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RAILROAD RESEARCH AND DEVELOPMENT

SUMMARY RESULTS OF 70 TON BOXCAR TESTING

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DEPARTMENT OF TRANSPORTATION ★ UNITED STATES OF AMERICA ★

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16. Abstract

Tests were performed at the Transportation Test Center in Pueblo, Colorado on a 70 ton boxcar with Barber S-2C trucks in April and May of 1981. The objective of the tests was to provide definition of the dynamic properties of the freight car for application to the validation of a mathematical model. The testing was conducted in two phases: static tests were performed on each truck to characterize their stiffness and damping properties; and vibration tests were performed on the complete boxcar, loaded and empty, to determine resonant frequencies. This report presents final results of the truck stiffness tests and preliminary results of vibration tests. A comparison is presented between tests results and the math model to be validated.

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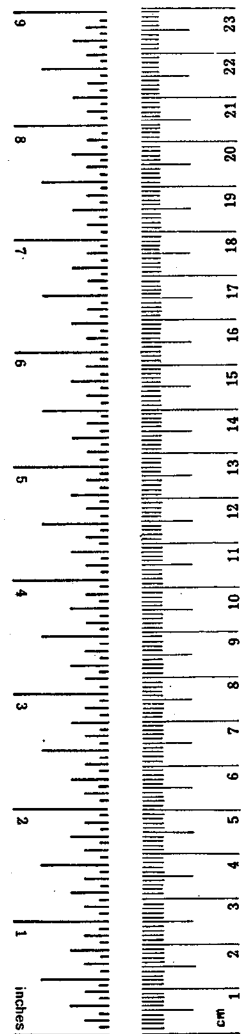
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

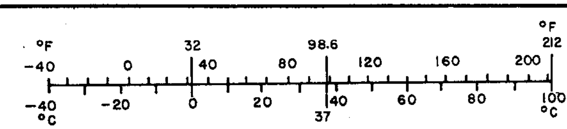
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	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



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EXECUTIVE SUMMARY

The Federal Railroad Administration (FRA) has sponsored a parallel effort of test and analysis of a 70 Ton Boxcar as part of its Freight Safety Research Program. The two fold objective of the program is (1) to develop and demonstrate an analysis validation procedure and (2) to provide an analytical capability of accurately predicting over-the-road response calculations. The analysis tool to be validated is the computer program FRATE, version FRATX1, as developed by the MITRE Corporation.

The testing, performed at the Transportation Test Center (TTC), Pueblo, Colorado, by Rail Dynamics Laboratory personnel, was conducted in two phases. The first was a characterization test performed on each truck to measure its stiffness and damping properties. These were quasi static tests performed by the application of sinusoidally varying forces and moments where the frequency of the sine was 0.1 or 0.2 Hertz which is well below any resonant conditions. The second phase consisted of vibration tests performed on the complete boxcar, loaded and empty, with and without friction snubbers.

This report presents final results from the Truck Characterization Test and a summary of preliminary results from the Boxcar Vibration Tests.

Truck Characterization Test

Both trucks were removed from the test boxcar and each tested separately. Test configurations were varied to show the effect of friction snubbers and freight car gross weight variation. The general test procedure was to support each truck in a test fixture in a manner duplicating service conditions, with truck wheels resting on fixed sections of rail. Loads were applied to the truck bolster at centerplate and side bearing through a fixture duplicating the carbody bolster interface.

Three types of tests were performed: (1) vertical loadings to obtain the vertical spring rates and snubber forces; (2) roll loadings to obtain roll spring/snubber values; and (3) lateral loadings to obtain the lateral spring/snubber values.

Boxcar Vibration Tests

Vibration tests were performed on the DOTX 503 70 Ton Boxcar using the Vibration Test Unit (VTU) located at the Transportation Test Center (TTC), Pueblo, Colorado. The tests were conducted in the period starting 6 May 1981 and ending 5 June 1981.

The basic objective of the vibration testing was to obtain definitions of the vibration characteristics of the boxcar and lading which can be used for comparisons with the respective analytical models. The vibration characteristics to be compared are the resonant frequencies, the deflection shape at each resonance, the damping associated with each resonance, nonlinear effects with respect to amplitude of motion and the effects of certain configurational changes.

There were nine resonant frequencies targeted for measurement. Five of these involved rigid body motions of the carbody on the truck suspension system with motions described as first roll, second roll, yaw, bounce and pitch. Two were the flexible carbody modes of torsion and vertical bending and two were resonances of the lading in lateral and vertical motion.

Tests were also conducted on the VTU simulating over-the-road conditions. These data, however, were not included in this preliminary analysis but will be reported on in the final boxcar vibration test report.

Summary of Test Results and Comparison with FRATE Model

A summary of test results with comparisons to the FRATE model is given in Table ES-1 for the truck characterization test and Tables ES-2 for the boxcar vibration test. From the truck test comparison it is seen that some parts of the FRATE model are too stiff and other parts are too soft. One general difference is that the FRATE model does not take into account the extent of change for load condition that is shown by the test results.

The first five modes of vibration test results without snubbers compares very well with FRATE analysis results with the exception of the second roll mode. The FRATE model did include a coulomb damping simulation of the friction snubbers using a friction force of 3,000 pounds. The truck characterization test showed the friction force to vary from 2,500 to 5,000 pounds depending on lading configuration. Despite the fact that the FRATE model included coulomb damping at a value within the range of test measurements, the FRATE results were closer to test results without snubbers than with snubbers. This leads to the initial conclusion that coulomb friction simulation needs to be modified in the FRATE program.

TABLE ES-1 COMPARISON OF FRATE MODEL WITH TRUCK CHARACTERIZATION TEST RESULTS

DESCRIPTIONS	TEST	FRATE	UNITS
<u>Vertical Spring Rates</u>			
Wheel & Side Frame: K(1) & K(3)	(1) 5.8 E5 (2) 9.2E5	.91E5 .91E5	LB/IN
Suspension: K(4)	(1) 3.82E4 (2) 4.96E4	4.8E5 4.8E5	LB/IN
<u>Lateral Spring Rates</u>			
Wheel & Side Frame: K(2)	(1) 1.34E5 (2) 2.16E5	.95E5 .95E5	LB/IN
Suspension: K(5)	(1) 1.50E4 (2) 3.77E4	.42E4 1.71E4	LB/IN
<u>Roll Spring Rates</u>			
Seated Center Plate:	(1) 1.86E7 (2) 2.81E7	(3)	LB IN/Rad.
Averaged Seated & Rocking Center Plate: KCPB	(1) 1.05E7 (2) 1.97E7	2.0E7 2.0E7	LB IN/Rad.
Side Bearing Contact: K(6)	(1) 4.18E7 (2) 5.23E7	7.5E7 7.5E7	LB IN/Rad.
<u>Friction Snubber</u>			
Local Structure: KLSB	(1) 0.63E5 (2) 2.34E5	1.0E7 1.0E5	LB/IN
Friction Force: MFSB	(1) 0.25E4 (2) 0.58E4	0.10E4 0.30E4	LB

(1) Empty Carbody Weight Condition
(2) 70 Ton Load Condition
(3) No Directly Comparable Model Spring

TABLE ES-2 RESONANT FREQUENCY COMPARISON OF FRATE MODEL WITH BOXCAR VIBRATION TESTS, HERTZ

MODE	FRATE PREDICTION	CONFIGURATION 1B (WITHOUT SNUBBERS)	CONFIGURATION 3 (WITH SNUBBERS)
1st Roll	0.7	.63 - .70	.68 - .88
2nd Roll	1.6	2.6	3.8
Yaw	1.7	1.7	2.4
Bounce	2.2	2.05	2.4
Pitch	2.9	2.77	3.8
Body Torsion	--	12.4	12.7
Body Bending	--	16.0	15.8 - 17.2
Lading Lat.	5.0	3.0	2.3 - 2.4
Lading Vert.	9.5	8.3	8.0 - 8.5

Conclusions

The truck characterization test was a success in that the desired data on truck stiffness and damping were obtained. The roll data were of particular value since their form and content are not available elsewhere and the results will be of considerable value in improving the accuracy of the FRATE model.

The boxcar vibration test was also considered a success on the basis of preliminary results. All targeted resonances were identified. The effects of snubbers and load conditions were measured.

Data processing problems were encountered in the analysis of the vibration test results without satisfactory solutions being obtained. The source of data processing problems are without doubt the very nonlinear characteristics of the boxcar suspension system and the type of accelerometer used in the test. All modal test techniques and computerized analysis methods in current use were developed for linear, lightly damped systems, and problems are to be expected in their application to freight cars. It is expected that at the completion of the analysis of the results of this testing it will be possible to make some positive recommendations on test and data analysis procedures.

The one recommendation to be made now is that a different accelerometer should be used. The ideal accelerometer will have a flat response from DC to 20 Hertz with roll-off of 6db per octave or greater and with linear range of $\pm$ 5.0g. The accelerometer must be capable of withstanding high frequency shock and vibration of 100g or greater.

A more detailed analysis of the vibration test results is continuing, following which the FRATE computer program will be validated using the test results as reference criteria. Reports will be issued presenting the final results of both of these efforts. The FRATE User's Manual will be updated to reflect the results of the validation.

1. INTRODUCTION

The Federal Railroad Administration (FRA) has sponsored a parallel effort of test and analysis of a 70 Ton Boxcar. The two fold objective of the program is (1) to develop and demonstrate an analysis validation procedure and (2) to provide an analytical capability of accurately predicting over-the-road response calculations.

The analysis tool to be validated is the computer program FRATE, version FRATX1, as developed by the MITRE Corporation. A description of FRATE can be found in the User's Manual, Reference 1.* The validation procedure which will be followed is contained in Reference 2.

All testing was performed at the Transportation Test Center (TTC), Pueblo, Colorado, by Rail Dynamics Laboratory personnel. The testing was performed according to the agreements contained in Reference 3 and the procedures specified in References 4 and 5. Key personnel in test performance were William Walters, Test Manager (FRA/TTC) and Danny Inskeep, Test Engineer (O&M/TTC). The testing was completed on June 5, 1981. The purpose of this report is to present a summary of preliminary test results with a comparison to the boxcar model in FRATE.

Testing was conducted in two phases. A characterization test was performed on each truck to measure its stiffness and damping properties. These were quasi static tests performed by the application of sinusoidally varying forces and moments where the frequency of the sine was 0.1 or 0.2 Hertz which is well below any resonant conditions. The second phase of testing was performed on the complete boxcar, loaded and empty, with and without friction snubbers.

The truck characteristics tests are presented in Sections 2.4, the boxcar vibration tests are presented in Section 3. Section 4 contains a comparison of the results from both tests to the FRATE boxcar model.

*The List of References is at the end of the report.

2. TRUCK CHARACTERIZATION TEST

The objective of the Truck Characterization Test was to obtain a measure of the stiffness and damping of the truck in a form that could be used to compare with values used in the boxcar model of Reference 1, in vertical, lateral and roll motions.

Both trucks of the test boxcar were tested with test configurations varied to show the effect of friction snubbers and freight car gross weight conditions. Due to recording problems, only data from the B truck was available for analysis, and consequently, only B truck data is presented in this report.

The general test procedure was to support each truck in a test fixture in a manner duplicating service conditions; i.e., the truck wheels resting on fixed sections of rail with loads applied to the truck bolster at centerplate and side bearing through a fixture duplicating the carbody bolster interface.

Three types of tests were performed: (1) vertical loadings to obtain the vertical spring rates and snubber forces; (2) roll loadings to obtain roll spring/snubber values and (3) lateral loadings to obtain the lateral spring/snubber values.

A sketch of the test configuration is shown in Figure 2.1. Roll moments were applied with differential loads in the two vertical actuators. Loads were applied with a sinusoidal variation in order to eliminate static friction effects. The frequency of load applications was 0.1 or 0.2 Hertz which was slow enough so as not to introduce any dynamic effects.

Measurements made during the testing are listed in Table 2-1. The data was recorded on analog tape and played back as load displacement plots with an x-y plotter. Figure 2.2 is a typical plot from the vertical test with the displacement measurement D15 plotted against the load measurement L19. Spring rate determinations were made from the slope of the load deflection line as indicated in Figure 2.2.

2.1 Truck Model Used in FRATE

A schematic of the truck math model is shown in Figure 2.3 with the spring damper notation for the B truck. This figure will be referred to in the ensuing discussion to relate measured values to model values. The A truck has a corresponding set of spring dampers with a continuing numbering system, i.e., K(7) in the A truck corresponds to K(1) in the B truck, K(8) to K(2), etc.

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NOTE: This sketch is not to scale and does not show all structural steel construction.

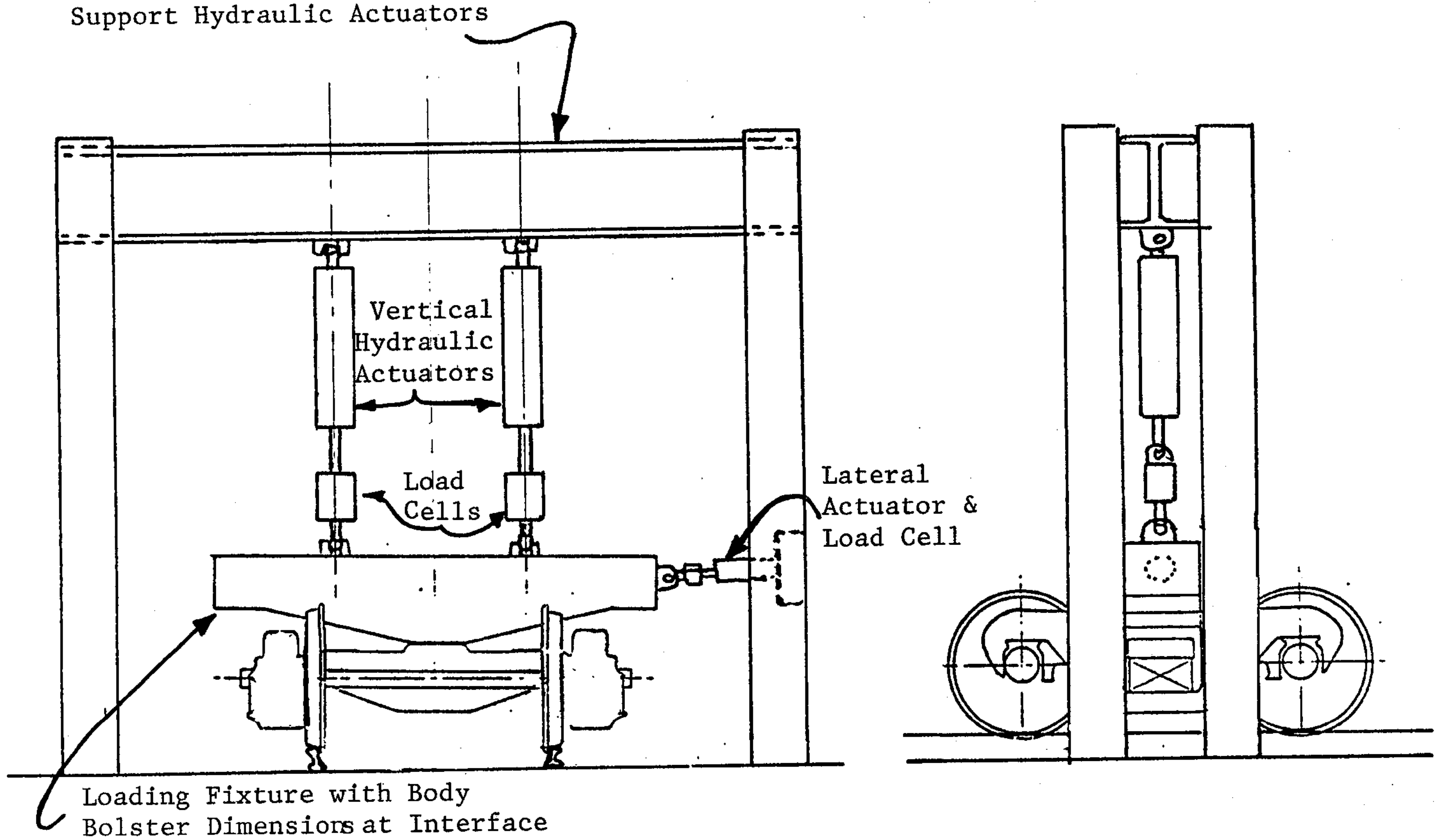


FIGURE 2.1 TRUCK CHARACTERIZATION TEST CONFIGURATION

TABLE 2-1 LIST OF MEASUREMENT TRUCK CHARACTERIZATION TEST

Measurement Number	Measurement Description
D-1	Center of truck bolster, vertical to ground
D-2	Right side of truck bolster, spring nest centroid, vertical to ground
D-3	Left side of truck bolster, spring nest centroid, vertical to ground
D-4	Center of right side frame at spring nest centroid, vertical to ground
D-5	Center of left side frame at spring nest centroid, vertical to ground
D-6	Front end of right side frame at axle 1 C_L, vertical to ground
D-7	Front end of left side frame at axle 1 C_L, vertical to ground
D-8	Rear end of right side frame at axle 2 C_L, vertical to ground
D-9	Rear end of left side frame at axle 2 C_L, vertical to ground
D-10	Truck bolster lateral to ground
D-11	Front end of right side frame at axle 1 C_L, lateral to ground
D-12	Front end of left side frame at axle 1 C_L, lateral to ground
D-13	Rear end of right side frame at axle 2 C_L, lateral to ground
D-14	Rear end of left side frame at axle 2 C_L, lateral to ground
D-15	Right side, vertical, carbody bolster to side frame
D-16	Left side, vertical, carbody bolster to side frame
D-17	Right side frame, lateral to ground at mid height of wear plate
D-18	Left side frame, lateral to ground at mid height of wear plate
L-19	Right side, carbody bolster, vertical load to ground
L-20	Left side, carbody bolster, vertical load to ground
L-21	Right side, carbody bolster, lateral load to ground
D-22	Right side carbody bolster, lateral displacement to ground

TABLE 2-1 LIST OF MEASUREMENTS TRUCK CHARACTERIZATION TEST (Concluded)

Measurement Number	Measurement Description
D-23	Right side carbody bolster, vertical displacement to ground
D-24	Left side carbody bolster, vertical displacement to ground
D-25	Front axle, left side, lateral displacement to ground
D-26	Rear axle, left side, lateral displacement to ground

NOTES:

D - are displacement measurements
L - are load measurements
CL - abbreviation for Center Line

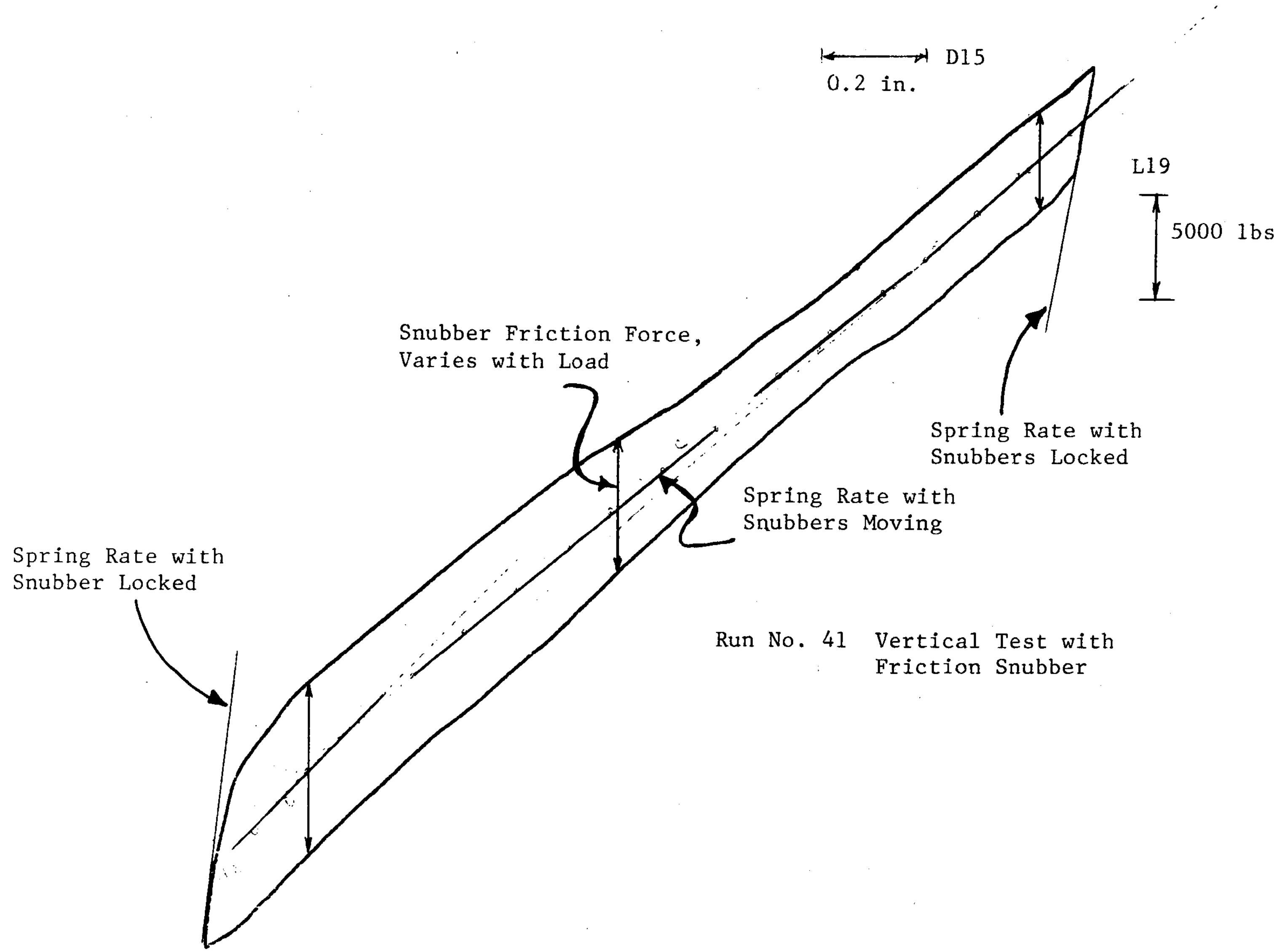
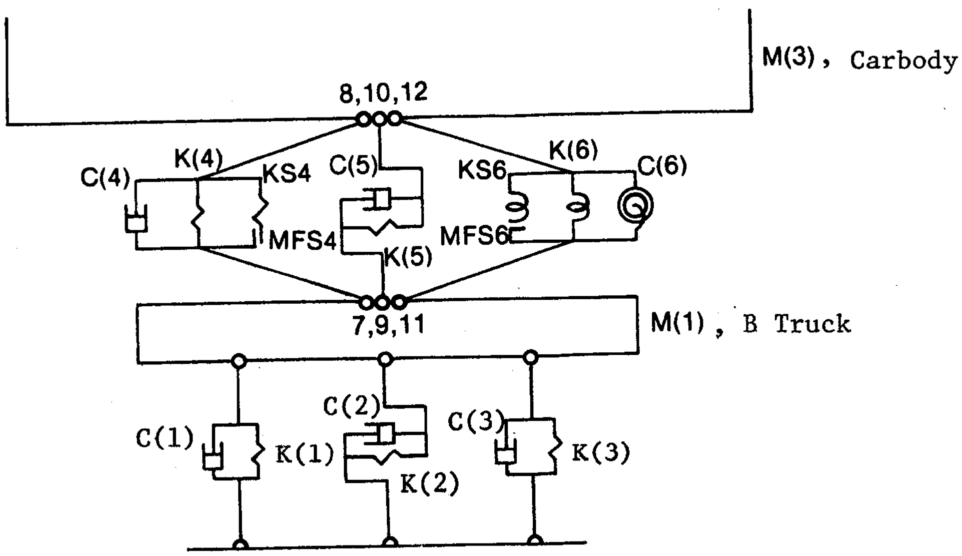
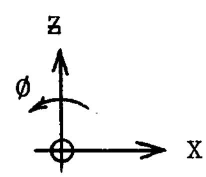


FIGURE 2.2 EXAMPLE LOAD-DEFLECTION PLOT TRUCK CHARACTERIZATION TEST





Z = Vertical

X = Lateral

Ø = Roll

FIGURE 2.3 FRATE TRUCK MODEL

Spring damper numbers (1) and (3) represent the vertical stiffness of track, wheels and side frame of each side and (2) represents the lateral stiffness of the total truck. The spring damper network for the truck suspension system is modeled differently in that the vertical, lateral and roll characteristics are represented separately by numbers (4), (5) and (6) respectively. KS4 and KS6 represent structure local to the snubbers and MFS4 and MFS6 the snubber friction forces.

2.2 Vertical Test Results

The deflections analyzed for determining vertical spring and damper rates were D4, D5, D15 and D16. The locations of these measurements are shown in Figure 2.4. Measurements D4 and D5 were used in developing the side frame--wheel set spring rates, corresponding to K(1), K(3), K(7) and K(9) in the FRATE model. The test results were output as load-deflection plots and are reproduced here in Figure 2.5 from the test with no snubbers and Figure 2.6 from the test with snubbers.

2.2.1 Vertical Stiffness of Wheels and Sideframe

Average values of spring rates were obtained in three steps. First, the average of the up and down stroke load-deflection plot was obtained, and the slope of these plots were measured graphically and tabulated as a function of load. Second, the average spring rates between right and left were obtained. Finally, the average spring rates between tests with and without snubbers were obtained. Figure 2.7 is a plot of the spring rates obtained in the second and third steps. This averaging process is diagramed below.

Spring Rate Averaging Process

1st Averaging	Up	Down	Up	Down	Up	Down	Up	Down
2nd Averaging	Right		Left		Right		Left	
3rd Averaging	With Snubbers				Without Snubbers			
	Final Values							

The load-deflection plots, Figures 2.5 and 2.6, need some discussion. Recall that these plots were obtained by a sinusoidal variation of the applied loads at a rate of 0.10 Hertz. The plots were made with an x-y plotter using pen and

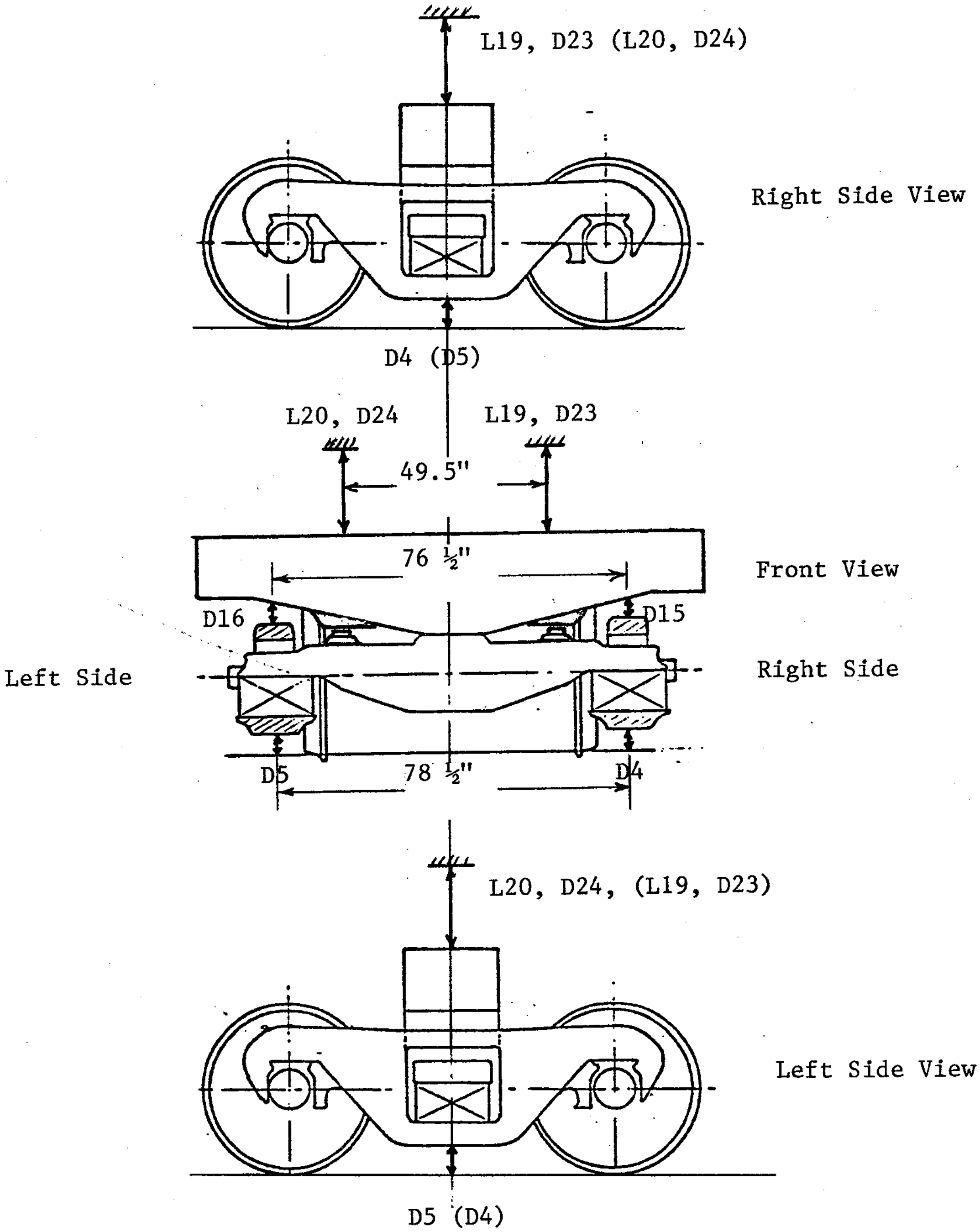


FIGURE 2.4 LOCATIONS OF LOAD APPLICATION AND DISPLACEMENT

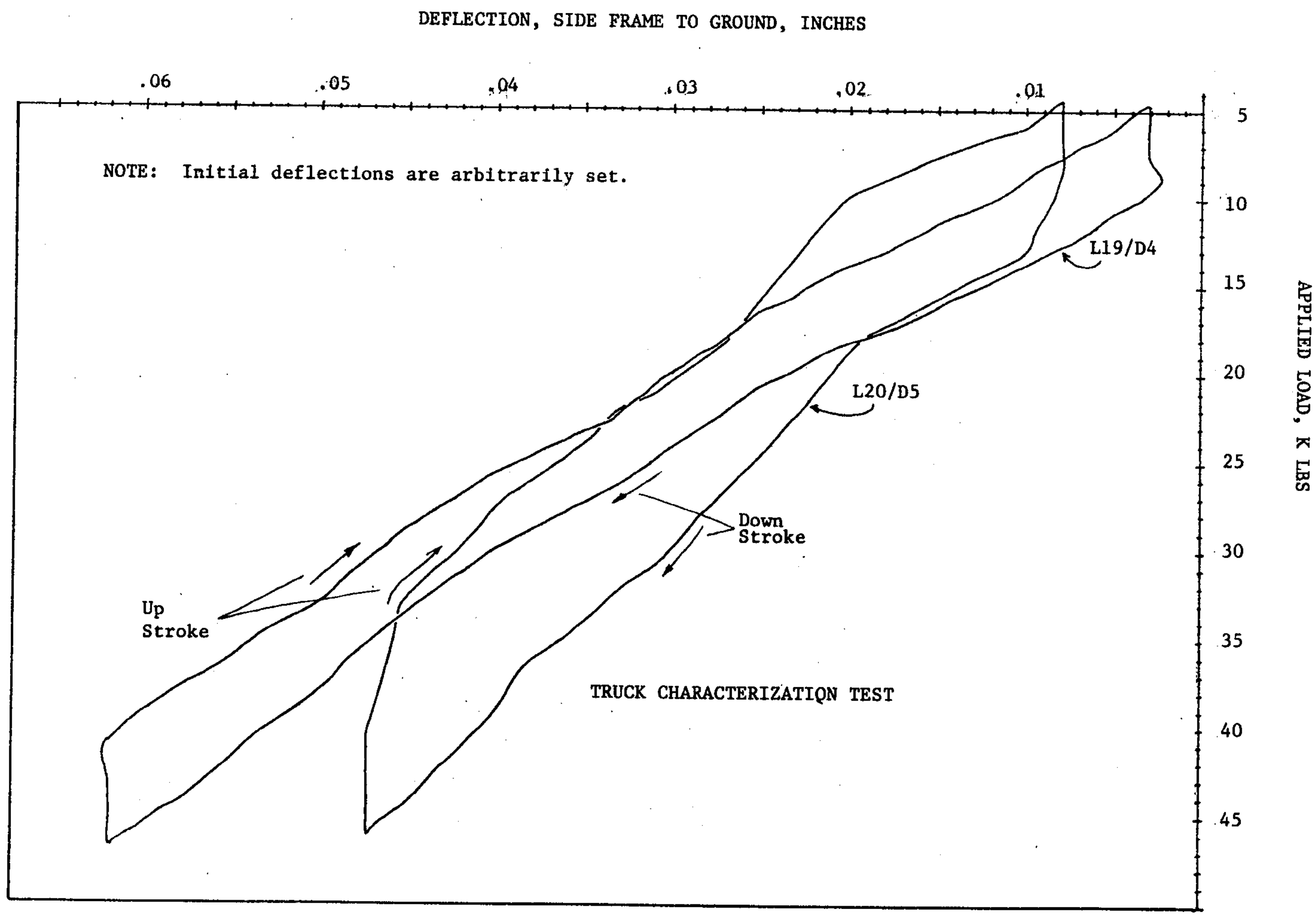


FIGURE 2.5 VERTICAL TEST, RUN 40, NO SNUBBERS, LOAD-DEFLECTION PLOT OF WHEELS AND SIDE FRAME

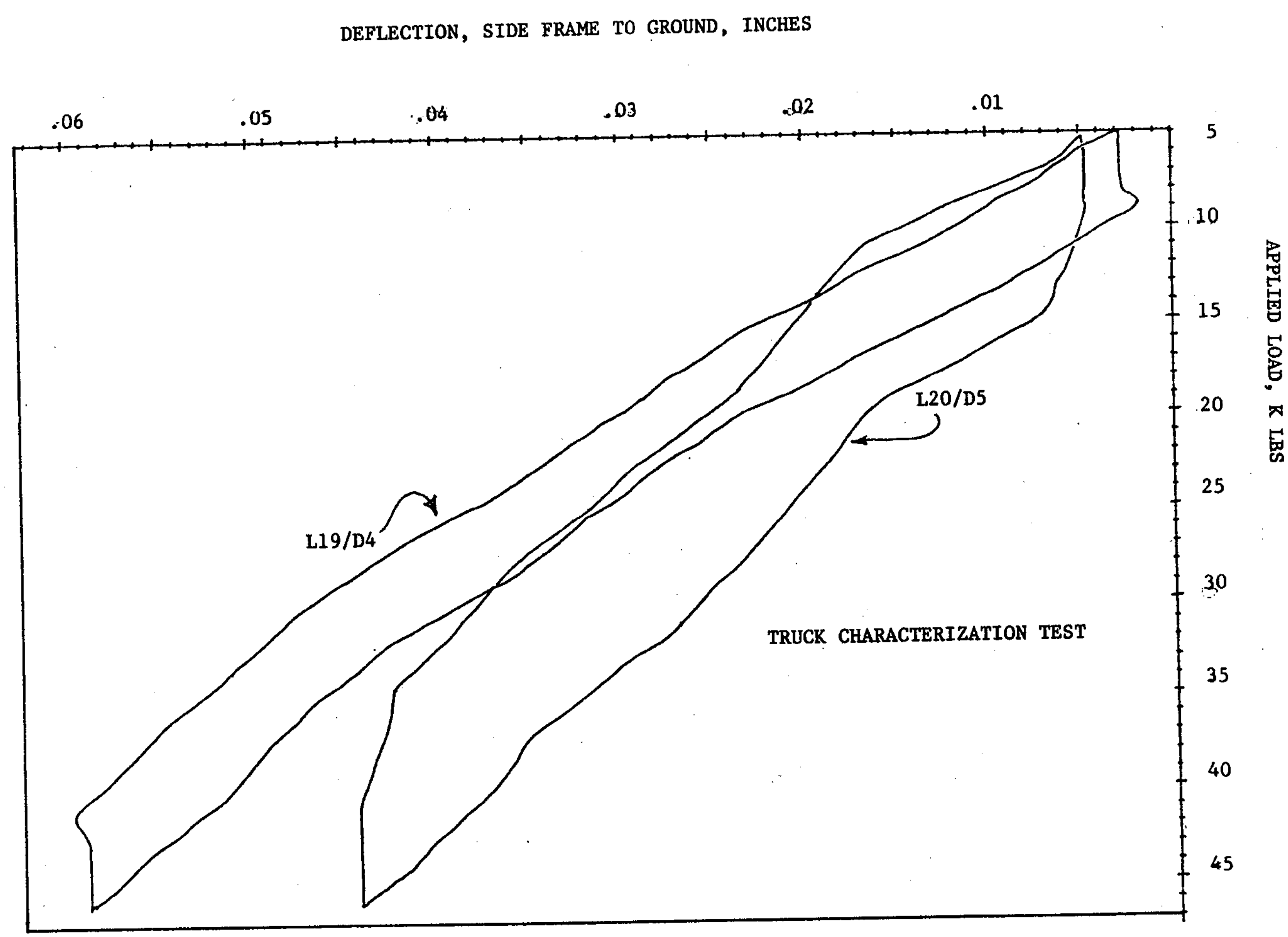


FIGURE 2.6 VERTICAL TEST, RUN 41, WITH SNUBBER, LOAD-DEFLECTION PLOT OF WHEELS AND SIDE FRAME

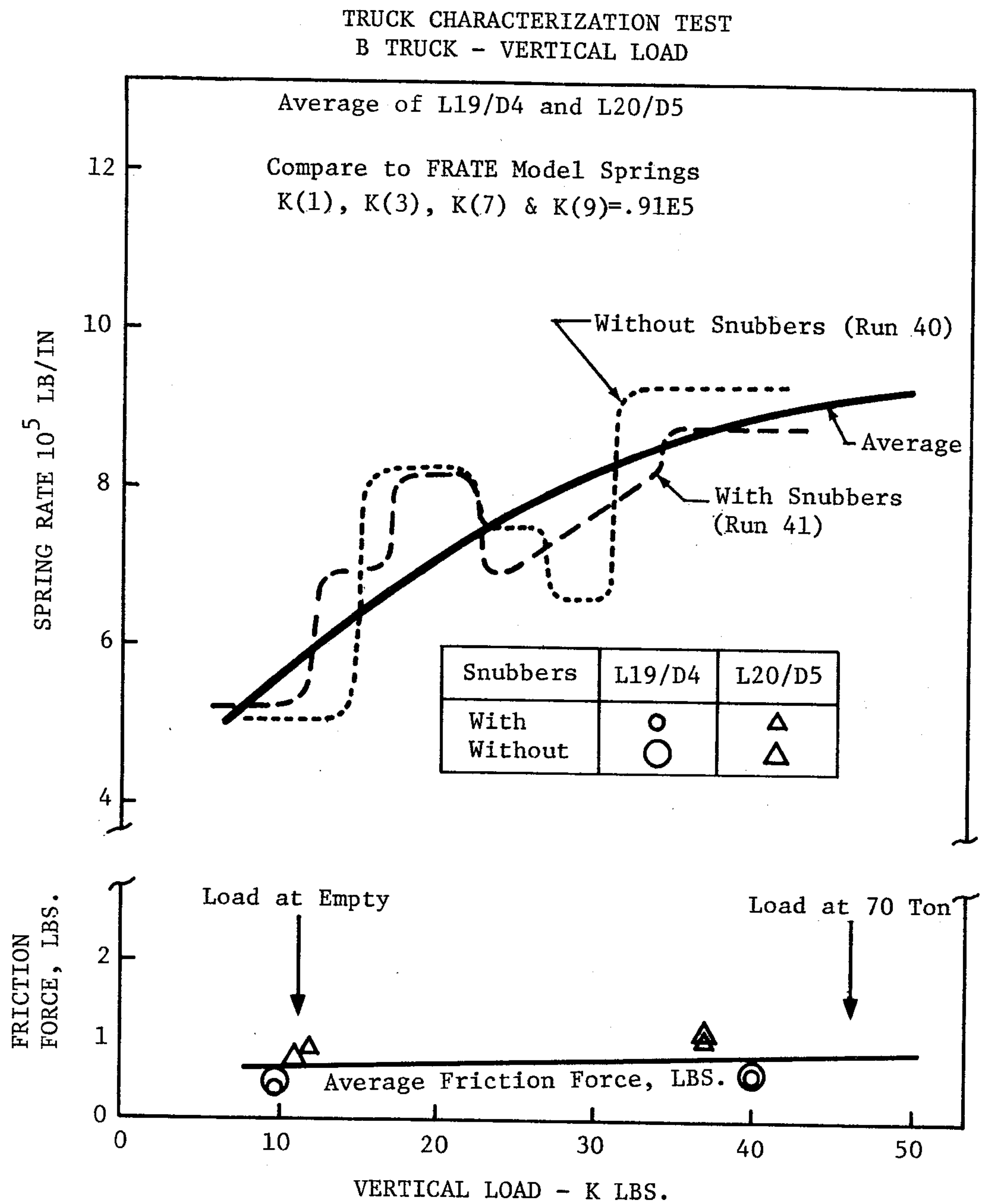


FIGURE 2.7 VERTICAL SPRING RATE—SIDE FRAME CENTER TO GROUND, PER TRUCK

ink, allowing the plotter to run for several cycles. Very little variation was noted from cycle to cycle. The left side, L20/D5, is seen to be generally stiffer than the right side and was found to be about twice as stiff at certain loads. Also, the hysteresis loop effect leads to the conjecture that the deflections are due to a combination of structure deformation and joint slippage, and that the joint friction in the left side is greater than the right side. It should also be noted that these spring rates are not significantly influenced by snubber condition.

The concluding remarks on Figure 2.7 are that the vertical stiffness of the wheels and side frame varies with load. Also because of the significant difference between right and left side it must be assumed that this stiffness will vary significantly from truck to truck as well as being unsymmetric.

2.2.2 Vertical Stiffness of Truck Suspension

Analysis of the data for truck vertical suspension stiffness paralleled that for wheel and side frame. Figure 2.8 and 2.9 present the load-deflection plots used and Figure 2.10 presents the spring rate and snubber force values finally obtained showing varation with load condition. Referring back to Figure 2.3, the spring rates in Figure 2.10 correspond to the spring rate K(4)/2 and the friction force corresponds to MFS4. The local structure spring, KS4, corresponds to the slope of the load-deflection plot at either end of the stroke from the test with snubbers, Figure 2.9. The test obtained values for KS4 were also obtained from the roll test and are presented in Section 2.4.

2.3 Lateral Test Results

The objective of the lateral tests was to obtain the lateral spring rates between the carbody bolster and side frames and between the side frames and ground. These correspond to K(5) and K(2) respectively in the FRATE model as shown in the schematic of Figure 2.3. The location and notation of the load application and deflection measurements are shown in Figure 2.11.

The tests were run with four different vertical loads. The combined loads applied by L19 and L20 were 25,000, 50,000, 75,000 and 93,000 pounds for runs 34, 35, 36 and 37 respectively. The loads applied by L19 and L20 varied in the course of each test as needed to keep the carbody fixture horizontal. This was done automatically by controlling L19 and L20 to maintain constant displacement of D23 and D24.

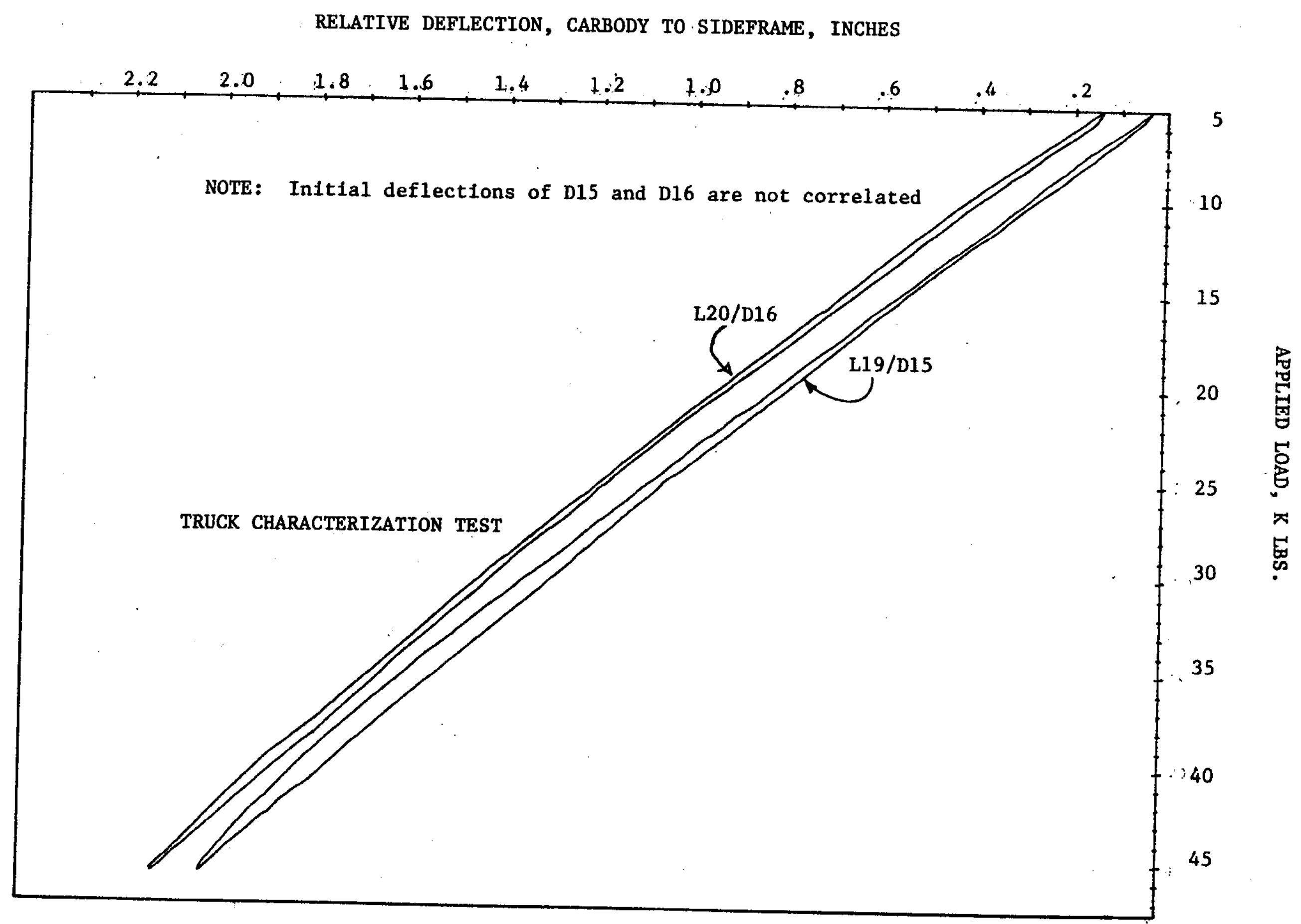


FIGURE 2.8 VERTICAL TEST, RUN 40, NO SNUBBER LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION, VERTICAL

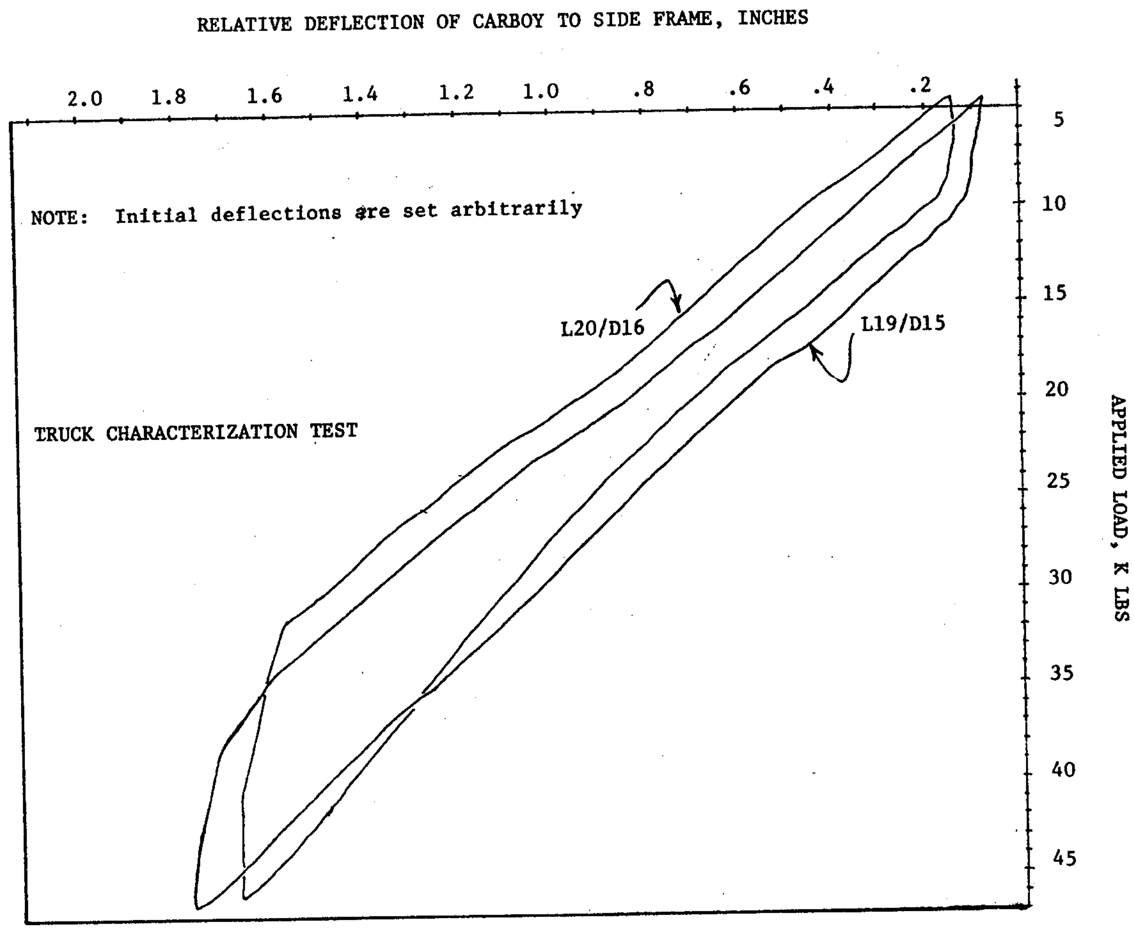


FIGURE 2.9 VERTICAL TEST, RUN 41 WITH SNUBBER, LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION, VERTICAL

TRUCK CHARACTERIZATION TEST
B TRUCK - VERTICAL LOAD

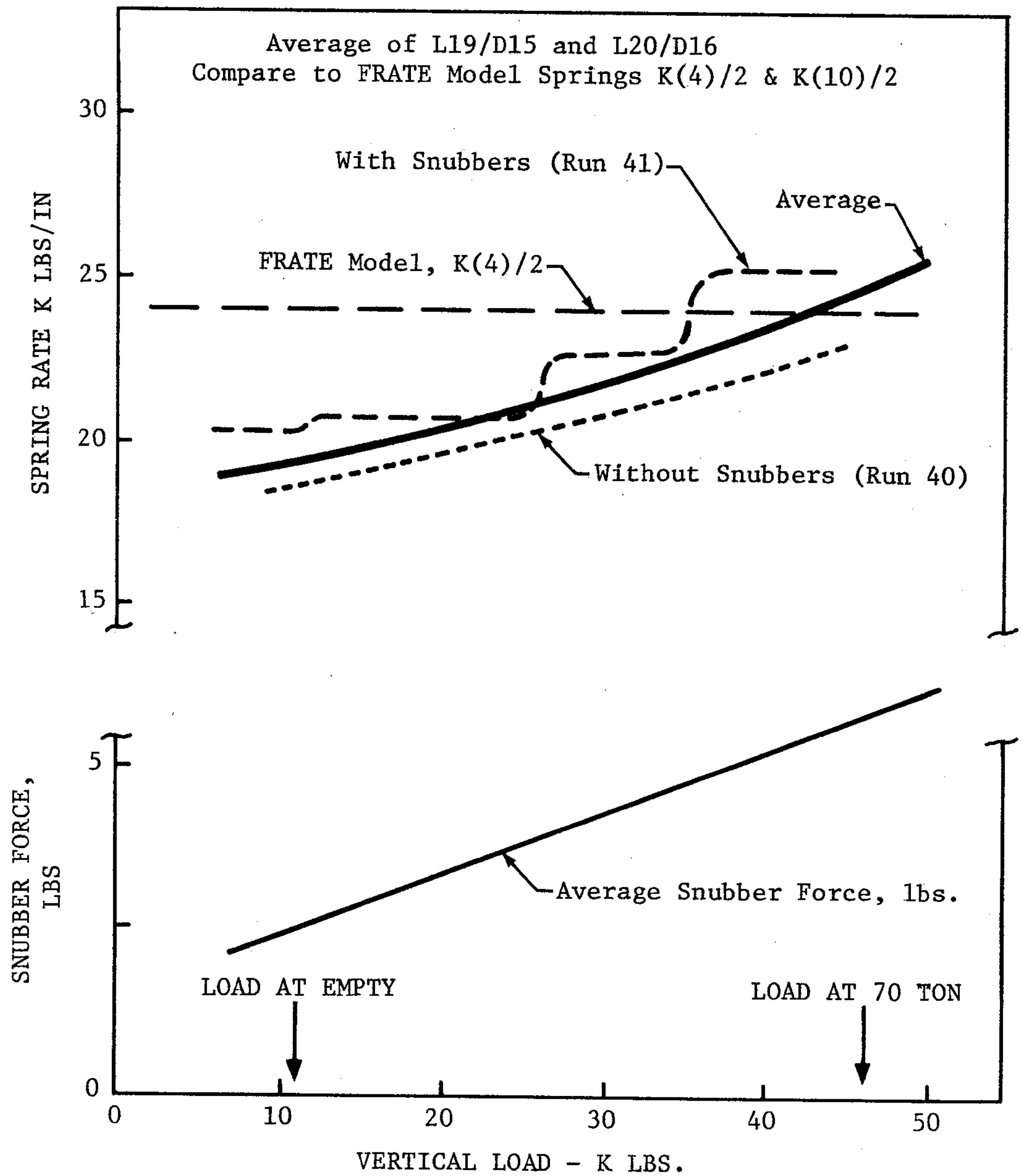


FIGURE 2.10 VERTICAL SPRING RATE, TRUCK SUSPENSION SYSTEM, PER SIDE

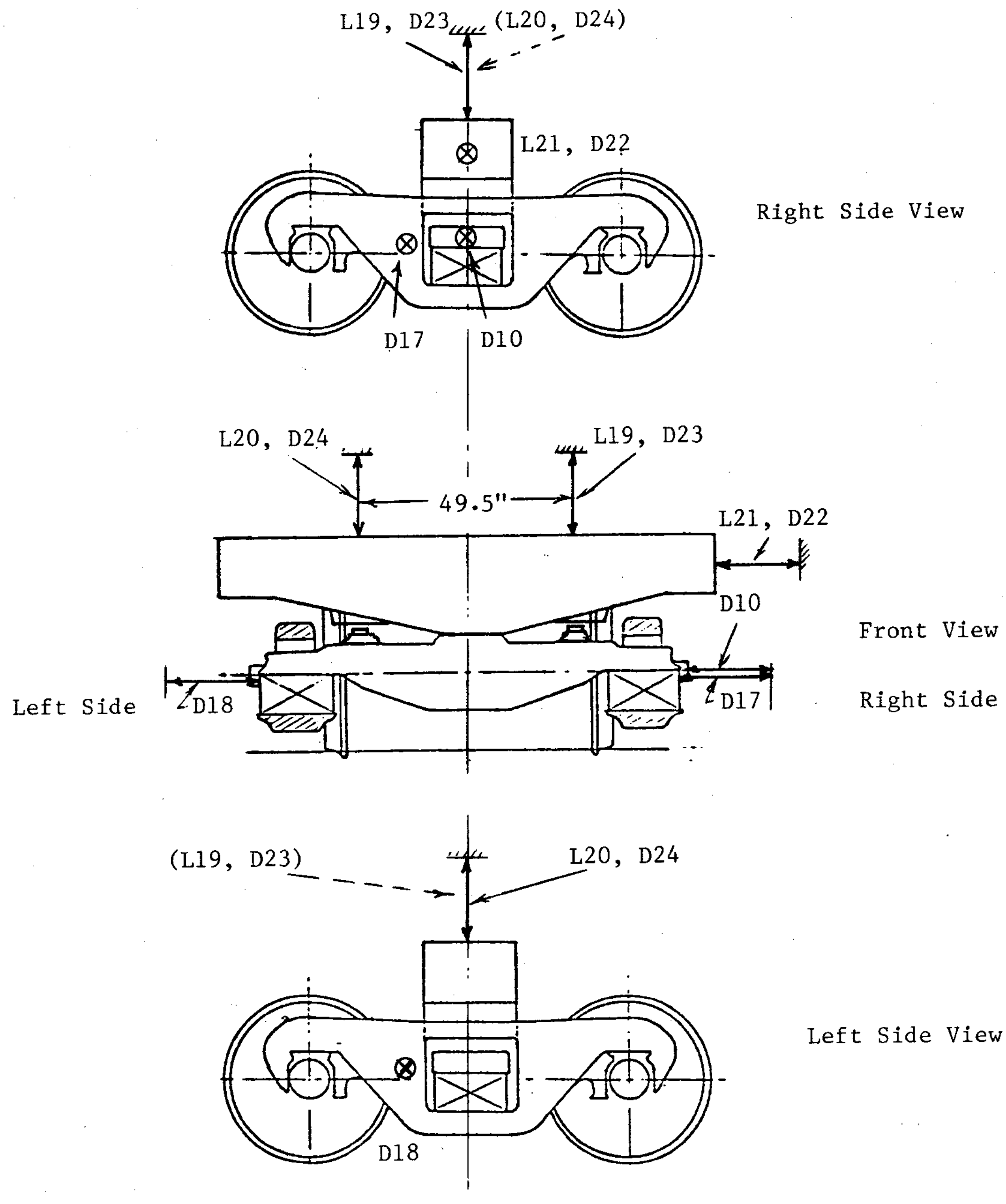


FIGURE 2.11 LOCATION OF LOAD APPLICATION AND DISPLACEMENT MEASUREMENTS FOR LATERAL TESTS

Lateral tests were performed with and without snubbers. However, in the tests with, the snubbers remained locked within the range of actuator load capability. Consequently, only the data from the lateral tests without snubbers were developed and are presented here.

The spring rate calculations to obtain values corresponding to K(2) and K(5) of the FRATE model were made as shown in Figure 2.12. The load-deflection plots of L21/D22, L21/D17 and L21/D18 used to determine these spring rates are shown in Figure 2.13, 2.14. and 2.15. (Note that the deflection scale used in Figures 2.14, and 2.15 differs by a factor of about 20 over Figure 2.13.)

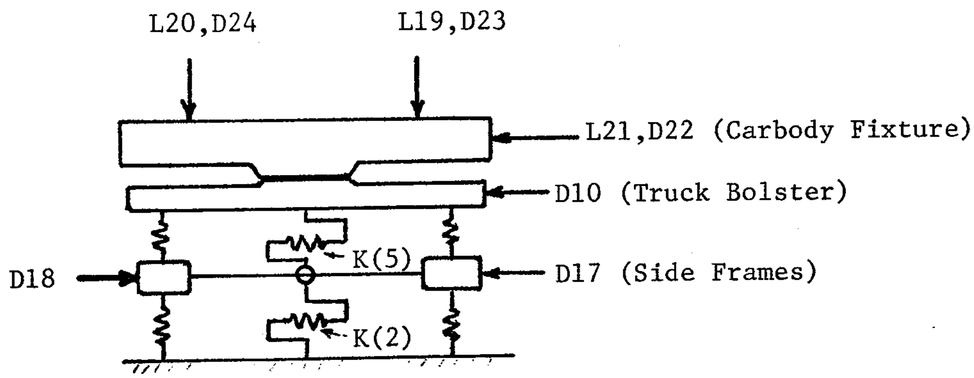
The slope of the load-deflection plot of L21/D22, Figure 2.13, was used to obtain the lateral spring rate of the total truck. There are two observations to be made from these plots which emphasize the nonlinear characteristics of the truck. First, the slopes, hence the spring rates, increase significantly with increase in vertical load. Second, there is no measurable movement at the start of each return stroke until some threshold force has been reached.

The L21/D17 and L21/D18 plots are similar to each other with the slopes in L21/D18 being consistently steeper (i.e., the left side of the truck is stiffer.) There is much larger change in these load-deflection plots between runs 34 and 35 than there is between runs 35, 36 and 37, probably because of a change in the way the truck parts move relative to each other. For example, with lateral load applied there results a rocking motion of the side frames the extent of which varies with both the size of the dead load on the truck and the size of the lateral load.

The slopes of the plots in Figures 2.14 and 2.15 were used to obtain the lateral spring rates plotted in Figure 2.16. The spring rates of the left and right sides are plotted separately to show the difference. These spring rates were combined with the total spring rates obtained from L21/D22 plots according to the relationship shown in Figure 2.12 to arrive at the truck suspension lateral spring rates plotted in Figure 2.17.

2.4 Roll Test Results

Referring back to the FRATE truck model in Figure 2.3, roll motions are accounted for by differential deflection between K(1) and K(3) and by deflection of the roll springs K(6). The roll testing was performed by applying loads at L19 and L20, Figure 2.18, oscillating at 0.1 Hertz and out of phase with each other.

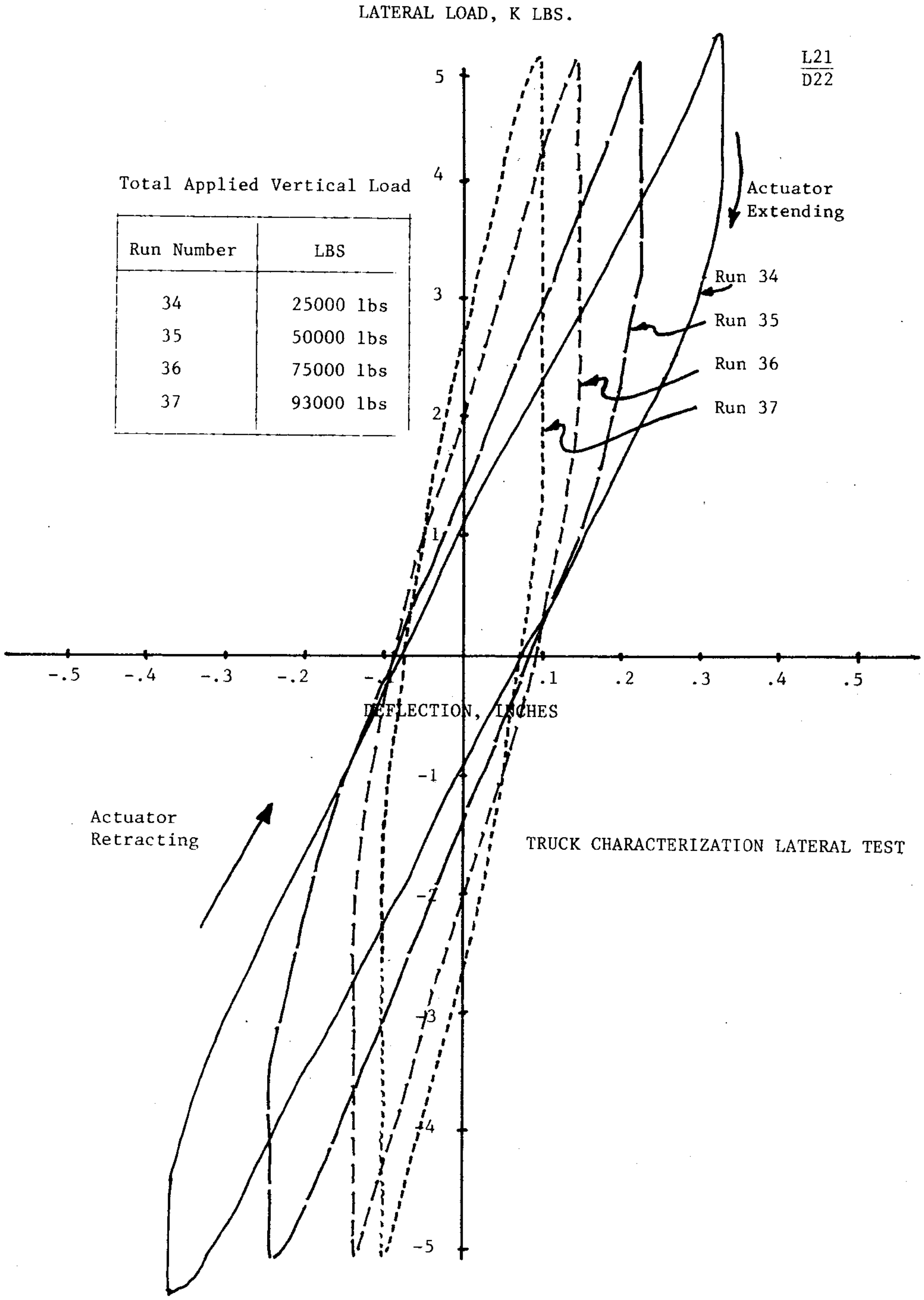


$$K_{total} = \frac{L21}{D22}$$

$$K(2) = \tfrac{1}{2}\left(\frac{L21}{D17} + \frac{L21}{D18}\right)$$

$$K(5) = \frac{1}{\dfrac{1}{K_{total}} - \dfrac{1}{K(2)}}$$

FIGURE 2.12 CALCULATION OF LATERAL SPRING RATES



Total Applied Vertical Load

Run Number	LBS
34	25000 lbs
35	50000 lbs
36	75000 lbs
37	93000 lbs

FIGURE 2.13 LOAD-DEFLECTION PLOTS FOR CARBODY, L21/D22

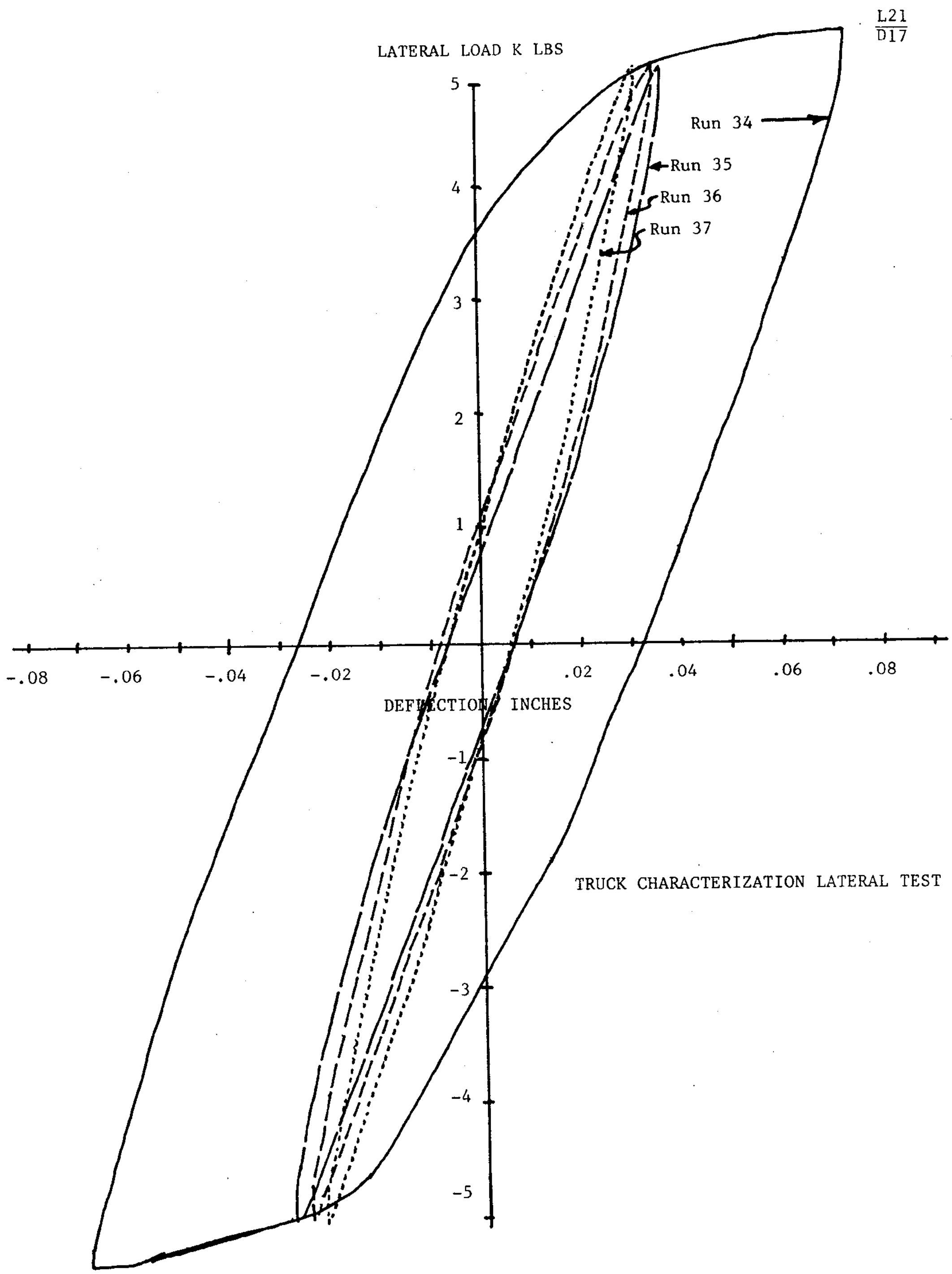


FIGURE 2.14 LOAD-DEFLECTION PLOTS FOR RIGHT SIDE FRAME, L21/D17

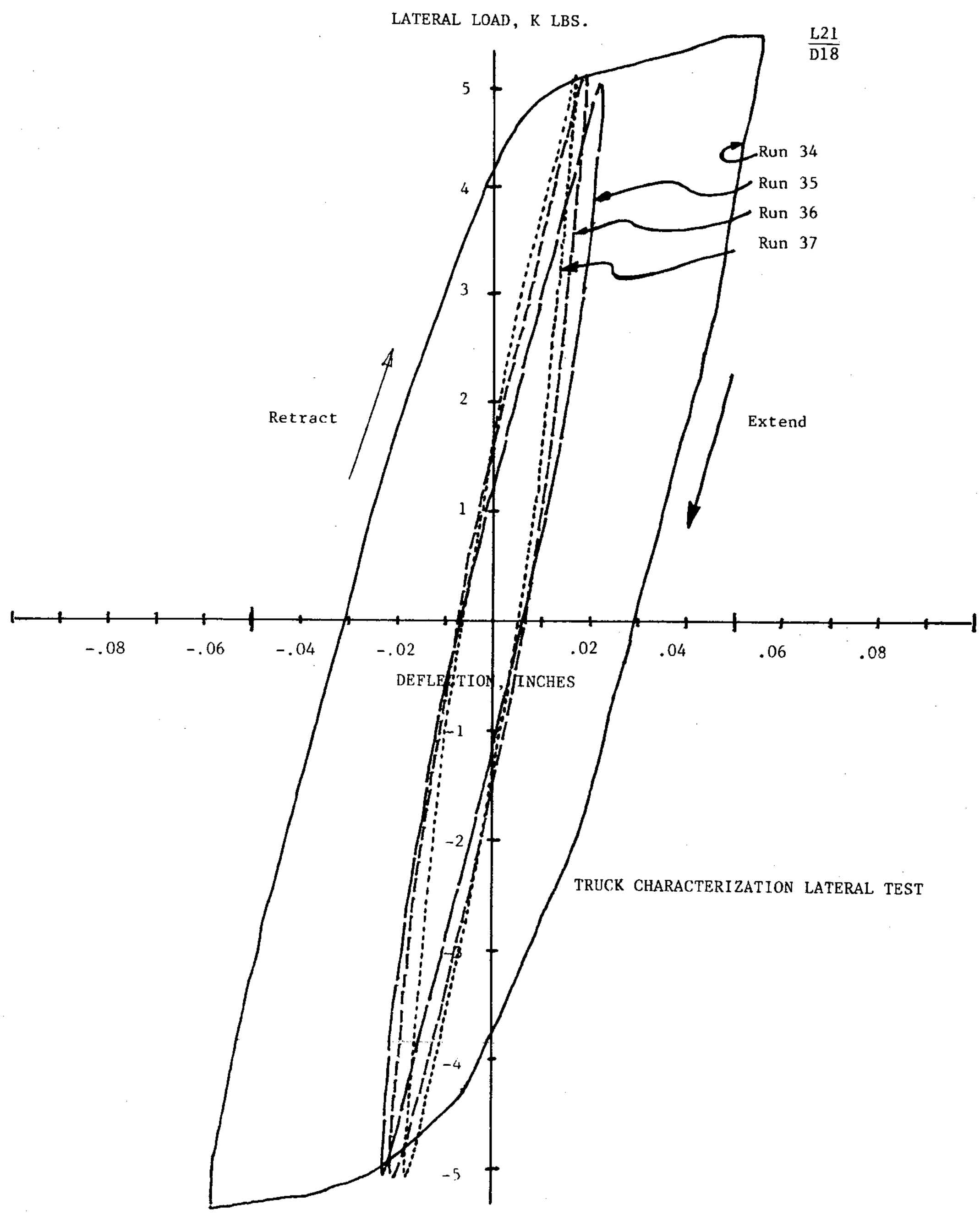


FIGURE 2.15 LOAD-DEFLECTION PLOTS FOR LEFT SIDE FRAME, L21/D18

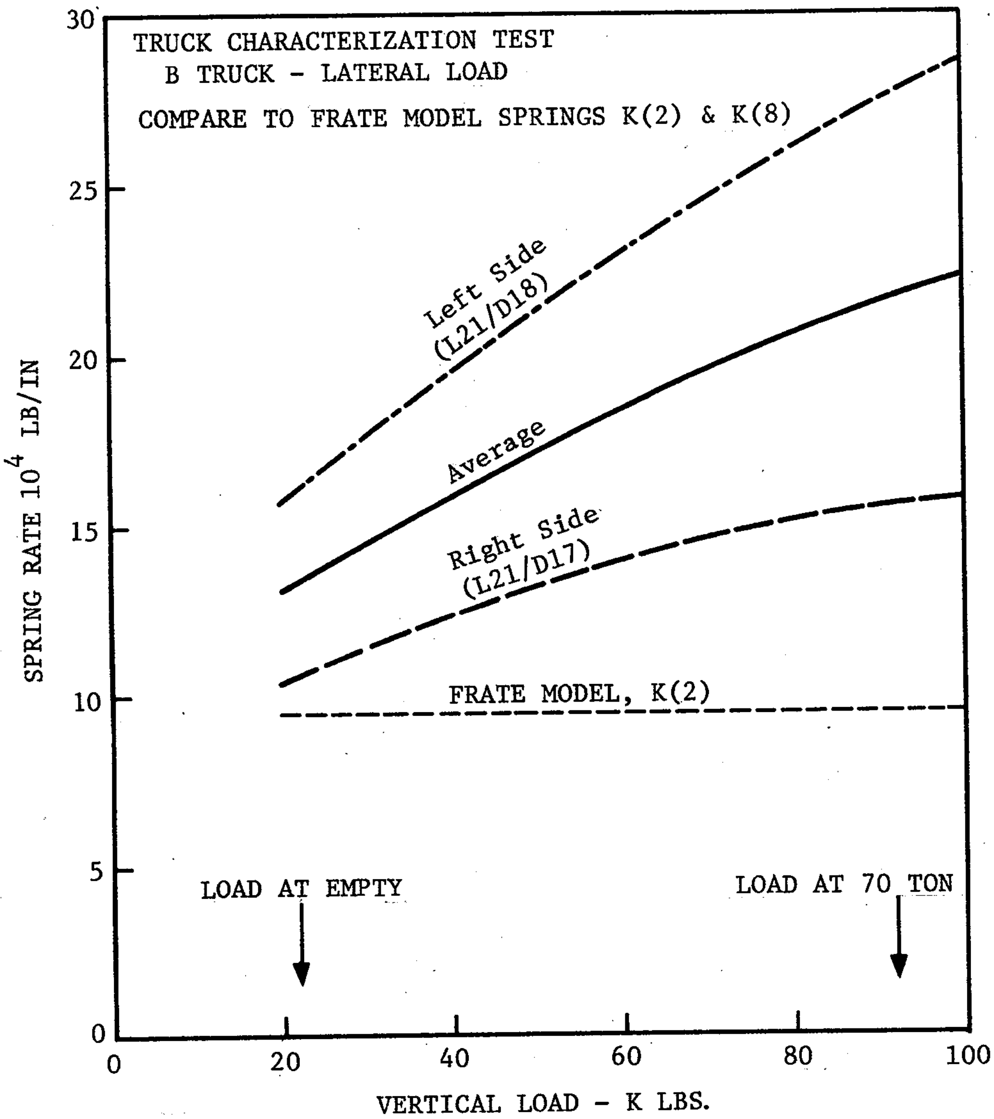


FIGURE 2.16 LATERAL SPRING RATE, SIDE FRAME TO GROUND, PER TRUCK

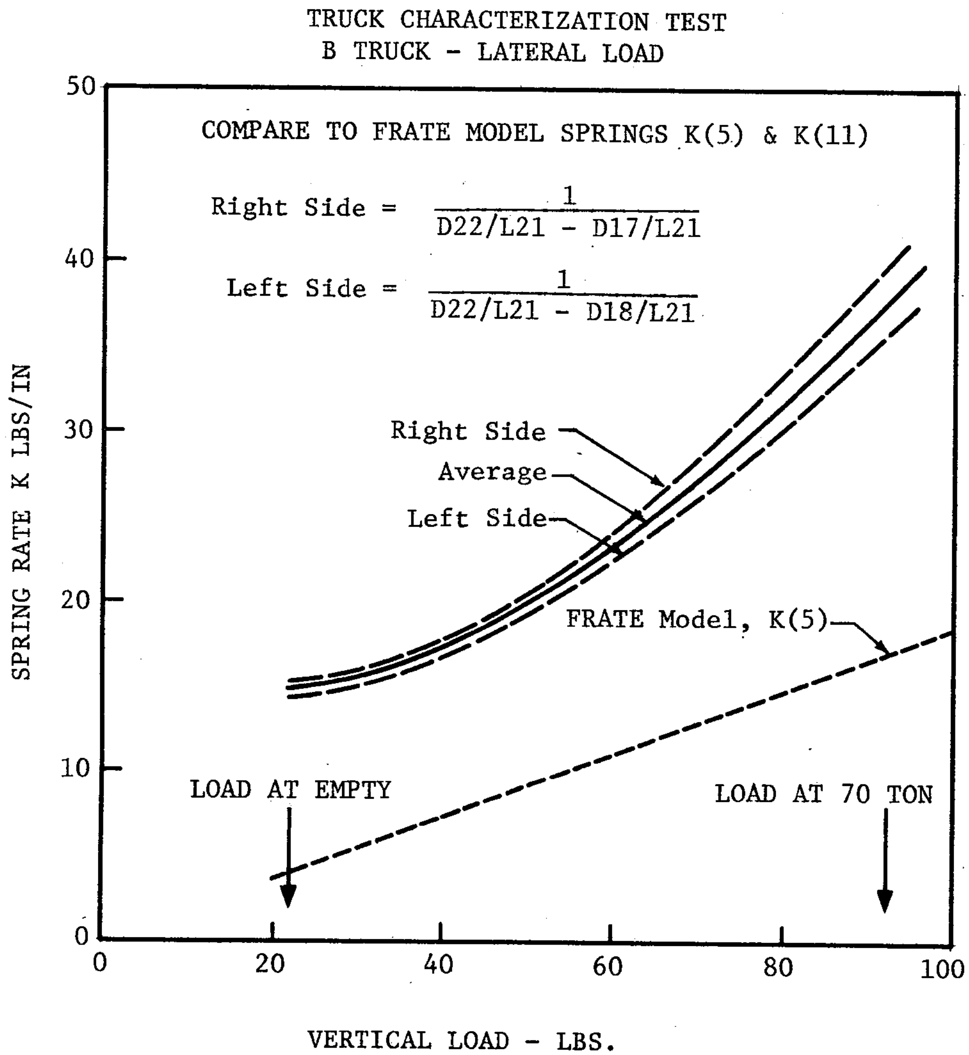
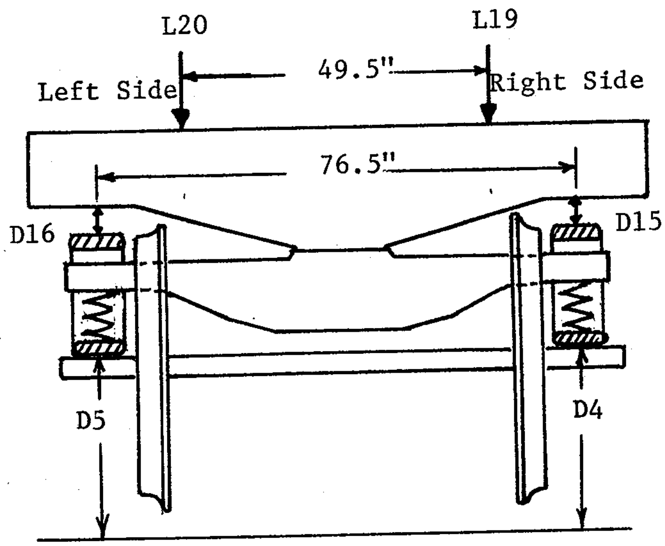


FIGURE 2.17 LATERAL SPRING RATE, TRUCK SUSPENSION SYSTEM WITHOUT SNUBBERS, PER TRUCK



1. Basic Equations

   - Applied Moment = (L20-L19) x 49.5/2, lb. in.
   - Angular Displacement = (D16-D15)/76.5, radians
   - Angular Spring Rate = $\frac{\Delta(L20-L19)}{\Delta(D16-D15)}$ x 1893.375, lb. in./rad.

$$= \frac{\Delta L20-\Delta L19}{\Delta D16-\Delta D15} \times 1893.375$$

2. Simplification

   - Assume $|\Delta L20| = |\Delta L19|$ and $|\Delta D16| = |\Delta D15|$
   - Then K $= \frac{\Delta L20}{\Delta L16} \times 1893.375 \approx \frac{\Delta L19}{\Delta L15} \times 1893.375$
   - Calculate and use average

FIGURE 2.18 CALCULATION OF ROLL SPRING RATES

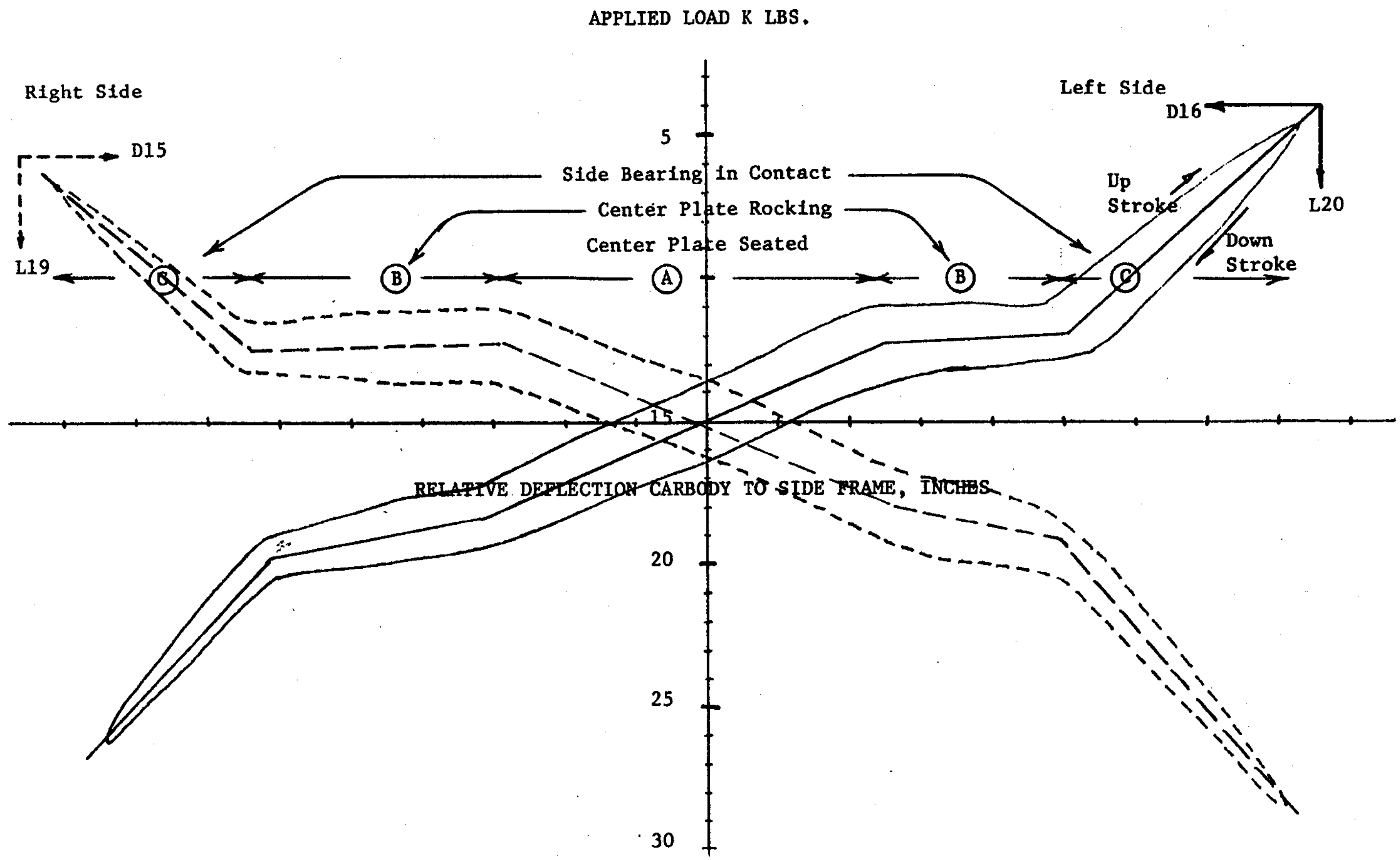
Two load levels were used; 15,000 ± 10,000 lbs and 25,000 ± 21,000 lbs (approximately). The resulting applied moment, (L20 - L19) x 49.5/2 and the relative roll deflection between carbody and side frame, (D16-D15)/76.5, are the moment-deflection data that would be used to derive roll spring rates. However, the data were available only as load-deflection plots of L19/D15 and L20/D16 without the ability to make the necessary instantaneous cross reference between them. The simplifying assumption was made, as noted in Figure 2.18, that the absolute values of the rates of change of L19 and D15 are approximately equal to L20 and D16 respectively. Roll K values can then be calculated separately using L19/D15 and L20/D16 and then averaged for final values.

The load-deflection plots for L19/D15 and L20/D16 for the two load conditions with and without snubbers are contained in the plots of Figures 2.19, 2.20, 2.21 and 2.22. The plots from the tests without snubbers, Figures 2.19 and 2.20 can be divided into five sections of constant slope. These constant slope sections are labeled A, B and C and have been identified with centerplate and side bearing positions as follows:

A	=	the centerplate is seated in the bowl
B	=	the centerplate is rocking; that is, the center plate is in contact with the bowl at the edge and is separated elsewhere
C	=	side bearing contact has been made and there is a two point contact between the carbody and truck bolster--the edge of the centerplate and the side bearing

The slope of the load-deflection plot in the region B is seen to be essentially zero which is what is expected in a rocking condition where the only restoring force (actually a moment) is the dead weight of the carbody. This is actually a negative spring rate since the restoring moment is the carbody weight times the lateral distance between the carbody center of gravity (cg) and the edge of the centerplate. Thus, the larger the roll angle the smaller the restoring moment.

The regions A and B are of special significance in that together they comprise a softening nonlinear spring. This correlates with the experimental observations of freight car characteristics in the rock and roll phenomena. The response curve of a softening spring is shown in Figure 2.23. The peak response of the systems will be larger in amplitude and occur at a lower frequency with a decreasing frequency sweep as compared to an increasing frequency. This matches freight car roll response which will typically be larger in amplitude and occur at a lower speed for decreasing speed as compared to increasing speed.



TRUCK CHARACTERIZATION TEST
RUN 38, WITHOUT SNUBBERS

FIGURE 2.19 LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION ROLL TEST MAXIMUM LOAD—25000 LBS.

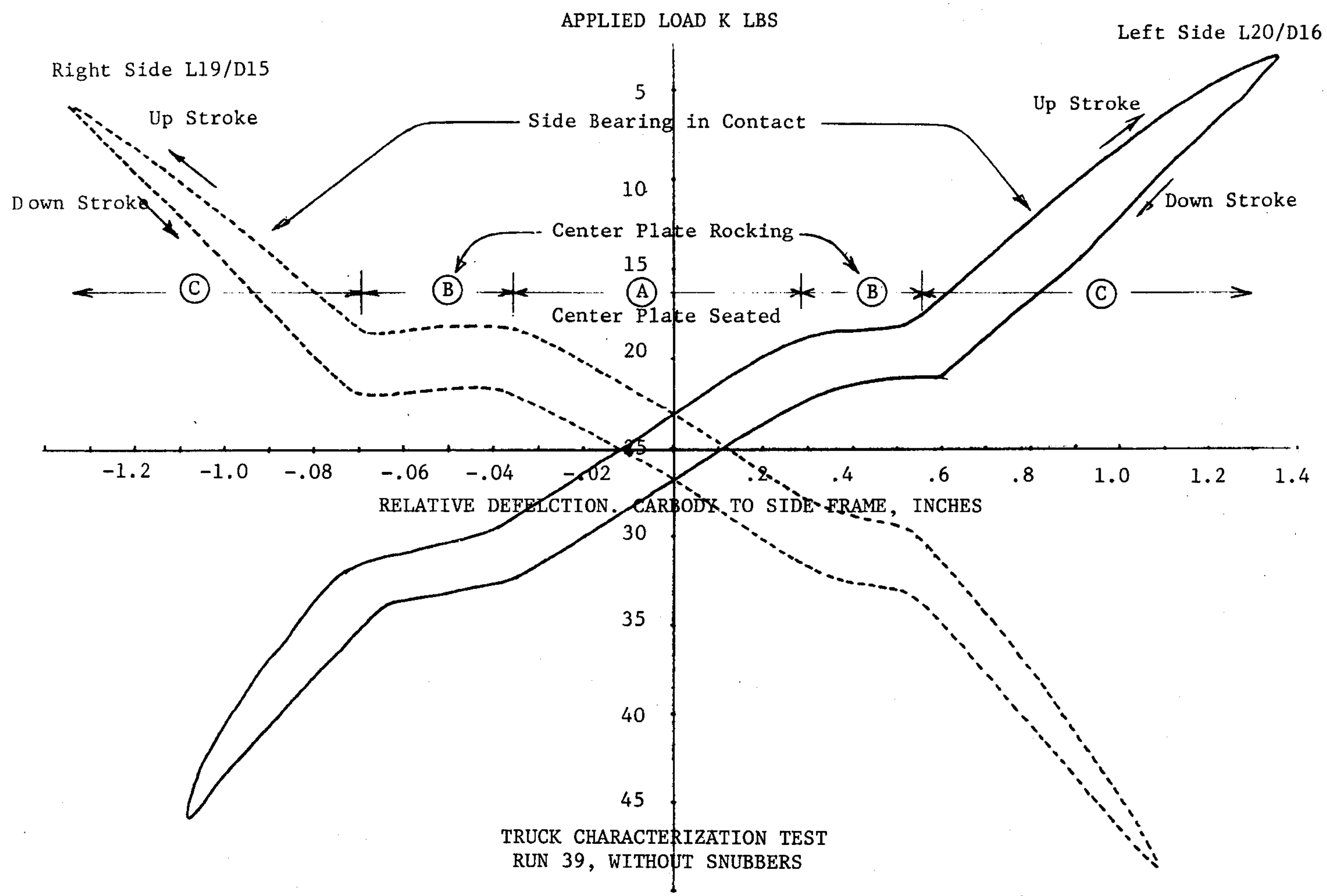


FIGURE 2.20 LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION ROLL TEST, MAXIMUM LOAD—46000 LBS.

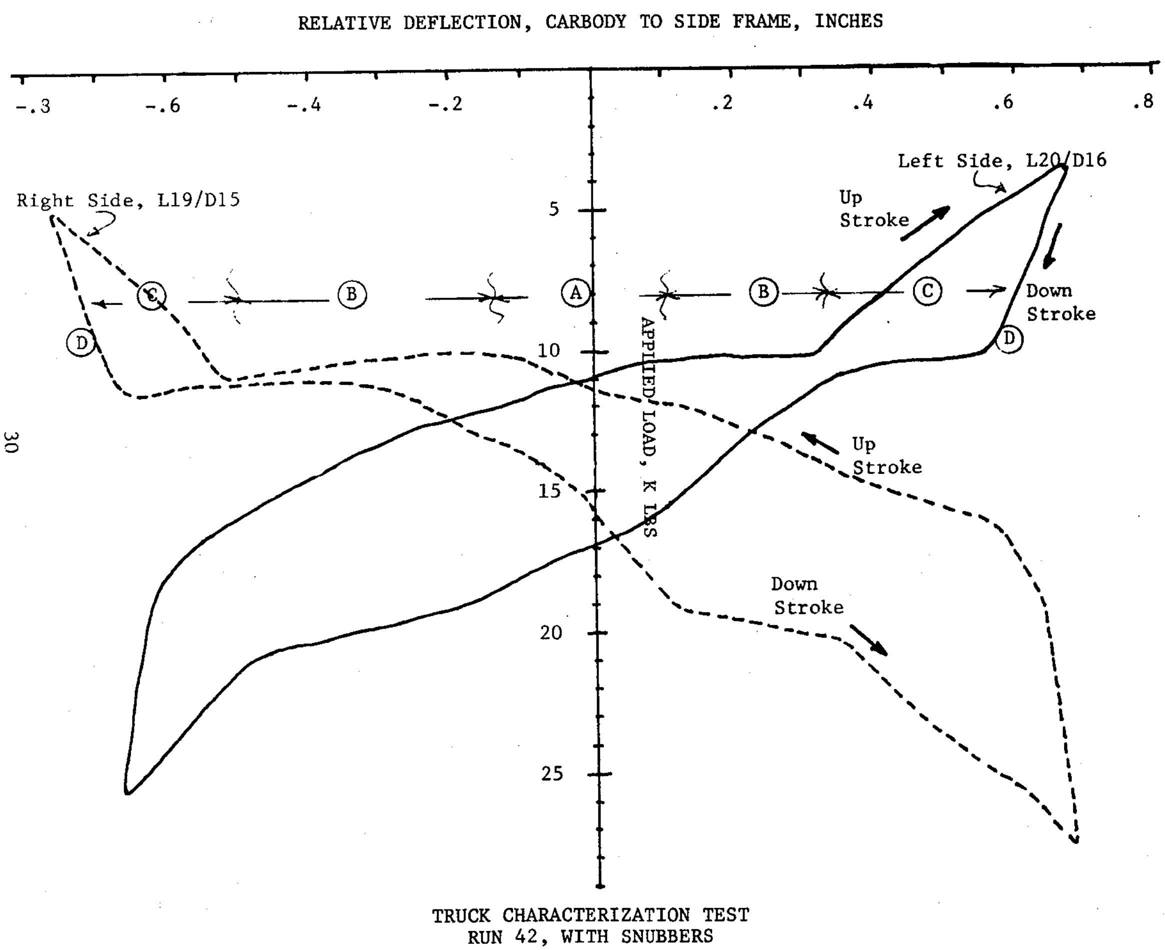
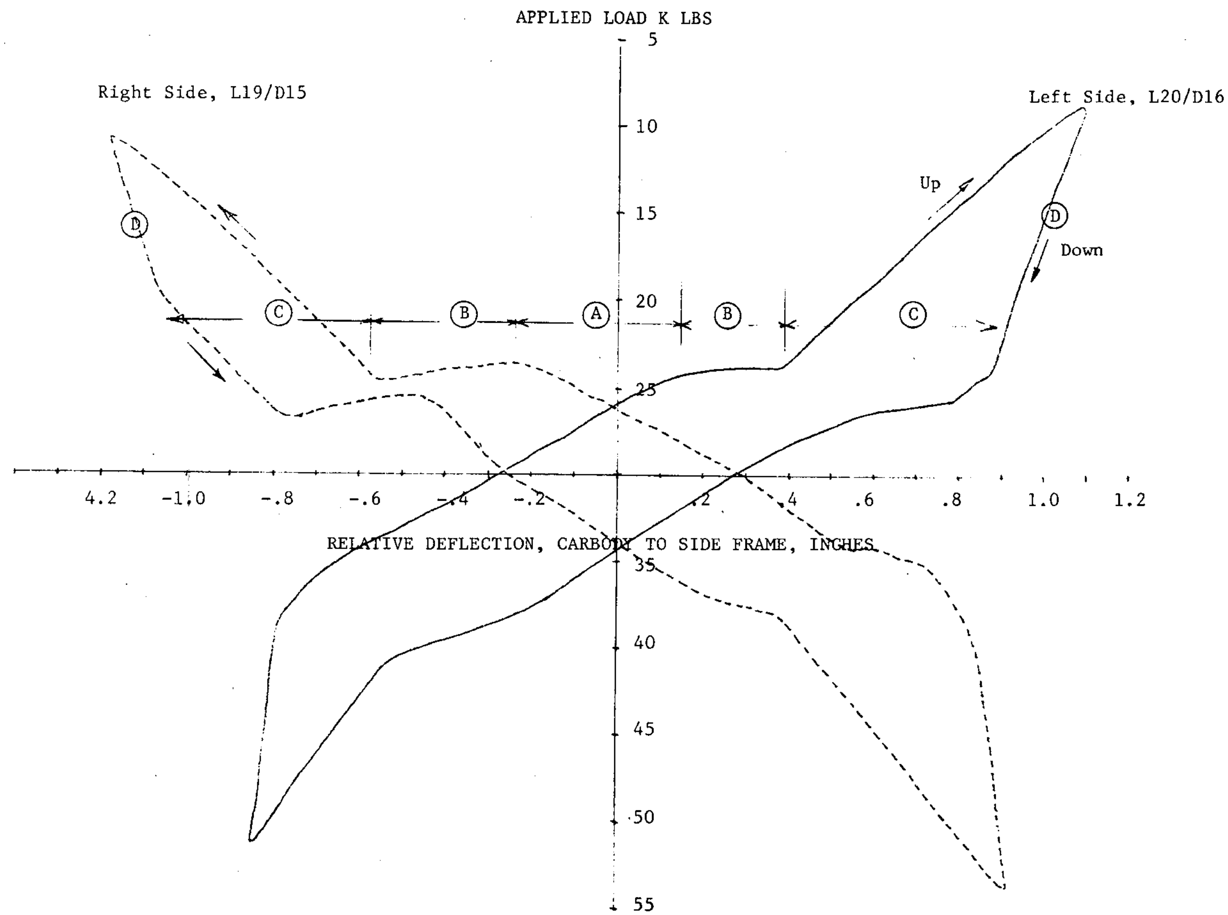


FIGURE 2.21 LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION ROLL TEST, MAXIMUM LOAD—25000 LBS.



TRUCK CHARACTERIZATION TEST
RUN 43, WITH SNUBBERS

FIGURE 2.22 LOAD-DEFLECTION PLOT OF TRUCK SUSPENSION ROLL TEST, MAXIMUM LOAD—46000 LBS.

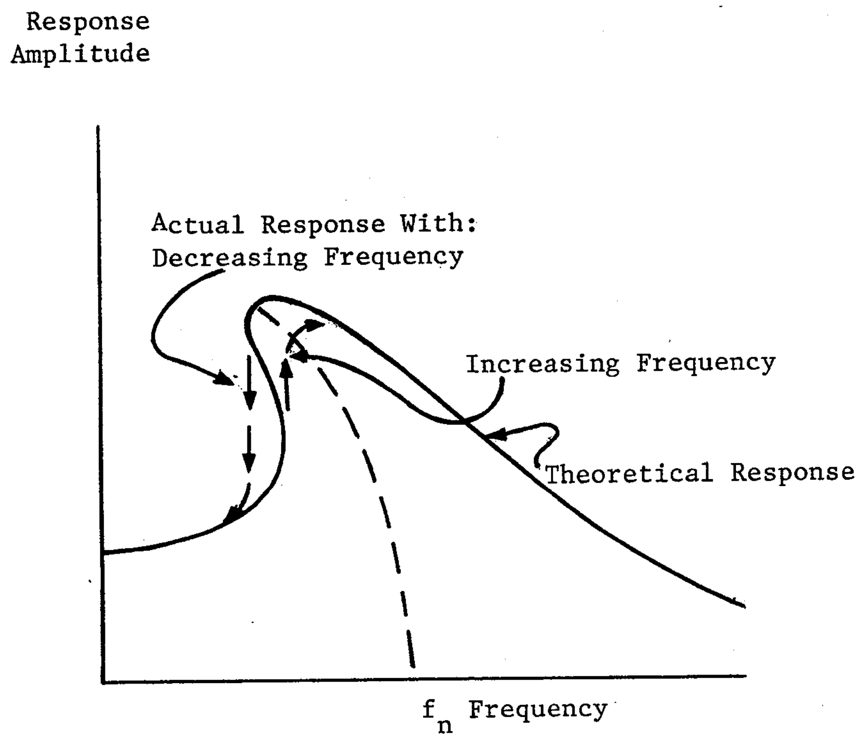


FIGURE 2.23 RESPONSE CURVE OF A NONLINEAR SOFTENING SPRING SYSTEM

The FRATE model uses an average value for the regions A and B and limits the region C to the point of centerplate lift. With centerplate lift the roll spring is set to zero.

The load-deflections for the tests with friction snubbers can also be regioned into the three conditions of seated centerplate, rocking centerplate and side bearing contact. The snubbers cause the load-deflection plots to be much less orderly. Also, there is an additional region labeled D which has been identified as the load-deflection curve when the snubbers are locked.

The slopes of the various regions were measured and plotted as spring rates against the vertical, dead weight load. Region B was omitted since it is roughly zero. However, the average spring rates over the combined A and B regions were determined and plotted as well as the A region by itself. These plots are presented in Figures 2.24, (Region A). Figure 2.25 (average over A and B), Figure 2.26 (Region C) and Figure 2.27 (Region D; locked snubbers). Data points used to establish average values are also plotted in these figures along with values of corresponding parameters used in the FRATE model.

All roll spring constant data to this point have been presented in lineal units of pounds per inch. Conversion from lineal to angular was made using the conversion factor K(angular) = K(lineal) x 1893.375. (Refer to Figure 2.18). Figure 2.28 is a summary plot of the truck suspension roll spring rates obtained in this test. A tabular summary is given in Table 2-2.

A reconstructed moment-deflection plot for the truck suspension system in roll is shown in Figure 2.29. The plot uses average A-B region values to side bearing contact and C region values beyond side bearing contact.

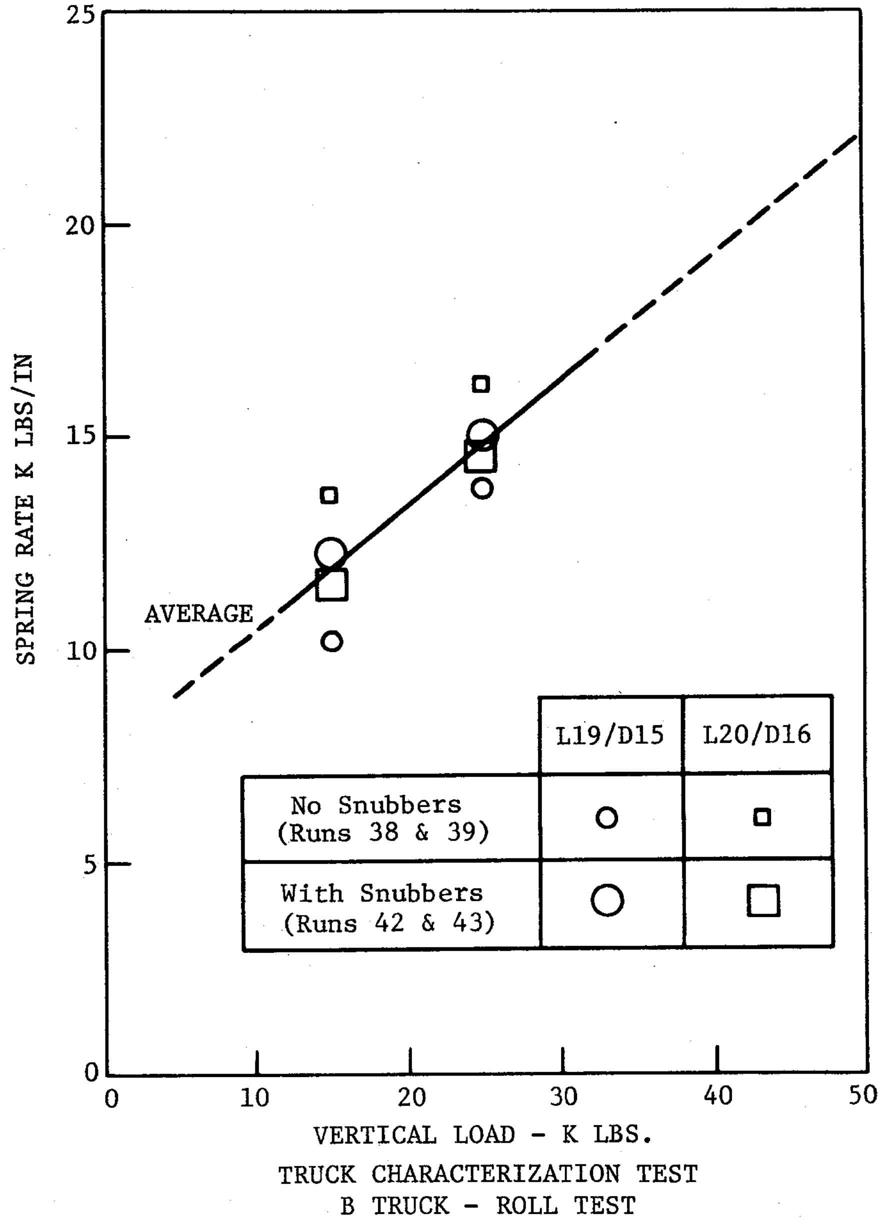
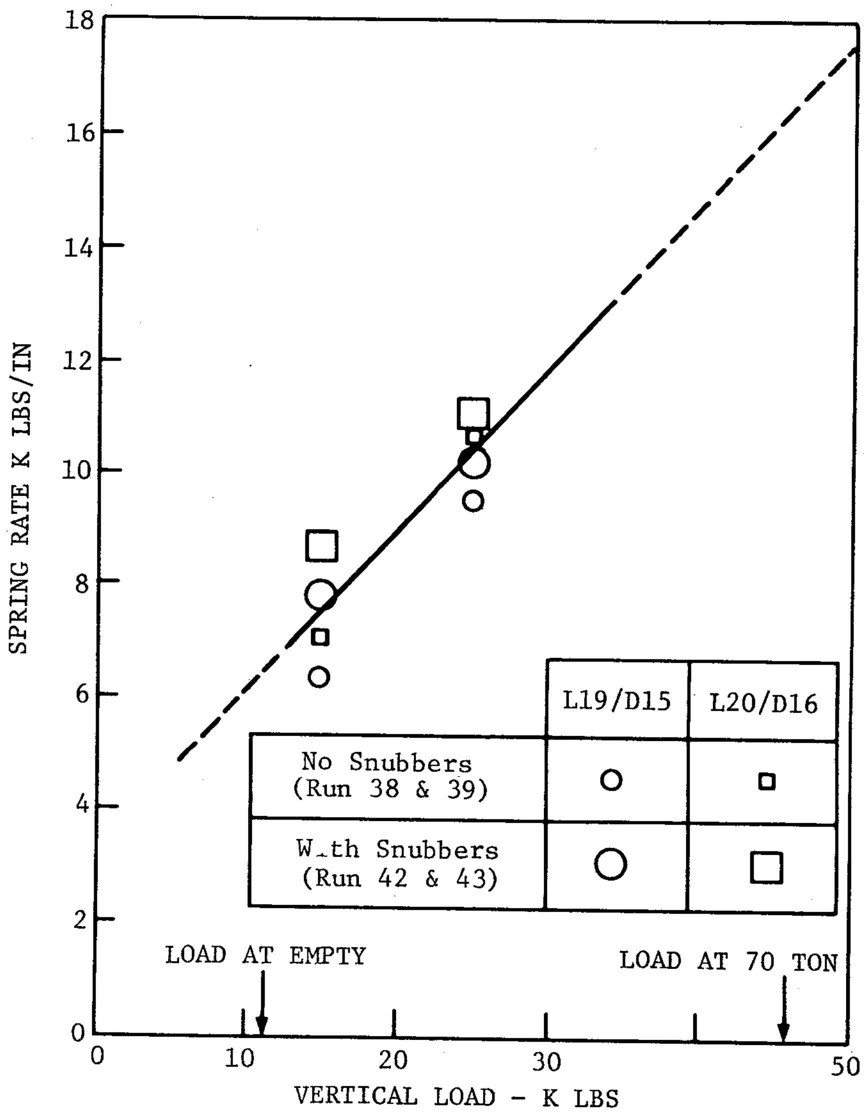


FIGURE 2.24 TRUCK ROLL SPRING RATES WITH SEATED CENTER PLATE (REGION A IN LOAD-DEFLECTION PLOTS)



TRUCK CHARACTERIZATION TEST, B TRUCK, ROLL

FIGURE 2.25 TRUCK ROLL SPRING RATES, AVERAGE OF SEATED AND ROCKING CENTERPLATE (REGIONS A & B)

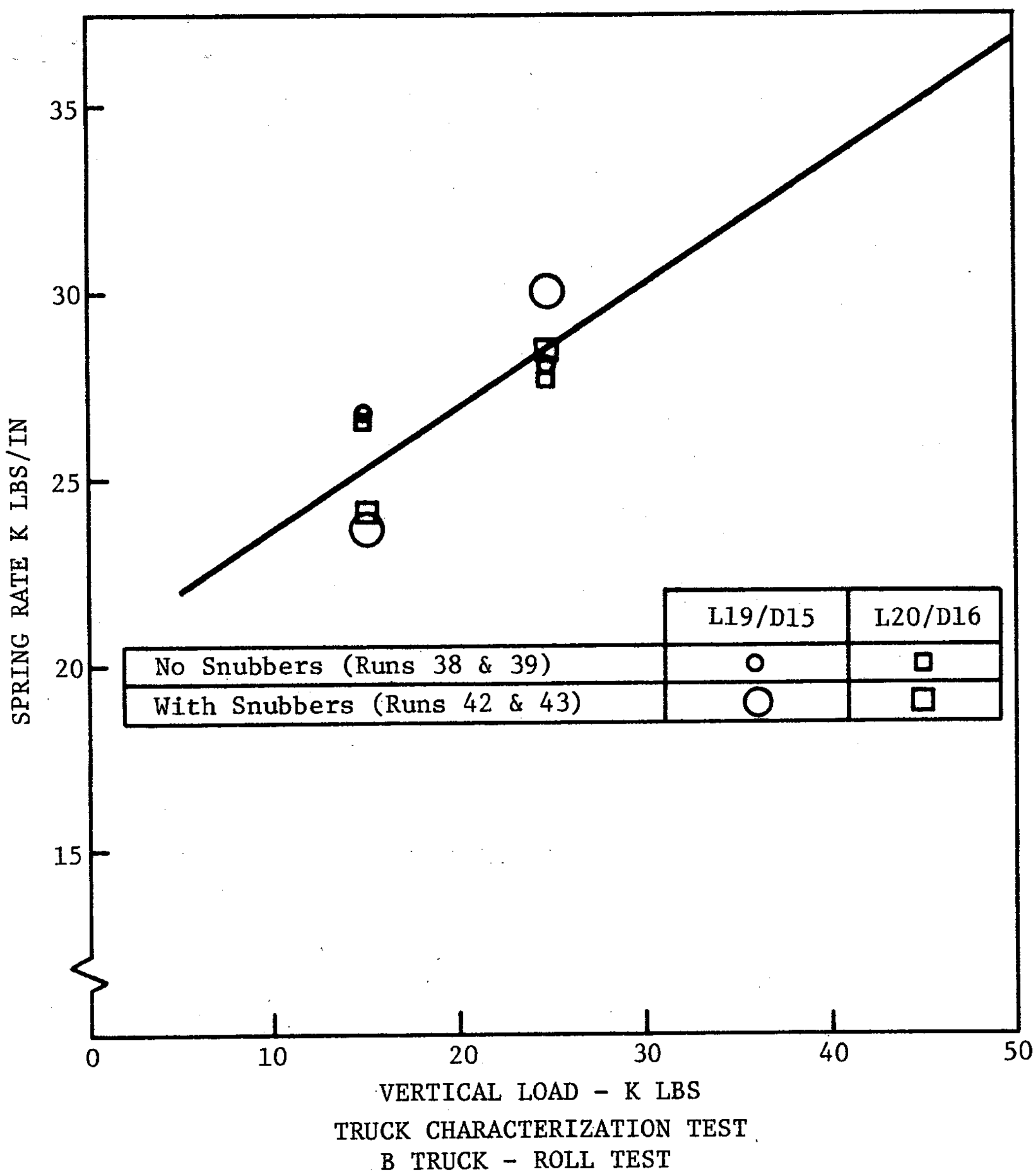
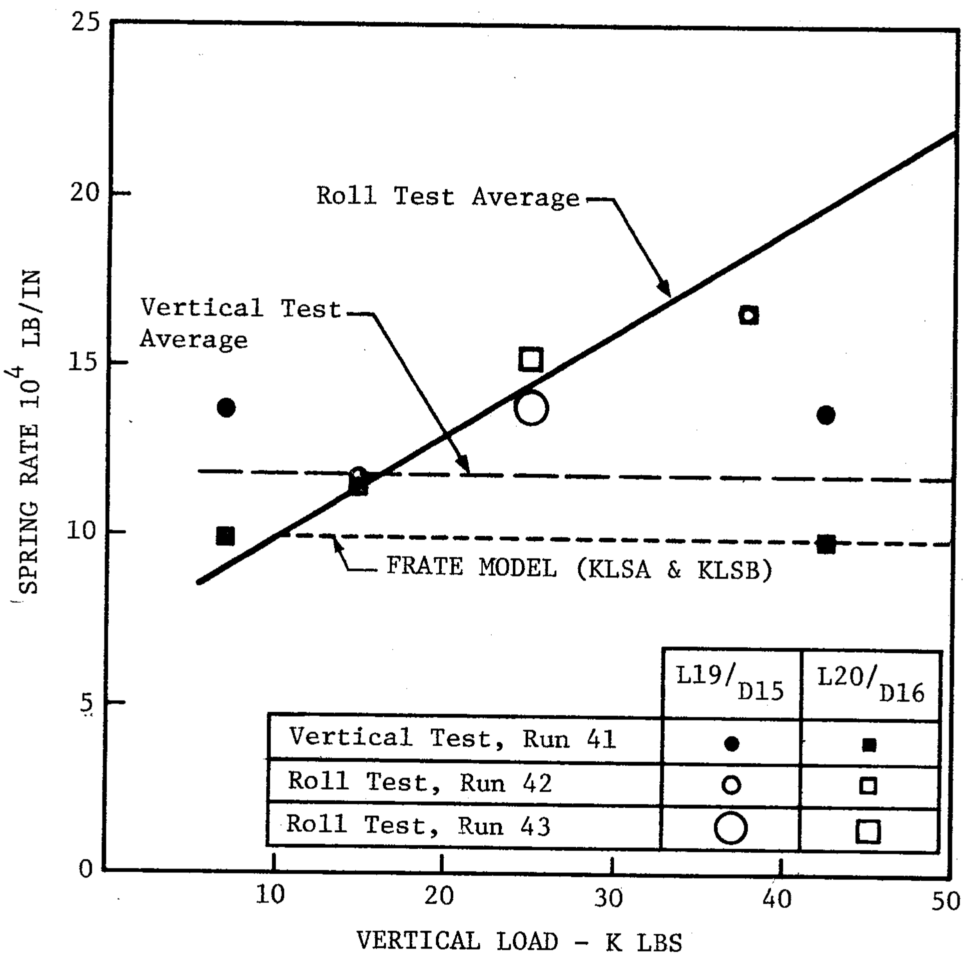


FIGURE 2.26 TRUCK ROLL SPRING RATES AFTER SIDE BEARING CONTACT (REGION C IN LOAD-DEFLECTION PLOTS)



TRUCK CHARACTERIZATION TEST
B TRUCK WITH SNUBBERS

FIGURE 2.27 SPRING RATES WITH LOCKED SNUBBERS

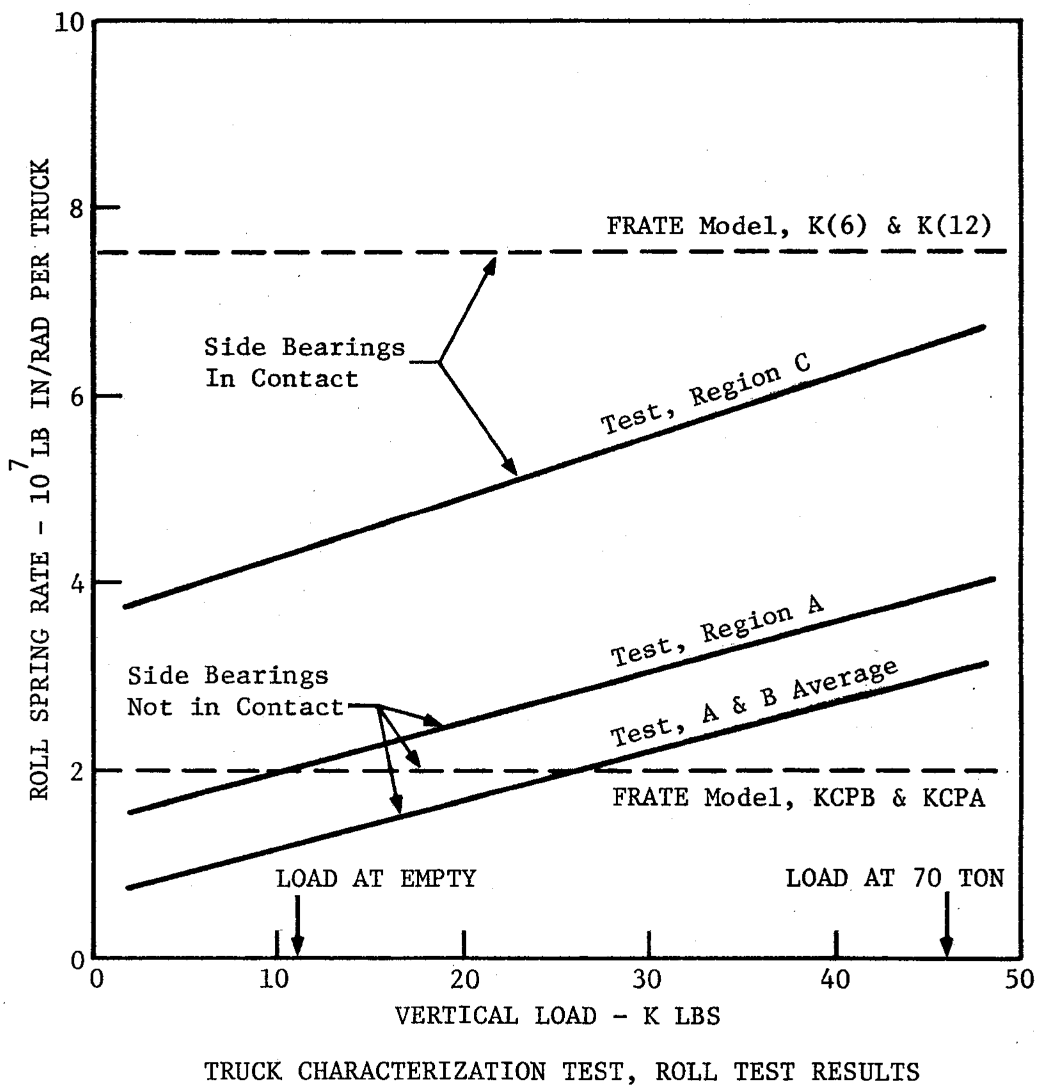
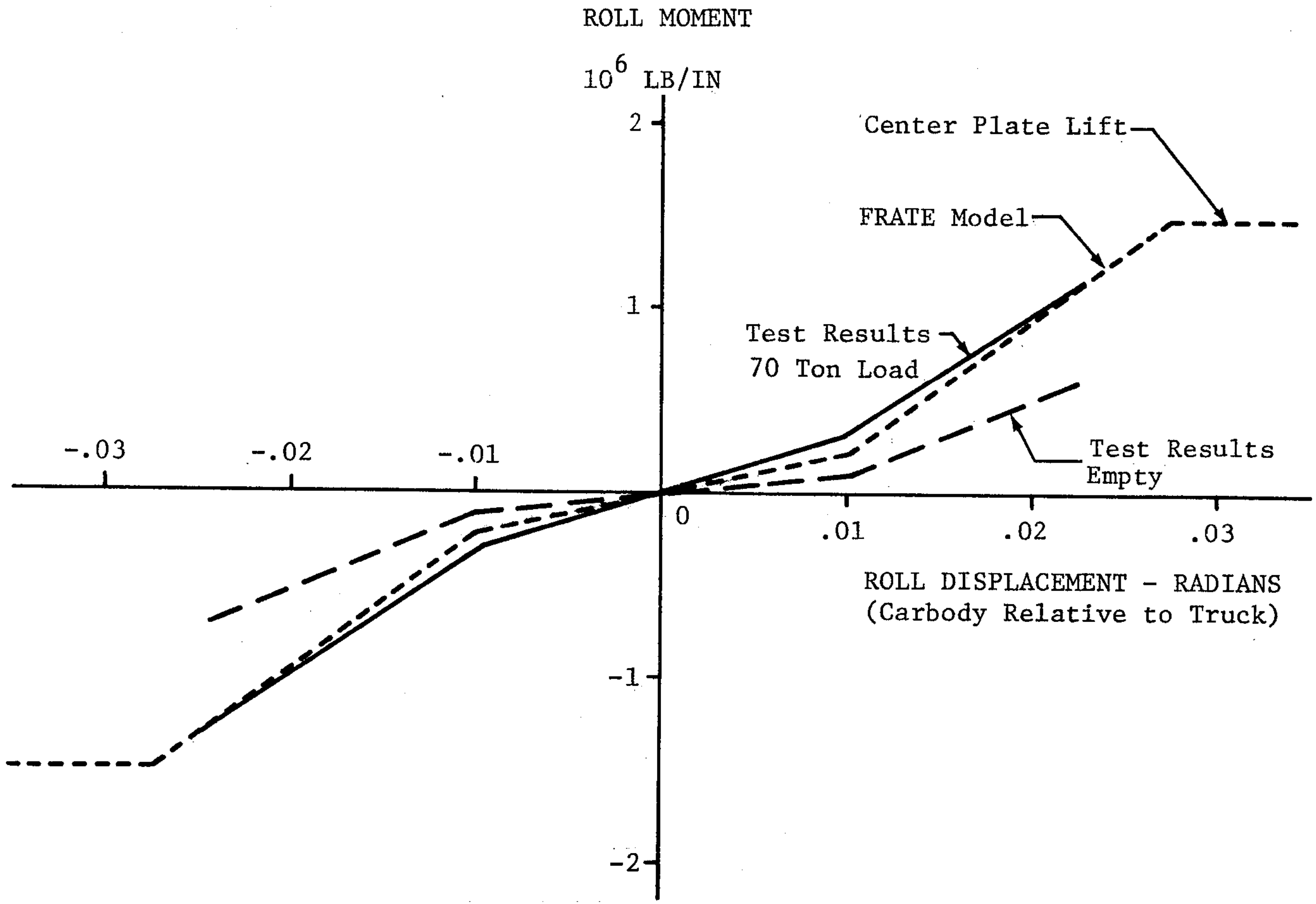


FIGURE 2.28 SUMMARY RESULTS TRUCK ROLL SPRING RATES



TRUCK CHARACTERIZATION TEST
B TRUCK - ROLL TEST

FIGURE 2.29 RECONSTRUCTED MOMENT—DEFLECTION PLOT,TRUCK SUSPENSION SYSTEM IN ROLL.

TABLE 2-2 CONVERSION OF LINEAL TO ROLL SPRING RATES: ROLL TEST

Item	Lineal (LB/IN)	Roll (LB IN/RAD)[2]	FRATE (LB IN/RAD)
Seated Center Plate			
15 K LB Vertical Load	11.847E3	2.243E7	--
25 K LB	14.842E3	2.810E7	--
Rocking Center Plate[1]			
15 K LB	7.514E3	1.423E7	2.0E7
25 K LB	10.409E3	1.971E7	
Side Bearings in Contact			
15 K LB	24.337E3	4.608E7	7.5E7
25 K LB	27.619E3	5.229E7	

[1] Combined Average of Seated and Rocking Center Plate

[2] Conversion Factor = ½ x 49.5 x 76.5 = 1893.375

3. SEVENTY TON BOXCAR VIBRATION TESTING

Vibration tests were performed on the DOTX 503 70 Ton Boxcar at the Transportation Test Center (TTC), Pueblo, Colorado. The Vibration Test Unit (VTU) located in the Rail Dynamics Laboratory (RDL) was used. The tests were conducted in the period starting 6 May 1981 and ending 5 June 1981.

The tests were performed according to the basic requirements set forth in the Requirements Document of Reference 6 and further detailed in Memorandum of Agreement, Reference 3 and the Implementation Plan, Reference 5.

3.1 Vibration Test Objectives

The basic objective of the vibration testing was to obtain definitions of the vibration characteristics of the boxcar and lading which can be used for comparisons to the respective analytical models. Two types of testing were performed:

1. Resonance
2. Response

The objectives of the resonance testing were to identify resonant frequencies, to define the deflection shape at each resonance, to obtain a measure of the damping associated with each resonance, to measure nonlinear effects with respect to amplitude of motion and to determine the effects of certain configurational changes.

The objectives of the response testing were to obtain a measure of responses on and in the boxcar to simulations of two track profile conditions and simulation of one hunting condition.

For the lading, in addition to the model validation objectives, the relative performance of two shipper designs were to be evaluated.

3.2 Vibration Test Description

The VTU is a vibration test facility designed for testing of railroad cars. It has 12 hydraulic actuators, one under each wheel driving vertically and one opposite each axle driving laterally. Motions of the actuators are controlled by a digital computer. All actuators are controlled by a master time function with the capability of independent control of relative amplitude and relative phase angle on each actuator. All actuators can be operated simultaneously thus giving a system capability of combined lateral, vertical, pitch, roll and yaw motions. The motions at the wheels described by the car traveling over a specific track profile at a specified speed can be simulated either with mathematical functions, actual profile data or a random signal generator.

The testing covered by this report was of two kinds. The first consisted of sinusoidal and random motion inputs for the purpose of identifying and defining resonance modes: sinusoidal testing was the primary procedure. Random was used on a trial basis as a potentially quicker, cheaper test method.

The second kind of testing performed was to simulate two track profile conditions and to simulate body hunting motions, and measure the resulting response motions of the boxcar and contents.

The purpose of the modal testing was to provide a data base against which the FRATE model could be compared and corrected. The purpose of the track condition simulation was to provide a second measure of the accuracy of the FRATE program. Included in the study of the boxcar was the study of the lading response and the effects of the change in packaging.

Primary measurement of input and response were made with Endevco Model 2262-25M15, $\pm$ 25g, Piezoresistive accelerometers. Accelerometer numbering systems and location descriptions are given in Appendix A. Other measurements made consisted of Trans-Tek LVDT Models 245-000 and 246-000, two gyros to indicate carbody roll angles and pressure transducers. Video cameras were used to monitor and record visible motions of the car and lading.

The accelerometer data was recorded on digital tape for post test data processing and on analog tape as back up. The data was filtered with 30 Hz low pass filters prior to analog-to-digital (A/D) conversion. The analog recording was made with 125 Hz low pass filters.

Preliminary identification of resonant frequencies and deflection shapes was obtained by sight and sound observations of the boxcar motions during the conduct of the tests.

There were three kinds of quick look data available for analysis immediately following the completion of each run. There was a "strip chart" from a pen and ink recorder with selected measurements which could be monitored in real time. There was an oscillograph with selected measurements using self-developing paper which was available for analysis within a few minutes after the completion of each run. A Hewlett Packard Analyzer was available for plotting response spectra or transfer functions of one or two selected measurements. A fourth source of data was transfer functions, generally referred to as Bode plots by the RDL, generated with the RDL's PDP 11/60 Data Reduction System Computer. These were made of selected measurements and were to be available on an overnight basis.

There were four configurations tested: 1A and 1B were loaded with corrugated paper cartons, with and without snubbers, 2 was empty without snubbers and 3 was loaded with stretch wrap packages with snubbers. Table 3-1 and Figure 3.1 provide some basic weight and dimensional data on the lading and boxcar. Configurations 1A, 1B and 3 had the same pallet configuration. This consisted of 56 pallets, stacked two high--28 stacks--with four layers of packages in the bottom pallets and five layers in the top pallets. The pallets were placed in the boxcar up against the side walls starting with the corner pallets up against the ends of the boxcar. The open aisle between the rows of pallets was filled with dunnage as was the left-over space at the center of the car.

3.3 Summary of Results, Boxcar Vibration

A summary of the resonant frequencies identified is shown in Table 3-2. The run numbers which are to be used for modal data analysis are noted with each resonant frequency. The testing and results obtained are discussed in this section for each kind of mode.

3.3.1 First Roll Mode

The first roll mode is a rolling motion of the carbody about a center of rotation that is somewhere near the horizontal plane of the top of rail. This is the mode normally associated with the staggered rail rock and roll phenomena. The frequency is amplitude dependent in that the frequency will become lower as the amplitude is increased. This checks with the results of the truck characterization test in roll where it was found that for small amplitudes the centerplate is seated and the roll stiffness is greater than for larger amplitudes where the centerplate is rocking. The snubbers tend to have the similar effect of raising the resonant frequency at small amplitudes and lowering it for large amplitudes. Also, because of the

TABLE 3-1 BOXCAR CONFIGURATION DATA

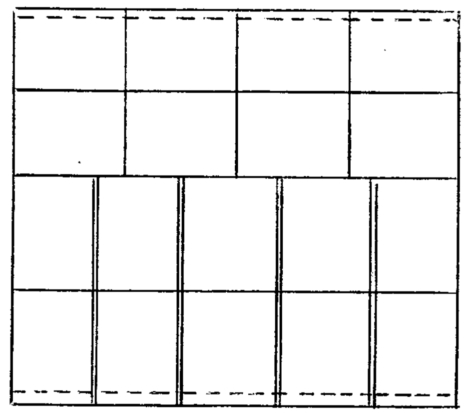
CONFIGURATION NUMBER	LADING PACKAGE	GROSS WEIGHT (LBS)	SNUBBERS
1A	Paper Box	181700	Active
1B	Paper Box	181700	Removed
2	Empty	61600	Removed
3	Stretch Wrap	180560	Active

Lading - Canned Dog Food
Paper Box - 48 cans/package
- 9 packages/layer
Stretch Wrap - 24 cans/package
- 18 packages/layer

Boxcar Dimension

Approximate Inside Length = 50 ft.
Width = 9 ft.
Height = 11 ft.
Volume = 5300 cu.ft.
Truck Spacing = 40 ft. 10 in.
Car Floor = 43.5 in. above top of rail
Cg Height/Empty = 53 in. above top of rail

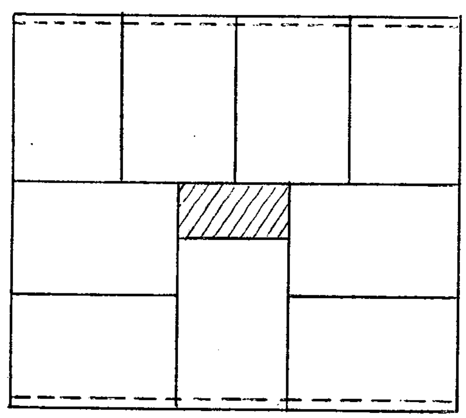
Stretch Wrap - 18 packages/layer, 48 x 42 inch footprint



Approximate size each package is:

12 x 9 x 8 1/8 in.

Paper Box - 9 packages/layer, 48 x 42 inch footprint



Approximate size each package is:

18 x 12 x 9 in.

Pallet Surface = 48 x 40 inches

Layers of lading arranged in mirror image pattern in alternate layers.

Pallets Double Stacked in Boxcar With:

- -4 Layers in Lower Pallet
- -5 Layers in Upper Pallet

FIGURE 3.1 LADING ON PALLET CONFIGURATIONS

TABLE 3-2 TEST FREQUENCY SUMMARY - 70 TON BOXCAR VIBRATION

MODE	CONFIGURATION 1A	CONFIGURATION 1B	CONFIGURATION 2	CONFIGURATION 3
1st Roll	.62 - .95[1] (Runs 9b & 59)	.63 - .70 (Runs 30 & 32)	.82 - .95 (Runs 64 & 72)	.68 (Run 101) .88 (Run 102)
2nd Roll	N.A.	2.6 (Run 35)	2.9 - 3.1 (Run 66)	3.8 (Run 103)
Yaw	N.A.	1.7 (Run 36)	2.6 (Run 73)	2.4 (Run 97)
Bounce	2.6 (Run 1)	2.05 (Run 39)	3.8 (Run 78)	2.4 (Run 108)
Pitch	3.75 (Run 20)	2.77 (Run 42)	4.34 (Run 80)	3.8 (Run 106)
Body Torsion	12.6 (Run 3)	12.4 (Run 37)	13.5 (Run 81)	12.7 (Run 99)
Body Bending	16 (Run 18)	(Need Bode, Run 40)	17.6 (Run 82)	15.8 (Vis.) 17.2 (HP) (Run 105)
Lading Lat.	1.9 - 2.6[1] (Run 5)	3.0 (Run 36)	--	2.3 - 2.4 (Run 97)
Vert.	7.5 - 8.1, 17-18 (Run 18)	8.3 (Run 40)	--	8 -8.5 (Run 105)

[1] Amplitude Dependent
N.A. - Not Available (snubbers remained locked)

Vis = Visual Observation
HP = Hewlett Packard Analyzer

nonlinear softening characteristics in roll, it was found that the resonant frequency appears at a lower frequency in a down sweep than in a up sweep. Thus, the first roll frequency for the loaded cases covers the relatively wide frequency band of 0.62 to.0.95 Hertz.

3.3.2 Second Roll Mode

The second roll mode motion consists of rigid body roll of the carbody about a center of rotation that is somewhere near the center of gravity of the carbody and contents. The mode is strongly influenced and subdued by the snubbers to the extent that in Configuration 1A it was not found. In Configuration 1B, where the change from 1A to 1B was the removal of the snubbers, the frequency was found to be 2.6 Hertz.

The change from Configuration 1B to 2 was to go from loaded to empty condition. The reduced weight would result in increased frequency (by a factor equal to the square root of the ratio roll inertias) except that the weight reduction also results in a softening of the suspension systems. The square root of weight change is about 1.6 and the square root of stiffness change, from truck characterization test data, is about 0.75. Thus, the projected frequency change from Configurations 1B to 2 is from 2.6 to 3.1 Hertz which is quite close to the test data showing the second roll to be between 2.9 and 3.1 Hertz.

The change from Configuration 1B to 3 consisted in changing the lading packaging from corrugated cardboard cartons to stretch wrap plastic packaging and the addition of the snubbers. The weight change was a decrease of 1140 pounds which is less than a one percent change and would not have any effect on the second roll resonance. It was, therefore, assumed that the frequency increase from 2.6 to 3.8 Hertz was due to the snubbers. This would mean that the snubbers caused an effective stiffness for this mode of about 210 percent.

In the search for this second roll mode for Configuration 1A, it was noted that the roll motion went through four distinct phases.

1. Below 0.8 Hertz, the motion was similar to first roll.

2. Between 0.9 and 1.2 Hertz, the motion seemed to be a rocking of the carbody on the centerplates.

3. Between 1.5 and 1.6 Hertz, the boxcar was in a lateral translational motion.

4. Above 2.4 Hertz, the motion was largely the second roll motion of carbody about its own center of gravity.

These motions are sketched in Figure 3.2. At the time of the testing of Configuration 1A it was concluded that the second roll resonance had not been reached and that it would be necessary to go to higher frequencies as well as larger input amplitudes. However, because of the difficulties encountered and attributed to the snubbers it was decided to discontinue this search and rely on the results of Configuration 1B and 3. Configuration 1B would be without snubbers and an easier test. Configuration 3 should be essentially the same as Configuration 1A.

3.3.3 Yaw Mode

The yaw mode was similar to the second mode in that the snubbers were very influential. In Configuration 1A the snubbers remained locked throughout the planned yaw testing. A roll test was run between two of the yaw tests in order to exercise the snubbers and wear off any accumulation of rust but no effect was seen in the yaw testing.

As in the second roll test, it was decided to forego further yaw mode testing in Configuration 1A, relying on results of subsequent testing to provide sufficient data for identification of the mode. The frequency was found to be 1.7 Hertz for Configuration 1B, 2.6 Hertz empty and 2.4 Hertz in Configuration 3.

Without snubbers the yaw mode was found to be very lightly damped. A good measure of resonant frequency and modal damping was obtained with decay testing. That is, excitations near resonance with a quick stop of shaker motion and measurement of the resulting decaying oscillations.

3.3.4 Bounce Mode

The bounce mode, again as in the second roll and yaw modes, was difficult to find because of the snubber action. It was concluded that with snubbers the bounce mode was in the 2.4 to 2.6 Hertz range with a one end pitch mode near 3.0 Hertz. The one end pitch would appear during the bounce test as the snubbers at one end would lock up resulting in small motions at that end and large motions at the other.

Without snubbers the bounce frequency was found to be at 2.05 Hertz with lading and 2.8 Hertz empty.

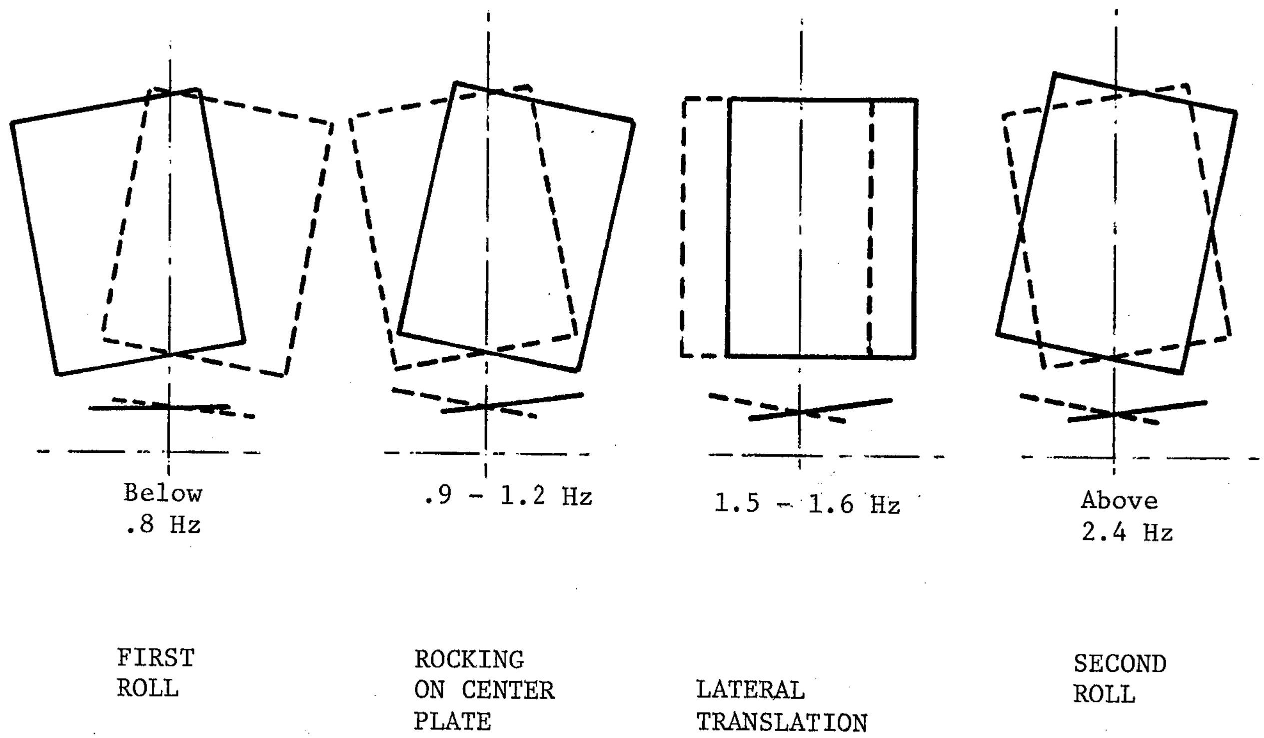


FIGURE 3.2 FOUR CHARACTERISTIC ROLL MOTIONS, CONFIGURATION 1A

There was an occurrence in the bounce that was noted as quite significant. While testing in Configuration 3, the test input amplitude was increased from $\pm$ 0.2 inches to $\pm$ 0.3 inches since at the lower amplitude the bounce mode was not sustained. At $\pm$ 0.3 inches input, as the frequency was being increased, the snubbers unlocked at 2.19 Hertz and the motion very quickly jumped to larger amplitudes. The test was quickly aborted for fear of damage to the vehicle. The test was repeated at $\pm$ 0.23 inches with satisfactory results.

This occurrence brings to mind that it can be shown in theory that a dynamic system with coulomb damping when excited at its resonant frequency will increase in amplitude to infinity. Also, it has been noted that freight cars, in general, can respond violently in the bounce mode to track irregularities when traveling in the 45-50 mile per hour range.

3.3.5 Pitch Modes

The frequencies for the pitch modes were obtained without any significant occurrences. The frequencies obtained were 3.75, 2.77, 4.34 and 3.8 for Configurations 1A, 1B, 2 and 3 respectively. For the configurations without snubbers the frequencies were obtained from decay testing.

The results of the pitch tests shown that the lading change had no effect on resonant frequency, 3.75 compared to 3.8 Hertz and that the snubbers has a stiffening effect, 3.75 Hertz with snubbers compared to 2.77 without.

3.3.6 Carbody Torsion

The body torsion mode was found to be about 12.5 Hertz with the carbody loaded and 13.5 Hertz empty. These frequencies are close to each other considering the difference in the weight of the two configurations. The explanation lies in the fact that the lading has a first lateral resonance below 3 Hertz and a first vertical resonance below 9 Hertz. The motion of the floor of the boxcar in the torsion mode is a combination of roll and lateral translation, but the lateral translation is predominant. The result is that most of the lading does not move when the carbody is in its body torsion mode.

However, during the yaw mode testing and during the hunting simulation, which consisted of a yaw motion input, there was a visible amount of torsional deformation of the boxcar floor. This deformation occurred in the loaded cases in the frequency range of the lading lateral resonance. It must, therefore, be concluded that torsional flexibility must be included in the modeling of boxcars and lading despite the relatively high carbody torsion mode frequency.

3.3.7 Carbody Bending

The body bending mode was found to be in the 16 to 18 Hertz range. As with body torsion mode, body bending is apparently not significantly different with or without lading.

There was some difficulty in exciting both the body torsion and body bending modes because excitation was applied at the rail and because the truck suspension systems tended to isolate the carbody. This was especially true in the loaded case without snubbers (1B).

3.3.8 Lading Resonant Frequencies

The lading resonant frequencies were found to be in the 2 to 3 Hertz range in the lateral direction and 7 to 8 Hertz in the vertical. There was little apparent difference in resonant frequencies between the corrugated paper and stretch wrap packaging. This was not as expected because the stretch wrap is a tighter and stiffer package. One explanation is that the lading resonance may be due primarily to the flexibility of the pallets.

3.3.9 Random Vibration Testing

The random vibration tests were performed with the objective of using this as a method for extracting modal data. It has the potential of taking less time and cost than sinusoidal modal testing. The random test was found to be useful in the body bending and body torsion tests but not for the other lower frequency resonances. Some of the problems encountered will probably be shown to be due to the type of accelerometers used and to limitations of the data processing and data analysis systems at the VTU.

4. COMPARISION OF TEST RESULTS WITH FRATE MODEL

Comparison of the FRATE model with tests results are given in Table 4-1 for the truck characterization test and Table 4.2 for the boxcar vibration test. From the truck test comparison it is seen that some parts of the FRATE model are too stiff and other parts are too soft. One general difference is that the FRATE model does not take into account the extent of change for load condition that is shown by the test results.

The first five modes of vibration test results without snubbers compares very well with FRATE analysis results with the exception of the second roll mode. The FRATE model did include a coulomb damping simulation of the friction snubbers using a friction force of 3,000 pounds. The truck characterization test showed the friction force to vary from 2,500 to 5,000 pounds depending on lading configuration. Despite the fact that the FRATE model included coulomb damping at a value within the range of test measurements, the FRATE results were closer to test results without snubbers than with snubbers. This leads to the initial conclusion that coulomb friction simulation needs to be modified in the FRATE program.

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TABLE 4-1 COMPARISON OF FRATE MODEL WITH TRUCK CHARACTERIZATION TEST RESULTS

DESCRIPTION	TEST	FRATE	UNITS
Vertical Spring Rates			
Wheel & Side Frame:			
K(1) & K(3)	(1) 5.8 E5	.91E5	LB/IN
	(2) 9.2 E5	.91E5	
Suspension:			
K(4)	(1) 3.82 E4	4.8E4	LB/IN
	(2) 4.96 E4	4.8E4	
Lateral Spring Rates			
Wheel & Side Frame:			
K(2)	(1) 1.34 E5	.95E5	LB/IN
	(2) 2.16 E5	.95E5	
Suspension:			
K(5)	(1) 1.50 E4	.42E4	LB/IN
	(2) 3.77 E4	1.71E4	
Roll Spring Rates			
Seated Center Plate:	(1) 1.86 E7	(3)	LB.IN/Rad
	(2) 2.81 E7		
Averaged Seated & Rocking CP:			
KCP6	(1) 1.05 E7	2.0E7	LB.IN/Rad
	(2) 1.97 E7	2.0E7	
Side Bearing Contact :			
K(6)	(1) 4.18 E7	7.5E7	LB.IN/Rad
	(2) 5.23 E7	7.5E7	
Snubber Local Structure:			
KS6	(1) .63 E5	1.0E5	LB/IN
	(2) 2.34 E5	1.0E5	

(1) Empty Carbody Weight Condition

(2) 70 Ton Load Condition

(3) No Directly Comparable Model Spring

TABLE 4-2 RESONANT FREQUENCY COMPARISON OF FRATE MODEL WITH BOXCAR VIBRATION TESTS, HERTZ

MODE	FRATE PREDICTION	CONFIGURATION 1B (WITHOUT SNUBBERS)	CONFIGURATION 3 (WITH SNUBBERS)
1st Roll	0.7	.63 - .70	.68 - .88
2nd Roll	1.6	2.6	3.8
Yaw	1.7	1.7	2.4
Bounce	2.2	2.05	2.4
Pitch	2.9	2.77	3.8
Body Torsion	--	12.4	12.7
Body Bending	--	16.0	15.8 - 17.2
Lading Lat.	5.0	3.0	2.3 - 2.4
Vert.	9.5	8.3	8.0 - 8.5

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Observations

The Truck Characterization Test

The truck characterization test was a success in that the desired data on truck stiffness and damping were obtained. The roll data was of particular value since its form and content are not available elsewhere and the results will be of considerable value in improving the accuracy of the FRATE model.

Although the results of the truck test were satisfactory, the test set up left something to be desired. It appeared to be what it was, a one shot, minimum cost test. It is felt that the general test method is a good one which can provide valuable truck properties data accurately and at relatively low cost. If there are to be other trucks tested, it is recommended that the test fixtures and set up be more carefully designed and of a more permanent sort and that some time be provided in the development of test technique and data processing procedures.

Boxcar Vibration Test

The boxcar vibration test was also considered a success on the basis of preliminary results. All targeted resonance were identified. The effects of snubbers and load conditions were measured.

Shaker System

The operation of the Shaker System was in general very dependable. Problems encountered were solved expeditiously. The testing was completed on schedule.

Data Processing

There were problems encountered in the data processing which were not solved at the time the tests were completed.

The main form of processed data was phase angle and relative amplitude of response measurements referenced to an input. These transfer functions frequently could not be deciphered to identify resonant frequency. The phase angle plots were generally erratic. It was also found to be difficult to do the circle fit Argand Plots for the modal analysis.

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A preliminary assessment put the source of data processing problems on the very nonlinear characteristics of the boxcar suspension system and on the type of accelerometer used in the test.

The nonlinear aspects of freight car suspension systems are a fact of life that need to be dealt with when performing dynamic tests. All modal test techniques and computerized analysis methods in current use were developed for linear, lightly damped systems. Problems are to be expected. It is expected that at the completion of the analysis of the results of this testing it will be possible to make some positive recommendations on test and data analysis procedures.

5.2 Recommendations

The one recommendation to be made now is that a different accelerometer should be used. The ideal accelerometer will have a flat response from DC to 20 Hertz with roll-off of 6db per octave or greater and with linear range of $\pm$ 5.0g. The accelerometer must be capable of withstanding high frequency shock and vibration of 100g or greater.

Program Continuation

A more detailed analysis of the vibration test results is continuing. The next planned effort will be to validate the FRATE computer program using the test results as reference criteria and following the procedure mentioned in the introduction of this report and detailed in Reference 2.

Final Comment

A final comment has to do with the personnel at the RDL. Each participant in the test program knew his job and did it well, but the outstanding characteristic was a willingness to turn to and accomplish what ever was needed.

REFERENCES

(1)
G. Kachadourian, "Users Manual for FRATX1 and FRATF1 Freight Car Dynamic Analysis Computer Programs," Report Number FRA/ORD-81-54, The MITRE Corporation, McLean, Virginia, July 1981.

(2)
G. Kachadourian, "Validation Plan for a 70 Ton Boxcar Model," MTR-79W00434, The MITRE Corporation, McLean, Virginia, December 1979.

(3)
Memorandum of Agreement, Rail Dynamics Laboratory, Vibration Test Unit, 70 Ton Boxcar Vibration and Lading Resonance Test, VTU-MOA-81-02-04, February 1981.

(4)
Implementation Plan, Rail Dynamics Laboratory, Vibration Test Unit, 70 Ton Truck Characterization Test Program, VTU-IP-81-01, February 1981.

(5)
Implementation Plan, Rail Dynamics Laboratory, Vibration Test Unit, 70 Ton Boxcar and Lading Test Program, VTU-IP-81-02-01, April 1981.

(6)
G. Kachadourian, "Vibration Test Requirements for a 70 Ton Boxcar, Revision 1, FRA/ORD-81-55, The MITRE Corporation, July 1981.

APPENDIX A
Measurements Numbering and Location

APPENDIX A
Measurements Numbering and Location

TABLE A-1 ACCELEROMETER LOCATIONS

NUMBER	MEASUREMENT DESCRIPTION
(1Z) A1AZ	Input Accel, Vertical Actuator 1A, Left Side
(2Z) A1BZ	Input Accel, Vertical Actuator 1B, Right Side
(3X) A1CX	Input Accel, Lateral Actuator 1C
(4Z) A2AZ	Input Accel, Vertical Actuator 2A, Left Side
(5Z) A2BZ	Input Accel, Vertical Actuator 2B, Left Side
(6X) A2CX	Input Accel, Lateral Actuator 2C,
(7Z) A3AZ	Input Accel, Vertical Actuator 3A, Left Side
(8Z) A3BZ	Input Accel, Vertical Actuator 3B, Right Side
(9X) A3CX	Input Accel, Lateral Actuator 3C

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TABLE A-1 ACCELEROMETER LOCATIONS (Continued)

(10Z) A4AZ	Input Accel, Vertical Actuator 4A Left Side
(11Z) A4BZ	Input Accel, Vertical Actuator 4B, Right Side
(12X) A4CX	Input Accel, Lateral Actuator 4C
A13Z	Vertical Accel, B Truck, Left Sideframe
A14X	Lateral Accel, B Truck, Left Sideframe
A15Z	Vertical Accel, B Truck, Left End of Truck Bolster
A16X	Lateral Accel, B Truck, Left End of Truck Bolster
A17Z	Vertical Accel, B Truck, Right Sideframe
A18Z	Vertical Accel, B Truck Bolster, Right Side
A19Y	Longitudinal Accel, B Truck Bolster Center
D20Z	Vertical Displacement, B Truck, Left Side, Sideframe to Truck Bolster
D21Z	Vertical Displacement, B Truck, Right Side, Sideframe to Truck Bolster
A22Z	Vertical Accel, A Truck Left Sideframe
A23X	Lateral Accel, A Truck Left Sideframe

TABLE A-1 ACCLEROMETER LOCATIONS (Continued)

A24Z	Vertical Accel, A Truck Bolster, Left End
A25X	Lateral Accel, A Truck Bolster, Left End
A26Z	Vertical Accel, A Truck, Right Sideframe
A27Z	Vertical Accel, A Truck Bolster, Right End
A28Y	Longitudinal Accel, A Truck Bolster, Center
D29Z	Vertical Displacement, A Truck, Left Side Sideframe to Truck Bolster
D30Z	Vertical Displacement, A Truck, Right Side, Sideframe to Truck Bolster
A31X	Lateral Accel, Top, Left Side of Carbody, @ B Truck Center Line
A32X	Lateral Accel, Bottom, Left side of Carbody, @ B Truck Center Line
A33Z	Vertical Accel, Bottom, Left Side, of Carbody, @ B Truck Center Line
A34Z	Vertical Accel, Bottom, Right Side of Carbody, @ B Truck Center Line
A35X	Lateral Accel, Top, Left Side of Carbody, @ Carbody Center
A36X	Lateral Accel, Bottom, Left Side of Carbody, @ Carbody Center
A37Z	Vertical Accel, Bottom, Left Side of Carbody @ Carbody Center
A38Z	Vertical Accel, Bottom, Right Side of Carbody, @ Carbody Center

TABLE A-1 ACCLEROMETER LOCATIONS (Continued)

A39X	Lateral Accel, Top, Left Side of Carbody, @ A Truck Center Line
A40X	Lateral Accel, Bottom, Left Side of Carbody, @ A Truck Center Line
A41Z	Vertical Accel, Bottom, Left Side of Carbody, @ A Truck Center Line
A42Z	Vertical Accel, Bottom, Right Side of Carbody, @ A Truck Center Line
A43Z	Vertical Accel, Top of Lading, Right Side B Truck Center Line
A44Z	Vertical Accel, Top of Lading, Left Side @ B Truck Center Line
A45X	Lateral Accel, Top of Lading, Left Side @ B Truck Center Line
A46Z	Vertical Accel, Bottom of Top Pallet of Lading, Left Side @ B Truck Center Line
A47X	Lateral Accel, Bottom of Top Pallet of Lading, Left Side @ B Truck Center Line
A48Z	Vertical Accel, Bottom of Bottom Pallet of Lading, Left Side @ B Truck Center Line
A49Z	Vertical Accel, Inside Bottom of Lower Pallet of Lading, @ B Truck Center Line
A50Z	Vertical Accel, Top of Lading, Left Side, @ Carbody Center
A51X	Lateral Accel, Top of Lading, Left Side, @ Carbody Center

TABLE A-1 ACCELEROMETER LOCATIONS (Concluded)

A52Z	Vertical Accel, Bottom of Lower Pallet of Lading, Left Side, @ Carbody Center
A53Z	Vertical Accel, Top of Lading, Right Side, @ A Truck Center Line
A54Z	Vertical Accel, Top of Lading, Left Side, @ A Truck Center Line
A55X	Lateral Accel, Top of Lading, Left Side, @ A Truck Center Line
A56Z	Vertical Accel, Bottom of Top Pallet of Lading, Left Side, @ A Truck Center Line
A57X	Lateral Accel, Bottom of Top Pallet of Lading, Left Side, @ A Truck Center Line
A58Z	Vertical Accel, Bottom of Lower Pallet of Lading, Left Side, @ A Truck Center Line

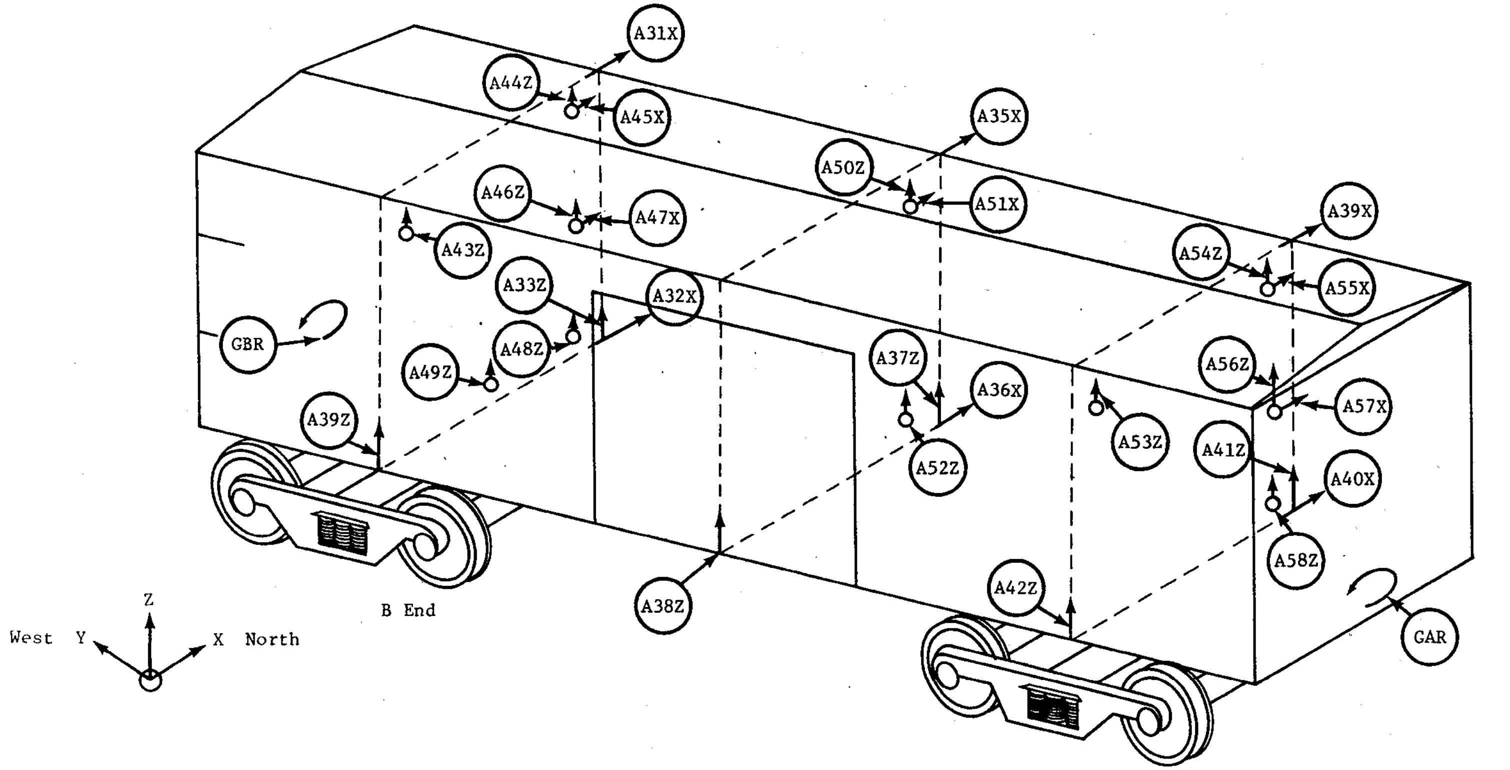


FIGURE A.1 MEASUREMENT LOCATION ON CARBODY AND LADING—70 TON BOXCAR

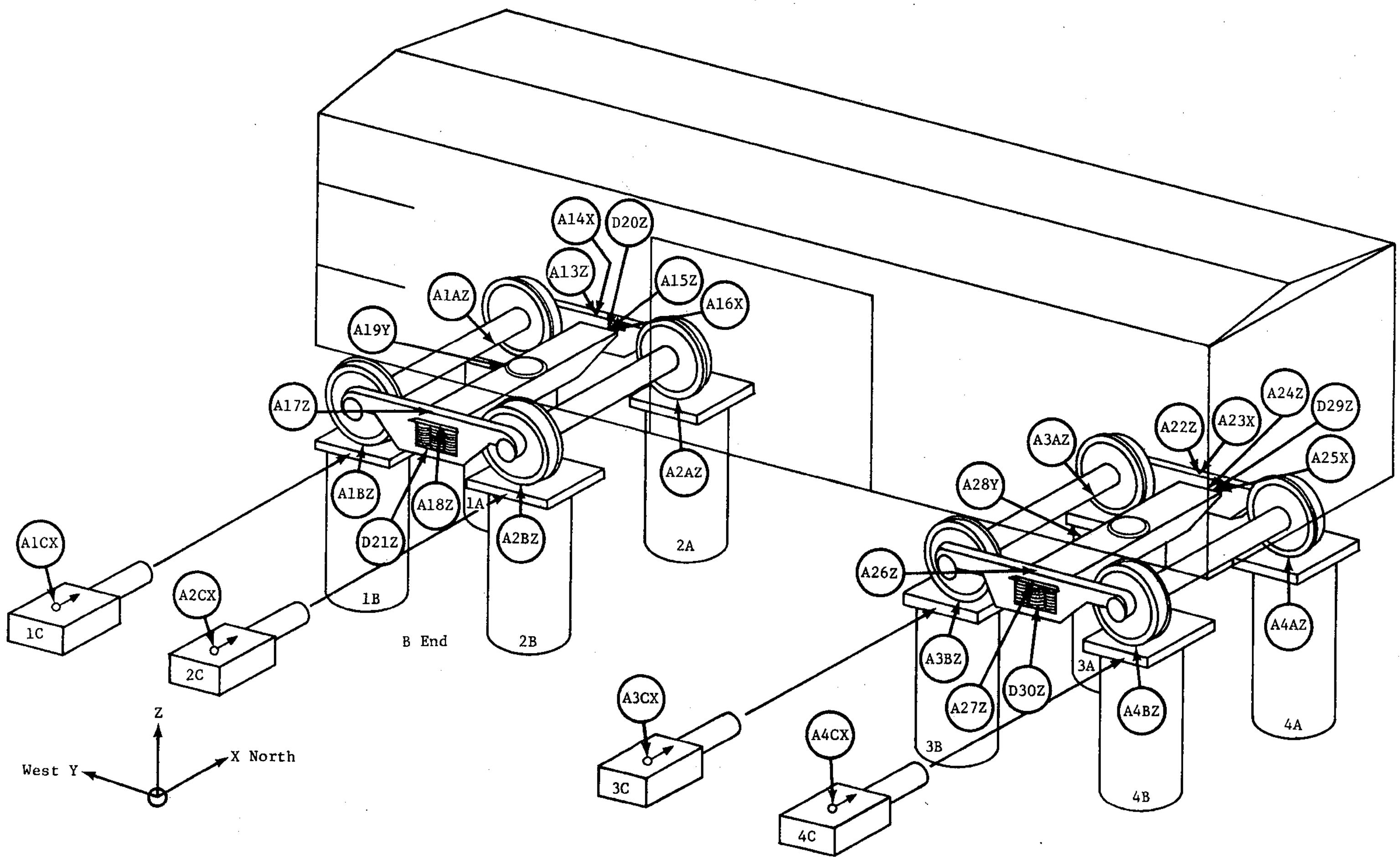


FIGURE A.2 MEASUREMENT LOCATION ON ACTUATORS AND TRUCKS—70 TON BOXCAR

APPENDIX B
Test Log: Boxcar Vibration Test

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (1)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.*	INPUT Amp.	INPUT Freq.(Hz)	RESULTS NOTES
CONFIGURATION 1A						
05/06/81	1	Torsion & Lading	3	± .2g	3.5-20	Doors Closed Max Amp 11.8Hz, 90° ∅ 13.0Hz
	2	"	"	"	"	Doors Open 11.8, 12.4 Lading - 3.5 Lat., 8.8 & 16 Vert
05/07/81	3	(Runs 1 & 2 Repeated because of Data Recording Problem)			6.0-20	Max. Amp / 90° ∅ / 90° ∅ 31/32 Doors Open 12.3 Hz Q=3.8 / 12.3Hz / 15.0Hz
	4					Doors Closed 12.3 Q=4.4 / 12.9 / 15.5
05/07/81	5	Yaw & Lading	3	±.05in	.5-60	Yaw mode not found - snubbers locked Lateral Lading - 2.6 Hz - 90° 2.69 Max. Q=6.2
	6	"	"	±.15in	.5-2.7-.5	
05/08/81	-	Exercised Vehicle in Roll to break in Snubbers				Yaw mode not found - snubbers remained locked.
	7	Same as 6		+.30	.5-2.5-.5	
05/08/81	8	1st Roll	2	.05in	.5-1.3	Snubbers moving between .94 & .98 Hz apparent resonance - .96 Hz

* The actuator numbering system and configuration notation is defined at the end of this appendix.

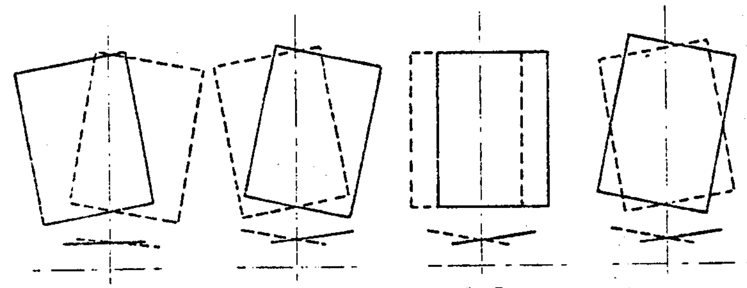
TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (2)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.*	INPUT Amp.	INPUT Freq.(Hz)	RESULTS NOTES
05/08/81	9a	1st Roll	2	$\pm$.15in	.5-1.3	Up Sweep - Snubbers broke & 90° at .766
	9b	-same-		$\pm$.15in	1.3-.5	Down Sweep - Snubbers broke at ~ 1.2 Hz Max Amp - .65 Hz 90° Ø - .68 - .73 Hz
	10	-same-		$\pm$.25in	.5-1.3	Snubbers broke at .62 Hz 90° Ø .68 - .72 Hz (filters at 10 Hz - effect not apparent)
	11	2nd Roll	2	$\pm$.1in	.5-3.0	Apparent resonance .95 Hz (1st Roll)
	12	-same-		$\pm$.3in	.5-3.0	Snubber breaks at .796 Hz (1st Roll)
	13	-no test-				
	14	2nd Roll	2	$\pm$.2in	.5-3.0-.5	.9 Hz Up, .85 Hz Down (1st Roll)

From visual observations the carbody has four phase characteristics through these runs

1. In-Phase W Input
2. Roll About Draft Gear
3. Lat. Translation
4. Roll About Body C.G.



Below .8 Hz | .9-1.2 Hz | 1.5-1.6 Hz | Above 2.4 Hz

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (3)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.*	INPUT Amp.	INPUT Freq.(Hz)	RESULTS NOTES
05/08/81	15	Bounce	4	$\pm$.1in	1.0-5.0	A snubber starts to move at 3.4 Hz B at 3.6 Hz, 90° at 4.0
	16& 17	-same-		$\pm$.2in	1.0-5.0	A snubber starts at 2.6 Hz switches to B at 3.0 Hz, both ends 3.8 Hz
	18	-same-		$\pm$.3g	.5-20	Lading Vert. - 7.8-8.0 Hz Roof Panel - 13.4; side panel 12.8; both - 12.1, 15.8, 17.0
05/11/81	19	Pitch	5	$\pm$.1in	2-8	Snubber broke at 4.46 Hz 90° at 7 Hz
	20	-same-		$\pm$.2in	2-8	Snubber broke at 3.63 Hz 90° at 3.8-4.6 Hz
	21	Bounce	4	$\pm$.2in	2-5	Repeat 16 with smaller Δf, (.2→.1) Bounce 2.6 Hz B end Pitch 3.0-3.2 Hz Bounce 3.5 Bounce $\leq$ 2.6 One End Pitch 3.0 Lading Vertical 7.8-8.0 Body Bending 12-14, 16
05/12/81	22	Hunting	8	$\pm$.2in	1.2-2.4	Lateral lading movement starting 2.2 Hz
	23	-same-		$\pm$.4in	1.2-2.4	Lateral lading 1.7 Hz start 2.0 Impacting side of car floor bends at door opening
	24	-same-		$\pm$.6in	1.2-2.4	

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (4)

DATE	RUN NO.	ACTUATOR Mode	No.*	INPUT Amp.	Freq.(Hz)	RESULTS NOTES
05/12/81	25	Staggered Rail	6	.1 in rec. sine	.7-.25	Max Amp at .45 - .40 Hz rec. sin. (.90 - .80 Hz sine eq.) (25 - 22.5 mph)
	26	-same-		.3in	.55-.25	Max Amp - .30 Hz
	27	-same-		.2in	.55-.25	Snubber start .45 stop .35
	28	B End	9	.25in RMS	Random	(shakers 1A, 1C, 2A & 2C)
	29	A & B Ends	9A	.125in RMS	Random	(shakers 1A, 1C, 2A, 2C, 3A, 3C, 4A & 4C)
CONFIGURATION 1B - Loaded With Corrugated Carton Lading - No Snubbers						
05/14/81	30	1st Roll	1	$\pm$.05in	.5-1.4	90^{o} ~ .7Hz
	31	-same-		$\pm$.15in	.5-1.4	Aborted due to excessive amplitudes resulting in damaged accels on side frames
	32	-same-		$\pm$.10	.4-1.4-.4	Large motions-including lateral movement of bolsters and gib impact Max at .66 Hz up, .60 Hz down
	33	2nd Roll	2	$\pm$.2in	1-4 Hz	1.4 Hz - gib bump, computer fail-incomplete run
	34	-same-		$\pm$.05in	1-4-1	Computer fail
	35	-same-		Repeat		Gib contact - Max 2.6 Hz 2.57 Hz
	36	Yaw	3	$\pm$.05in	.5-4.0-.5	Yaw resonance: Up - 1.59 Hz Down - 1.88-1.68 Hz Lateral lading 2.5-3.5 Hz
	37	Torsion	3	$\pm$.2g	6-20	No torsion resonance visable

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (5)

DATE	RUN NO.	ACTUATOR Mode	No.*	INPUT Amp.	Freq. (Hz)	RESULTS NOTES
05/14/81	38	Bounce	4	±.2in	1-5	Heavy spring bottom LVDT Lost - auto cut-off
	39	Bounce (Decay)	4	±.05in	2.05 (Decay)	decay reading - 2.05 Hz ξ = .0038
	40	Bending & Lading	4	±.3g	5-20	Lading bounce mode - 8.3 Hz Body benidng 12-13, 16-18
	41	Pitch (Decay)	5	±.1in	3.0	2.77 Hz decay = .047
	42	-same-		±.1in	2.8	
05/15/81	43	Staggered Rail Profile	6	±.1in	.7-.45	Sine input by mistake
	44	-same-		.2in	.7-.45	Rec. sine - Max at .30 Hz input (.6Hz response)
	45	-same-		.1in	.7-.45	Rec. sine - Max at .3 Hz input
	46	1st Roll	2	±.05in	.7 Hz	f,decay = .67 Hz & 2.5 Hz

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (6)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.	INPUT Amp.	INPUT Freq. (Hz)	RESULTS NOTES
05/15/81	47	Random 2 End	10	.125in RMS	Random 10 Hz Roll-Off	60 cycle noise, repeat run
	48			-same-		
	49	Hunting Simulation	8	±.2in	1.2-2.4	
	50	-same		±.1in	1.2-2.4	
	51	Yaw (Decay)	3	±.1in	1.2-1.6	f, decay - 1.67 Hz ξ = .047*
	52	1-COS Hump	7	.5in	2.5	
	53	-same-		1.0in	2.5	Unplanned repeated humps
	54	-same-		1.0in	2.5	
	55	-same-		1.0in	2.77	
	56	-same-		1.0in	3.1	Max response
	57	-same-		1.0in	2.5	(Discontinued because of spurious shaker inputs)
	58	2 End Random	9A	.125in rms	Random 10 Hz Roll-Off	(Shakers 1, 2, 3, 4 - A & C)
	Replaced snubbers for repeat of 1st test with change in dwell time					
	59	1st Roll		±.15in	.4-1.4-.4	Dwell time doubled to 16 sec. Snubbers broke at .761 Up lock at .645 Down

* ξ = damping ratio, C/C_c

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (7)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.	INPUT Amp.	INPUT Freq. (Hz)	RESULTS NOTES
CONFIGURATION 2: Empty, No Snubbers						
05/21/81	60	1st Roll (Decay)	1	±.05in	1.0	f, decay = .94 ξ = .088
	61	-same-		±.13in	.9	= .92 = .100
	62	-same-		±.15in	.75	= .87 = .07
	63	-same-		±.15in	.85	= .82 = .04
	64	-same-		±.15in	.02	= .81-.86 = .047
	65	2nd Roll (Decay)	2	±.10in	3.0	= 2.7 = .04-.10
	66	-same-		±.05	2.7	= 2.9-3.1 = .24-.06
	67	Staggered Rail	6	.1ins	.8-.25	Max amp at .4 Hz
	68	-same-		.2in	.8-.25	Max apt at .4 Hz
	69	-same-		.3in	.8-.25	Very large amp at .45 Fall off after .40
	70	1st Roll	1	±.10in	.4-1.4	Max amp .95 Hz 90^{o} phase .95-1.0 Hz
	71	-same-				
	72	1st Roll	1	±.05in	.4-1.4	Max amp .92-.97 Hz
05/22/81	73	Yaw (Decay)	3	±.2in	2.0	Decay f = 2.62 ξ = .022
	74	-same-		±.08in	2.6	f = 2.58 ξ = .02
	75	Hunting	8	±.10in	1.5-3.5	Max amp 2.5-2.6 Hz Coupled Yaw & 2nd Roll 2.6-3.5 Hz
	76	-same-		±.20in	1.5-3.5	Gib contact 2.4 Hz Max amp 2.5-2.6 Hz Coupled Yaw & 2nd Roll starts 2.7 Hz

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (8)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.	INPUT Amp.	INPUT Freq. (Hz)	RESULTS NOTES
05/22/81	77	Bounce (Decay)	1	±.15in	2.0	f = 3.85 ⟶ 3.92 Hz
	78	-same-		±.013	3.8	f = 3.79 ξ = .008-.018 ≈ .01
	79	Pitch (Decay)	5	±.165	3.5	f = 4.35
	80	-same		±.015	4.3	f = 4.34 ξ = .011-.021 ≈ .015
	81	Random	10	.125in RMS	20 Hz Roll Off	Body torsion = 13.5 Hz
	82	Random	4	.125in RMS	20 Hz Roll Off	Body bending = 17.6 Hz
	83	Track Irreg.	7	1.0in	3.4	1-COS track hump.
	84	-same-		1.0in	4.0	
	85	Random	10	.125in RMS	20 Hz Roll Off	Short repeat run 81 for Hewlett-Packard analysis.

TABLE B-1
TEST LOG - 70 TON BOXCAR VIBRATION TEST (9)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.	INPUT Amp.	INPUT Freq. (Hz)	RESULTS NOTES
CONFIGURATION 3 - Loaded Boxcar with Stretch Wrapped Packaging - With Snubbers						
06/02/81	86	Staggered Rail	6	.10in Rect.	.8-.2 sine	Snubber start ~.45 Hz (Shaker thump-.7 Hz) Max amp .40 Hz Out .35 Hz
	87	-same-		.20in Rect.	.8-.2 Sine	Snubber start - .50 Hz Run abort due to computer
	88	-Repeat Run-87				Snubber start - .50 Hz; Max amp.-.40 Hz out at .35 Hz (Shaker thump at end)
	89	-same-		.30in	.8-.2	Snubber start - .60 Hz; Max amp. -.40 Hz out at .35 Hz
	90	-same-		.50in	.8-.2	Snubber start - .80 Hz; Max amp. .40 Hz Side bearing contact - .45 Hz Max amp - .35 Hz out - .30 Hz (Shaker thump at end)
	91	Hunting Simulation	8	$\pm$.2in	1.2-2.4	Lading bounce starts at 1.9 Hz
	92	-same-		$\pm$.4in	1.2-2.4	Large lading motion - impact car side starts at 1.7 Hz
	93	-same-		$\pm$.6in	1.2-2.4	Lading impact starts at 1.45 Hz
	94	Track Irreg. (1-COS hump)	7	1.0in	2.5	Pitch & bounce motions dead beat, Lading lateral - 3 Hz, Lading vert. - 6 Hz, Carbody vert & lat - 13.5 Hz.
	95	-same-		1.0in	3.0	
	96	-same-		1.0in	3.4	

TABLE B-1

TEST LOG - 70 TON BOXCAR VIBRATION TEST (10)

DATE	RUN NO.	ACTUATOR Mode	ACTUATOR No.	INPUT Amp.	INPUT Freq. (Hz)	RESULTS NOTES
06/03/81	97	Yaw	3	$\pm$.05in	1-5	Lading lat. - 2.3-2.4 Hz, Q= 10
	98	Body Torsion & Lading	3	$\pm$.2g	6-20	Floor twist - 7 Hz 10.5, 10.8, 12 - side panels 17.1 door vert
	99	Rerun 90 with open door wedged and clamped				Door - 12.0 Hz Side panel 11.5
	100	1st Roll	1	$\pm$.05in	.4-1.5	1.02 Hz - snubber squeaked twice
	101	-same-		$\pm$.15in	.4-1.5	Snubbers broke - .71 Hz A end snubbers lock up .9 Hz B end 1.2 Hz
	102	2nd Roll	2	$\pm$.2in	.5-3	B snubbers broke at .9 Hz Lateral translation - 1.4-1.7 Hz 2nd Roll ?
	103	-same-		$\pm$.1in	1-5	
	104	Bounce	4	$\pm$.2in	1-5	Snubbers start - 2.6 Hz Violent A motion 2.8 Hz (pitch) Bounce 3.7 Hz
	105	Bending	4	$\pm$.3g	5-20	Lading vert - 8-8.5 Hz Side panels - 12, 12.5, 13.7 Body Bending - 15.8 Door panels - 17-17.3
06/04/81	106	Pitch	5	$\pm$.2in	2-8	Snubbers broke- 3 Hz 90° phase 3.8-4.4 Hz
	107	Bounce	4	$\pm$.3in	1-4	Snubber broke - 2.19 Hz Quickly went into very large amplitude, LVDT failure - abort test
	108	-same-		$\pm$.23in	1-5	Snubber broke - 2.19 Max amp & 90° 2.39 Hz

TABLE B-2 SHAKER CONFIGURATIONS

Shaker Config.	B Truck						A Truck					
	Axle 1			Axle 2			Axle 3			Axle 4		
	1A	1B	1C	2A	2B	2C	3A	3B	3C	4A	4B	4C
1	1	-1	0	1	-1	0	1	-1	0	1	-1	0
2	.5	-.5	1	.5	-.5	1	.5	-.5	1	.5	-.5	1
3	0	0	1	0	0	1	0	0	-1	0	0	-1
4	1	1	0	1	1	0	1	1	0	1	1	0
5	1	1	0	1	1	0	-1	-1	0	-1	-1	0
6	1	-1	0	1	-1	0	1	-1	0	1	-1	0
7	1	1	0	1	1	0	1	1	0	1	1	0
8	0	0	1	0	0	1	0	0	-1	0	0	-1
9	1	0	1	1	0	1	0	0	0	0	0	0
10	1	-1	0	1	-1	0	-1	1	0	-1	1	0

Shaker Notations: A = left side vertical
B = right side vertical
C = lateral

Shaker heads are in-phase if + and π-phase if -.