consequently, it was most meaningful to assume flange to flange motions for long enough durations to reach steady state conditions. This was generally found to be within 10 cycles, which is a realistic figure with respect to hunting motions found in service.

3.0 RESULTS

3.1 Response to Vertical Coincident Pulse

In this analysis the boxcar was caused to transverse a hump in the track roadbed simulated by a 1-cosine shape on each rail as shown in Figure 3. The hump length and track speed were tuned for two maximum response conditions: (1) the vertical bounce mode at about 2.2 Hertz and (2) the pitch mode at about 2.8 Hertz. The height of the hump was 2.0 inches which is the track profile deviation permitted for class 4 track (40-60 mph) by the Track Safety Standards of Reference 5.

Figures 5 through 8 are presented as representative time history response plots. These results are from the computer run which maximizes the vertical response. Figure 5 is the time history of the pitch deflection of the carbody showing that even though this was a run with maximum condition of vertical motion there was also a lot of pitching motion. This coupled vertical and pitch motion is also seen in Figure 6 which plots time histories of vertical acceleration of the carbody at each truck. The vertical acceleration time history of each of the four compliant lading masses is shown in Figure 7. The maximum acceleration of 2.26 g's predicted for the carbody and 2.35 g's for lading would probably lead to lading damage.

Figure 8 presents the time history of vertical wheel-rail forces for one side of each truck. The wheel forces on the trailing truck, the A truck in this case, reach the highest and lowest values; there is separation at one point and the force actually exceeds twice the static force at another.

Figure 9 shows the effect of variations of the pulse length. For very short pulse lengths the lading response is seen to peak up to about 3.6 g's. Although this is an unusually severe g load for lading, pulse lengths this short (less than 10 feet), are not likely

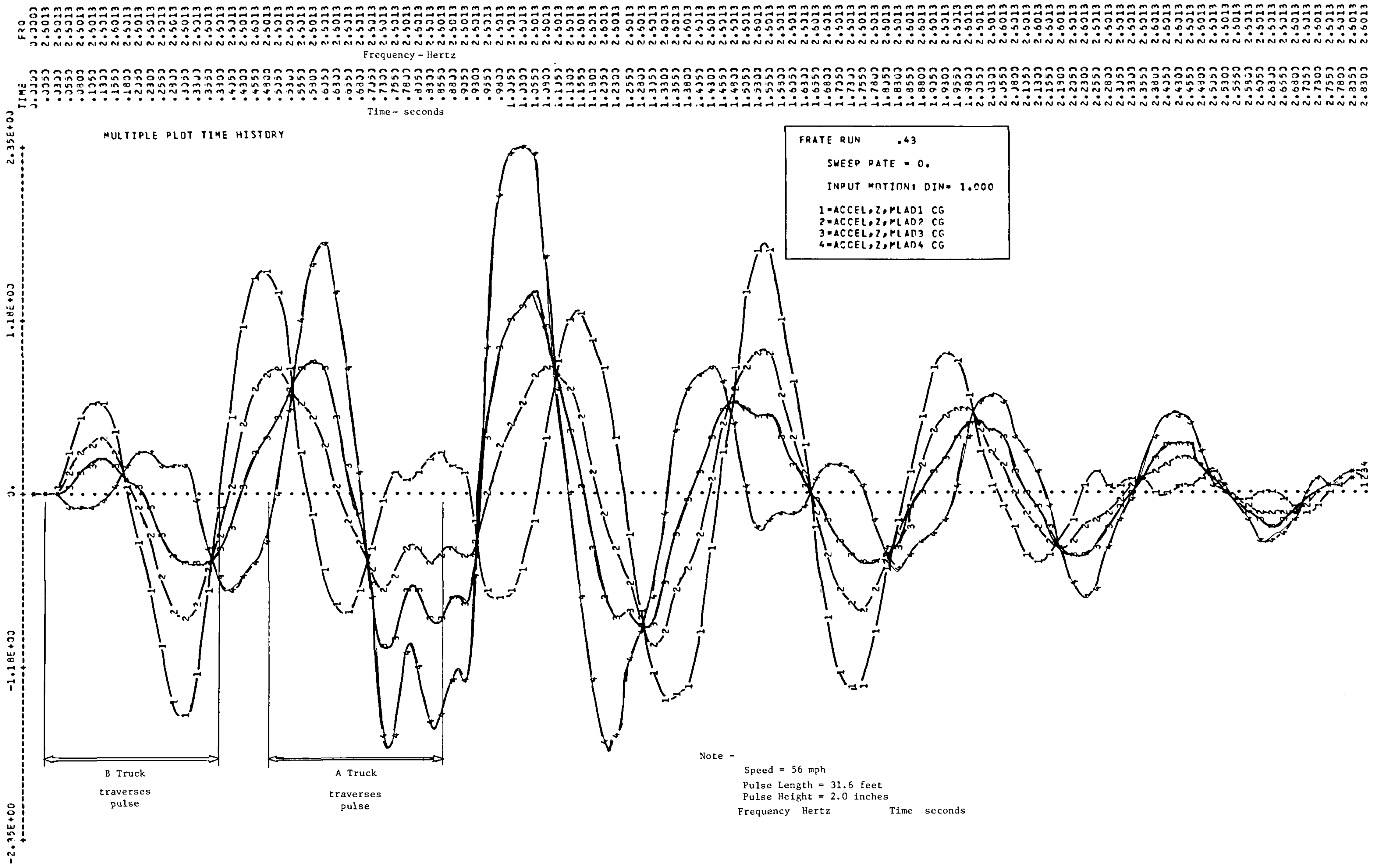


FIGURE 7
ACCELERATION RESPONSE OF COMPLIANT LADING AT C.G., IN Z DIRECTION VERTICAL COINCIDENT PULSE

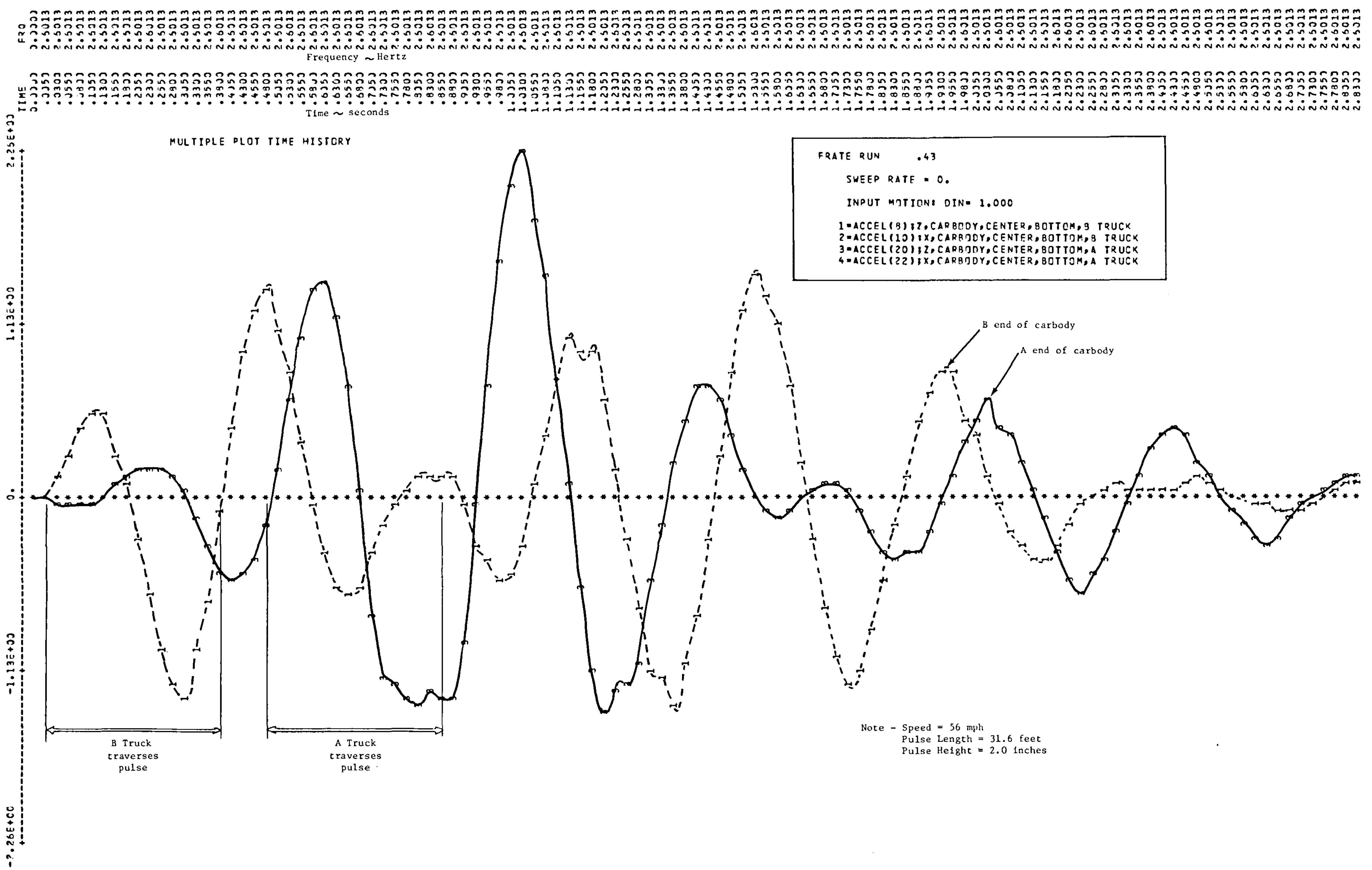


FIGURE 6
CARBODY ACCELERATION RESPONSE TO VERTICAL COINCIDENT PULSE

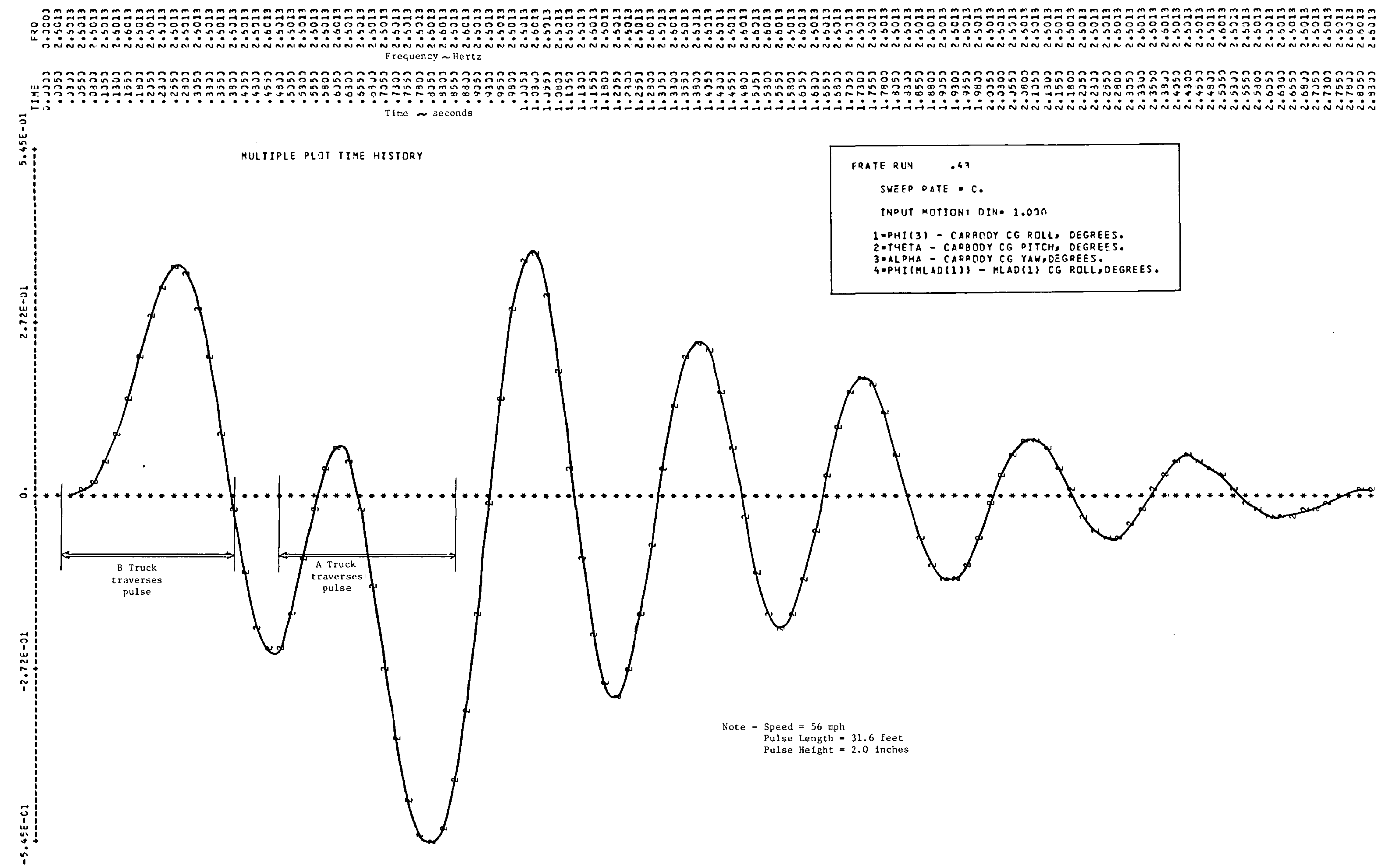
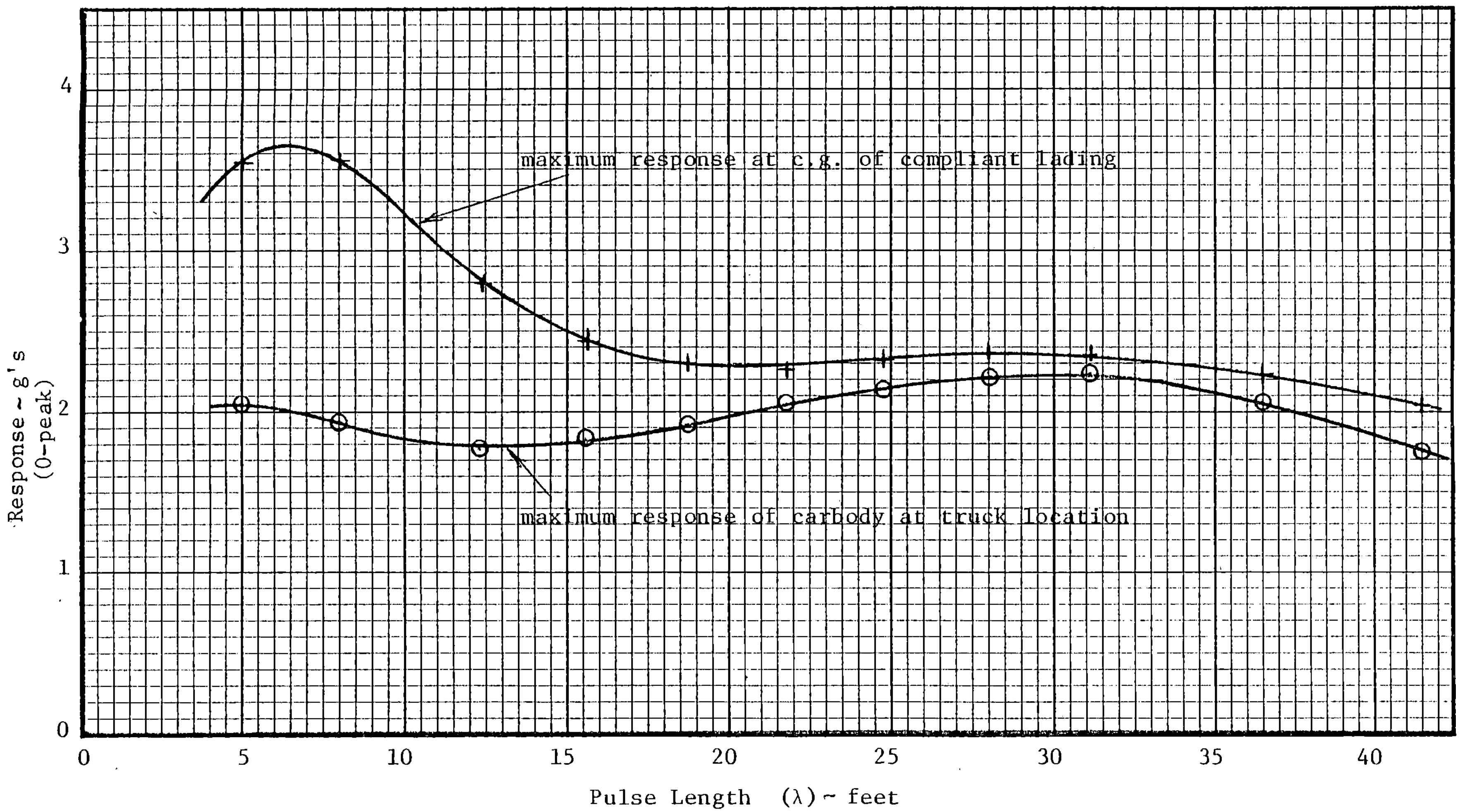


FIGURE 5
CARBODY PITCH MOTION RESPONSE TO VERTICAL COINCIDENT PULSE



Speed = 60 mph
Pulse Height = 2.0 inches

FIGURE 9
VERTICAL ACCELERATION RESPONSE OF CARBODY AND LADING TO COINCIDENT VERTICAL PULSE

to occur. For pulse lengths greater than 15 feet lading response is relatively unaffected by change in pulse length.

Figures 10 and 11 show the effect of speed variations on response. The pitch response peaks at 52 mph and the vertical response peaks at about 55 mph. These peak speeds are close to the predictions of 50.3 and 59.2 mph respectively based on the uncoupled natural frequencies of the boxcars as discussed in Section 2.2.1.1.

In general, the response of the boxcar to a 2.0 inch hump in the track profiles is seen to be undesirably severe. This is believed to be in agreement with service experiences. Further analyses to explore the effect of changes to the model to reduce responses are recommended.

3.2 Response to Cross Level Pulse

It was anticipated that the single cross level pulse would result in the same general responses as for the jointed track, rock and roll simulations but at significantly lower levels. Consequently, one exploratory run was made and compared to rock and roll results. The comparisons, Table V, show the single cross level pulse to be relatively mild and no further runs were made.

3.3 Results of Analyses for Response to Jointed Rail

The cusping condition of staggered joint rail found in service was simulated in this analysis with rectified sine waves phase shifted 90° between rails (refer to the discussion in Section 2.2.1.1). A cross level offset of 0.75 inches was used in all runs. This value of cross level was taken from the Association of American Railroads' "Specifications for Testing Special Devices to Control Stability of Freight Cars."

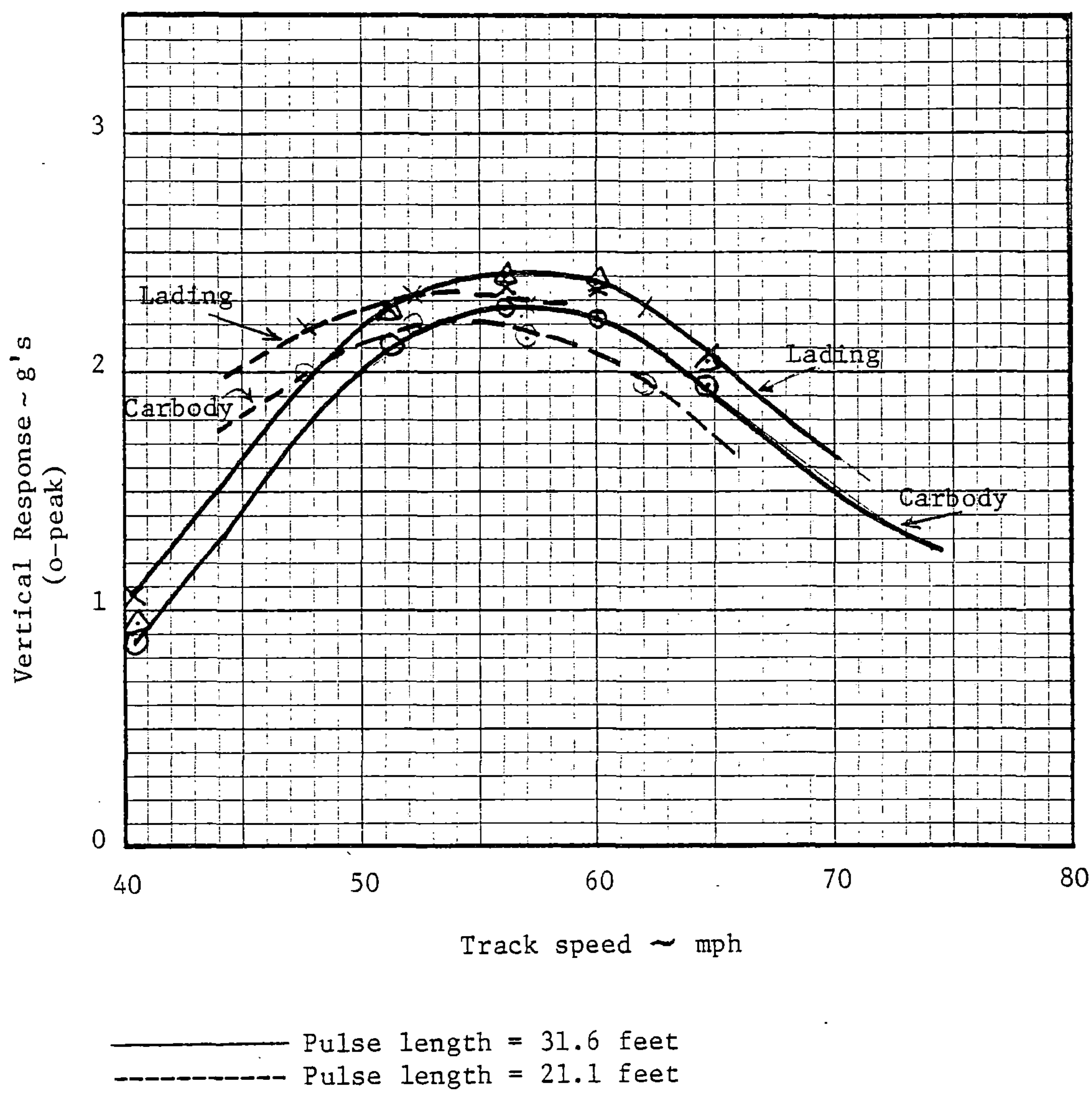


FIGURE 10
VERTICAL ACCELERATION RESPONSE OF CARBODY AND LADING TO VERTICAL COINCIDENT PULSE, FOR TWO PULSE LENGTHS

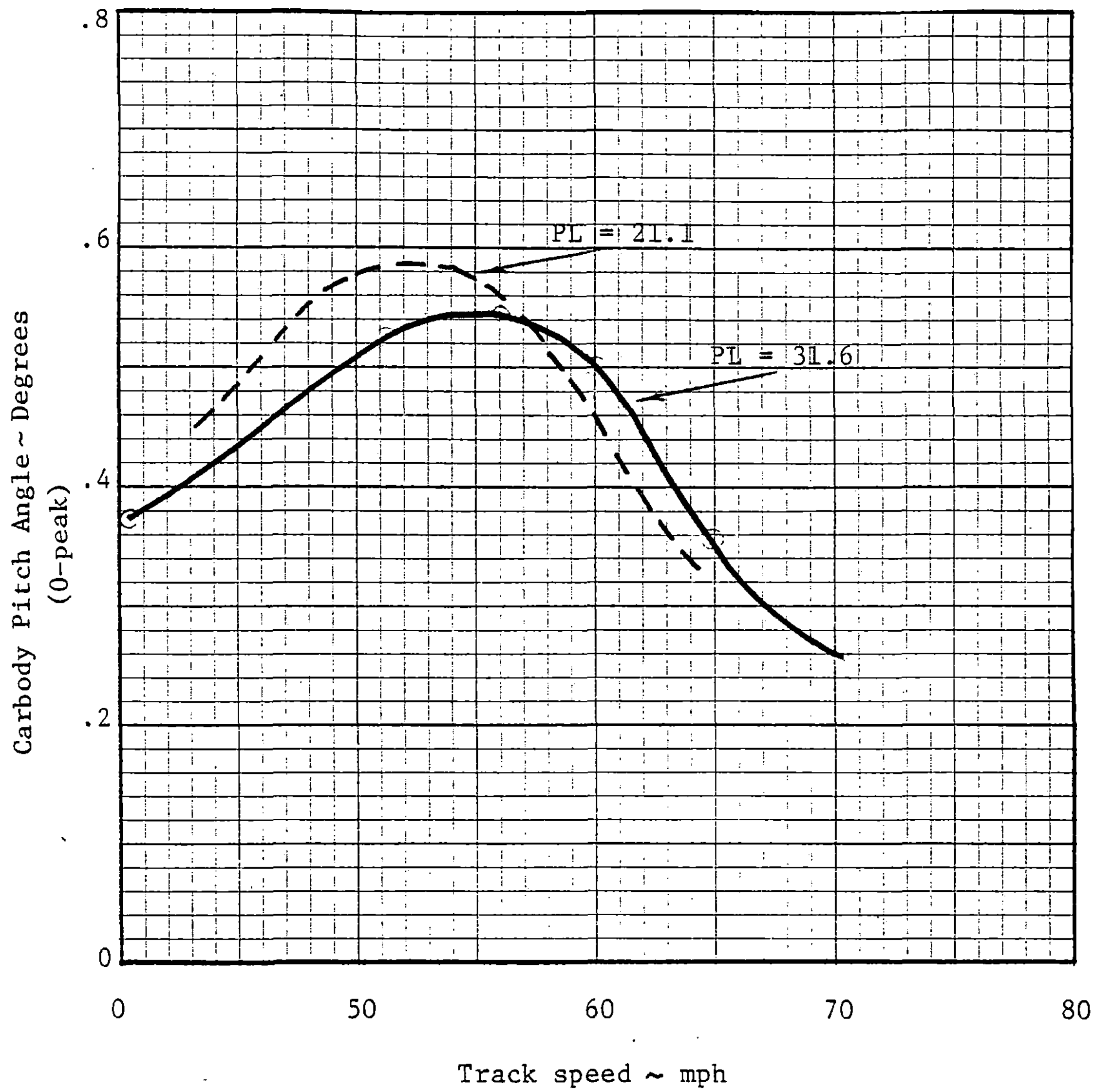


FIGURE 11
CARBODY PITCH MOTION DUE TO COINCIDENT VERTICAL PULSE FOR TWO PULSE LENGTHS

TABLE V

COMPARISONS OF RESPONSES FROM CROSS LEVEL PULSE AND ROCK AND ROLL SIMULATION

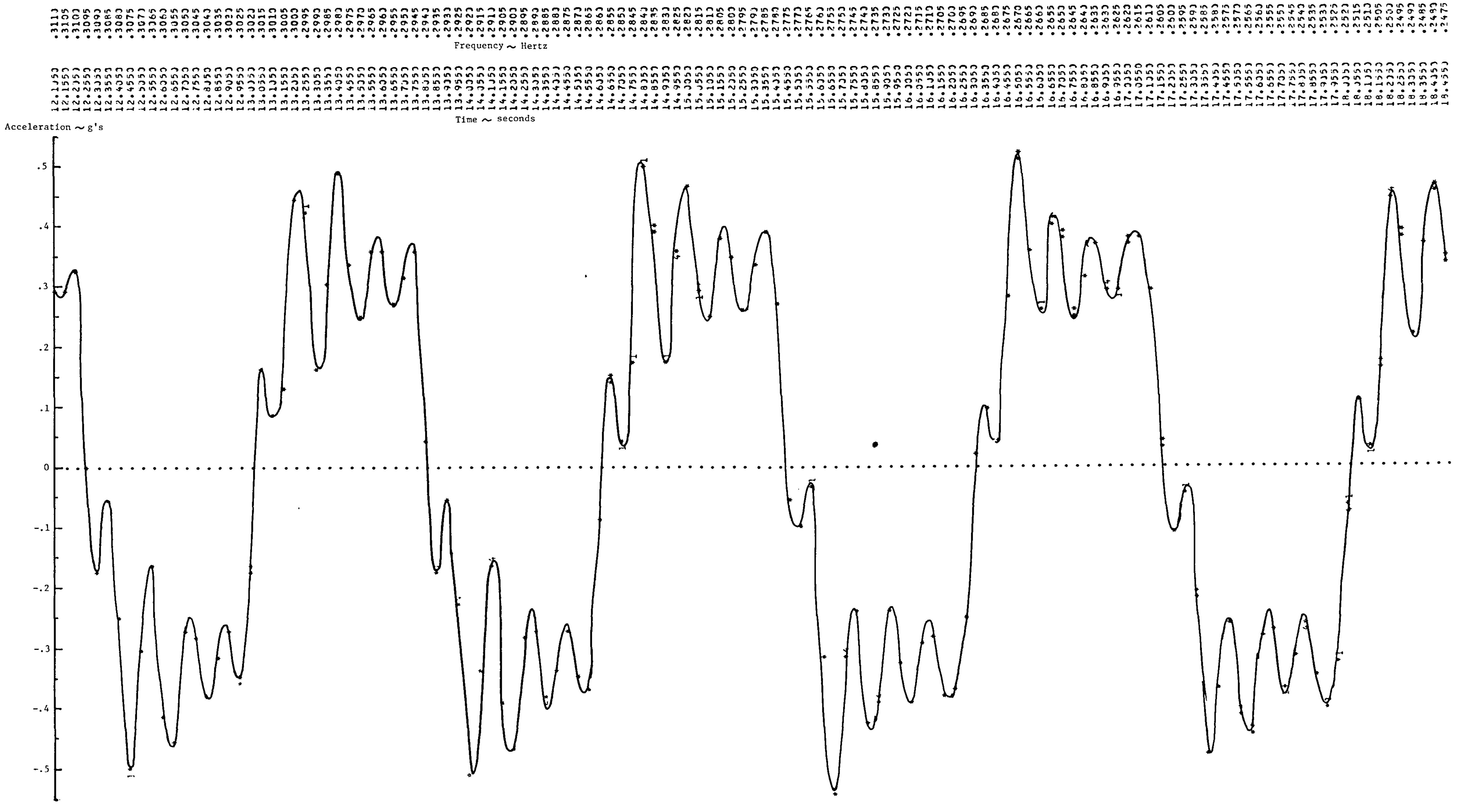
Response Parameter	Cross Level Pulse	Rock & Roll (Constant Speed max response)
Carbody Roll Angle, degrees	1.94	5.58
Lading Acceleration, g's	.335	.66
Lading Deflection, inches	3.67	9.4
Wheel-Rail forces, lbs.		
max vertical	76900	97400
min vertical	16918	0
max lateral	10766	23376
L/Vo (L/44126)	.24	.53

Representative time history plots from the most severe responses encountered are shown in Figures 12, 13 and 14. In this run the freight car was decelerating at the rate of 0.53 mph/sec. through the critical speed. Figures 12 and 13 are vertical and lateral accelerations of the c.g of the compliant lading. Both of these responses show the higher frequency responses resulting from the impacts after wheel lift. Figure 14 presents plots of wheel-rail forces and gives some indication of the severity of the wheel lift.

There were two types of runs performed for the jointed rail analyses: (1) constant speed and (2) deceleration runs. The constant speed runs were made for speeds between 12 and 70 mph. The deceleration runs were made at three different deceleration rates from 23 to 10 mph. Figure 15 summarizes the maximum carbody roll angles from all of the runs. The constant speed runs put the critical speed at 19 mph and the maximum roll angle at 5.58 degrees, single amplitude. The speed of 19 mph with 39 foot rail corresponds to a carbody roll frequency of .71 Hertz. The uncoupled natural frequency calculated in Appendix B lies between .67 and .97 Hertz.

With the deceleration of .53 mph/sec the critical speed drops to about 14.1 mph and the roll angle reaches a maximum of 6.55 degrees. This result correlates with similar results in field measurements attributed to the nonlinear characteristics of freight car trucks.

At the higher deceleration rates of 1.33 and 2.9 mph/sec, critical speeds and maximum angles are 16.1 and 13.7 mph and 3.5 and 2.66 degrees. It would appear that with these higher deceleration rates there is not enough time or cycles of input to build up to a full resonant condition. For example, when decelerating at 1.33 mph/sec there were 10 cycles of roll motion from the start, at 30 mph, to the maximum



Note - 0.75 inch cross level offset

FIGURE 13
LATERAL RESPONSE OF COMPLIANT LADING TO STAGGERED JOINT RAIL

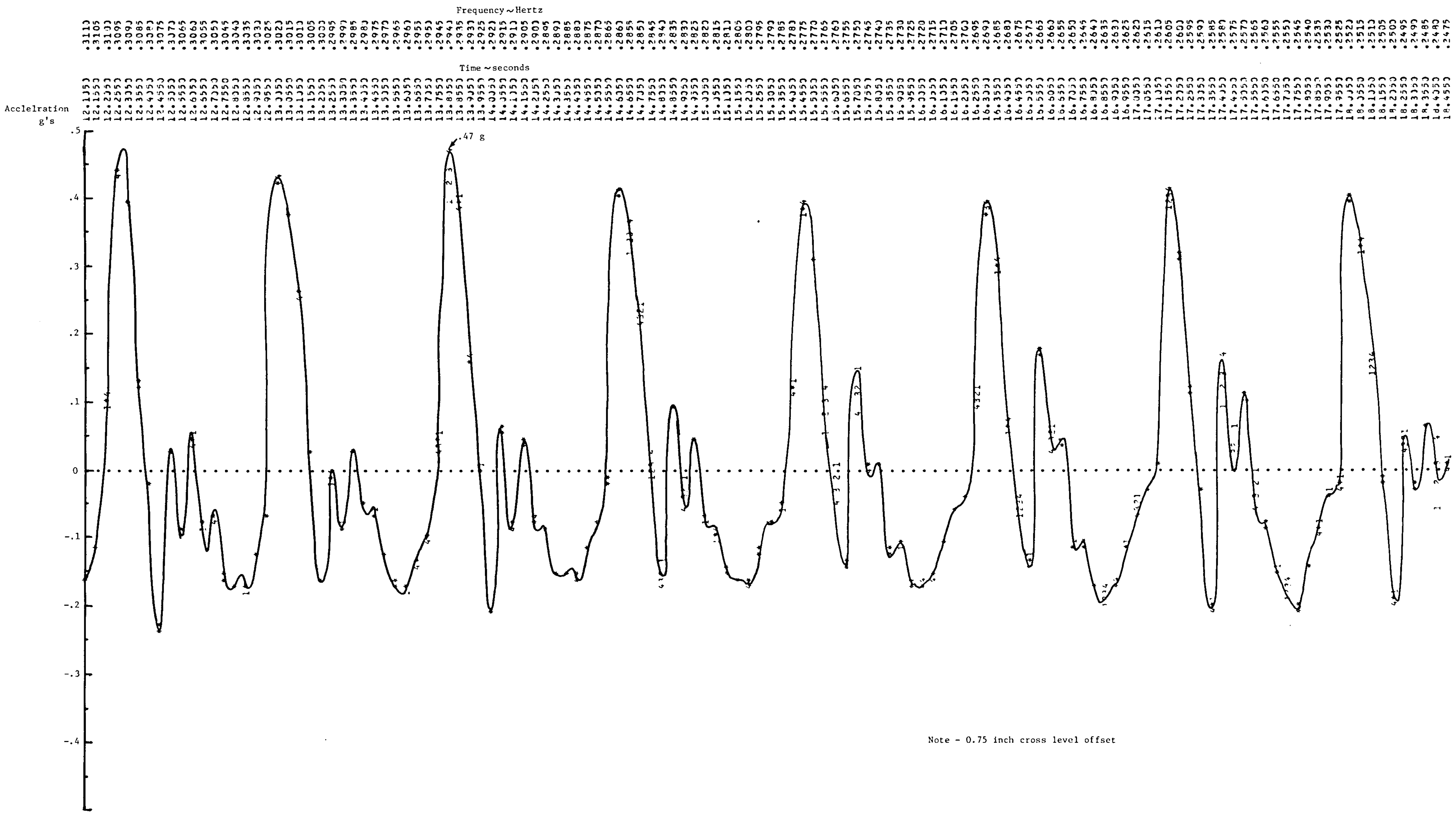


FIGURE 12
VERTICAL RESPONSE OF COMPLIANT LADING TO STAGGERED JOINT RAIL

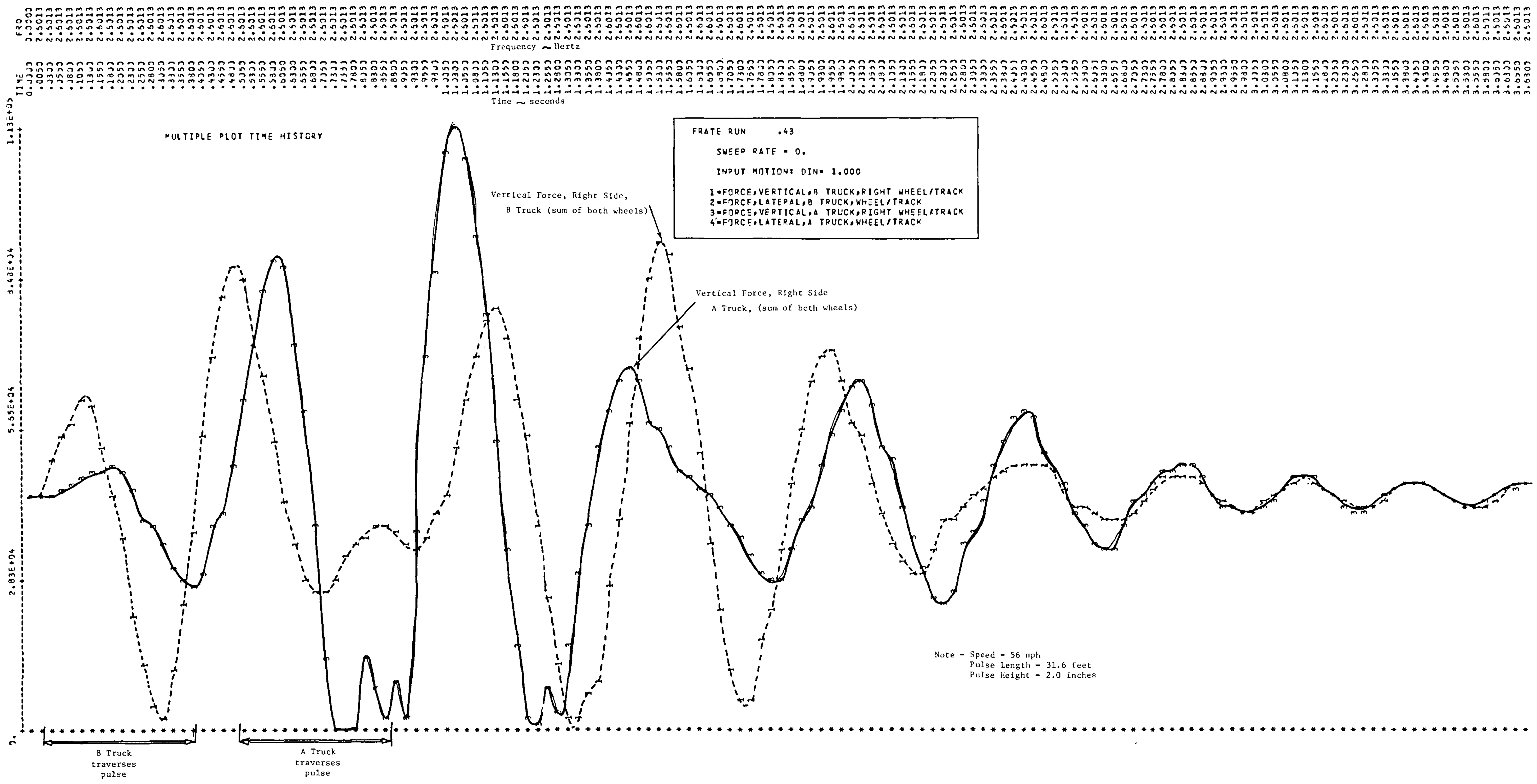


FIGURE 8
WHEEL-RAIL FORCES DUE TO VERTICAL COINCIDENT PULSE

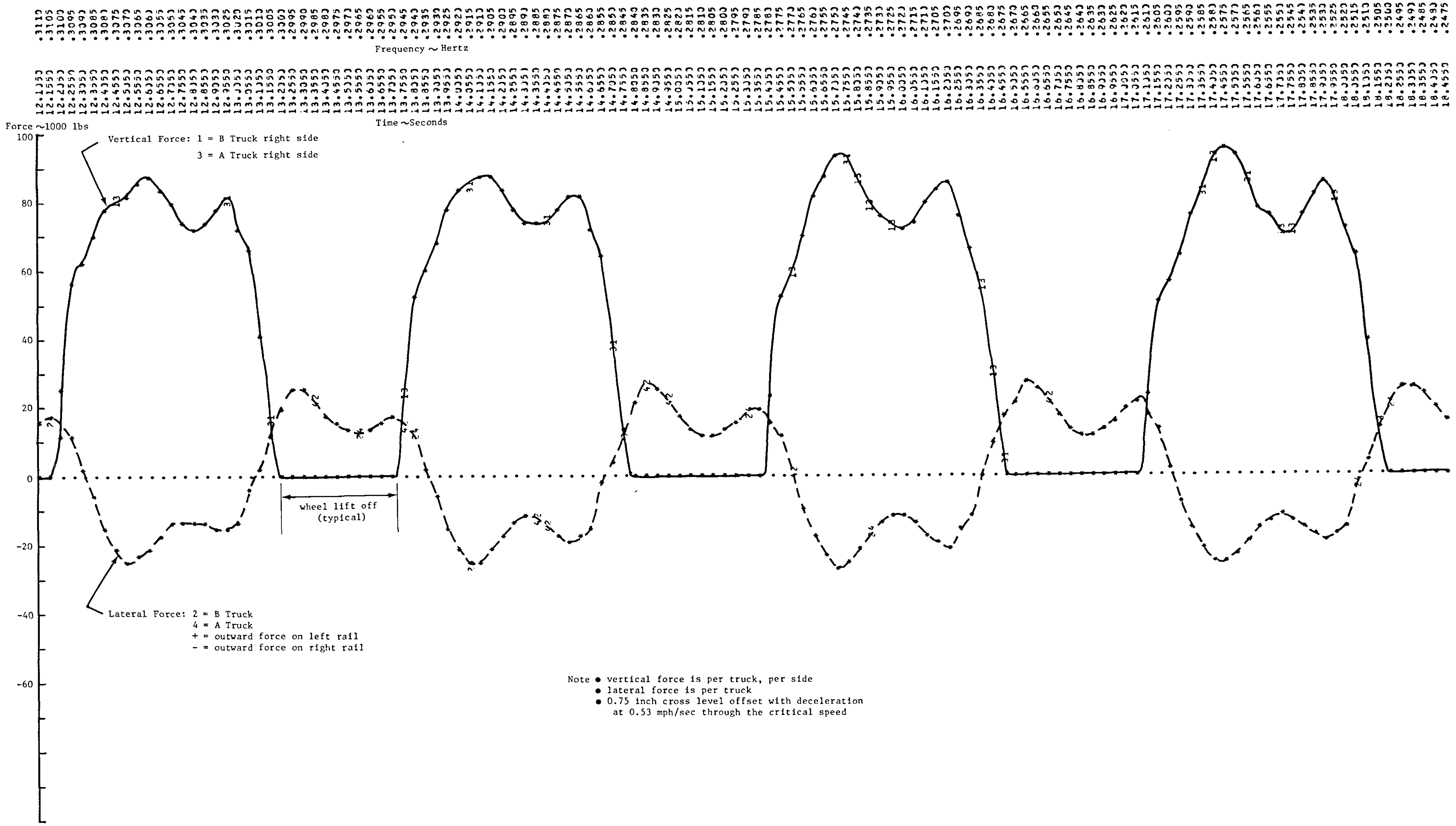
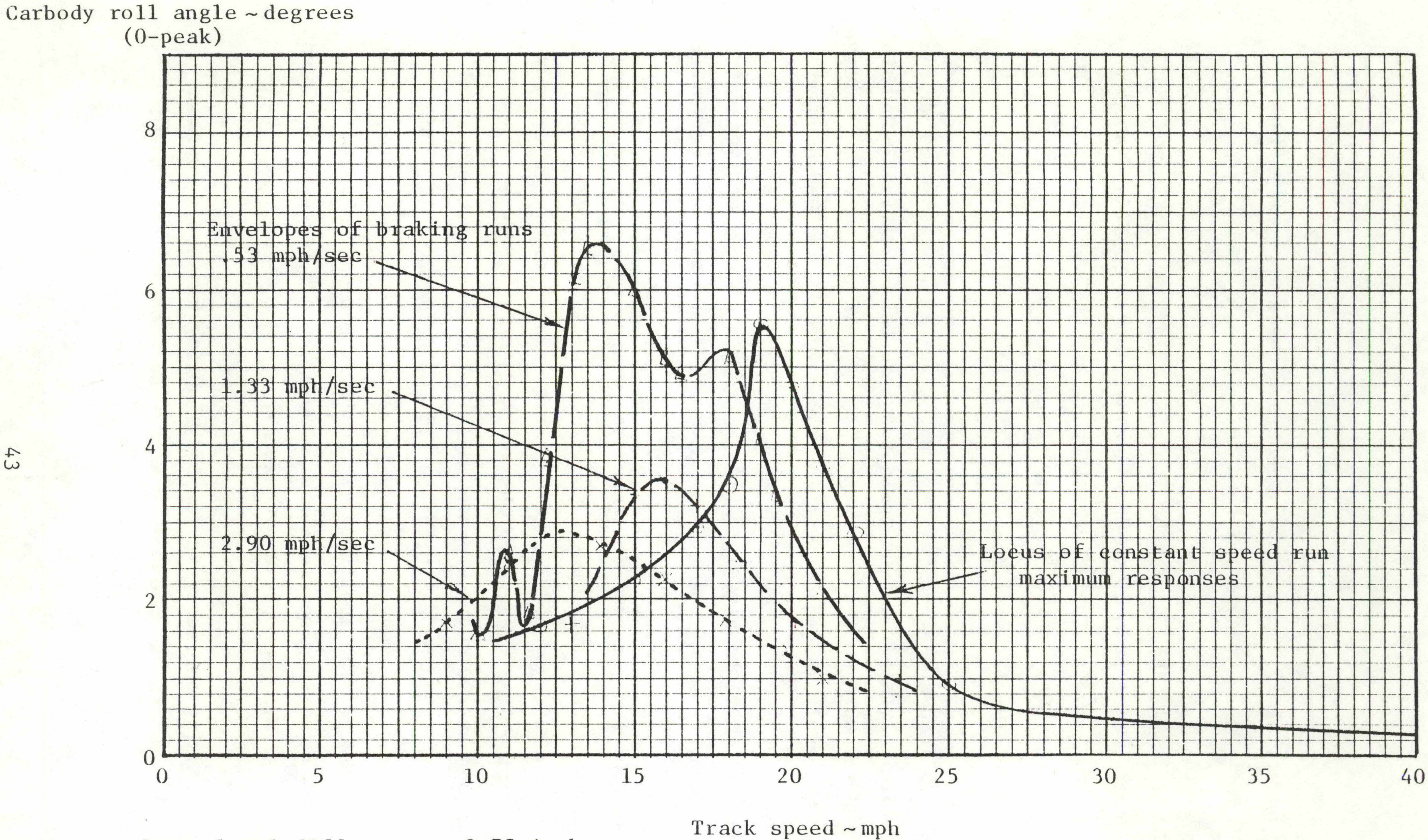


FIGURE 14
WHEEL-RAIL FORCES DUE TO STAGGERED JOINT RAIL



Cross level difference = 0.75 inches

FIGURE 15
CARBODY ROLL RESPONSE TO STAGGER JOINTED 39 FOOT RAIL

response point, at 15.1 mph. At 2.9 mph/sec there were only five cycles of roll motion imposed by the track.

Figures 16, 17 and 18 present the results of constant speed runs made at track speeds from 12 to 70 miles per hour. This data shows that the primary loads are in the region of the carbody low center roll mode, peaking between 19 and 21 mph. Maximum response values from all runs are summarized in Table VI.

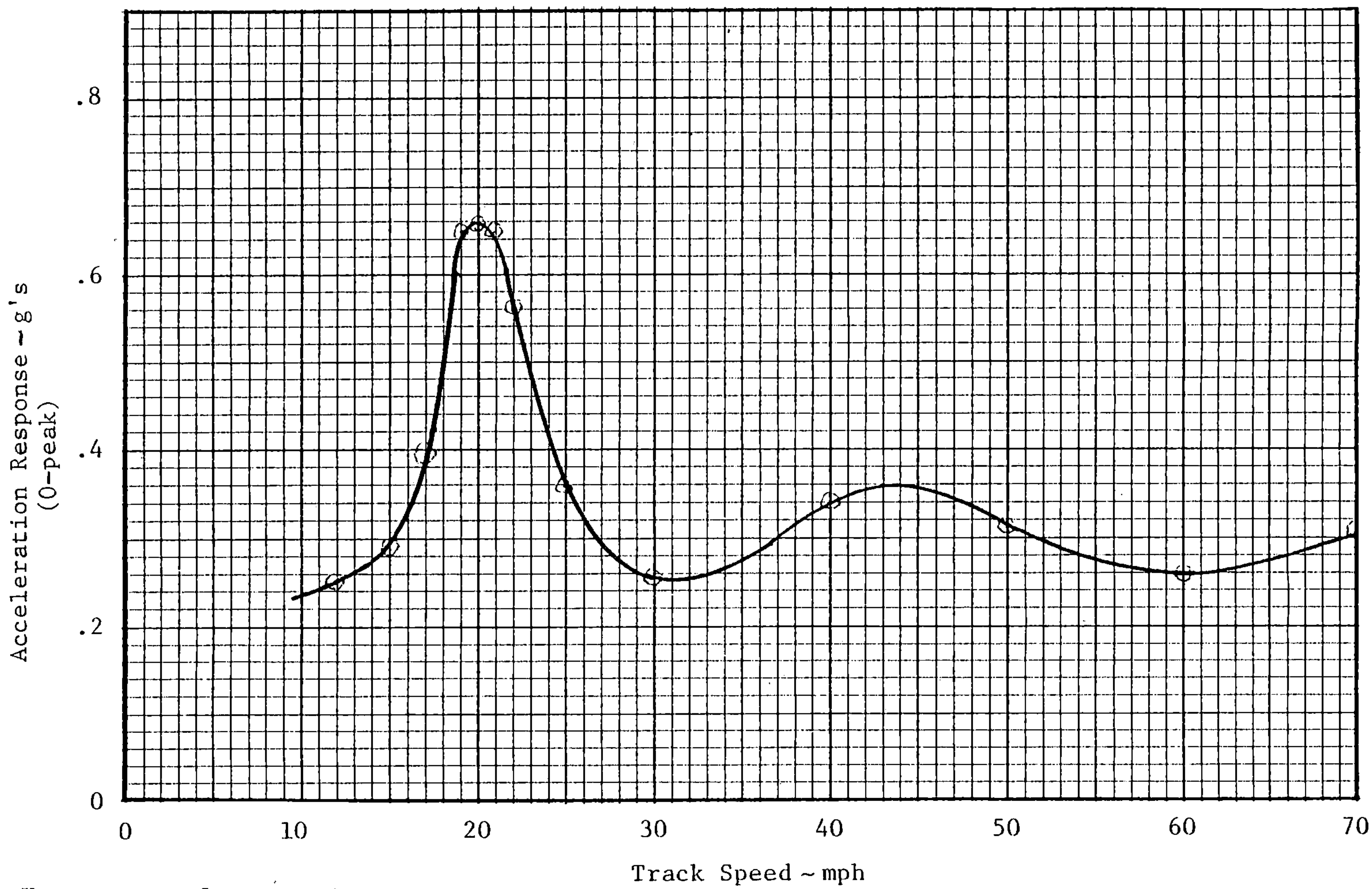
The 0.75 inch crosslevel offset runs resulted in responses which were prohibitively severe. One would conclude that additional snubbing is required for safe operations either in the form of increased friction snubbing (the analysis used a friction snubber force of 3000 pounds) or the addition of hydraulic snubbers or other roll damping devices. Since the 0.75 inch crosslevel results were prohibitive, analyses at larger crosslevels were felt to be unproductive and were not performed.

3.4 Results of Analyses for the Response to Simulated Hunting Motion

In this analysis two types of body hunting motions were reproduced: (1) carbody yaw and (2) carbody high center roll. In carbody yaw body hunting, the trucks are caused to move laterally in sinusoidal motions at 180 degrees phase angle relative to each other at the yaw natural frequency. In the carbody high center roll, the trucks were caused to move laterally in phase at the high center roll natural frequency. The amplitude of the input motions in both cases was 1.62 inches peak to peak. (This is an arbitrary value, 125 percent of an estimated average on class 4 track.)

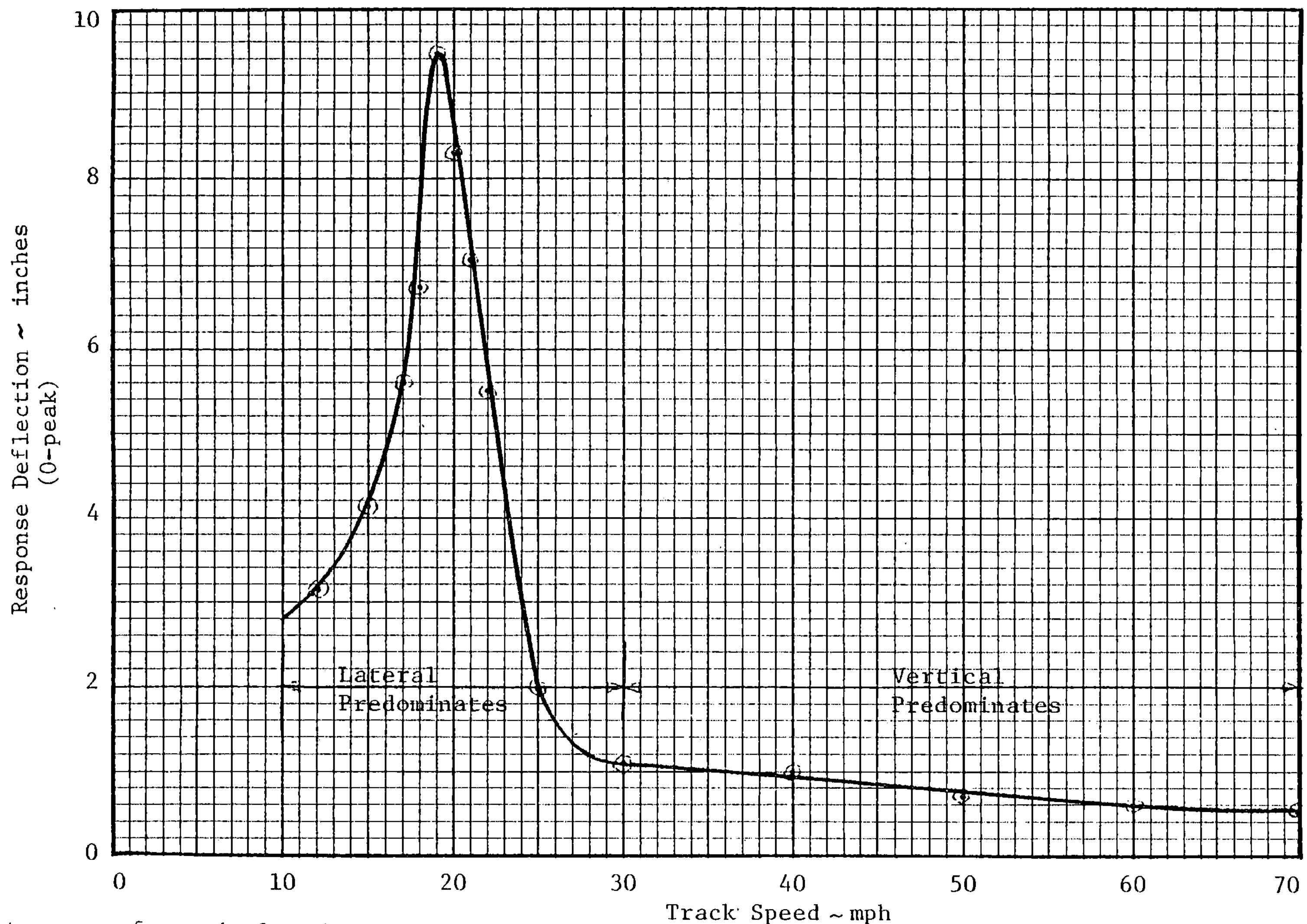
The g loading on lading in yaw body hunting reached a maximum value of .64 g which is undesirably high. Deleterious effects will result on track, car and lading as wheels become worn.

The responses for the high center roll body hunting case are very low, probably because the boxcar configuration used had a relatively low center of gravity which minimized coupling between lateral motions of the trucks and roll of the carbody. The results of the hunting simulation analyses are summarized in Table VII.



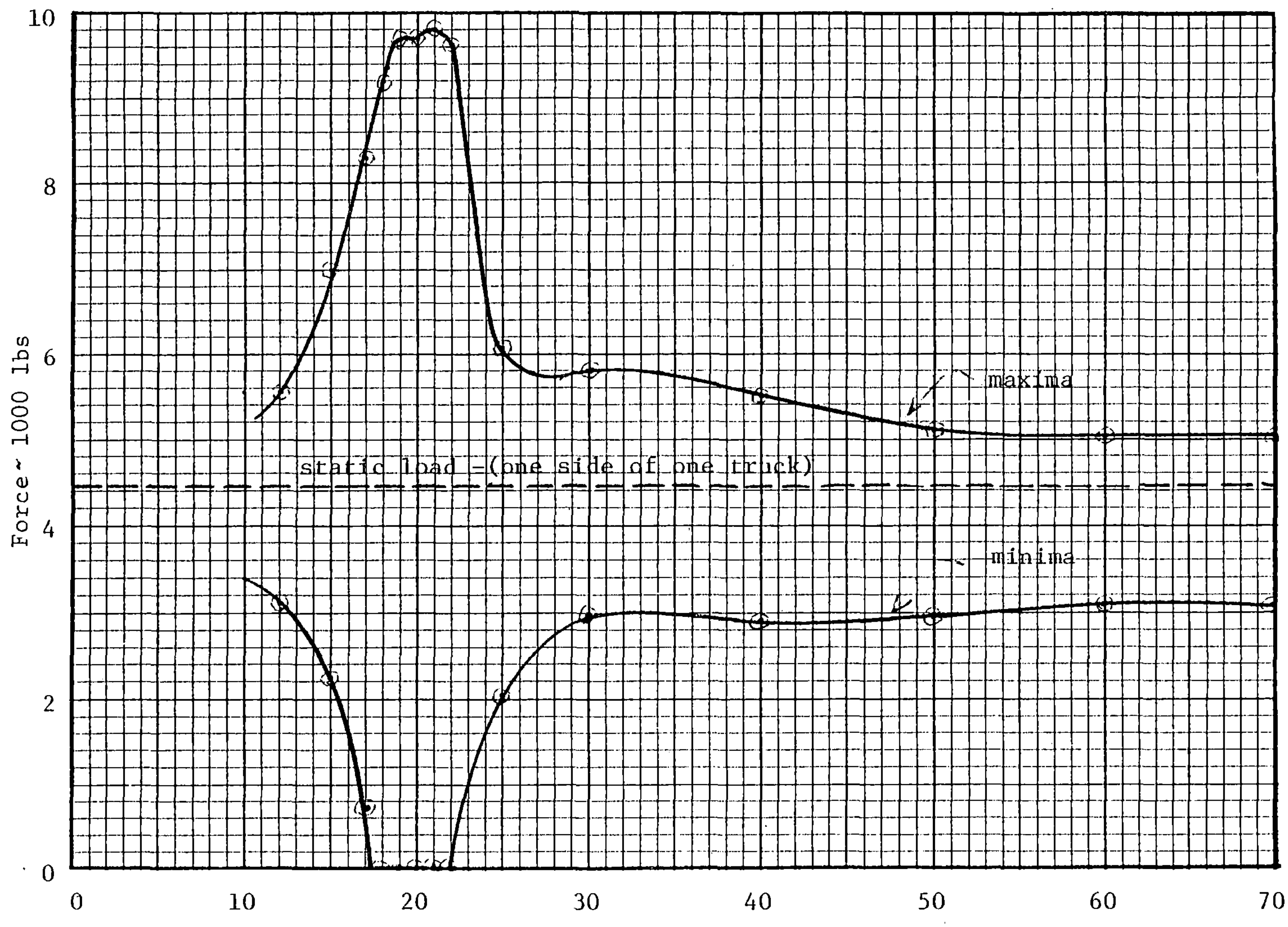
Vector sum of vertical and
lateral accelerations.

FIGURE 16
ACCELERATION RESPONSE AT CORNER OF LADING TO
39 FOOT STAGGER JOINTED RAIL



Vector sum of vertical and lateral deflections.

FIGURE 17
DEFLECTION RESPONSE AT CORNER OF LADING TO 39 FOOT STAGGER JOINTED RAIL



0.75 inch cross level input

FIGURE 18
VERTICAL WHEEL-RAIL FORCES IN RESPONSE TO CONSTANT SPEED RUNS OVER 39 FOOT STAGGER JOINTED RAIL

TABLE VI

RESULTS OF JOINTED RAIL ANALYSES

(ROCK AND ROLL SIMULATION)

MAXIMUM RESPONSE VALUES

Response Parameter	Run Speed Condition			
	Constant Speed	Deceleration - mph/sec		
		.53	1.33	2.90
Speed at maximum response (mph)	1.90	14.1	16.1	13.7
Frequency at maximum response (Hertz)	.71	.53	.60	.52
Carbody roll angle (degrees)	5.58	6.55	3.50	2.66
Lading acceleration (g's)	.66	.74	.57	.40
Lading deflection (inches)	9.4	11.3	6.81	5.21
Wheel-Rail forces (lbs.)				
maximum vertical	97400	96800	91100	76700
minimum vertical	0	0	0	13806
maximum lateral	23376	27104	23680	16874
L/V_o (V_o = 44126)	.53	.62	.54	.30

TABLE VII

SUMMARY OF MAXIMUM RESPONSES FOR BODY HUNTING CONDITION
(Assumed sinusoidal lateral motion at wheel-track interface within gage clearance limit of 1.62 inches)

RESPONSE PARAMETER	Yaw Mode Body Hunting		Roll Mode Body Hunting (Ambient values, no max.)
	max. wheel force condition	max. yaw angle condition	
Frequency of Motion, Hertz	1.8	1.5	2.46
Carbody Max. Acceleration, g's	.45	.40	.34
Carbody Yaw or Roll Angle, degrees (single amplitude)	.33	.41	.38
Compliant Lading Accel. g's (at corner)	.64	.54	.32
Compliant Lading Displacement, inches (at corner)	1.97	2.34	.56
Wheel Rail Forces - lbs.			
max. vertical	31,700	31,100	28,620
min. vertical	19,654	20,300	22,260
max. lateral	12,046	11,400	9,540
L/Vo (Vo=25,360 lb.)	.48	.45	.38
L/V (V instantaneous)	.38	.37	.33
Hunting Critical Speed* new/worn wheel - mph	63/32	53/27	87/43

*This is the hypothetical critical hunting speed to result in body hunting at the carbody resonant frequency. These results are not a prediction of hunting, but a calculation of dynamic responses on the assumption that hunting has occurred.

TABLE VIII

SUMMARY OF PREDICTED RESPONSES

Response Measure	Track Input Motion: Jointed Track	Vertical Pulse	Hunting
Carbody z, g's	.46	2.3	-
Carbody x, g's	.33	-	.45
Carbody Roll, degrees	6.55	-	.41
Carbody Pitch, degrees	-	.59	
Carbody Yaw, degrees	-	-	.41
Lading z, g's	.47	3.6	.64
Lading x, g's	.54	-	.58
Lading Defl., inches	11.3	3.6	2.3
Wheel-rail Forces			
Vert. max., lbs.	97,400	108,000	31,700
Vert min., lbs.	0	0	19,654
Lateral max., lbs.	23,376	-	12,046
L/V	.53	-	.48
Input Condition			
Amplitude, inches	.75	2.0	.81
Speed, mph	14.1	55-60	30-60
Frequency, Hertz	.53	2.2-2.8	1.5-1.8

NOTE: The acceleration and displacement values are all single amplitude.

4.0 SUMMARY AND RECOMMENDATIONS

The FRATE computer program has been shown to be useful for study of the dynamic response of a boxcar with compliant lading to service conditions. Three basic types of vehicle responses were studied: (1) the vertical and roll 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. It was shown what particular conditions caused the maximum responses and values of those responses. The responses included accelerations, displacements and forces. A summary comparison of responses is shown in Table VIII.

In the truck simulation of this version of FRATE the vertical and roll spring/damper characteristics are separated. This was done by having a single vertical spring/damper at the center of each truck and a separate roll moment spring/damper for roll motions. A friction snubber force of 6,000 pounds per truck was assumed. The friction snubber force was in effect only after the side bearing gap, 0.25 inches, had been exceeded and was in effect for the up stroke only, this being a duplication of the actual operating condition. The friction snubber force of 6,000 pounds per truck corresponds to 3,000 pounds per snubber. Half this value was used in Model 2, the hunting case, because of the lightweight configuration .

Of the three conditions studied the vertical track irregularities resulted in the largest acceleration responses and wheel-rail forces. Maximum accelerations were 2.3g on the floor of the carbody and 3.6g at the c.g. of the compliant lading. Wheel-rail vertical forces reached a maximum of 108,000 pounds, which is 2.4 times the static load. These responses are higher than expected and may be so because the 2.0 inch track irregularity assumed may be higher than that found in actual service conditions.

The rock and roll response to jointed rail was also higher than expected. The .75 inch cross level difference resulted in responses which were greater than the AAR Special Device Test allowables. For example, a maximum roll angle of 6.55 degrees was calculated compared to 3.0 degrees maximum allowed by the test specification. Also, lateral accelerations of .45 g's were calculated compared to an allowable .35g. However, the larger responses are not altogether unreasonable considering (1) that there were no snubbing devices other than the friction dampers in the spring nest assembly and (2) that a relatively low value of friction snubbing was used.

The hunting responses were in general as expected with the acceleration loading on the compliant lading being of a high enough g level to be of concern. There are two avenues of approach in seeking to reduce body hunting responses. One is to lower the resonant frequencies of the carbody. This has two effects: (1) it causes an additional separation of the carbody frequency from the truck hunting critical frequency thus reducing the occurrence and (2) since the amplitude of the hunting motion is bounded by gage clearance the inertial forces will reduce by the ratio of frequency squared. (This is the reverse application of hunting characteristics of lightly loaded freight cars, which have higher resonant body frequencies by virtue of their light weight and tend to have more severe hunting conditions than fully loaded cars.) The other approach to reducing body hunting is to increase damping. Neither approach is easy.

Based on the results of analyses performed for this report, three recommendations are made. The recommendations pertain to a 70 ton box car with standard side bearings with 1/4 inch clearance each side, friction snubbers equivalent to snubber force of about 3,000 lbs. per side and no hydraulic snubbers.

1. It is recommended that a lower frequency, i.e., a softer primary suspension system, be used. The softer suspension will have two benefits: (1) the occurrence of body hunting motions will be decreased and the acceleration loading under a body hunting condition will be lowered, and (2) ride improvement will be realized over vertical track irregularities. The model of this analysis used D5 spring group characteristics. Use of the D7 spring group would have the desired softening effect. It is recommended that the railroads and their suppliers continue the trend for softer, larger deflection primary suspension systems.

2. It is recommended that the freight car truck designers find a way to increase the effectiveness of snubbers against roll motions and at the same time decrease snubber forces. This paradoxical change is possible because snubbers, both hydraulic and friction, are active only after side bearing contact is made. If the side bearing gap is made smaller, an increase in snubbing will be effected with no change to the snubber. A second way in which the recommendation can be implemented is to devise a snubber which is active before side bearing contact is made.

3. It is recommended that snubber forces be made as small as possible. Properly done, this would require a compromise between rock and roll responses and responses due to vertical irregularities. Put another way, an optimum snubber for rock and roll will cause severe loadings due to vertical irregularities and optimum snubber size for the vertical irregularities will result in unacceptable rock and roll characteristics.

APPENDIX A

LADING MODEL

APPENDIX A

LADING MODEL

The modeling of the flexibly mounted lading and the combined, rigid carbody and lading is outlined here.

The lading consists of rectangular cardboard shippers containing cylindrical cans filled with a product of a specified density. These shippers are arranged in layers, each layer consisting of the same number of shippers, mounted on a wooden pallet dimensions 40" x 98" x 6". Within the boxcar a specified number of these identical pallet-mounted stacks of shippers are arranged symmetrically. Both geometric and weight restrictions are adhered to in sizing the lading.

The compliant portion of the lading is assumed to consist of the topmost 40 percent of the lading stacks. Four such compliant lading masses are incorporated in the boxcar, each one of which is permitted three degrees of freedom--vertical, lateral and roll. The mechanical properties of these four flexible lading masses are separately calculated from the remainder of the lading which is assumed intergally mounted to the carbody.

Determination of Mass Properties

Analyses were performed with two loading conditions and are referred to here as Model 1 and Model 2. Model 1 had a total lading weight of 112842 pounds loaded in the boxcar to a height of 91.5 inches above the floor. Model 2 had a total lading weight of 38814 pounds loaded to a height of 34.5 inches above the floor. These loadings represent 72 and 25 percent of the maximum allowable. In each model there were four pieces of the lading separated out as shown in Figure A-1, and spring mounted. The mass density of the material in the cans was assumed to be the same as for water.

Using the empty box car properties given in Reference 4 the following mass properties were compiled.

Empty Carbody

W = 46400 pounds

h_{cg} = 23.15 inches (above floor)

I_{roll} = .41516 lb. in. sec.^{2}

I_{pitch} = .428E7 lb. in. sec.^{2}

I_{yaw} = .4205E7 lb. in. sec.^{2}

Carbody and Rigid Lading

Model 1

W = 135170 lb. = 350.2 lb. sec.^{2}/in.

h_{cg} = 37.15 in. (above plane of center board)

I_{roll} = .821E6 lb. in. sec.^{2}

I_{pitch} = .111E8 lb. in. sec.^{2}

I_{yaw} = .111E8 lb. in. sec.^{2}

Model 2

W = 76934 lb. = 199.3 lb. sec.^{2}/in.

h_{cg} = 21.8 in. (above plane of center bowl)

I_{roll} = .514E6 lb. in. sec.^{2}

I_{pitch} = .658E7 lb. in. sec.^{2}

I_{yaw} = .656E7 lb. in. sec.^{2}

Spring Mounted Lading Masses

Model 1

W = 6018 lb. = 15.6 lb. sec.^{2}.in. (each)

h_{cg} = 61.65 in. (above floor)

y_{cg} = 262.4, 60.4, -60.4, -262.4 inches (lading masses 1 - 4, measured from carbody midpoint)

I_{roll} = .147E5 lb. in. sec.^{2}

Model 2

W = 2070 lb., = 5.36 lb. sec.^{2}/in.

h_y = 14.88 in. (above floor)

y_y = 262.4, 60.4, -60.4, -260.4 inches (lading masses 1 - 4, measured from carbody midpoint)

I_{roll} = .47E4 lb. in. sec.^{2}

Total Weight on the Rail

Model 1

Carbody and rigid lading	135170
Sprung lading (4)	24072
Trucks (2)	17239
Total	176481 lb.

Model 2

Carbody and rigid lading	76934
Sprung lading (4)	8280
Trucks (2)	17239
	102453 lb.

Determination of Springs and Dampers Supporting Compliant Lading Mass

The natural frequencies and amplification factors assumed for the compliant lading were as follows:

lateral resonance:	$f = 5.0$ Hertz	$Q = 6.25$
vertical resonance:	$f = 9.5$ Hertz	$Q = 4.2$
Roll resonanace:	$f = 15.0$ Hertz	$Q = 8.3$

These assumed values are based on information given in Reference 3 on the results of lading test measurements.

The spring damper values were calculated on a single degree of freedom basis using the following relationships:

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \qquad C_c = 2\sqrt{KM} \qquad \frac{1}{Q} = 2\frac{C}{C_c}$$

where M, f, and Q are known
and K and C are to be solved for

For example solve for the lateral spring/damper for the Model 1 lading.

Given:

$M = 15.6$ lb. sec.^{2}/in.

$f = 5.0$ Hertz

$Q = 6.25$

Solve for:

$K = M\,(2\pi f)^2$

$= .154E5$ lb./in.

$C = \frac{1}{Q}\sqrt{KM}$

$= 78.4$ lb. sec./in.

Determination of Lading Data for FRATE Input

The input lading data required for FRATE then can be summarized as follows:

MLAD = mass of each flexible lading = 15.60 lb.sec.^{2}/in. model 1
= 5.36 lb.sec.^{2}/in. model 2

INLAD = roll moment of inertia of each flexible lading
= .148E5 lb.in.sec.^{2} model 1
= .470E4 lb.in.sec.^{2} model 2

DLAD = longitudinal distances between the boxcar/rigid lading c.g. and each flexible lading c.g.
= 262.4, 60.6, -60.6, -262.4 inches, models 1 and 2

RLAD = lateral distances between the boxcar/rigid lading c.g. and each flexible lading c.g.
= 0., 0., 0., 0. inches, models 1 and 2

KLAD = vertical, lateral and roll stiffness of each flexible lading mass

vertical = .556E5 lb./in., model 1
= .191E5 lb./in., model 2

lateral = .154E5 lb./in., model 1
= .529E4 lb./in., model 2

roll = .131E9 in.lb./rad., model 1
= .418E8 in.lb./rad., model 2

CLAD = Vertical, lateral and roll damping of each flexible lading mass

vertical = 223. lb.sec./in., model 1
= 76.8 lb.sec./in., model 2

lateral = 78.4 lb.sec./in., model 1
= 27.0 lb.sec./in., model 2

roll = .167E6 in.lb.sec./rad., model 1
.532E5 in.lb.sec.rad., model 2

HL = height of the flexible lading c.g. above the c.g. of the combined boxcar/rigid lading c.g.
= 42.8 inches, all four lading masses, model 1
= 12.0 inches, all four lading masses, model 2

Bx = lateral corner distance = 51.0 inches, models 1 and 2

By = longitudinal corner distance = 40.2 inches, models 1 and 2

Bz = vertical corner distance = 15.5 inches, model 1
5.35 inches, model 2

In addition, this appendix also gives the equations for the following input parameters:

M(3) = mass of the combined boxcar/rigid lading
= 350.2 $lb.sec.^2/in.$, model 1
= 199.3 $lb.sec.^2/in.$ model 2

I(3) = roll moment of inertia of the combined boxcar/rigid lading
= .821E6 $in.lb.sec.^2$, model 1
= .514E6 $in.lb.sec.^2$, model 2

I(10)= pitch moment of inertia of the combined boxcar/rigid lading
= .111E8 $in.lb.sec.^2$, model 1
= .658E7 $in.lb.sec.^2$, model 2

I(11)= yaw moment of intertia of the combined boxcar/rigid lading
= .111E8 $in.lb.sec.^2$, model 1
= .656E7 $in.lb.sec.^2$, model 2

The vertical geometric constraint for the lading is

$$(6 + 9.48n) < 125''$$

The weight constraint for the lading is

Max. lading weight $< [220,000 - 2(8620) - 46,400] = 156,360$ lb.

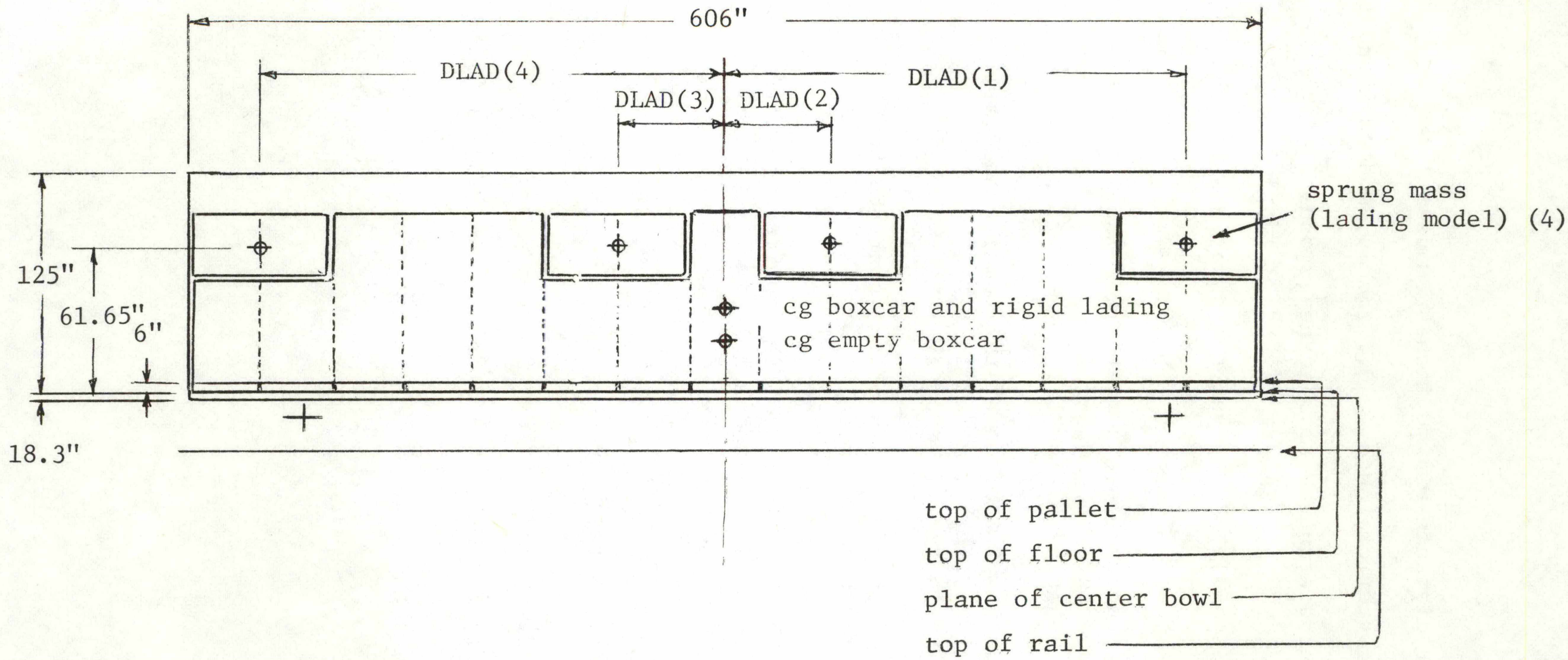


FIGURE A-1
BOX CAR—LADING CONFIGURATION, MODEL 1

APPENDIX B

CARBODY NATURAL FREQUENCIES

APPENDIX B

CALCULATION OF THE CARBODY UNCOUPLED NATURAL FREQUENCIES

The carbody is modeled as a rigid body integrally incorporating the total lading. The total lading is used in these calculations rather than the rigid lading because the dynamically mounted lading uncoupled natural frequencies are well above the uncoupled carbody natural frequencies. (The lading uncoupled natural frequencies are 9.5, 5 and 15 cps for the vertical, lateral and roll directions respectively.) The carbody is symmetrically mounted on the trucks permitting the carbody to be modeled as shown in this appendix for the purposes of calculating its uncoupled natural frequencies.

Because of the nonlinear truck properties included in the model some of the natural frequencies will change depending on the amplitude of motions. There are two conditions to cause this. One is that the truck suspension is stiffer when the amplitudes of motion are low and the snubbers are "locked-up". The other condition is the bilinear truck roll spring representing conditions with and without side bearing contact.

<u>Vertical Uncoupled Natural Frequency (fv)</u>

$$f_v = \frac{1}{2\pi} \sqrt{\frac{2Kv}{M}} \quad \text{cps}$$

Kv = vertical stiffness of one truck

$$\frac{1}{Kv_1} = \frac{1}{K(1) + K(3)} + \frac{1}{K(4)} \quad \text{lb/in}$$

for large amplitudes

$$\frac{1}{Kv_2} = \frac{1}{K(1) + K(3)} + \frac{1}{K(4) + Ks4}$$

for small amplitudes

M = mass of the carbody plus the lading, lb sec^2/in

Model 1	Model 2
K(1) = K(3) = .91E5	
K(4) = .48E5	all K's the same
Ks4 = .24E6	
M = 350.2 + 4 x 15.6	M = 199.3 + 4 x 5.36
= 412.6 lb. $sec.^2$/in.	= 220.74
Kv1 = .380E5	
Kv2 = .1103E6	
Vertical Natural Frequency, large motions	
f_{v1} = 2.16 Hz	f_{v1} = 2.95 Hz
Vertical Natural Frequency, small motions	
f_{v2} = 3.68 Hz	f_{v2} = 5.03 Hz

Pitch Uncoupled Natural Frequencies

$$f_p = \frac{1}{2\pi}\sqrt{\frac{2Kv \times (L/2)^2}{I_p}}$$

L = 474, inches

Kv = as calculated for vertical nautral frequencies

$2Kv1 \times (L/2)^2$ = .42688E10 (large motions), lb. in./radian

$2Kv2 \times (L/2)^2$ = .12391E11 (small motions), lb. in./radian

Model 1	Model 2
$I_p = .111E8 \times \frac{412.6}{350.2}$	$I_p = .656E7 \times \frac{220.74}{199.3}$
$= .1308E8$	$= .7266E7 \text{ lb. in. sec.}^2$

Pitch Natural Frequencies

Large Motion

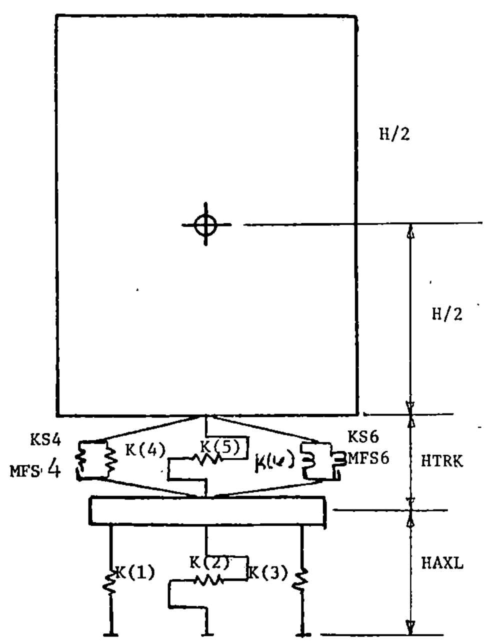
$f_{p1} = 2.88$ Hz	$f_{p1} = 3.86$ Hz

Small Motions

$f_{p2} = 4.90$ Hz	$f_{p2} = 6.57$ Hz

Roll Uncoupled Natural Frequencies

Single degree of freedom approximations were made with the assumption that the centers of rotation would be at the top of the rail for low center roll and at the center of gravity of the loaded boxcar for high center roll.



Low Center Roll

$$K_R = \cfrac{2}{\cfrac{1}{(K(1) + K(3)) \ x \left(\frac{R(1)}{2}\right)^2 + K(2) \ x \left(\frac{HAXL}{2}\right)^2} + \cfrac{1}{K(6) + K(5) \ x \left(HAXL + \frac{HTRX}{2}\right)^2}}$$

$$I_R = I(3) \ x \ \frac{M(3) + 4 \ x \ MLAD}{M(3)} + (M(3) + 4 \ x \ MLAD)$$

$$x \quad (H/2 + HTRK + HAXL)^2$$

High Center Roll

$$K_R = \cfrac{2}{\cfrac{1}{(K(1) + K(3)) \ x \left(\frac{R(1)}{2}\right)^2 + K(2) \ x \ (H/2 + HTRK + HAXL/2)^2} + \cfrac{1}{K(6) + K(5) \ x \ (H/2 + HTRK/2)^2}}$$

$$I_R = I(3) \ x \ \frac{M(3) + 4 \ x \ MLAD}{M(3)}$$

Model 1	Model 2
K(1) = .91E5 = K(3) lb./in.	
K(2) = .95E5 lb./in.	K(2) = .35E6 lb./in.
K(5) = .16E5 lb./in.	K(5) = .67E4 lb./in.
K(6) = .6185E8 lb. in./rad	(for large motions)
KCP6 = .20E8 lb.in./rad	(for small motions)
R(1) = 58 in.	
HAXL = 16.5 in.	
HTRK = 9.0 in.	
H/2 = 37.15 in.	H/2 = 21.18 in.

Low Center Roll - large amplitude

K_R = .9624E8 in. lb./rad	K_R = .9486E8 in. lb./rad
I_R = .2587E7 lb. in. $sec.^2$	I_R = .1050E7 lb. in. $sec.^2$

Low Center Roll - small amplitude

K_R = .46265E8	K_R = .4064E8

High Center Roll - large amplitudes

K_R = .2151E9	K_R = .1308E9
I_R = .9673E6	I_R = .5693E6

Low Center Roll - small amplitude

K_R = .920E8	K_R = .5847E8

Model 1			Model 2		
I(3)	=	.821E6 in. 1.b sec.^{2}/in.	I(3)	=	.514E6 in. lb. sec.^{2}/in.
M(3)	=	350.2 lb. sec.^{2}/in.	M(3)	=	199.3 lb. sec.^{2}/in.
MLAD	=	15.6 lb. sec.^{2}/in.	NKAD	=	5.36 lb. sec.^{2}/in.

Roll Natural Frequencies, Hertz.

$$f_r = \frac{1}{2\pi}\sqrt{\frac{K_R}{I_R}}$$

	Model 1	Model 2
low center roll		
large amplitudes	.97	1.51
small amplitudes	.67	.99
high center roll		
large amplitudes	2.37	2.41
small amplitudes	1.55	1.61

<u>Yaw Uncopled Natural Frequencies</u>

$$K_y = \frac{1}{\frac{1}{K(2)} + \frac{1}{K(5)}} \times \left(\frac{L}{2}\right)^2$$

$$I_y = I(11) \times \frac{M(3) + 4\ \text{MLAD}}{M(3)}$$

$$f_y = \frac{1}{2\pi}\sqrt{\frac{K_y}{I_y}}$$

		Model 1	Model 2
K(2)	lb./in.	.95E5	.35E6
K(5)	lb./in.	.16E5	.67E4
L	in.	474	474
I(11)	$lb.in.sec.^2$	.111E8	.656E7
M(3)	$lb.sec.^2/in.$	350.2	199.3
MLAD	$lb.sec.^2/in.$	15.6	5.36
K_y	in.lb./rad	.1538E10	.7385E9
I_y	$lb.in.sec.^2$	.1308E8	.7266E7
f_y	Hertz	1.73	1.60

APPENDIX C

COMPUTER COST ESTIMATES

Computational costs consist essentially of the calculation and display of the responses. Because of the advances made in the CDC computers actual calculation costs have been reduced to the point where results printout are a major cost factor and cannot be ignored.

The computational cost of performing this analysis was estimated by assuming each run consisted of three cost elements. The first was based on the value of SBU's (system billing units). Two priorities were available: P4 for fast turnaround at the cost of $1.15 per SBU and P2 for overnight running at $.95 per SBU. Second are the input/output costs, mostly dependent on the amount of output. The third cost included terminal connect time costs, the cost of transmitting, data to and from the 7600 in Minneapolis and an estimate of miscellaneous service and rental costs.

There were a total of 80 runs made in the performance of the analyses for this report with the following cost total breakdown:

I/O	$886
Calculation	565
Terminal Connect	400
Total Cost (80 runs)	$1851

The number of simulated seconds and the number of plotted curves are considered in arriving at a cost per simulated second for each curve. Every run produced 40 curves; 4 input motions, 20 body acceleration responses; 4 body rotational displacements, 4 lading accelerations, 4 lading deflections and 4 forces at the wheel/rail interfaces. Then, if a 6 second simulation run costs $30, the cost was \$5.00 per simulation second or 12.5¢ per simulation second per plot.

APPENDIX D

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