FRA-76-7.I

REPORT NO. FRA/ORD-76/303,I

TRAIN-TO-TRAIN REAR END IMPACT TESTS
Volume I - Pre-Impact Determination of Vehicle Properties

R.L. Anderson
P.L. Cramer

ULTRASYSTEMS, INC.
The Dynamic Science Division
1850 West Pinnacle Peak Road
Phoenix AZ 85027

MARCH 1977

FINAL REPORT

Prepared for
U.S. DEPARTMENT OF TRANSPORTATION
FEDERAL RAILROAD ADMINISTRATION
Office of Research and Development
Washington DC 20590

Train-to-Train

Vol. I

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
FRA/ORD-76/303,I		

4. Title and Subtitle	5. Report Date
TRAIN-TO-TRAIN REAR END IMPACT TESTS - Volume I - Pre-Impact Determination of Vehicle Properties	March 1977
	6. Performing Organization Code
	8. Performing Organization Report No. DOT-TSC-FRA-76-7,I

7. Author(s)	
R. L. Anderson, P. L. Cramer	

9. Performing Organization Name and Address	10. Work Unit No. (TRAIS)
Ultrasystems, Inc.* The Dynamic Science Division 1850 West Pinnacle Peak Road Phoenix AZ 85027	RR628/R7323
	11. Contract or Grant No. DOT-TSC-840-1
12. Sponsoring Agency Name and Address	13. Type of Report and Period Covered
U.S. Department of Transportation Federal Railroad Administration Office of Research and Development Washington DC 20590	Final Report June 1974 - Dec 1975
	14. Sponsoring Agency Code

15. Supplementary Notes

*Under Contract to: U.S. Department of Transportation
Transportation Systems Center
Kendall Square
Cambridge MA 02142

16. Abstract

Nine train-to-train rear end impact tests were performed by the Dynamic Science Division of Ultrasystems, Inc., at DOT's Transportation Test Center under contract with the Transportation Systems Center which is conducting the program for the Federal Railroad Administration.

This final report documents these nine tests.

Volume I, Pre-Impact Determination of Vehicle Properties, summarizes the vehicle properties obtained prior to the impact tests. These vehicle properties were used in computer simulation of the impact tests and included weights, pitch moments of inertia, spring rated, vertical center of gravity location, and linear dimensions.

Volume II, Impact Test Summaries, describes the impact tests. The impact tests were remotely controlled with impact speeds ranging from 3 to 30 mph. An array of approximately 20 highspeed cameras and 50 channels of data, including accelerations, strains, and displacement, documented the impacts.

Volume III, Impact Test Summaries Appendix, is an appendix to Volume II. It contains the original data of the impact test.

17. Key Words	18. Distribution Statement
Train-to-Train Imapct Tests, Impacts, Baseline Impact Tests, Impact Tests, Rear End Train Collisions	DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	100	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

This final report, Volume I, summarizes the determination of vehicle properties used in simulation of the train-to-train rear impacts which complemented the train-to-train impact test program conducted by the Dynamic Science Division of Ultrasystems, Inc., for the Transportation Systems Center under direction of the Federal Railroad Administration; Contract DOT-TSC-840. The Contract Technical Managers for the program were Mr. Sam Polcari and Mr. How Wong who worked in conjunction with Dr. A. R. Raab, Program Manager, and Dr. Pin Tong, program consultant; all of Transportation Systems Center. Mr. Don Levine was the Federal Railroad Administration Sponsor.

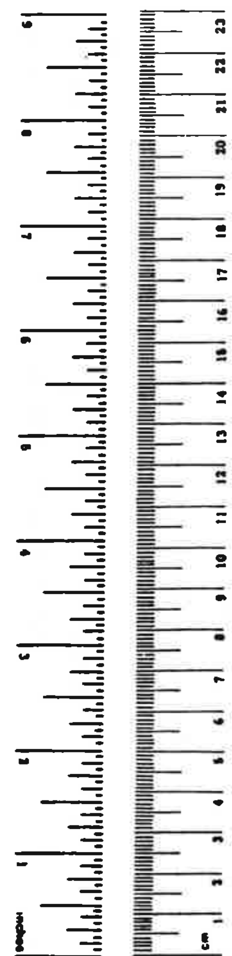
The program was devoted to determining the dynamic response characteristics of a series of rear-end train collisions, ranging from 3 mph to 30 mph.

The opinions and findings expressed in this publication are those of the authors and not necessarily those of the Transportation Systems Center.

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



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

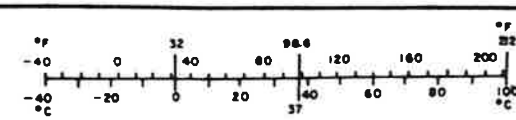


TABLE OF CONTENTS

TABLE OF CONTENTS (CONTD)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (CONTD)

LIST OF ILLUSTRATIONS (CONTD)

LIST OF TABLES

1. INTRODUCTION

1.1 BACKGROUND

As part of the effort to decrease the loss of life, the injury rate, and the extent of property damage due to train collisions, derailments, and other accidents, the Federal Railroad Administration (FRA) is pursuing a program to study the crashworthiness of rail vehicles and techniques for occupant injury minimization. In support of this effort, the DOT/Transportation Systems Center (TSC) provided assistance in the organization, conduct, and analysis of train-to-train impact tests relative to locomotive cabs, directed toward minimizing occupant fatalities and injuries during rear-end collisions. The impact tests were performed by the Dynamic Science Division of Ultrasystems, Inc. under contract with TSC (Contract No. DOT-TSC-840).

In the eight-year period from 1966 to 1973, there were 332 reported rear-end collisions. Seventy-two of these were responsible for 51 fatalities and 112 injuries to locomotive cab occupants. The FRA safety effort in this area is focused on determining why the impacted car, usually a caboose, in many instances overrides and crushes the locomotive cab during its post-impact trajectory while sustaining limited or no damage itself. This work is also aimed at determining the crushing forces exerted on the cab and the manner in which it fails, so that appropriate structural improvements or other energy management countermeasures may be developed.

The objective of the test series was to generate data which provide basic information on train-to-train dynamic interaction. These data include information on:

a) Locomotive frontal deformation

b) Force levels on the locomotive, caboose, and car in front of the caboose

c) Locomotive and caboose dynamics (trajectories, derailment)

d) Locomotive and caboose interaction (intrusion, buckling, crushing)

e) Possible injury modes of locomotive occupants

f) Fire hazards.

The data from these tests also provided the basis for refinement of mathematical computer models which simulated the dynamic behavior of the two trains during an impact. The basic test data, along with the results of computer simulations, will be applied to modify and delethalize the impacting vehicles and will be utilized in the planning of future crash energy management efforts.

The program consisted of 9 impact tests, ranging from 3 mph to 30 mph. For each test the trains were instrumented to measure forces, strain, acceleration, and displacements. These data along with the data from the high speed photography will provide the basis for investigating the above areas of interest.

A fundamental requirement for the mathematical model to be useful is that reasonable values of the physical properties are used to define the parameters in the mathematical model. With this in mind, a series of tests was devised to measure the dimensions and physical properties of some of the impact test cars and locomotives prior to the impact testing.

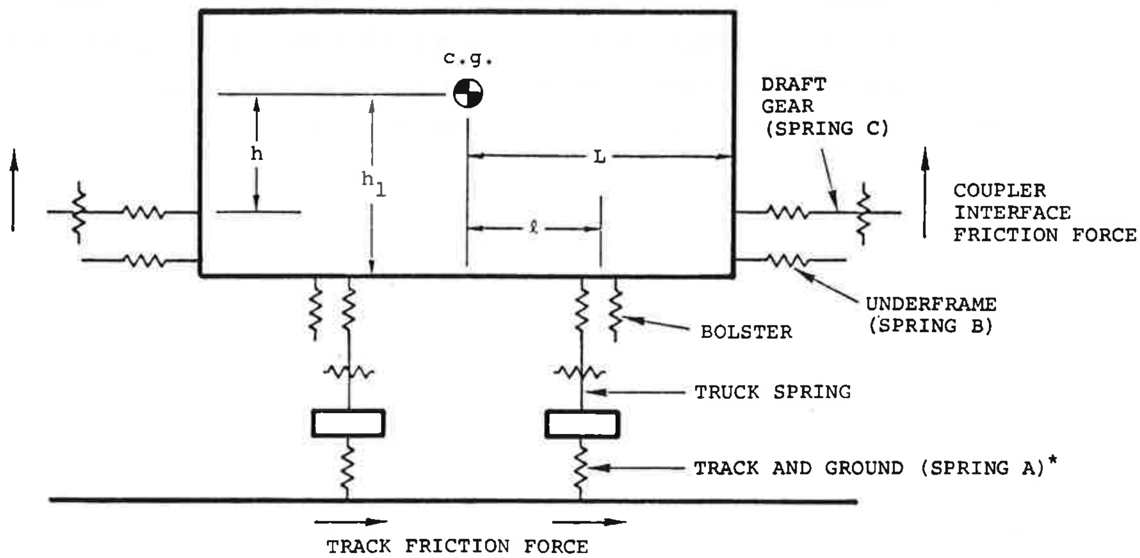
Volume I of this report summarizes the pre-impact physical property determination effort.

1.2 COMPUTER MODELING

The computer modeling was conducted independently by TSC and Washington University.

The modeling effort and the testing effort were integrated for the mutual benefit of both programs. The model provided a predicted response for a set of test conditions that was experimentally verified by test. The model was then upgraded to match experimental results more closely, and a new set of test conditions were defined which further verified the model.

A description of the computer modeling effort is beyond the scope of this report, but the TSC model did employ the modular formulation* developed at TSC. This method of formulation includes three-dimensional beam elements, various spring elements, rigid body elements, and modal elements. Figure 1 is an example of a schematic for the caboose model used in the simulation for the train-to-train impacts.



*SPRING LETERS ARE FOR TEXT REFERENCE.

FIGURE 1. MATHEMATICAL MODEL OF A CAR.

For the computer the cars are idealized and replaced by somewhat simplified elements and interconnections that can be conveniently described by mathematical expressions. The mass, inertial, and stiffness characteristics of each of these elements and interconnections must then be defined for the mathematical model to be used. The objective of the pre-test measurements described in this report is to measure and test for the characteristics required for the computer simulations. For instance, Section 3.8

*Tong, Pin, and Rossettos, J. H., "Modular Approach to Structural Simulation for Vehicle Crashworthiness Prediction," DOT Report No. DOT-TSC-NHTSA-74-7, March 1975.

defines the force deflection curve of the track and ground that is idealized by spring A, shown in Figure 1. Section 3.5 defines the underframe and draft gear force deflection characteristics idealized by spring B and C. Each measurement and test is aimed at filling out the required information for the computer analysis.

1.3 PROPERTY MEASUREMENTS

The test train consisted of three different types of vehicles; locomotive, caboose, and hopper car. The parameters for each car were obtained in a series of tests called pre-test measurements and include such things as weights, moments of inertia, dimensions, force deflection characteristics, etc.

2. FACILITIES AND DATA ACQUISITION

2.1 FACILITIES

Two basic facilities were utilized during the program. The Federal Railroad Administration's Transportation Test Center (TTC) near Pueblo, Colorado was the site where the tests were performed. The majority of the tests were performed at the Rail Dynamics Laboratory (RDL) and the Storage and Maintenance Building (SMB). Figure 2 is a map of the Transportation Test Center and Figure 3 is a picture of the operations area.

The truck stiffness, moments of inertia, and centers of gravity were tested in the RDL using the large overhead crane, the bridge pit, and a special "A-Frame" fixture built by TTC. The longitudinal and vertical spring rate tests were performed in the SBM using an indoor section of track.

The second facility utilized was the Dynamic Science Division of Ultrasystems, Inc., located in Phoenix, Arizona. Fabrication of equipment, calibration of instruments, and all data reduction were done at the Phoenix site.

2.2 DATA ACQUISITION

The data acquisition equipment for the pre-test measurements consisted basically of an SR4 strain indicator, a load cell, a series of linear potentiometers, dial indicators, and a hydraulic pressure gauge. Each instrument was calibrated either directly before the tests or shortly after the tests. The Appendix has a summary of the instrument accuracy and calibration information. A schematic of the location of the instruments used throughout the tests is included with each test summary.

Once the data was acquired, they were reduced into a form more useful for analysis, i.e., plots, force-deflection curves, structural stiffnesses, etc. Following each test, preliminary data were forwarded to TSC to be used in the computer modeling. This report summarizes the data acquisition and data reduction process along with the final results.

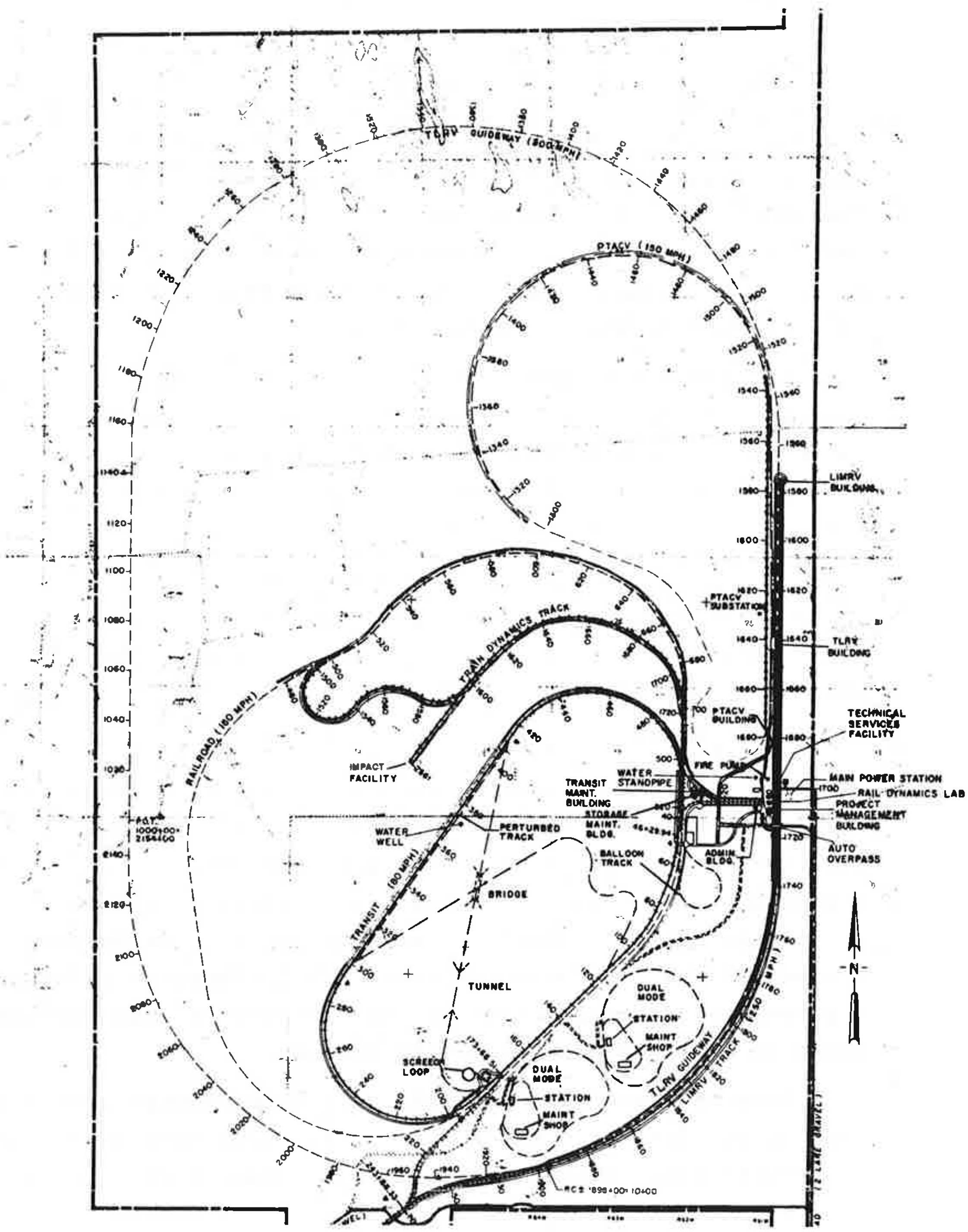


FIGURE 2. TRANSPORTATION TEST CENTER.

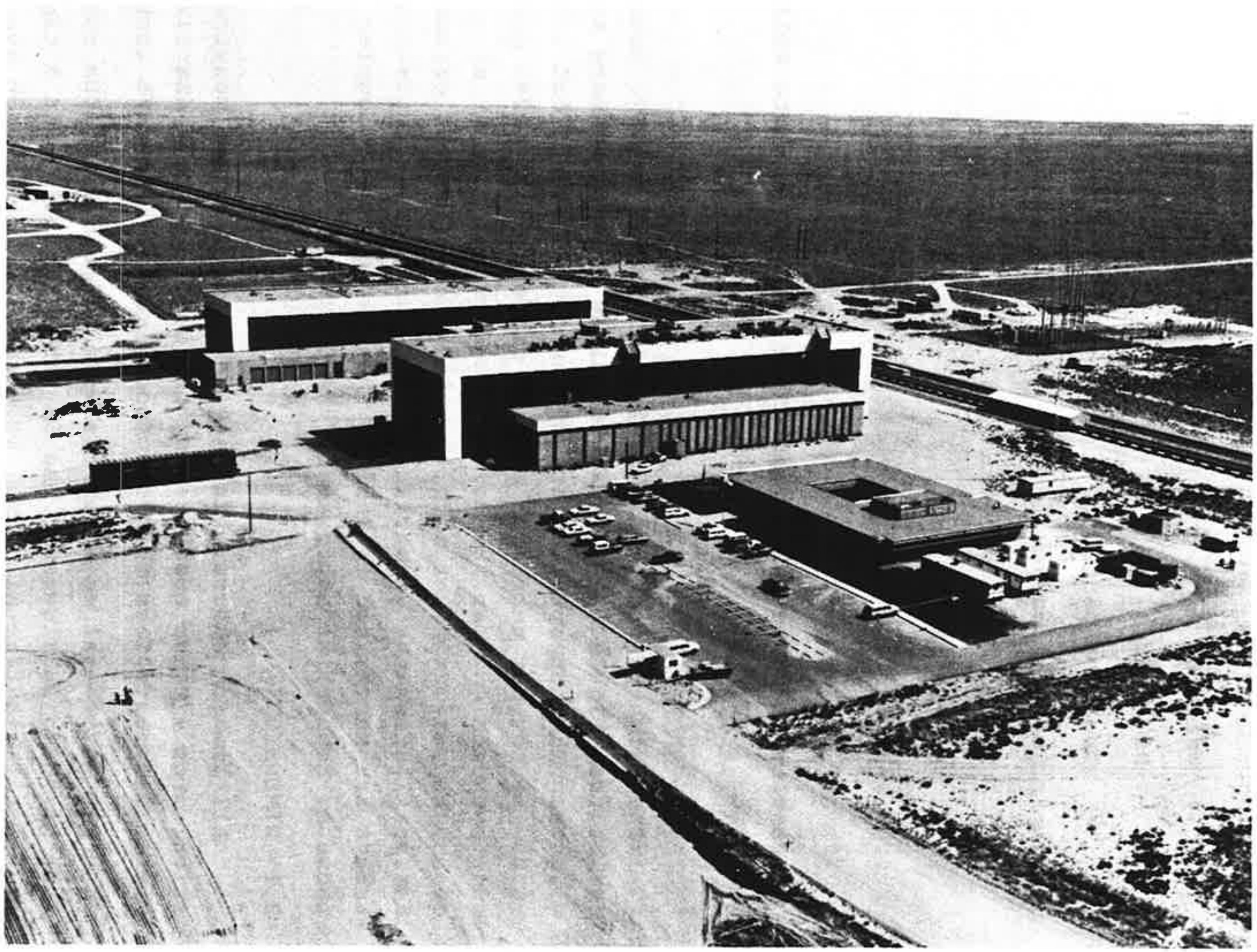


FIGURE 3. TRANSPORTATION TEST CENTER'S OPERATIONS AREA.

3. TEST AND MEASUREMENT SUMMARIES

The physical property tests and measurements included the following:

- Weights
- Dimensions
- Truck properties
- Vertical car stiffness
- Longitudinal car stiffness
- Center of gravity
- Pitch moment of inertia
- Vertical rail stiffness.

The results of these measurements are presented in the following sections.

3.1 WEIGHTS

The weights were obtained with the use of the TTC track scales (see Figure 4). Table 1 itemizes the car numbers and weights. The hopper cars were ballasted to a specified 155,000 ±4,000 pounds per car and a maximum difference of 1,240 pounds end-to-end on any car. Thus, the longitudinal center of gravity of any hopper was never more than 1.5 inches (or 0.4 percent of the total) from the geometric center. Weights of representative trucks are summarized in Table 2. The weight of a coupler (caboose or hopper) is approximately 400 pounds.

3.2 DIMENSIONS

All linear measurements were taken with a tape measure or ruler. For a range greater than one foot, accuracy was ±1/4 inch. For measurements less than one foot, accuracy was ±1/8 inch.

The horizontal measurements were referenced to the coupler face as shown in Figures 5 and 6. The knuckle was in a tightly closed position (total travel at knuckle edge = 1 inch average),

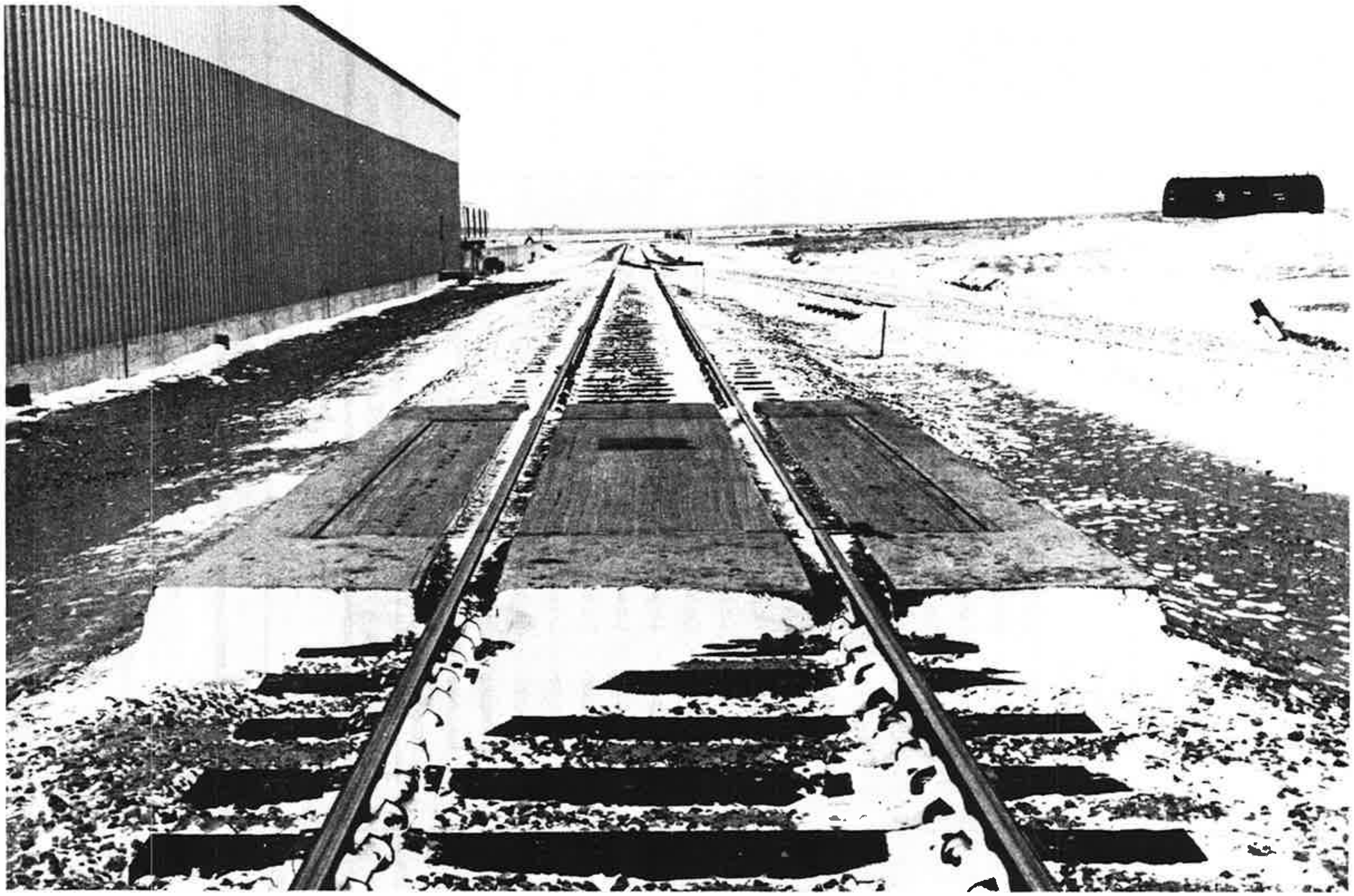


FIGURE 4. TRANSPORTATION TEST CENTER'S TRACK SCALES.

TABLE 1. VEHICLE WEIGHTS		
Type Vehicle	Serial Number	Total Weight (lb)
Locomotive - Dry	8003	248,173
Locomotive - Wet*	8003	259,849
Locomotive - Wet*	8670	266,058
Caboose	MP918	42,140
Caboose	MP912	42,948
Hopper - Loaded	536,506	157,660
Hopper - Loaded	536,605	157,840
Hopper - Loaded	536,614	153,080
Hopper - Loaded	536,631	157,800
Hopper - Loaded	536,843	151,540
Hopper - Loaded	537,119	159,500
Hopper - Loaded	537,508	158,620
Hopper - Empty	538,021	46,800
Hopper - Loaded	538,021	152,360
Boxcar - Loaded	142,075	150,680
Boxcar - Loaded	142,402	165,200
Boxcar - Loaded	142,508	156,400
Boxcar - Loaded	146,468	159,320
Boxcar - Loaded	146,702	147,160
Boxcar - Loaded	146,929	159,460
Boxcar - Loaded	147,317	143,480
Boxcar - Loaded	273,349	N/A
Boxcar - Loaded	274,877	147,160
Boxcar - Empty	276,567	46,280

*Includes 1,400 gallons of water for Test 9 only.

TABLE 2. TRUCK WEIGHTS	
Vehicle	Weight (lb)
Locomotive (estimate)	28,000
Caboose	7,060
Hopper	7,940
Boxcar	7,240

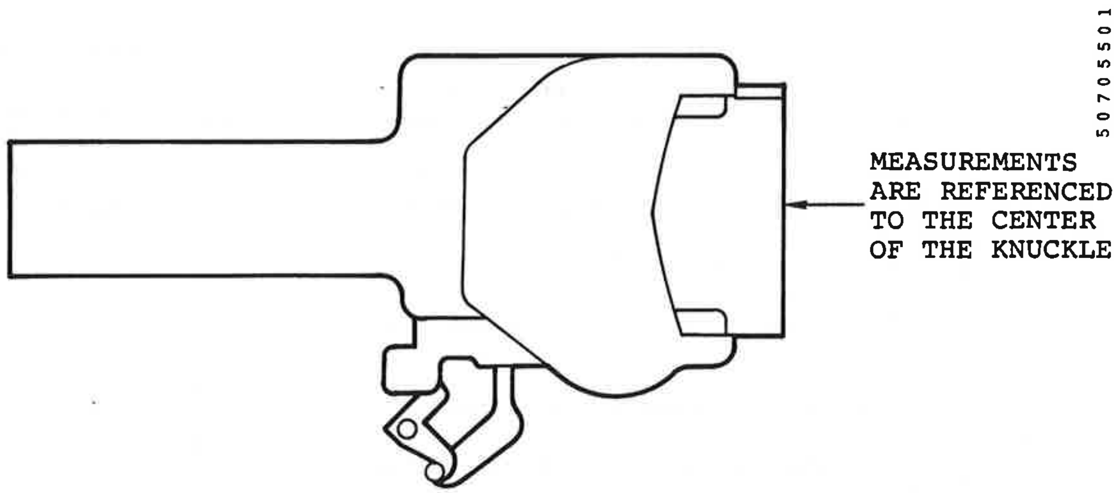


FIGURE 5. COUPLER MEASUREMENTS - SIDE VIEW.

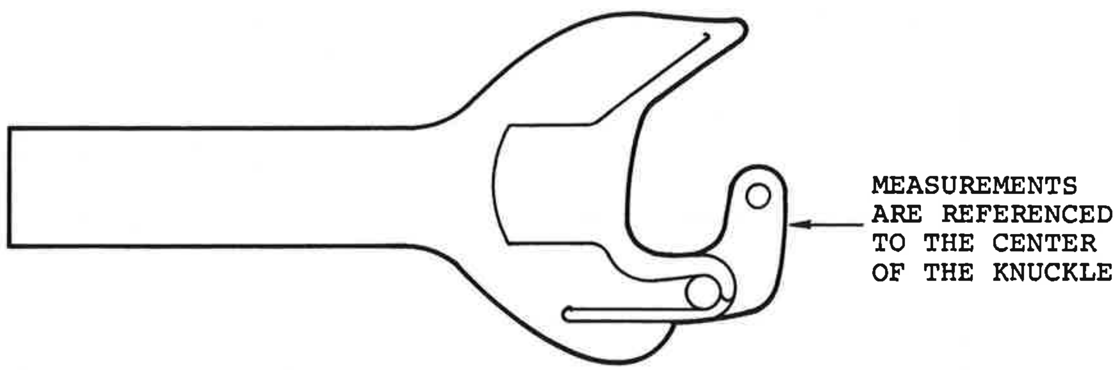


FIGURE 6. COUPLER MEASUREMENTS - TOP VIEW.

and the coupler was pushed by hand up against the draft gear. Vertical measurements were made with respect to a bar placed across the top of the rails directly below the point to be measured. The draft gear spring travel was measured as the total distance the coupler was allowed to move to put all draft gear components rigid with respect to the center sill. Tables 3 through 5 list the dimensions measured on three test vehicles. The locomotive trucks were rigidly attached to the bolster plate, thus no center pin height was obtained.

TABLE 3. CABOOSE DIMENSIONS (IN.)		MP912	MP918
Distance Between Coupler Faces		418.0	416.0
Length of Center Sill		375.0	372.0
Thickness of Buffer Casting (striker plate)		5.0	5.0
Distance Between Truck Centers		250.0	248.0
Longitudinal Coupler Slack (knuckles + draft gear)		2.25	2.5
Depth of Truck Bolster Dish		1.25	1.25
Draft Gear Spring Travel		2.25	2.5
Coupler Height Above Rail:	"A" end	32.0	31.5
	"B" end*	34.0	34.5
Bolster Height Above Rail:	"A" end	25.5	25.5
	"B" end	27.1	27.4
Sill Height Above Rail:**	"A" end	27.25	27.25
	"B" end	29.25	29.9
Coupler Height Above Bolster:	"A" end	6.5	6.0
	"B" end	6.9	7.1
Height of Center Pin Above Bolster Surface:	"A" end	N/A	6.5
	"B" end	N/A	7.25
Vertical Coupler Slack:	"A" end	3.5	3.25
	"B" end	2.9	3.9

*Impact end.
**Measured to bottom of flange next to the buffer casting.

TABLE 4. HOPPER NO. 538021 DIMENSIONS (IN.)		
Distance Between Coupler Faces		538.0
Length of Center Sill		491.0
Thickness of Buffer Casting (striker plate)		5.0
Distance Between Truck Centers		381.0
Longitudinal Coupler Slack (knuckles + draft gear)		2.75
Depth of Truck Bolster Dish		1.25
Draft Gear Spring Travel		3.0
Coupler Height Above Rail (empty car):	"A" end*	32.0
	"B" end	33.75
Bolster Height Above Rail (empty car):	"A" end	25.9
	"B" end	26.1
Sill Height Above Rail (empty car):**	"A" end	28.1
	"B" end	28.4
Coupler Height Above Bolster	"A" end	6.1
	"B" end	7.6
Height of Center Pin Above Bolster Surface:	"A" end	8.25
	"B" end	5.5
Vertical Coupler Slack:	"A" end	3.25
	"B" end	2.75

*Impact end.
**Measured to bottom of flange next to buffer casting.

TABLE 5. LOCOMOTIVE 8003 DIMENSIONS (IN.)	
Distance Between Coupler Faces	654.0
Length of Center Sill	600.0
Thickness of Buffer Casting (striker plate)	9.5
Distance Between Truck Centers	354.0
Longitudinal Coupler Slack (knuckles + draft gear)	1.1
Draft Gear Spring Travel (estimate)	2.5
Coupler Height Above Rail - Impact End	32.0
Bolster Height Above Rail	40.0
Sill Height Above Rail	43.0
Depth of Truck Bolster Dish	N/A
Coupler Height Above Bolster - Impact End	-8.0
Height of Center Pin Above Bolster Surface	Attached
Vertical Coupler Slack - Impact End	1.0

The hopper (538021) was loaded for Test 9. The following is a list of changed weights and dimensions due to addition of ballast.

Weight:	"A" end	75,500 lb
	"B" end	76,860 lb
Total Weight		152,360 lb
Coupler Height Above Rail* (impact end)		33.0 in.
Sill Height Above Rail:	"A" end**	26.9 in.
	"B" end	27.25 in.
Bolster Height Above Rail:	"A" end	24.75 in.
	"B" end	25.0 in.

*A shim plate was added to raise the coupler to maximum allowable by AAR specifications.
**Impact end.

When the car was loaded, the majority of the ballast was distributed on each end section with only a small amount in the middle section. The unloaded sill had a slight bend, as shown in Figure 7, and after loading, the sill was bent more toward the ends. (Hopper car compliance in bending is discussed later in Section 3.4.)

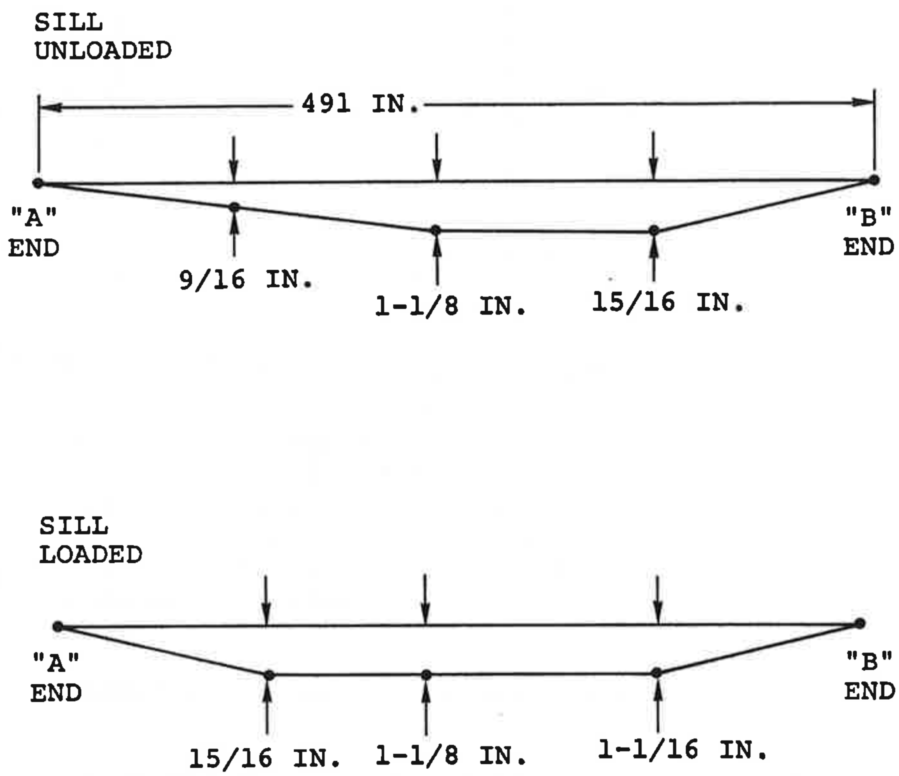
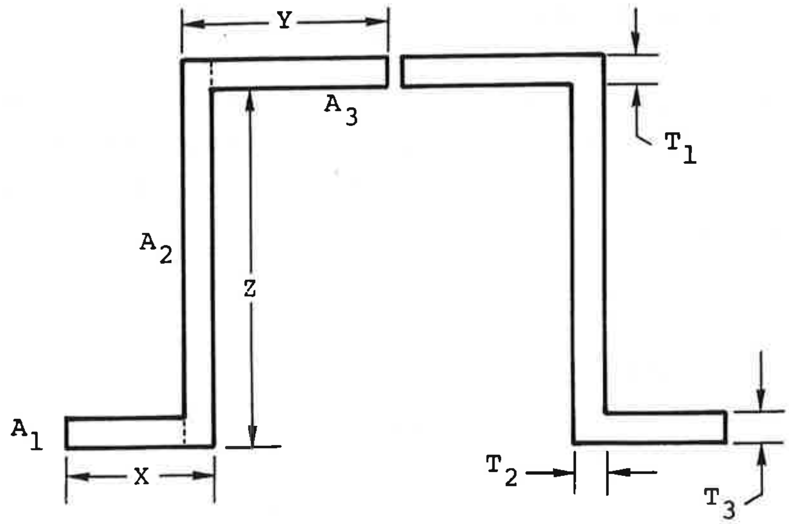


FIGURE 7. HOPPER SILL BEND.

The dimensions of the center sill for both Hopper (538021) and a Caboose (MP918) are shown in Figure 8. Both vehicles had severe rust, therefore, the figures shown represent an average measurement taken at several locations.



CABOOSE			HOPPER		
Z (IN.)	X (IN.)	Y (IN.)	Z (IN.)	X (IN.)	Y (IN.)
12.38	3.97	6.80	12.5	4.06	6.80
T_1 (IN.)	T_2 (IN.)	T_3 (IN.)	T_1 (IN.)	T_2 (IN.)	T_3 (IN.)
.500	.430	.775	.50	.455	.75

FIGURE 8. CROSS SECTION OF THE CENTER SILL.

The area moment of inertia of the center sill was calculated using three equations:

$$\bar{y} = \frac{\Sigma Ay}{\Sigma A} \tag{1}$$

where $\bar{y}$ = distance from reference axis to the centroid of the sill

A = area of a particular portion of the sill

y = distance to center of A from the reference axis and

$$I_x = [1/3\ b_1h_1^{\ 3} + 1/3\ b_2h_2^{\ 3} + 1/12\ b_3h_3^{\ 3} + A_3d^2]\ 2 \tag{2}$$

where I = total area moment of inertia of the sill about the reference axis

b = width of a particular area

h = height of a particular area

d = distance from center of A_3 to reference axis

and

$$I_x = I_c + A\bar{y}^2 \tag{3}$$

where I_c = area moment of inertia about centroidal axis of the total area

A = total area of sill

Figure 8 is a graphic representation of the cross section of the center sill of the hopper and the caboose.

Using the above equations, the values shown in Table 6 were calculated for the cross sectional area of the center sill (the sill was assumed to be symmetrical about the vertical centerline).

TABLE 6. MOMENT OF INERTIA OF CENTER SILL

Vehicle	Vertical Distance to Centroidal Axis-$\bar{L}$* (in.)	Area Moment of Inertia (I_c) (in.^{4})
Caboose MP918	6.71	597.0
Hopper 538021	6.77	616.0

*Measured vertically up from the bottom surface of the bottom flange.

3.3 TRUCK PROPERTIES

3.3.1 Requirements

The vehicle trucks have springs that cushion vertical motion of the car. During the impact test, the car was pushed up or down, with some of the vertical forces being taken in the springs. When the springs bottomed out, the forces were transferred into the rail through the bolster and side frames. The elastic constant of both the springs (coil or leaf) and the relatively rigid truck were required as parameters for the computer model. In addition, dimensions and weights were needed to define the mass-spring system. Tests were performed to obtain the stiffness of the truck springs and the stiffness of the combined bolster and side frame.

3.3.2 Test Procedure

The preparation for the test included calibrating the test equipment in Phoenix and building fixtures at TTC.

Calibration of the hydraulic system was accomplished using a load cell and meter to develop force-pressure curves for the pressure gauge. The calibration procedure is included in the Appendix. The fixtures that were fabricated at TTC include two 6-foot-long 18 WF96 I-beams with special attachments for hydraulic cylinders and mounting brackets for instruments.

The tests were conducted on both an ASF ride-control hopper truck and a Bettendorf caboose truck. Each truck was set on a section of rail in the RDL over a pit. One of the I-beams was attached to the rail support beams under the truck and the other was set on top of the bolster (see Figure 9). Two 100,000-pound hydraulic cylinders were used to connect the two beams. When hydraulic pressure was applied, the beams compressed the truck to give deflection versus force. Chains and cables were added to ensure a safe test.

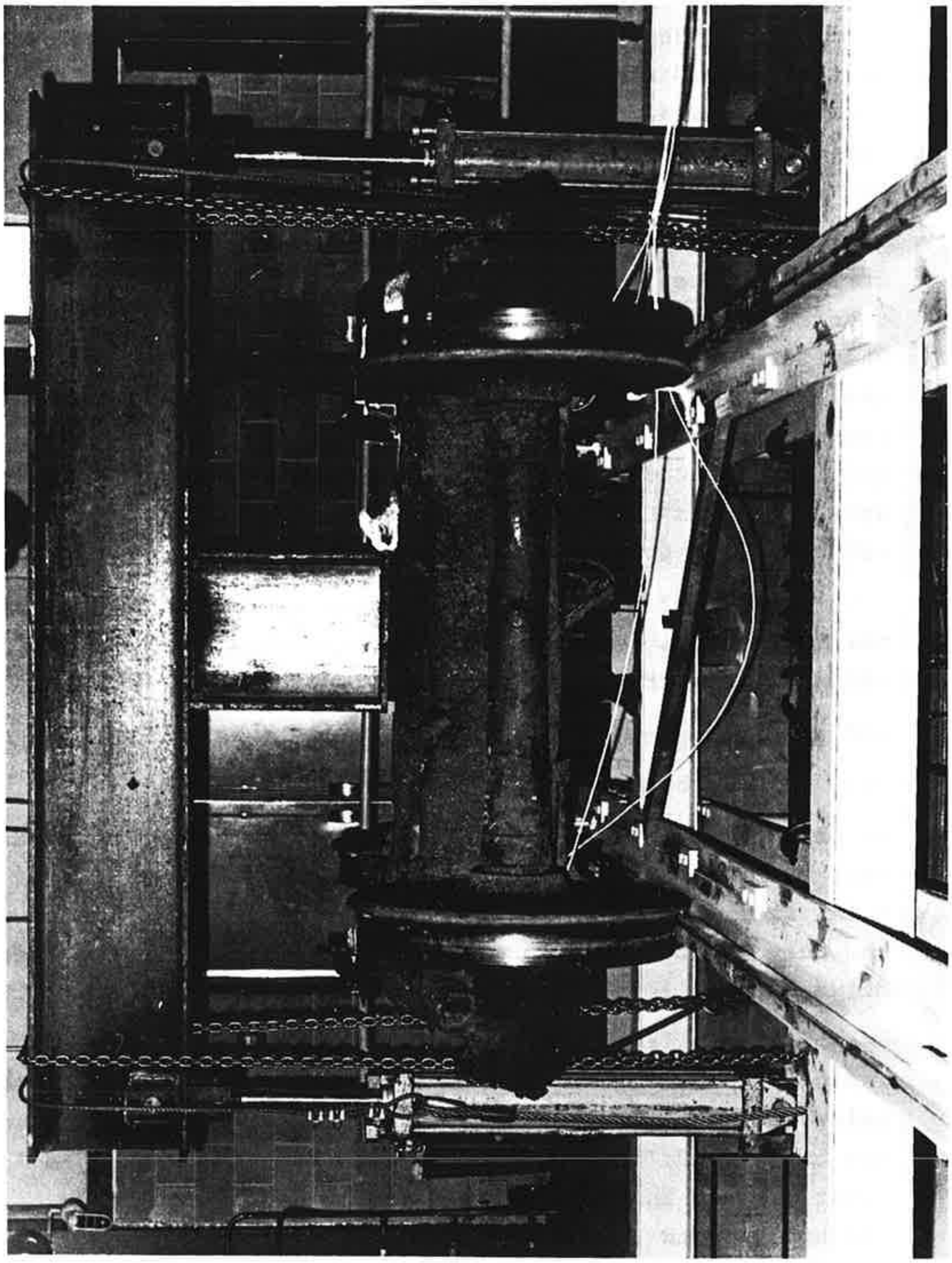


FIGURE 9. TRUCK STIFFNESS TEST CONFIGURATION.

The force was measured by means of a pressure gauge (see Figure 10) between the hydraulic pump (see Figure 11) and the cylinders. The pressure was converted to force by using the calibration curves shown in the Appendix. The deflections were measured using dial indicators which were mounted to the side frames for the spring rate of the springs or to the base of the rail for the spring rate of the bolster and side frames (see Figure 12).

Once the pressure-versus-deflection of the bolster was measured with the coil or leaf springs installed, the force deflection curve in pounds-force per inch was calculated. The spring stiffness was then calculated as the slope of the mean curve obtained as a result of several tests. If the first or last reading did not follow a linear pattern due to springs settling or bottoming out, they were not used to obtain the slope. The pressure gauge was found to be somewhat irratic for readings below 200 psi, thus force readings below 4,000 pounds were not included.

Figures 13 and 14 are the curves of force versus deflection for the caboose leaf springs and the hopper coil springs. The slope of the curve for the caboose springs obtained from points 1 and 2 is $\frac{40,000 - 6,460}{7.45 - 3.26}$ or 8,004 lb/in. Similarly, the hopper coil springs have stiffness of $\frac{80,000 - 43,693}{.725 - 0} = 50,078$ lb/in. To obtain the stiffness of the truck after the springs bottomed out, the caboose leaf springs were blocked with 5 in. x 4 in. x 9 in. steel blocks. The same test procedure was used to obtain the curves shown in Figure 15. The steel blocks (two) were set between the elipitcal springs as shown in Figure 16.

The compliance of the blocks was assumed negligible. A dial indicator was mounted on a member across the bridge beam directly below the center of the bolster. A vertical rod was attached to the bottom side of the bolster and extended down to the dial indicator. Thus, the deflection measured was the distance between the bolster and the bottom of the wheels (see Figure 17). To check for deflection of the rail under the wheels, another dial

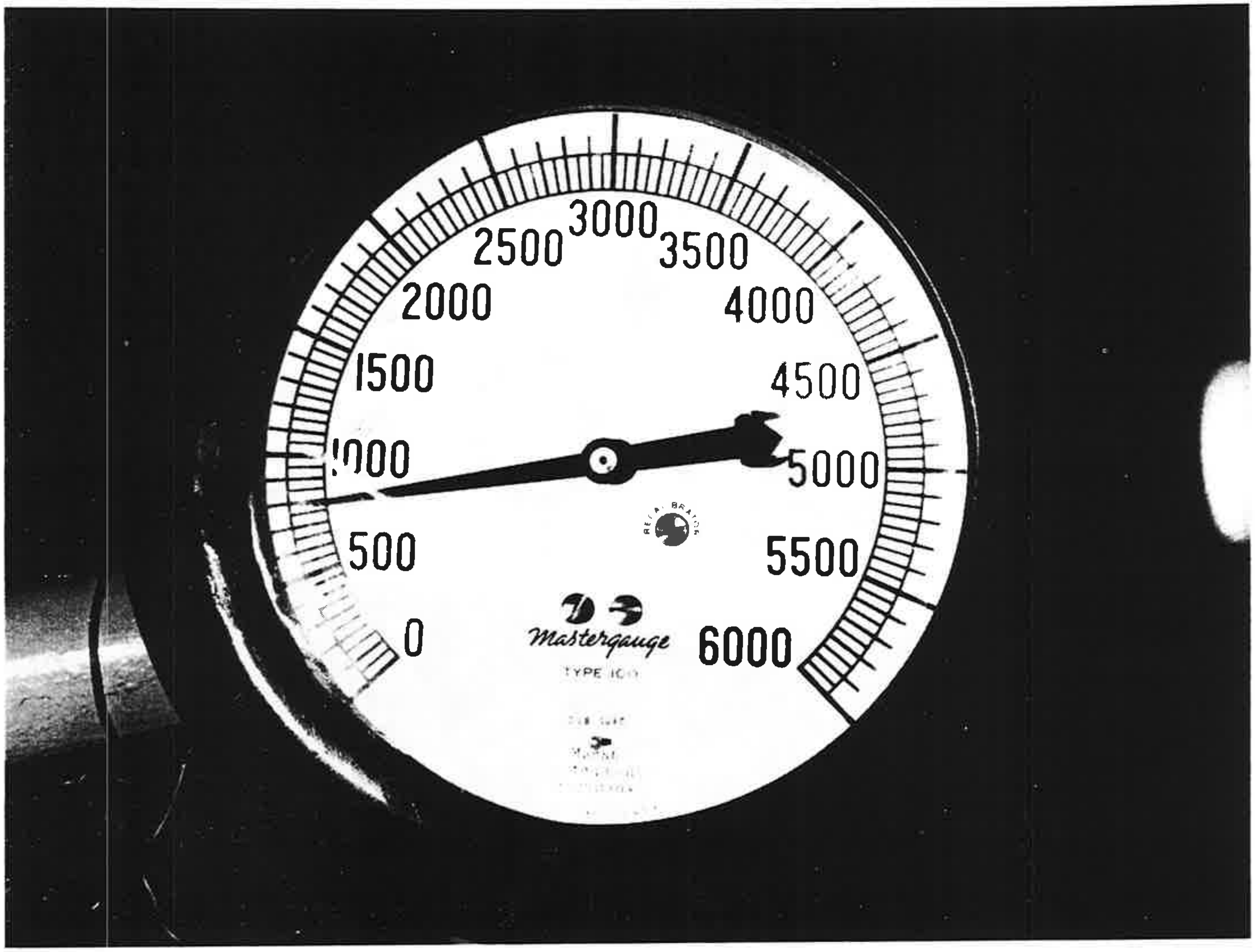


FIGURE 10. PRESSURE GAUGE FOR HYDRAULIC SYSTEM.

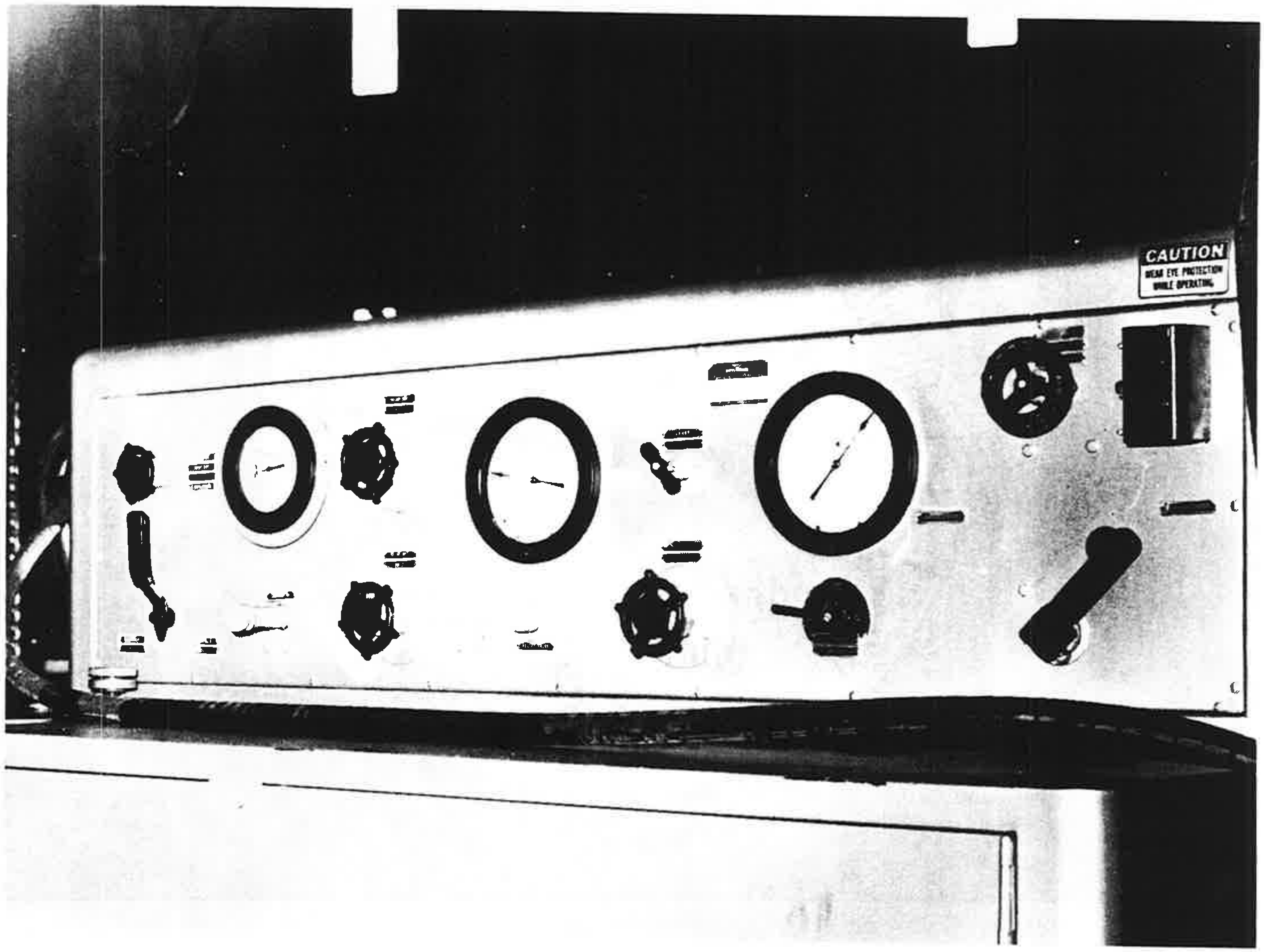


FIGURE 11. HYDRAULIC PUMP TO SUPPLY HYDRAULIC CYLINDERS.

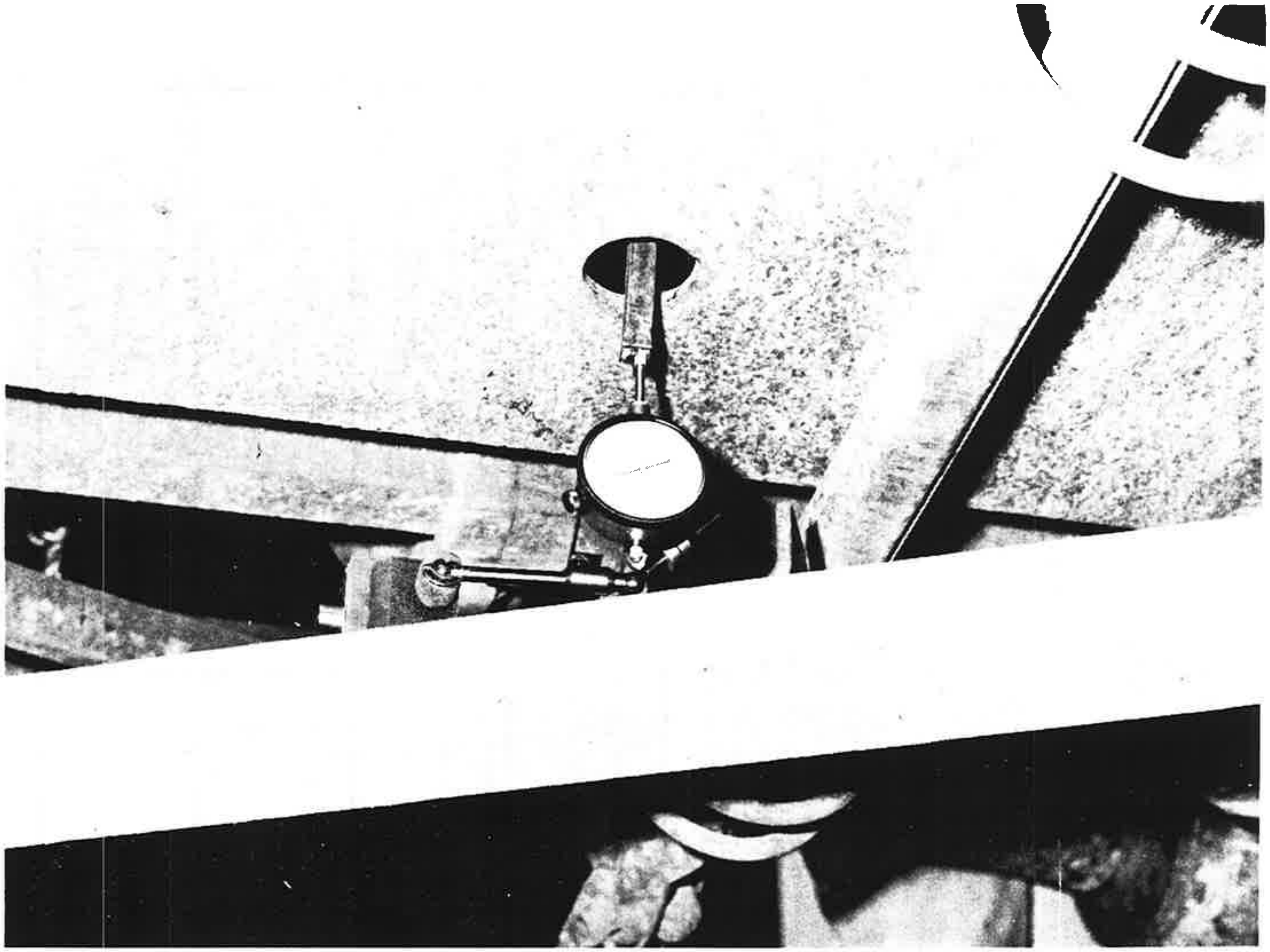


FIGURE 12. DIAL INDICATOR FOR MEASURING BOLSTER DEFLECTION.

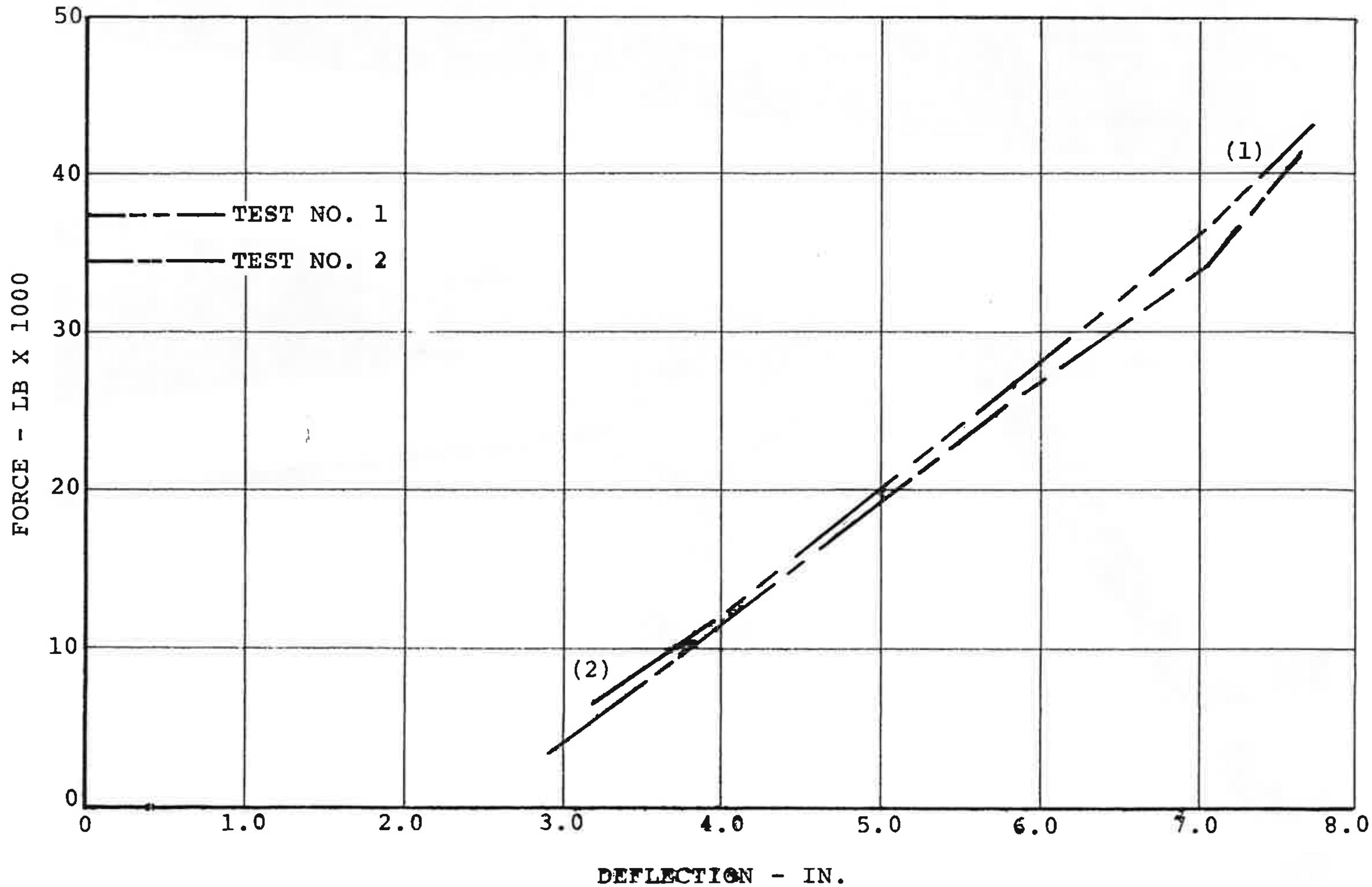


FIGURE 13. FORCE DEFLECTION OF CABOOSE TRUCK LEAF SPRINGS.

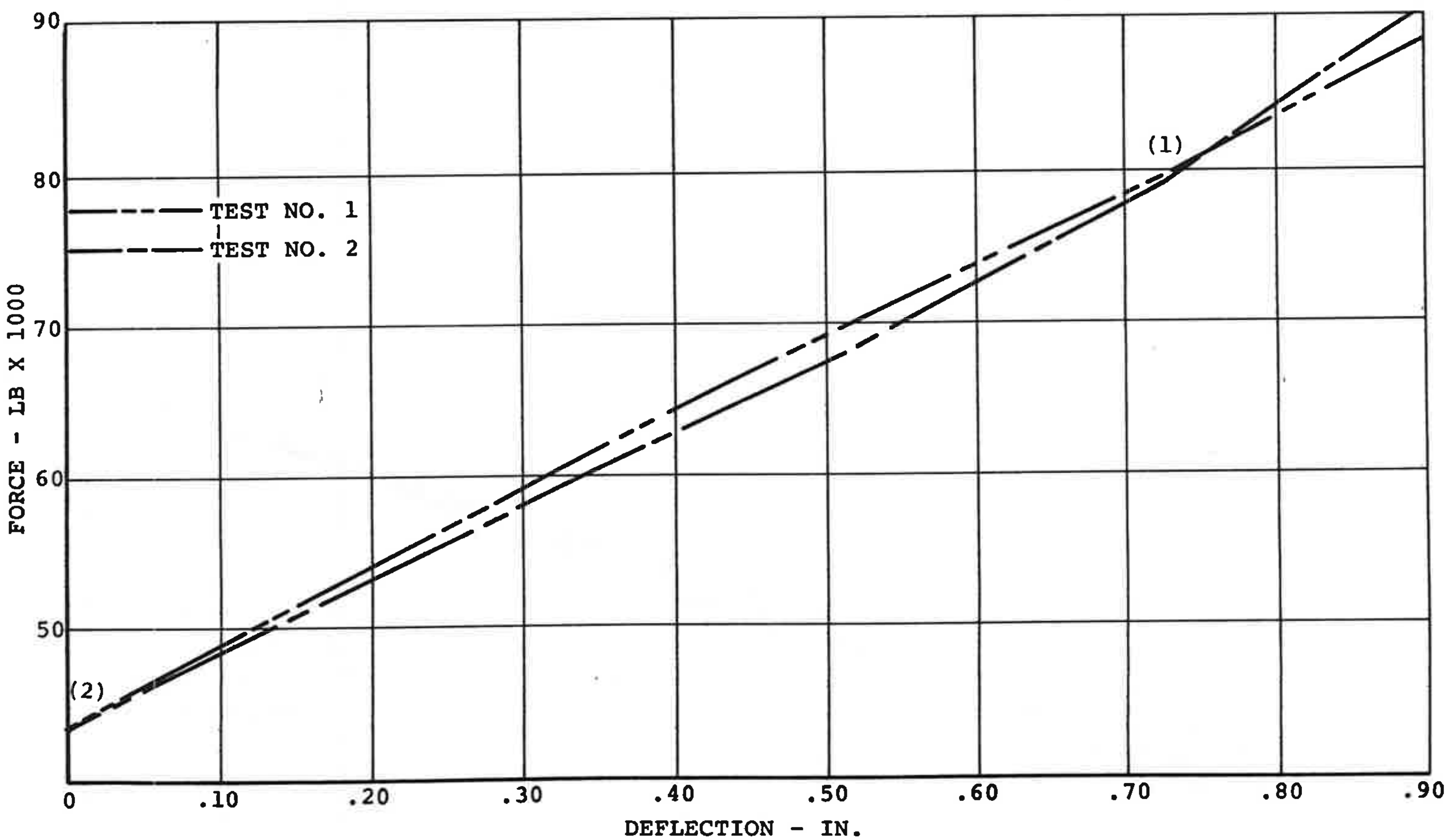


FIGURE 14. FORCE DEFLECTION OF HOPPER TRUCK COIL SPRINGS.

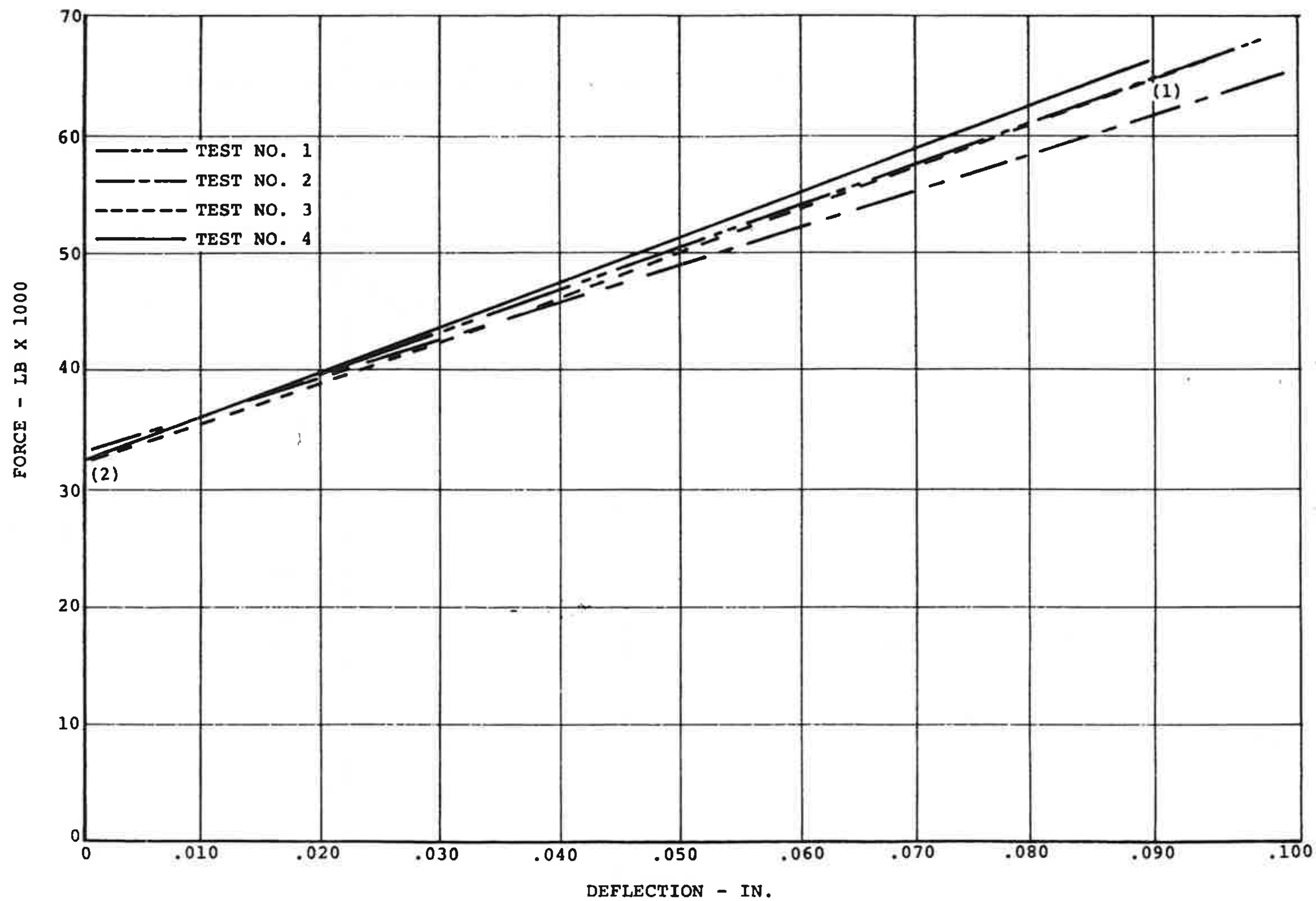


FIGURE 15. CABOOSE TRUCK BOLSTER AND SIDE FRAME FORCE DEFLECTION.

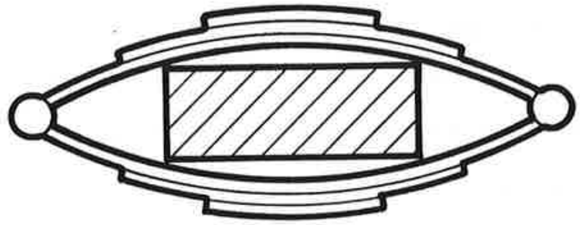


FIGURE 16. SCHEMATIC SHOWING HOW CABOOSE SPRINGS WERE BLOCKED.

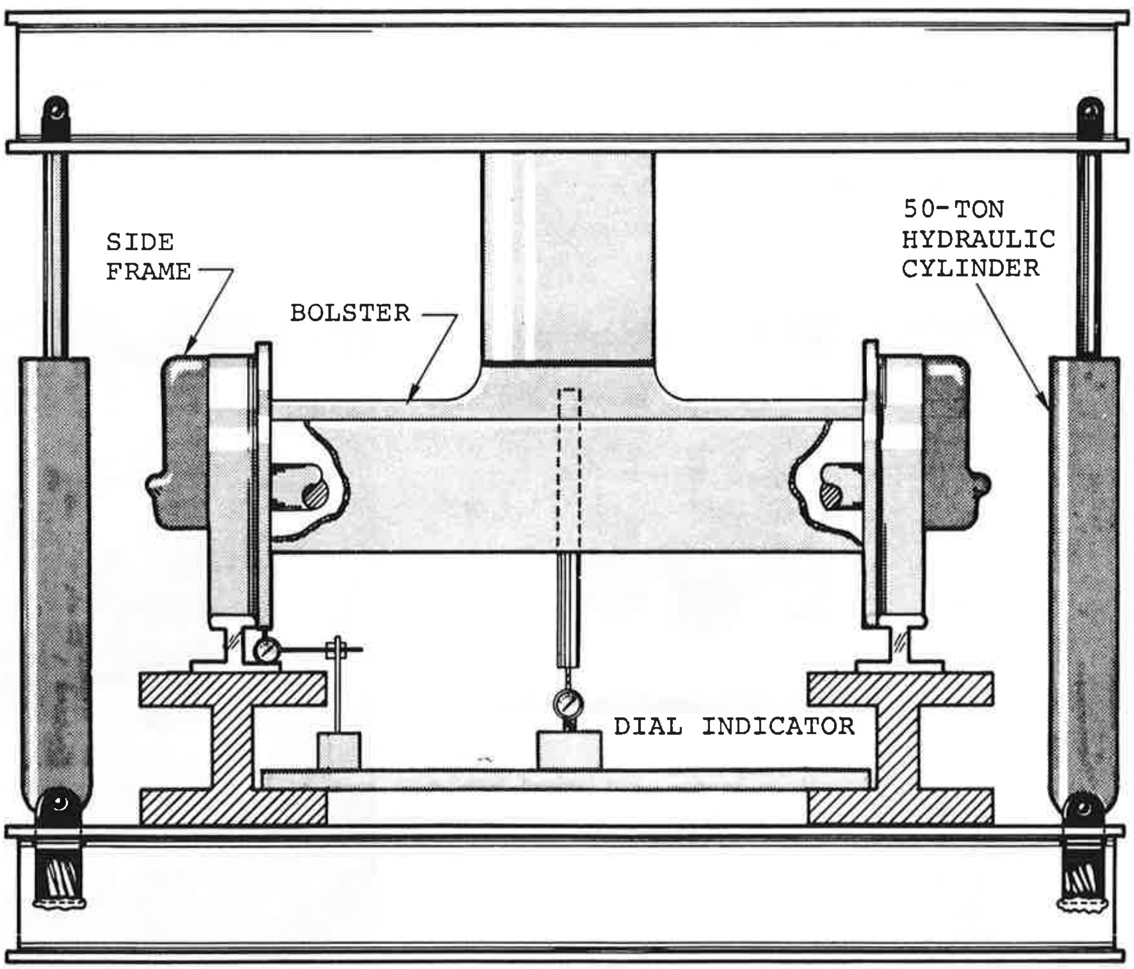
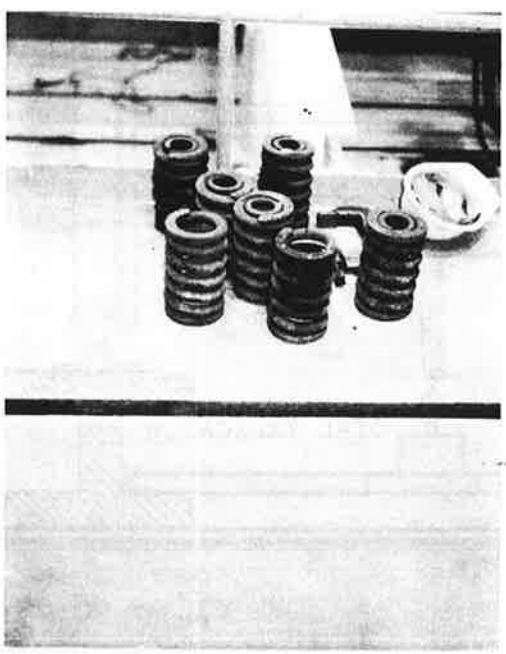


FIGURE 17. TRUCK BOLSTER AND SIDE FRAME STIFFNESS TEST CONFIGURATION

indicator was mounted to read vertical motion of the wheel during the test. The total rail deformation was less than 0.001 inch. The caboose stiffness was calculated as $\frac{65,000 - 32,000}{.089 - 0}$ or 370,786 lb/in.

The hopper truck had coil springs which were removed and replaced with 14 steel tubes 4 inches in diameter with a 1/4-inch wall (Schedule 40) and 8 inches long (see Figure 18). The tubes have a calculated stiffness of approximately 157 million lb/in. which is significantly stiffer than the rest of the truck. The same bolster and side frame deflection test was performed as on the caboose truck. The hopper truck was stiffer as seen in Figure 19. The hopper stiffness is calculated to be $\frac{120,000 - 44,800}{.167 - .074}$ or 809,735 lb/in.



ACTUAL SPRINGS REMOVED FROM TRUCK

7 TUBES PER SIDE REPLACED 7 COIL SPRINGS

FIGURE 18. HOPPER TRUCK SPRINGS AND TUBES.

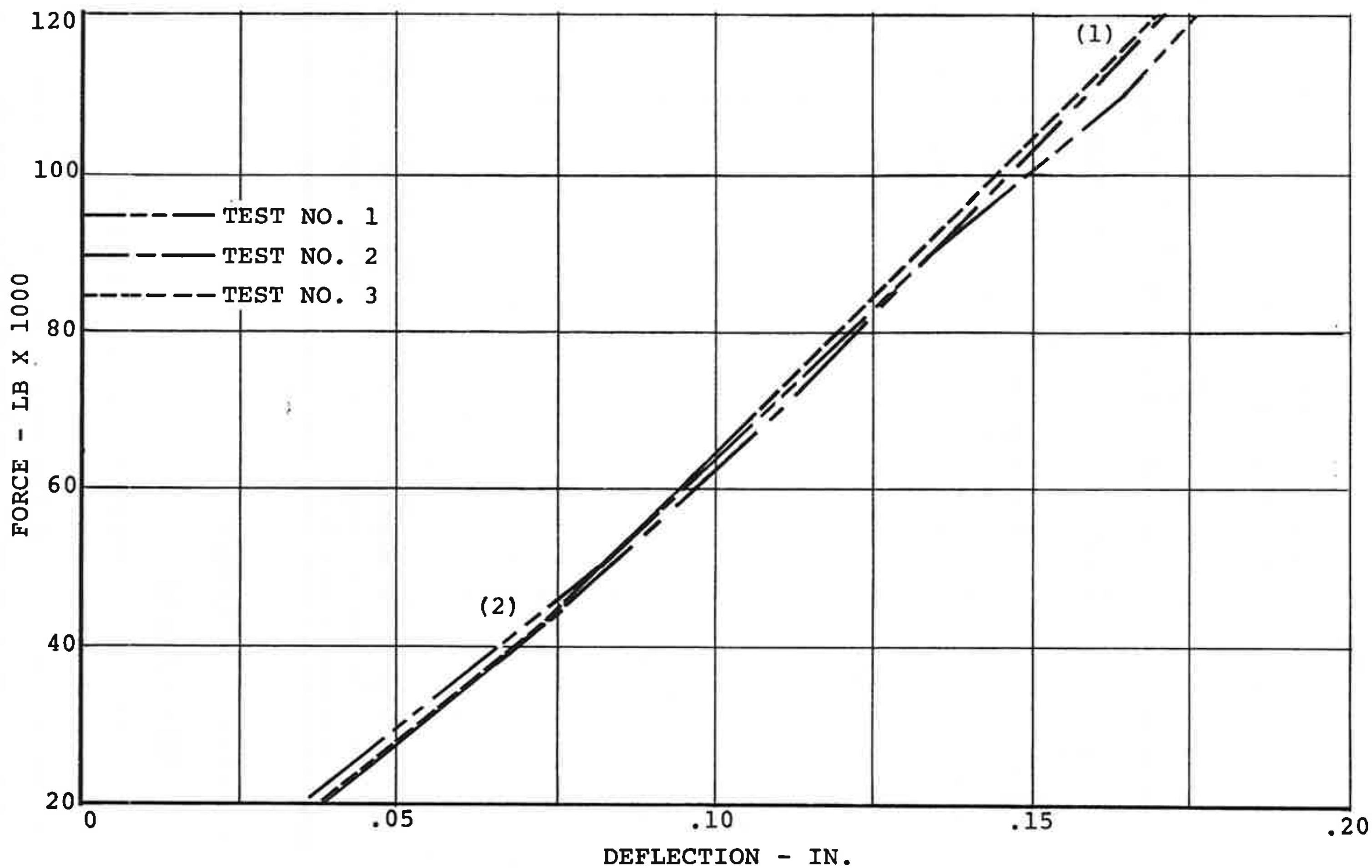


FIGURE 19. HOPPER TRUCK BOLSTER AND SIDE FRAME FORCE DEFLECTION.

3.3.3 Test Results

Table 7 is a summary of the results of the truck tests along with additional weights and dimensions.

TABLE 7. TRUCK PROPERTIES

Properties	Caboose Truck	Hopper Truck	Locomotive Truck
Vertical Stiffness	8,004 lb/in.	50,078 lb/in.	80,000 lb/in.
Bolster and Side Frame Stiffness	370,786 lb/in.	809,735 lb/in.	1 x 10^6 lb/in.
Truck Spring Travel to Bottom (average)	3.6 in.	2.2 in.	2.6 in.
Truck Spring Travel to Top (average)	1.5 in.	3.25 in.	3.0 in.
Inside Diameter of Bolster Dish	12.0 in.	14.0 in.	-
Bolster Dish Height	1.25 in.	1.25 in.	1.5 in.
Axle Size	4.25 x 8 in.	6.0 x 11.0 in.	-
Total Weight	7,060 lb	7,940 lb	28,000 lb

Note: The stiffness for the locomotive truck are estimates, scaled from measured car/truck stiffness, and the truck weight was extrapolated from known weights of trucks for similar locomotives.

3.4 VERTICAL STIFFNESS

3.4.1 Requirements

For some considerations, a railroad car is a rigid object, but when subject to high loads, it can deform. The computer model requires a knowledge of the force deflection characteristics of the car. The vertical or bending stiffness was obtained by changing the vehicle support points and measuring the change in deflection of the vehicle. The result gives a relationship between vertical force applied at the coupler and elastic vertical deflection of the car at various longitudinal stations.

3.4.2 Test Procedure

The vertical force deflection characteristic of the hopper (Serial No. 536843) and the caboose (MP918) were obtained by utilizing the weight of the vehicle as a downward force and lifting up on each end with hydraulic cylinders. Each cylinder was capable of lifting 100,000 pounds, thus with two, the loaded hopper (151,540 pounds) could easily be supported at the couplers.

At the beginning of the test, wooden blocks were placed under the center of gravity of the car such that the center of the vehicle was slightly higher than the truck bolsters. Next, two hydraulic cylinders, set in holes between the rail (see Figure 20), were used to push up on the couplers until the car was level. The car was then supported in three points, with forces of known value being applied by the cylinders. Five string-type linear potentiometers, or "string-pots," were attached to the sill and mounted to the ties directly under the car. Figure 21 is a schematic of the test configuration. As pressure was increased in the cylinders, the vertical deflection was measured at each station. As the ends of the car were raised, pneumatic jacks were used on each side rail at the longitudinal center of gravity position to stabilize the car (see Figure 22) by counteracting the rolling tendency of the car supported only by the couplers. For the hopper car, the cylinder pressure was used for the force readout. The caboose was light compared to the hopper, and because of the possible error in force readings below 4,000 pounds, a load cell was installed between the hydraulic cylinder and coupler and force readout directly from the load cell. Figure 23 shows the load cell between the cylinder and a special fixture built to safety transmit the force into the coupler.

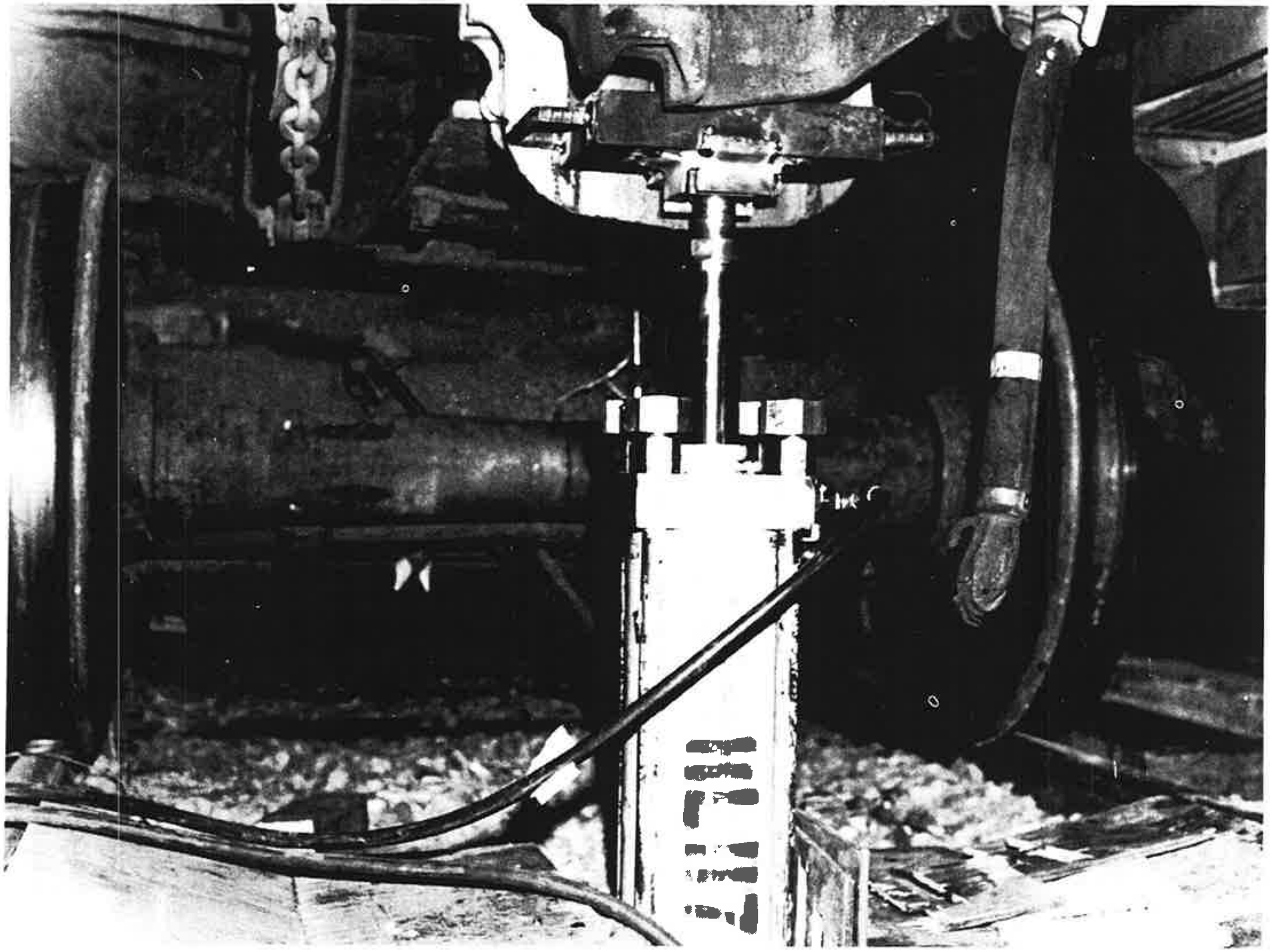


FIGURE 20. INSTALLATION OF HYDRAULIC CYLINDERS.

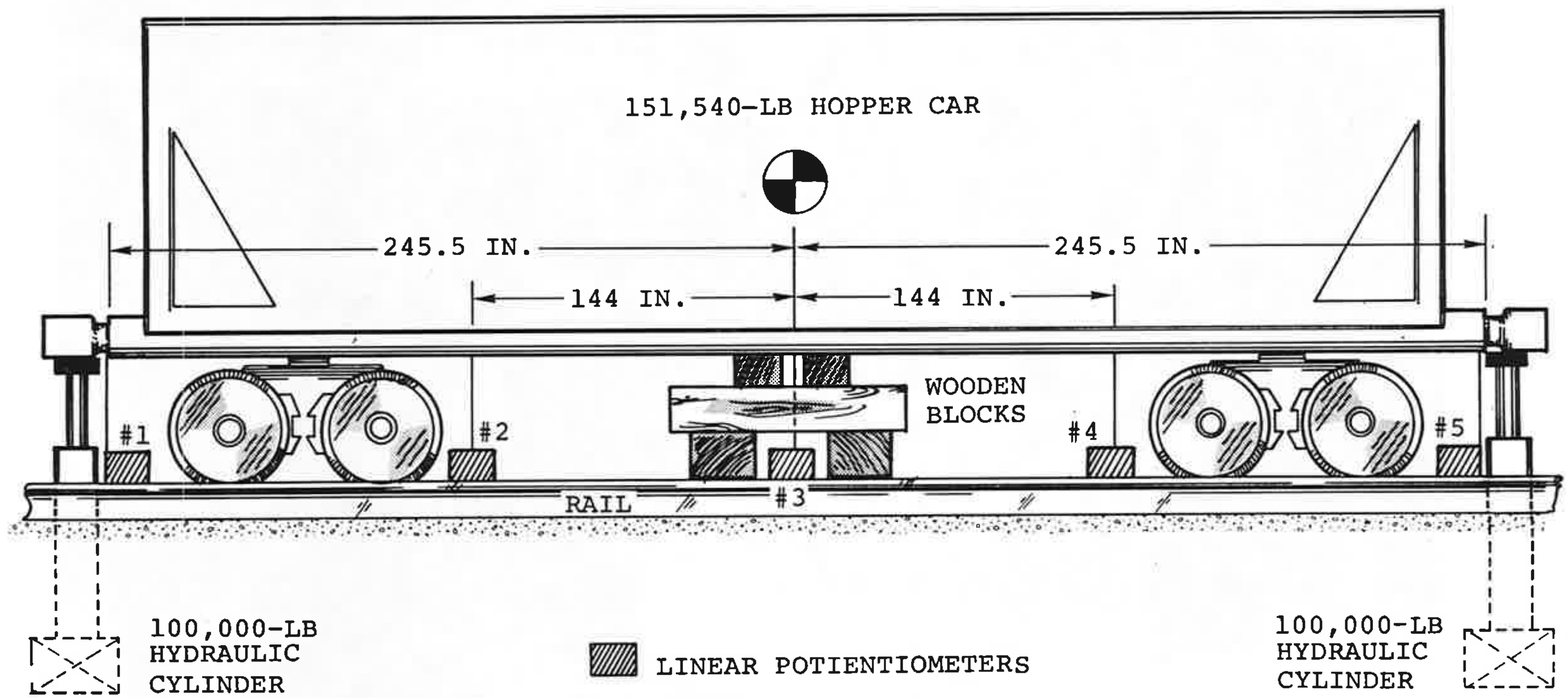


FIGURE 21. TEST SETUP FOR VERTICAL STIFFNESS.

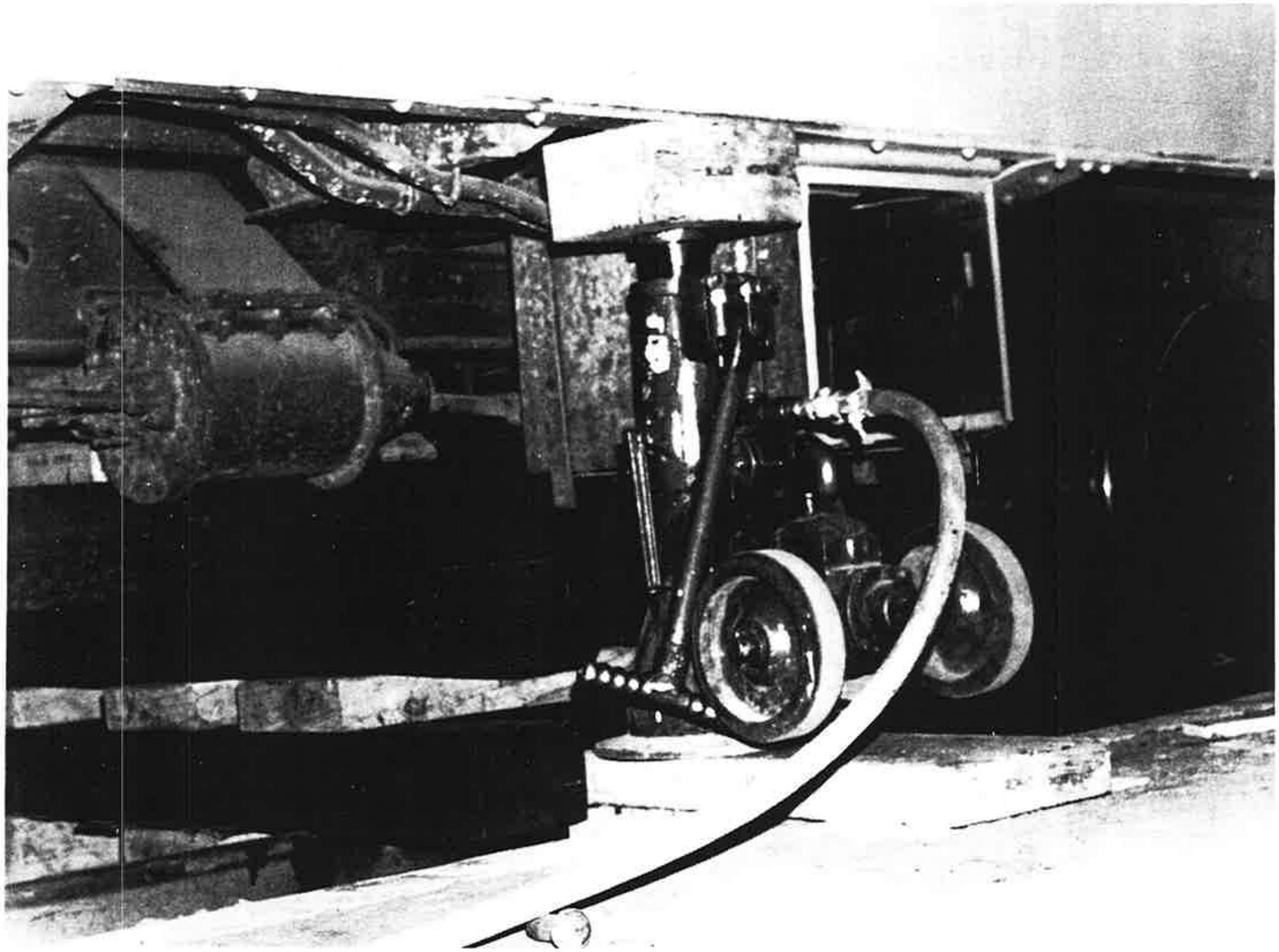


FIGURE 22. AIR JACKS USED TO PREVENT ROLL.

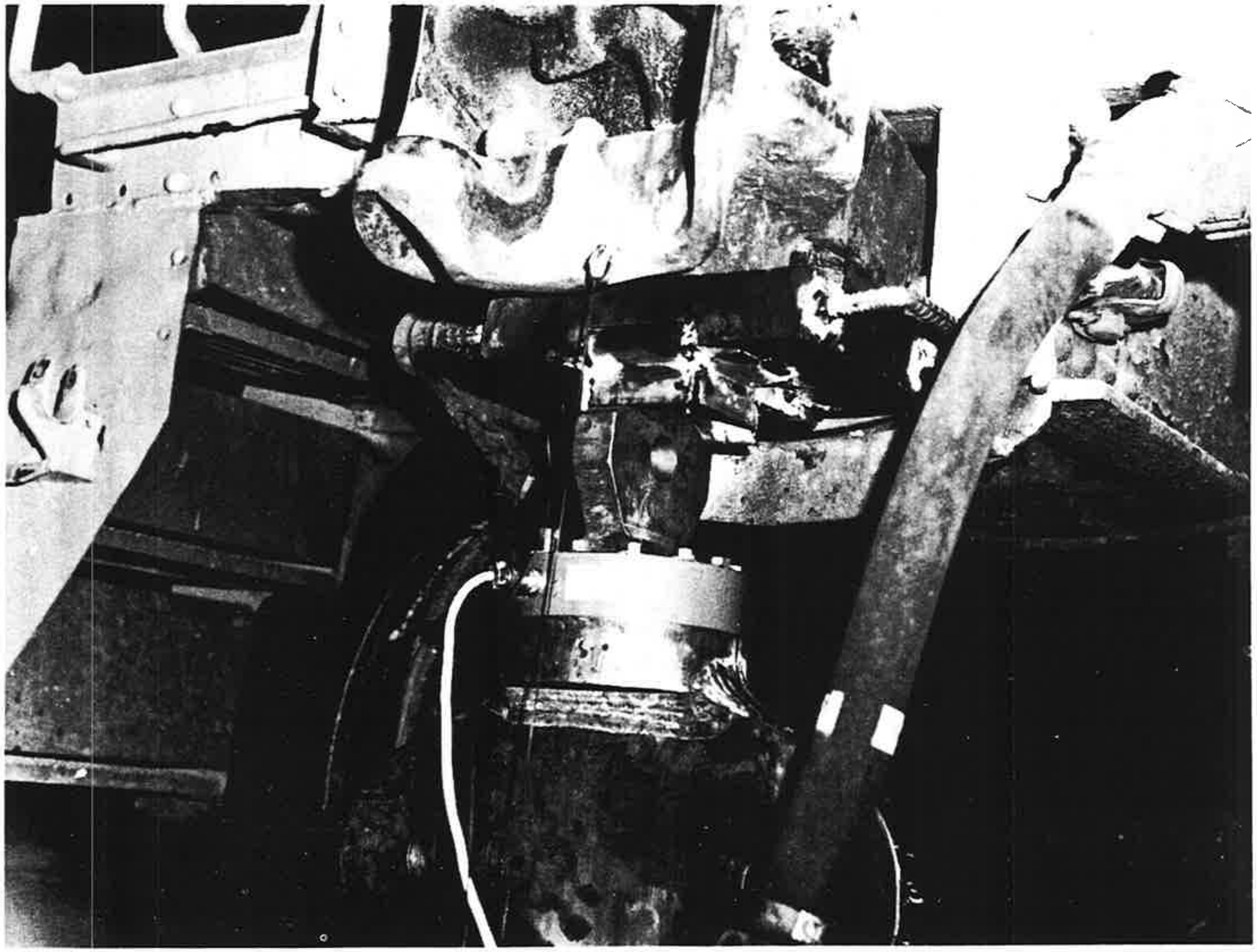


FIGURE 23. LOAD CELL USED ON CABOOSE VERTICAL STIFFNESS TEST.

The resultant output from these tests was a series of force-deflection curves for the five points measured. Since the car did not always lift evenly on each side, the two opposite displacements were averaged. For example, potentiometer Nos. 2 and 4 were averaged to obtain a mean displacement at a distance of 144 inches from the center of the hopper car. Next, the displacement of the center (if any) was subtracted from the average obtained above, giving total relative deflection to the force applied. A schematic of how the car was treated as a large beam is shown in Figure 24. Location A is the average of potentiometer Nos. 1 and 5.

Two assumptions were made to simplify the calculations and make the data generalized so it would be applicable to all similar cars:

1. The test cars were assumed to be symmetrical about the center of gravity in the longitudinal direction.

2. The small distance between the point of application of force on the bottom of the coupler and the attachment point of the linear potentiometers (Nos. 1 and 5) was assumed negligible.

The dimensions of the caboose and hopper linear potentiometers' mounting positions were measured with respect to the center of the vehicles as shown in Figure 25.

The stiffness, defined as vertical force divided by measured vertical deflection, shown in Figures 26 through 29 are:

	Vertical Stiffness - lb/in.	
Car Type	Location A	Location B
Caboose MP918	52,000	41,700
Hopper 536843	86,000	62,500

In both cars, the Location A potentiometer was attached near a sill cross brace or support structure, leaving the sill from Location A to Location B unsupported by any major structure of the car.

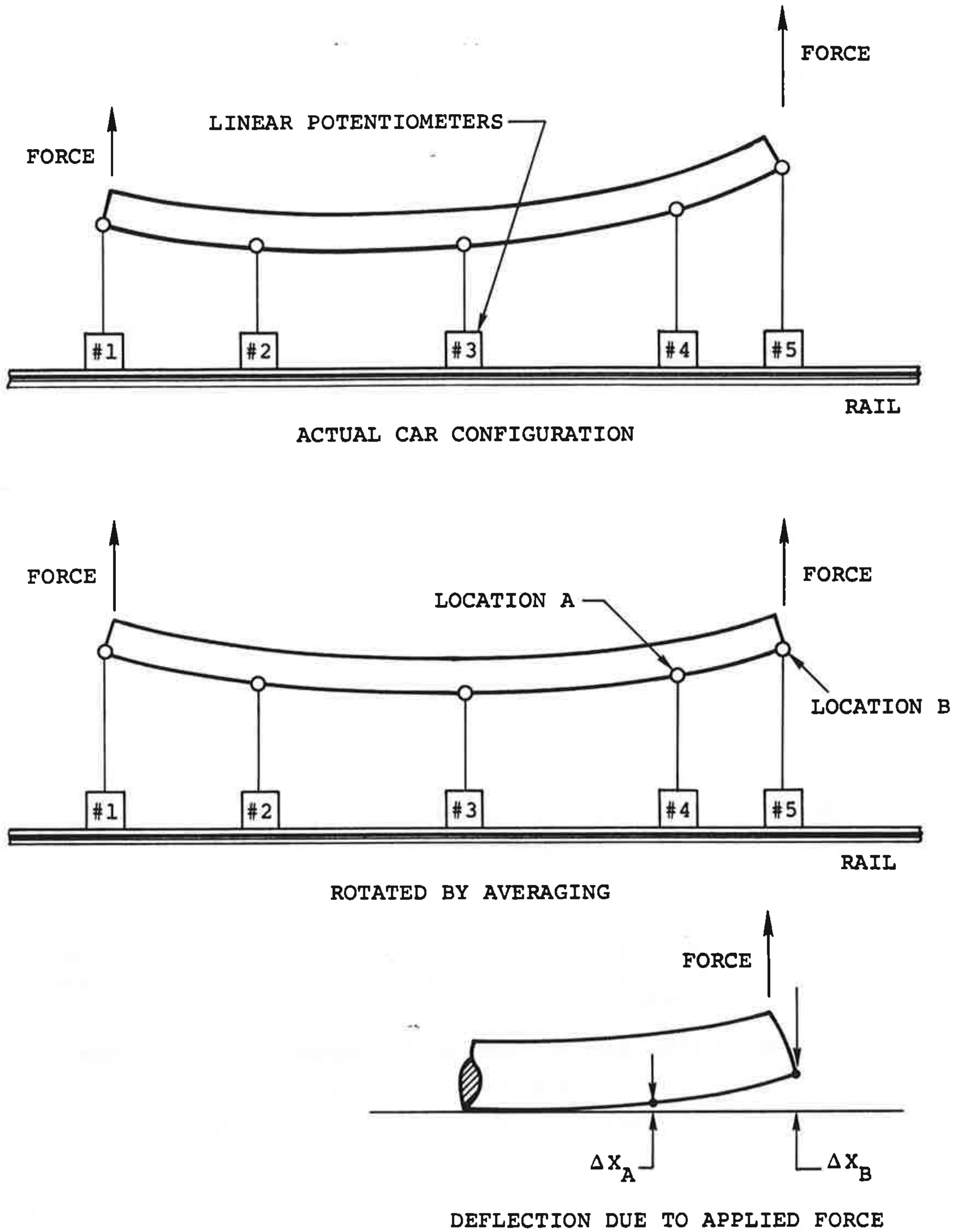


FIGURE 24. SIMPLIFICATION OF VERTICAL STIFFNESS TEST.

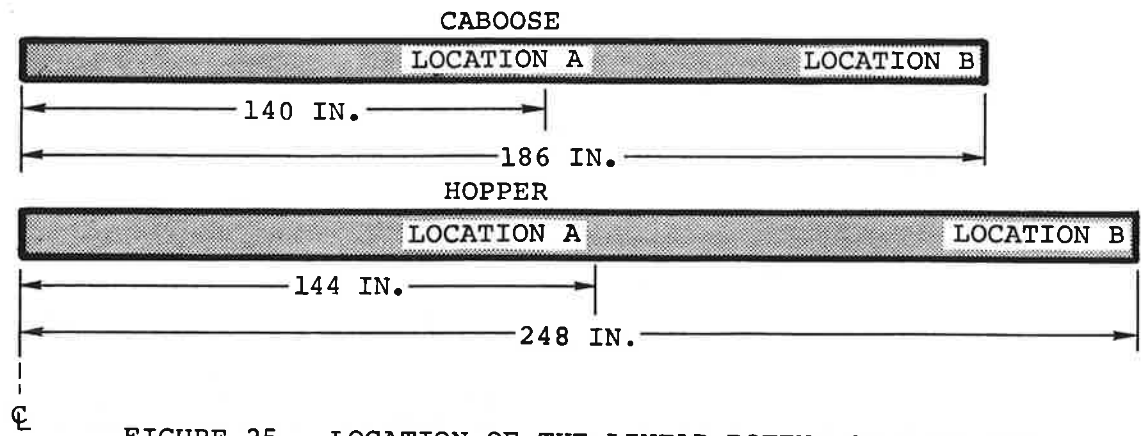


FIGURE 25. LOCATION OF THE LINEAR POTENTIOMETERS FOR VERTICAL STIFFNESS TEST.

Calculating a stiffness of this section of the sill (using the difference of ΔX_B and ΔX_A), the caboose center sill "end section" was 208,000 lb/in. and the hopper was 222,000 lb/in.

3.5 LONGITUDINAL STIFFNESS

3.5.1 Requirements

The computer modeling requires the stiffness of the test vehicles in both the vertical direction and the horizontal direction. The longitudinal stiffness was measured as the deflection rate due to a force applied to the center sill in the horizontal direction. Two assumptions were made for these tests:

1. Bending in the longitudinal direction was negligible.
2. Addition of ballast to the hopper car did not significantly change the stiffness obtained by measuring the empty car.

Both cabooses and one empty hopper car were tested, and curves of force versus deflection were obtained for each car. In addition to obtaining stiffness, the tests were used to calibrate strain gauges on the center sill of the cars. The strain gauges were applied to monitor the forces being transmitted through the cars during impact tests. Also tested for one caboose and one hopper was force deflection characteristics of the draft gear.

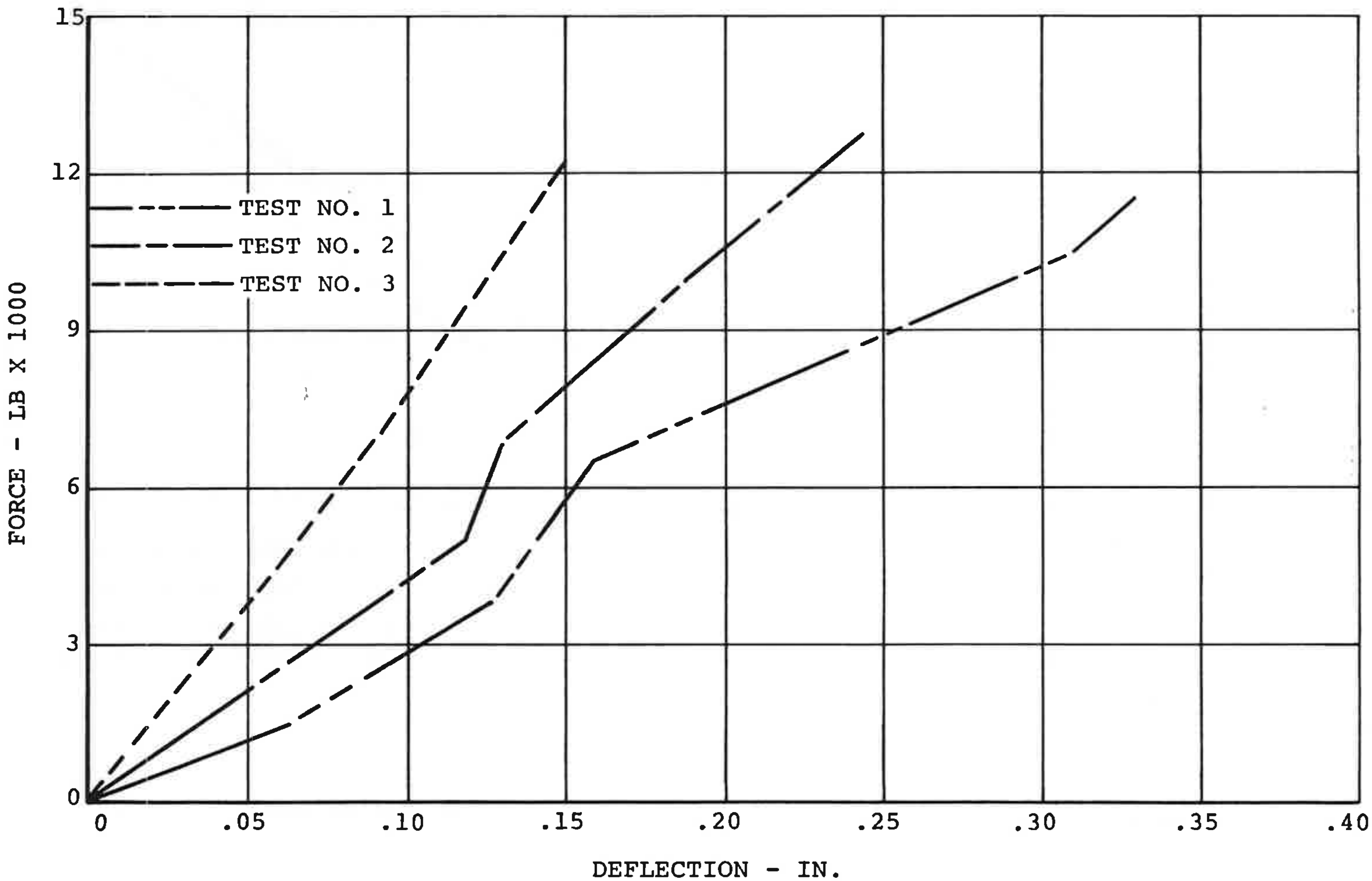


FIGURE 26. CABOOSE VERTICAL STIFFNESS LOCATION B.

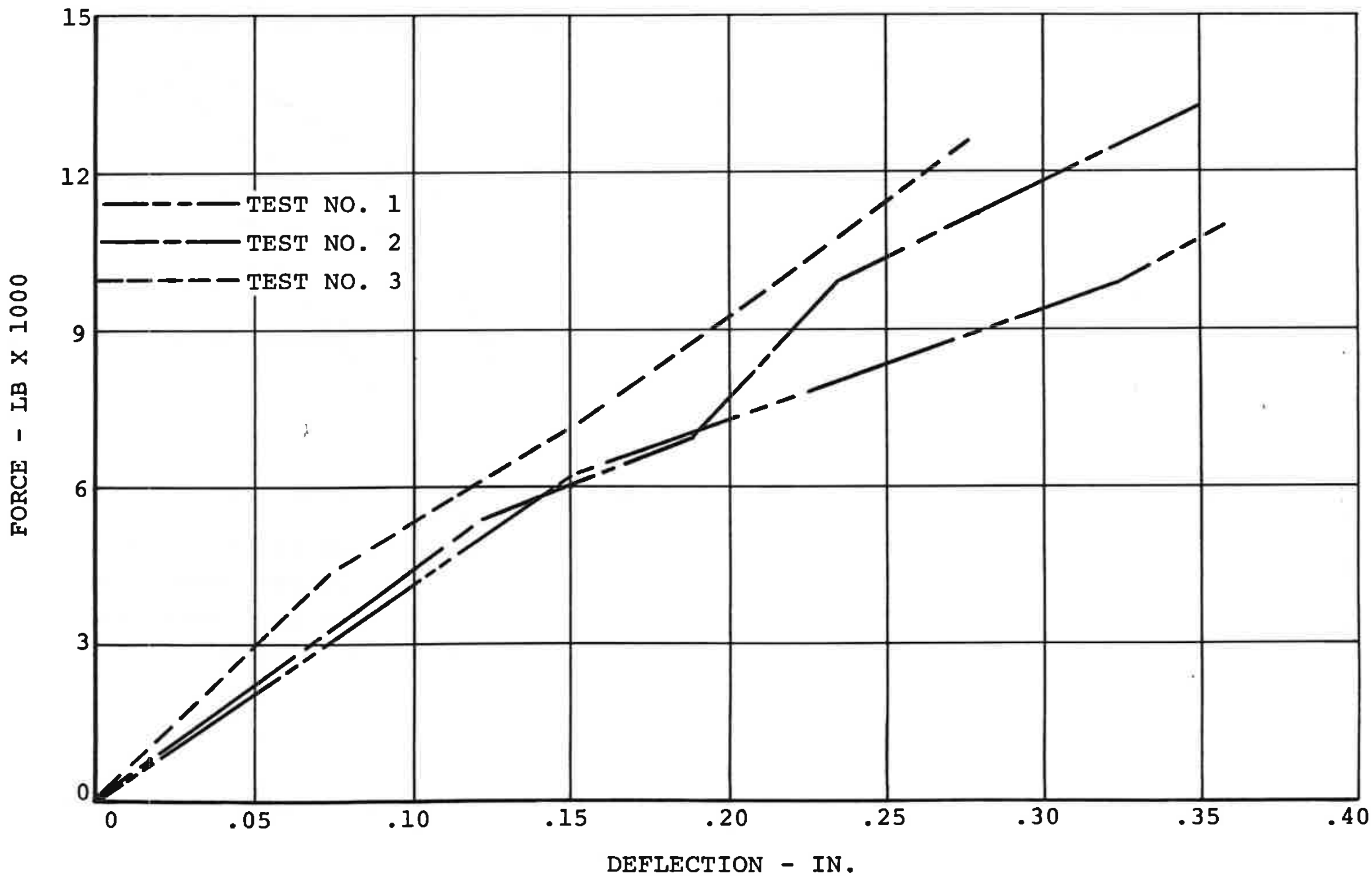


FIGURE 27. CABOOSE VERTICAL STIFFNESS LOCATION A.

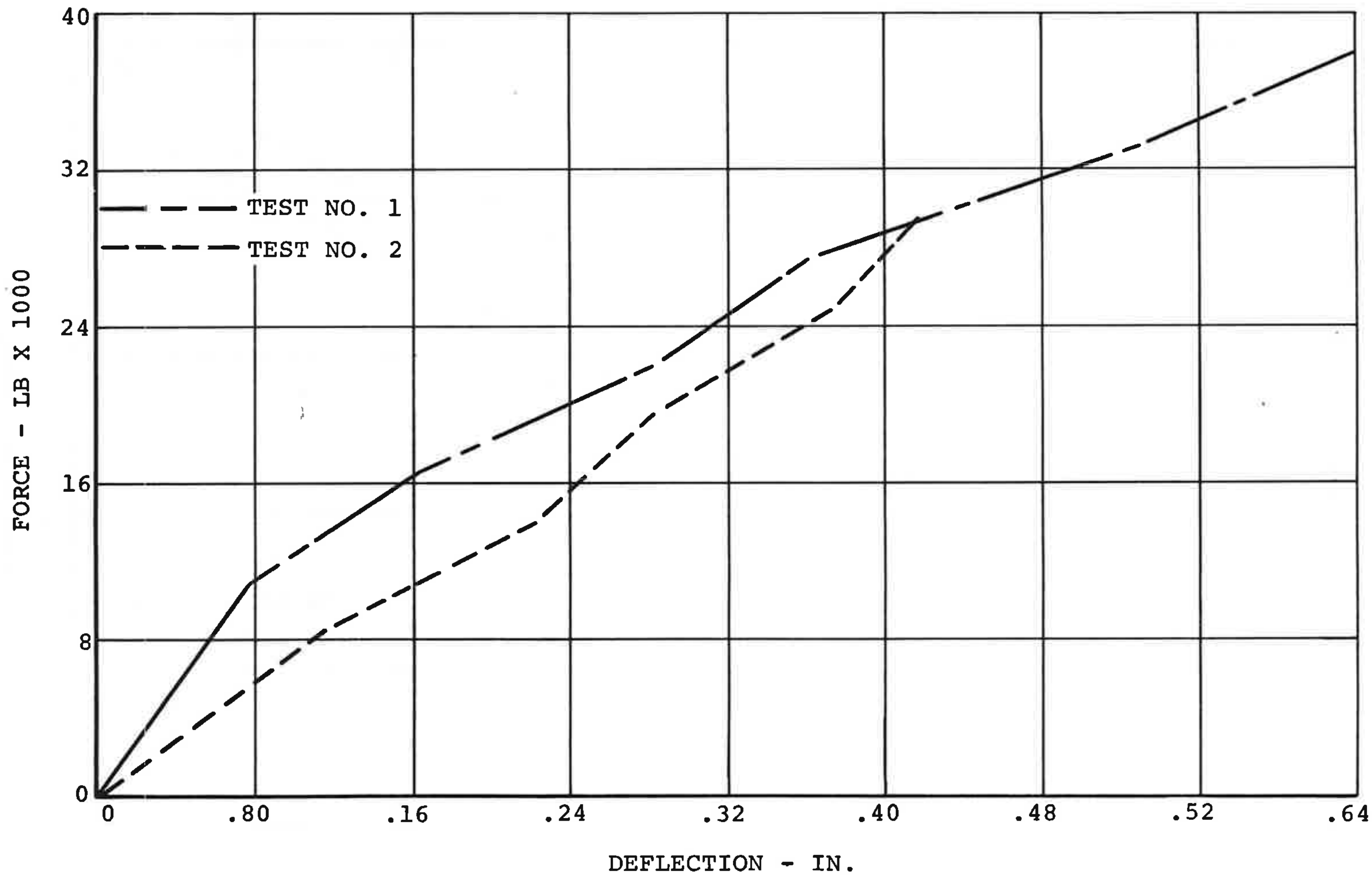


FIGURE 28. VERTICAL STIFFNESS HOPPER 843 LOCATION B.

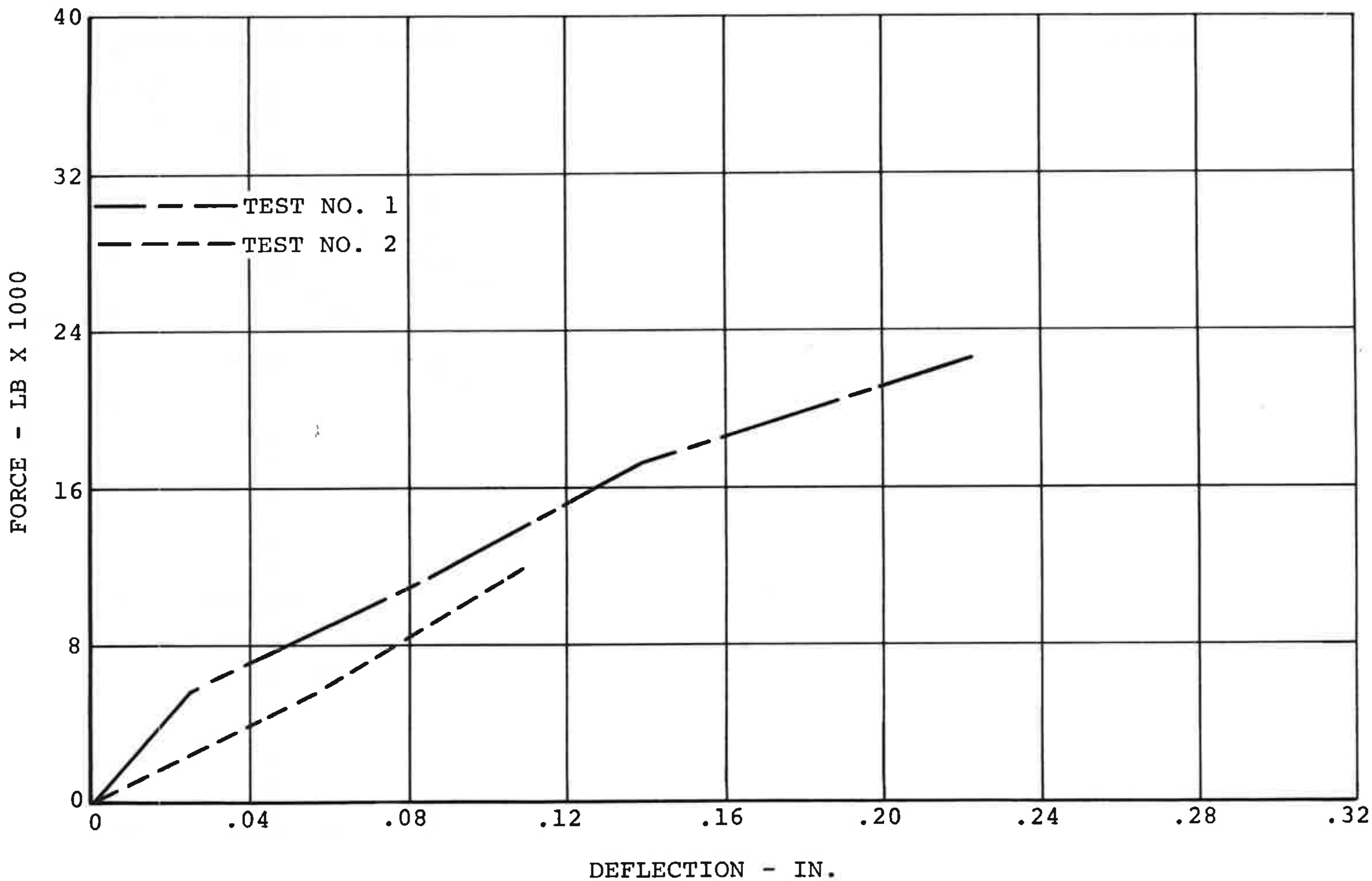


FIGURE 29. VERTICAL STIFFNESS HOPPER LOCATION A.

3.5.2 Test Procedures

The procedure for the longitudinal force deflection test utilized two 18-inch wide-flange I-beams set in a vertical position at each end of a test vehicle. The beams were connected to each other with a 1-1/2-inch wire rope in series with a 100,000-pound-capacity hydraulic cylinder. Figure 30 is a schematic of how the beams, cables, and cylinders were arranged to apply a 200,000-pound force through the center sill of a caboose. One cable connecting the beams was laid under the car and one went through the center of the car. Figure 31 shows the top cable through the hopper 538021. On the caboose, the top cable went through the walkway with both doors open. A 6-inch diameter thick-wall pipe was welded to the center of the beam and with the coupler removed, used to push against the draft gear to obtain the draft gear spring rate (see Figure 32). After the spring rate of the draft gear was obtained, the pipe was cut shorter and spacers were put around the pipe to allow the beam to push directly against the buffer casting. Strain gauges applied to the sill did not give any significant readings when force was being applied through the draft gear since the gauges were applied between the draft gear attachment point and the end of the center sill. Figure 33 is a photograph of the strain gauge on the center sill of the hopper.

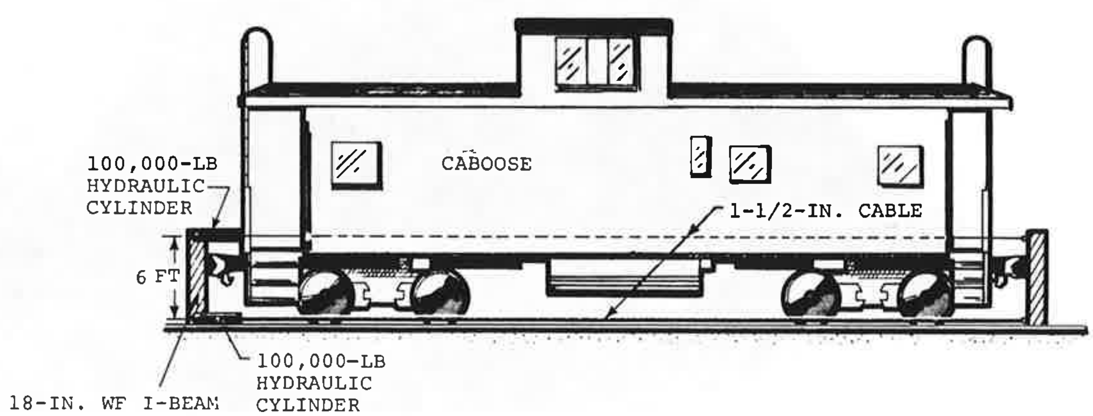


FIGURE 30. CONFIGURATION FOR LONGITUDINAL STIFFNESS TEST.

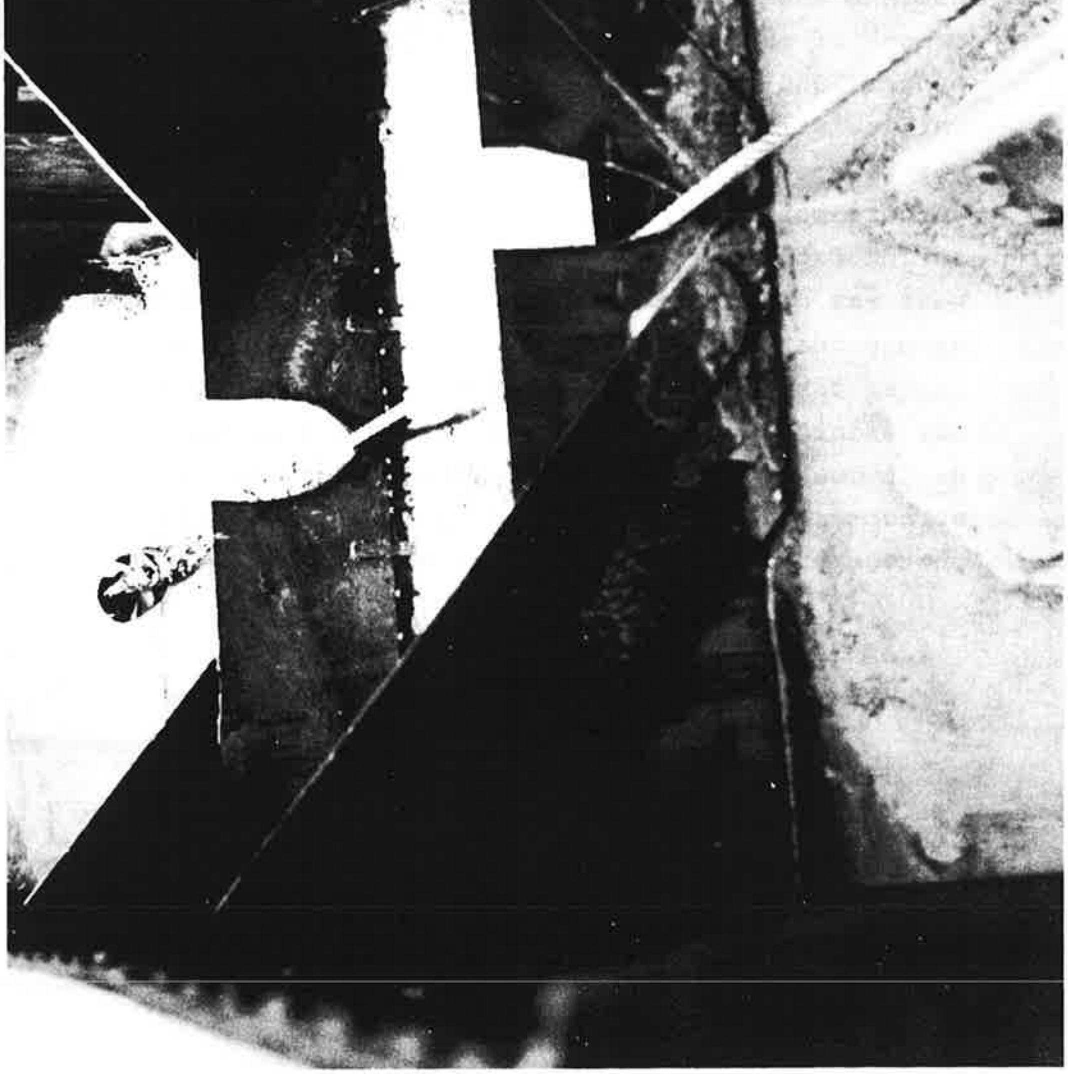


FIGURE 31. CABLE THROUGH CENTER OF HOPPER CAR.

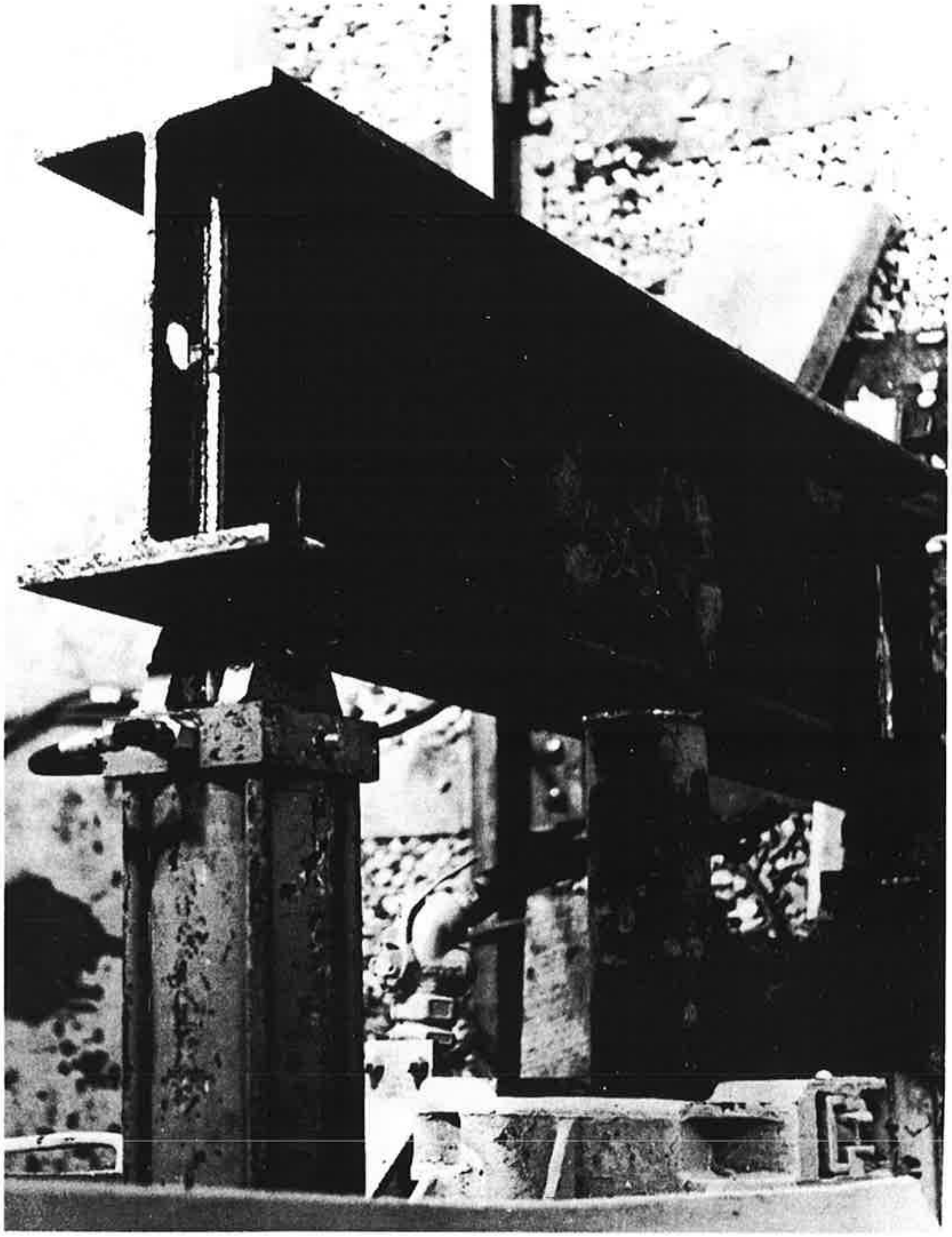


FIGURE 32. I-BEAM USED FOR LONGITUDINAL STIFFNESS TEST.

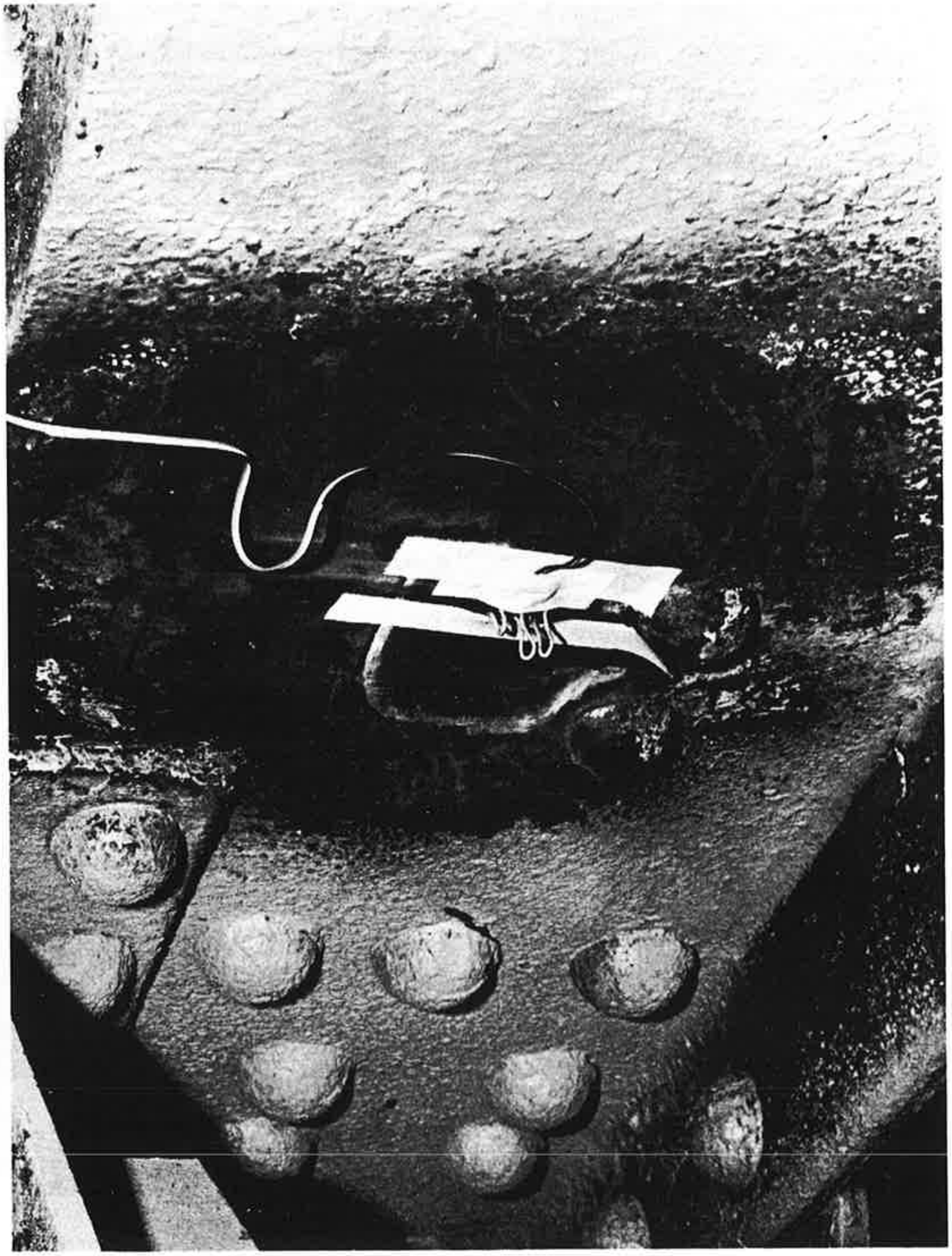


FIGURE 33. STRAIN GAUGE ON CENTER SILL OF HOPPER.

The deflection of the draft gear was measured by a linear potentiometer mounted horizontally and attached to a 2-1/4-inch-thick plate (see Figure 34). A potentiometer was positioned at each end, and by taking the difference between the two displacements, any translational movement of the car was eliminated. The draft gears were friction type and once compressed, they do not return immediately. Figure 35 is an example of the force-deflection characteristic of a hopper-type coupler manufactured by Miner Enterprises, Inc. The upper and lower curves of the characteristics represent the loading and unloading portions of force deflection, respectively. Since the draft gear utilizes a wedging friction for the dissipation of energy the loading force is higher than the unloading. When the load is slowly removed, the gear does not necessarily return back to its initial position. When the load is reapplied with the wedges in a partially deflected position, the load increases without significant deflection until the top curve is reached. When the wedges are jarred loose, the draft gear returns at a low load level. This property of the draft gear makes a static measurement of its force-deflection curve difficult since when the load is applied slowly, the wedges tend to hold the draft gear compressed even after the load is released. Successive tests then measure a different part of the draft gear curve since there is an initial compression of the draft gear.

For the data presented in Figures 36 and 37 of the draft gear tests, Test 1 gives the initial stiffness of the draft gear.

The second test involved loading the compressed draft gear and measuring the stiffer portion of the curve. Tests 3 and 4 represent testing the partially returned draft gear. For the caboose, there are two basic slopes shown which correspond to the two areas of the manufacturer's curve, shown in Figure 35. For the hopper car, the load applied did not get high enough to get on the steeper portion of the curve.

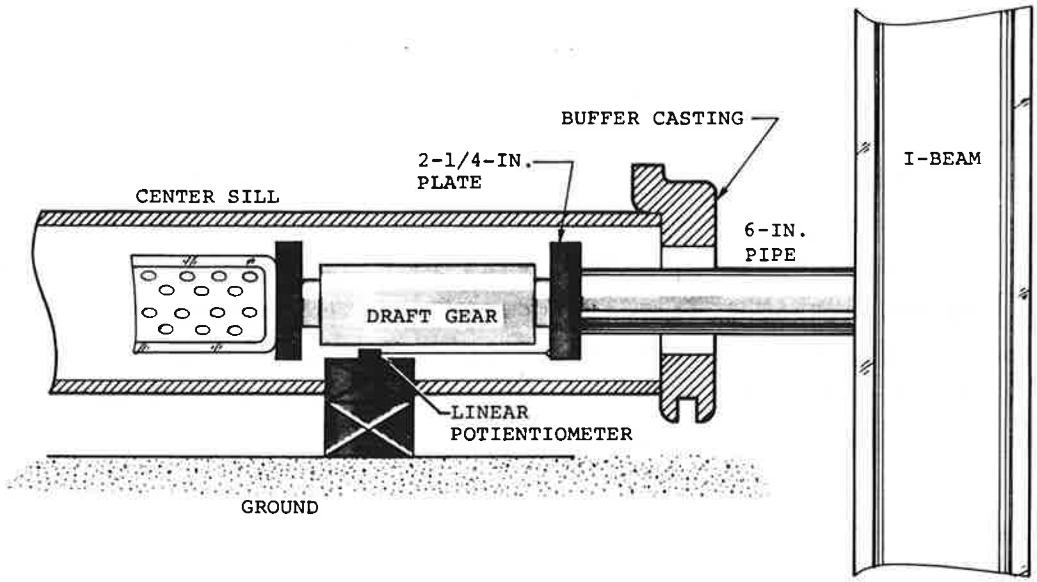


FIGURE 34. DRAFT GEAR DEFLECTION MEASUREMENT.

The stiffness was calculated by dividing the change in force by the corresponding measured deflection. Using Test 1 as the initial curve, the draft gear stiffnesses are:

Vehicle	Draft Gear Stiffness
Caboose MP918	255,500 lb/in.
Hopper 538021	493,800 lb/in.

The results of the force-versus-deflection tests performed on the caboose and hopper draft gear are shown in Figures 36 and 37.

Figures 38 and 39 are graphs of force versus deflection for the underframe of the caboose and hopper cars. The deflections were measured at the buffer casting and reflect the total change in length of the car as if the force was being input through the buffer casting. The force was transmitted from the buffer casting into the center sill through eleven rivets on each end. Thus, the stiffness listed on the following page includes the buffer casting and rivets.

CHARACTERISTIC COMPRESSION AND RELEASE PERFORMANCE
MINER FRICTION DRAFT GEAR, CLASS A-22-XL,
AS TESTED UNDER 27,000-POUND HAMMER.

TOTAL CAPACITY, FOOT-POUNDS,	22,500
ABSORBED BY FRICTION, FOOT-POUNDS,	19,800
PERCENT OF CAPACITY,	88
RECOIL, FOOT-POUNDS,	2,700
PERCENT OF CAPACITY,	12
MAXIMUM SILL PRESSURE, POUNDS,	300,000
RATIO OF MAXIMUM SILL PRESSURE TO CAPACITY, POUNDS PER FOOT-POUND.	14

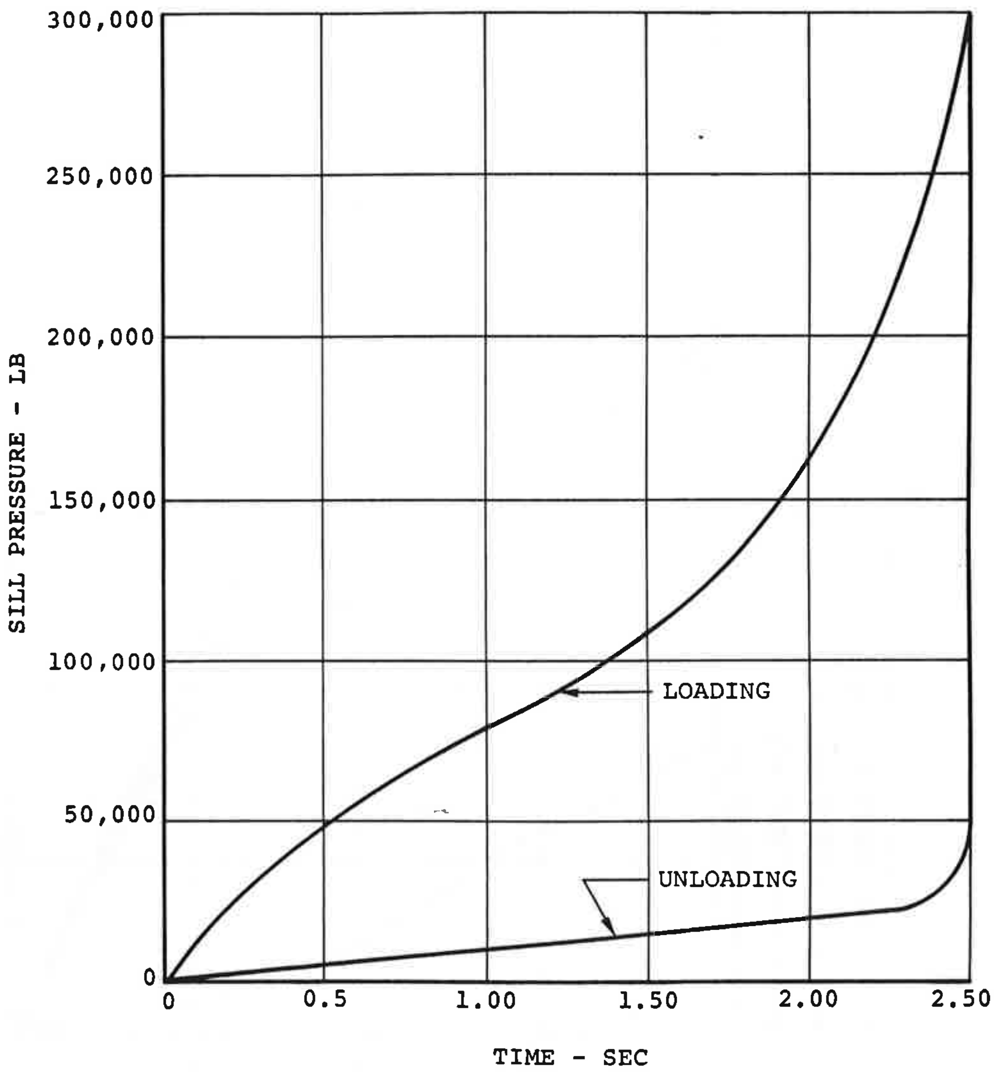


FIGURE 35. EXAMPLE FORCE-DEFLECTION CURVE OF A DRAFT GEAR.

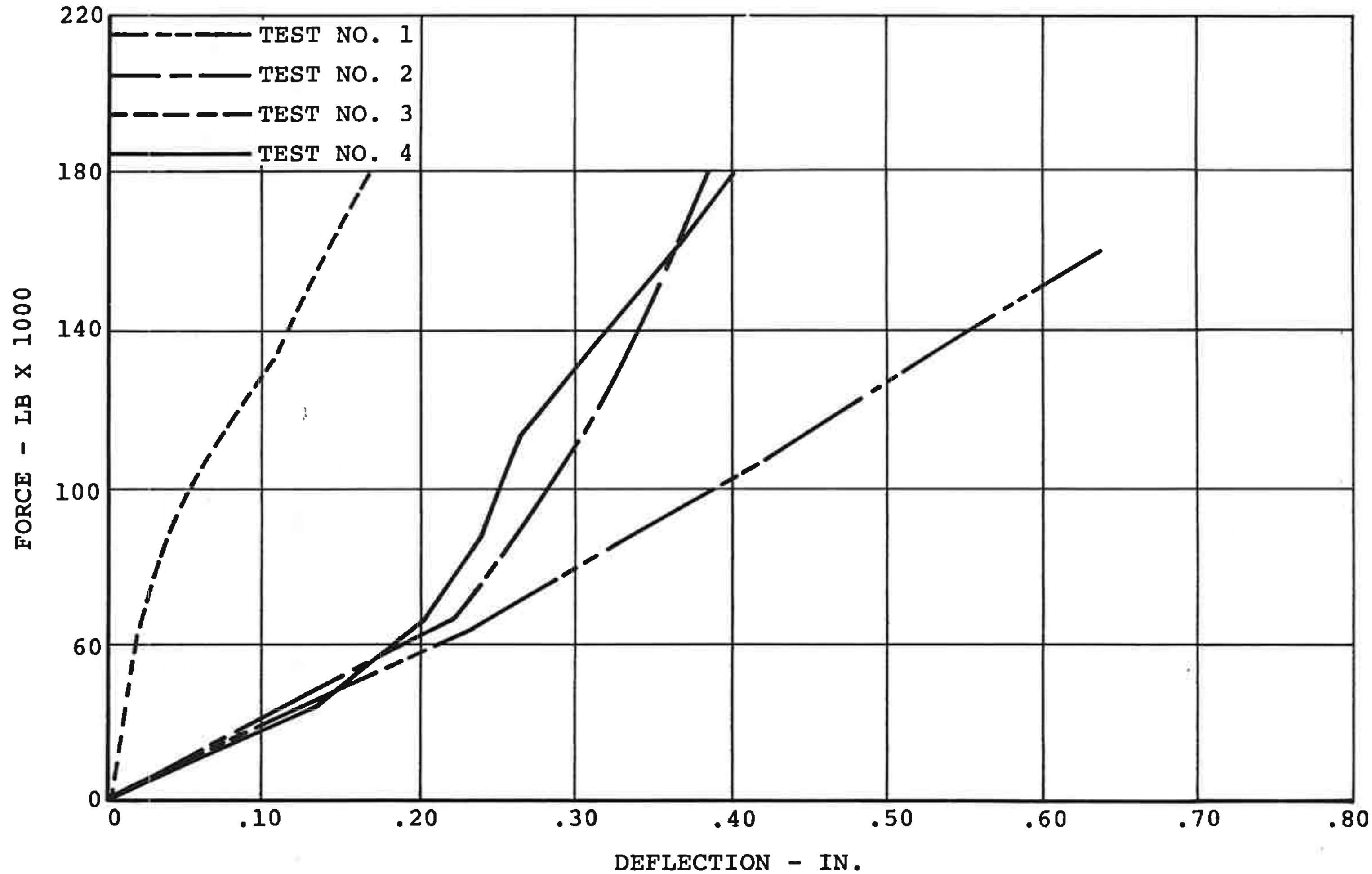


FIGURE 36. CABOOSE DRAFT GEAR FORCE DEFLECTION.

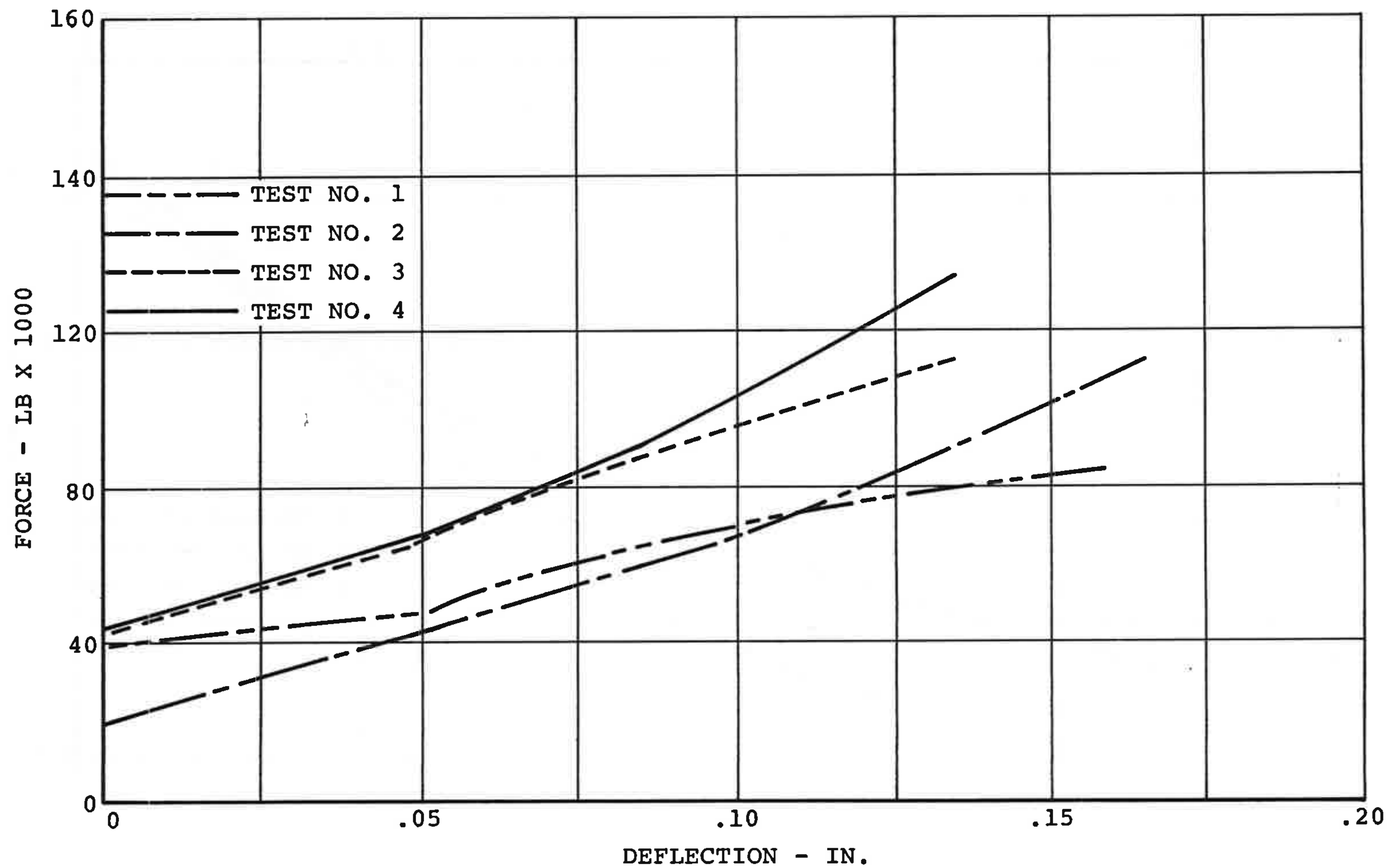


FIGURE 37. HOPPER DRAFT GEAR FORCE DEFLECTION.

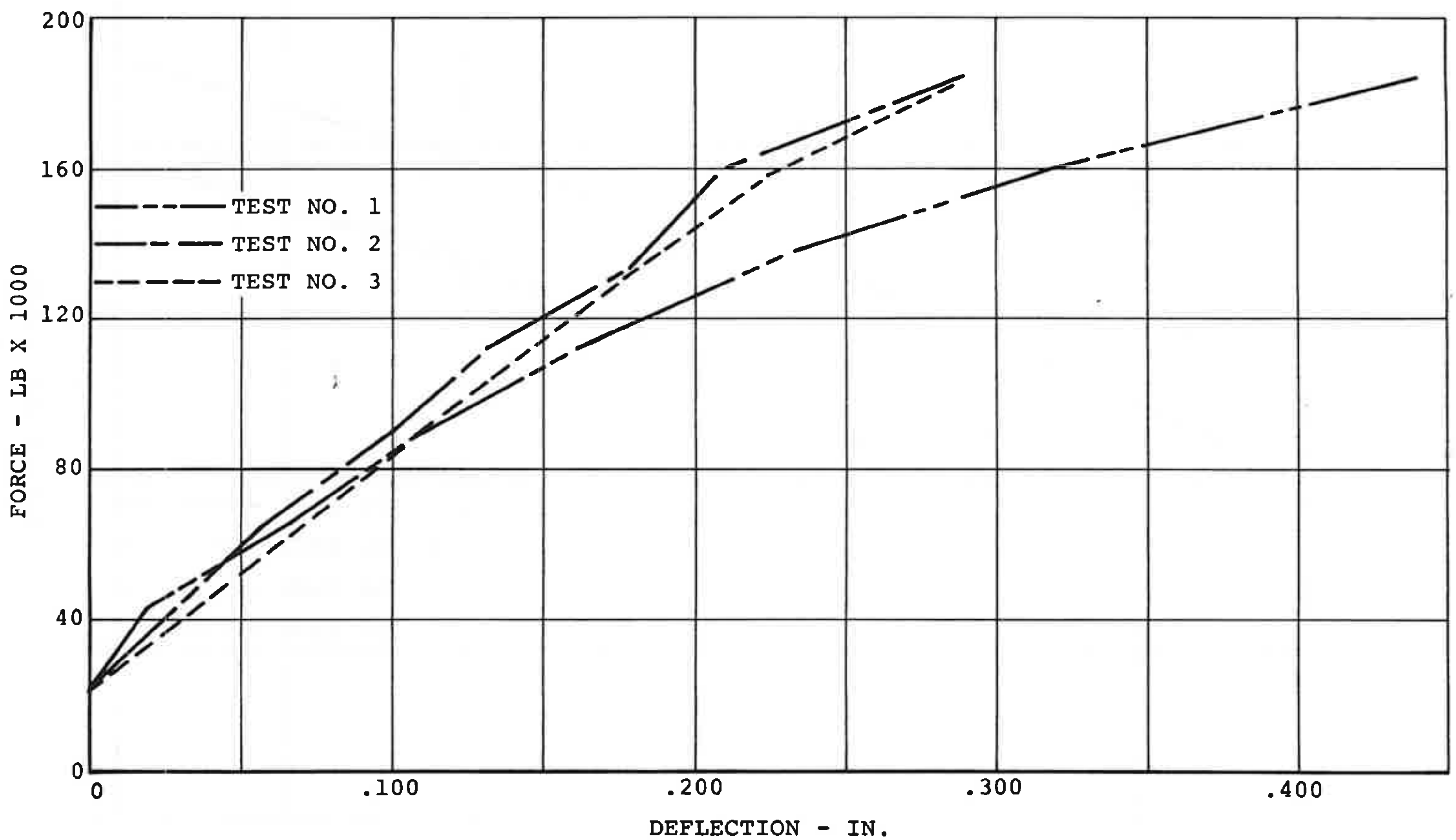


FIGURE 38. CABOOSE UNDERFRAME FORCE DEFLECTION.

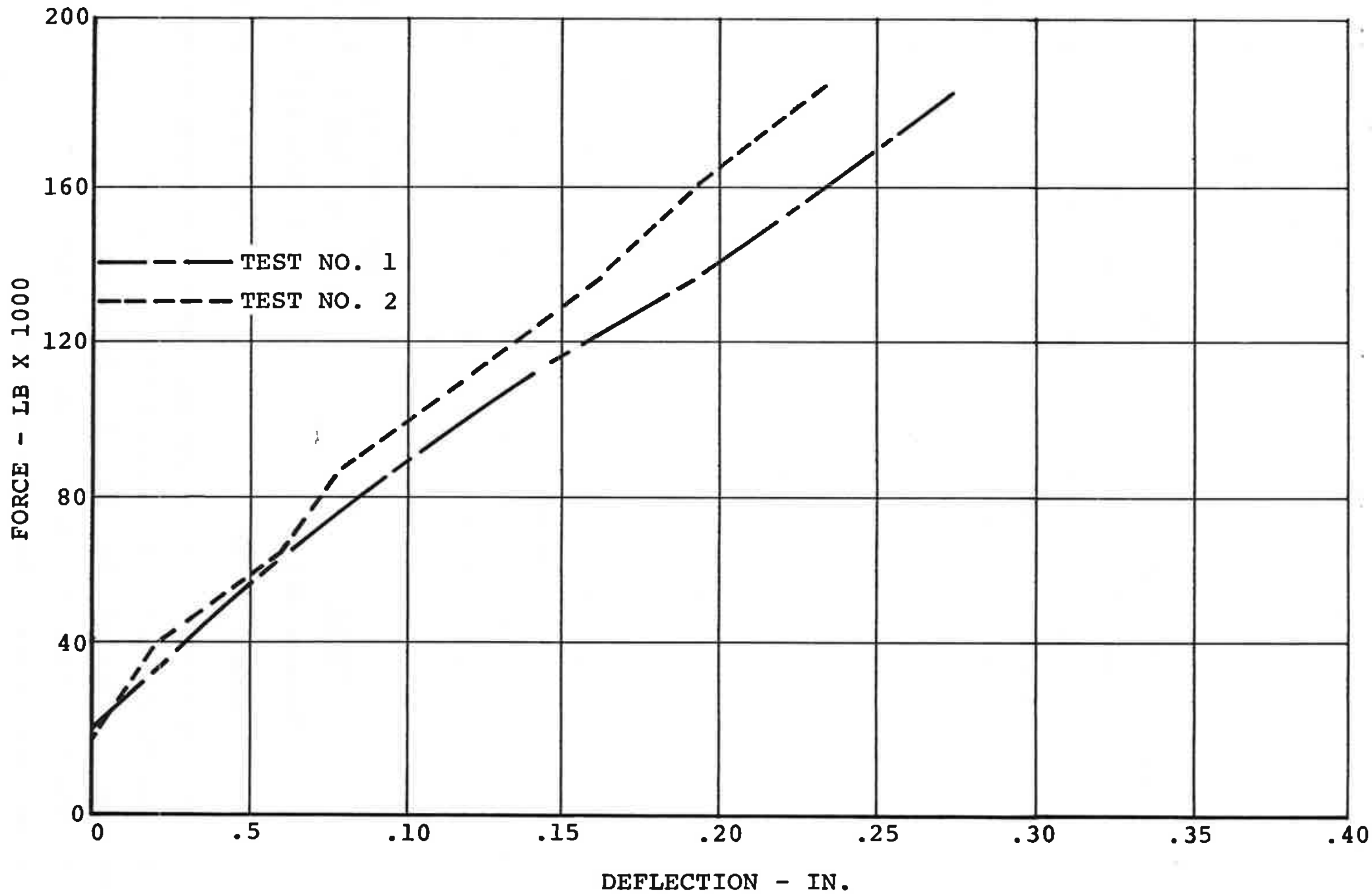


FIGURE 39. HOPPER UNDERFRAME FORCE DEFLECTION.

Vehicle	Longitudinal Stiffness
Caboose MP918	612,394 lb/in.
Hopper 538021	598,151 lb/in.

These values represent the average slope of a line drawn through the graphs. Note that the caboose is slightly stiffer due to the floor grates and additional support members tied into the buffer casting. The average diameter of each rivet was 7/8 inch, producing a total cross sectional area of 6.61 in.^{2}. This would allow a maximum force which would plastically deform the rivets at about 1/4 million pounds. The rivets would deform until the buffer casting would be pushing directly against the end of the sill. Once this occurred, the spring rate would be much higher. A second measurement was performed on the caboose to measure this deflection of the center sill only. A special fixture (see Figure 40) was welded to the sill at each end (20 inches from the buffer casting) and connected by a long steel rod in series with a dial indicator. The force was applied through the buffer casting as in previous tests, but only the change in length of the center sill was measured. The test was performed on both MP912 and MP918 with the results shown graphically in Figures 41 and 42. The tests performed on each car were repeatable and linear within 10 percent. The slope of a straight line drawn through the curves is as follows:

Vehicle	Longitudinal Stiffness of Center Sill
Caboose MP912	1.72×10^6 lb/in.
Caboose MP918	2.61×10^6 lb/in.

No good explanation was found for the differences between the caboose except those that are related to their age. Varying degrees of rust of rigidity of connectors between the center sill and the remainder of the caboose structure must account for the variations in their measured stiffness.

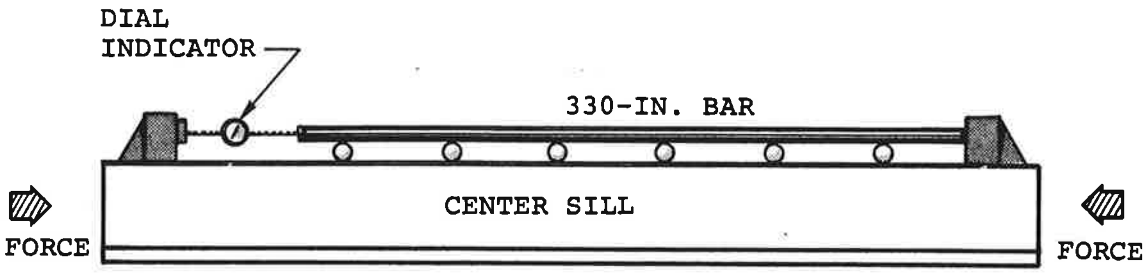


FIGURE 40. CENTER SILL DEFLECTION MEASURING DEVICE.

During the underframe spring constant tests, center sill strain gauges were monitored and strain calculated using the published gauge factor for the strain gauges. The resulting curves of force versus strain are shown in Figures 43, 44, and 45. A summary of the slopes is :

Vehicle	Center Sill Force-Strain Rate
Caboose MP912	565 lb/μin./in.
Caboose MP918	611 lb/μin./in.
Hopper 538021	618 lb/μin./in.

Force-strain values can also be calculated from the longitudinal deflection of the center sill. Using $\varepsilon = {}^{\Delta L}/L$ where ΔL is the actual change in length of the center sill and L = 332 inches (distance between mount points), the average strain for any given force can be obtained. A comparison of calculated average force-strain rate versus measured force-strain rate for the two cabooses is:

Vehicle	Calculated Force-Strain Rate	Measured Force-Strain Rate
Caboose MP912	643 lb/μin./in.	565 lb/μin./in.
Caboose MP918	972 lb/μin./in.	611 lb/μin./in.

The calculated average force-strain rate assumes a uniform strain throughout the total length of the center sill. The stiffness near the center of the car is probably higher than near the ends since the sill is stiffened in the center section by several cross braces which tie into the caboose.

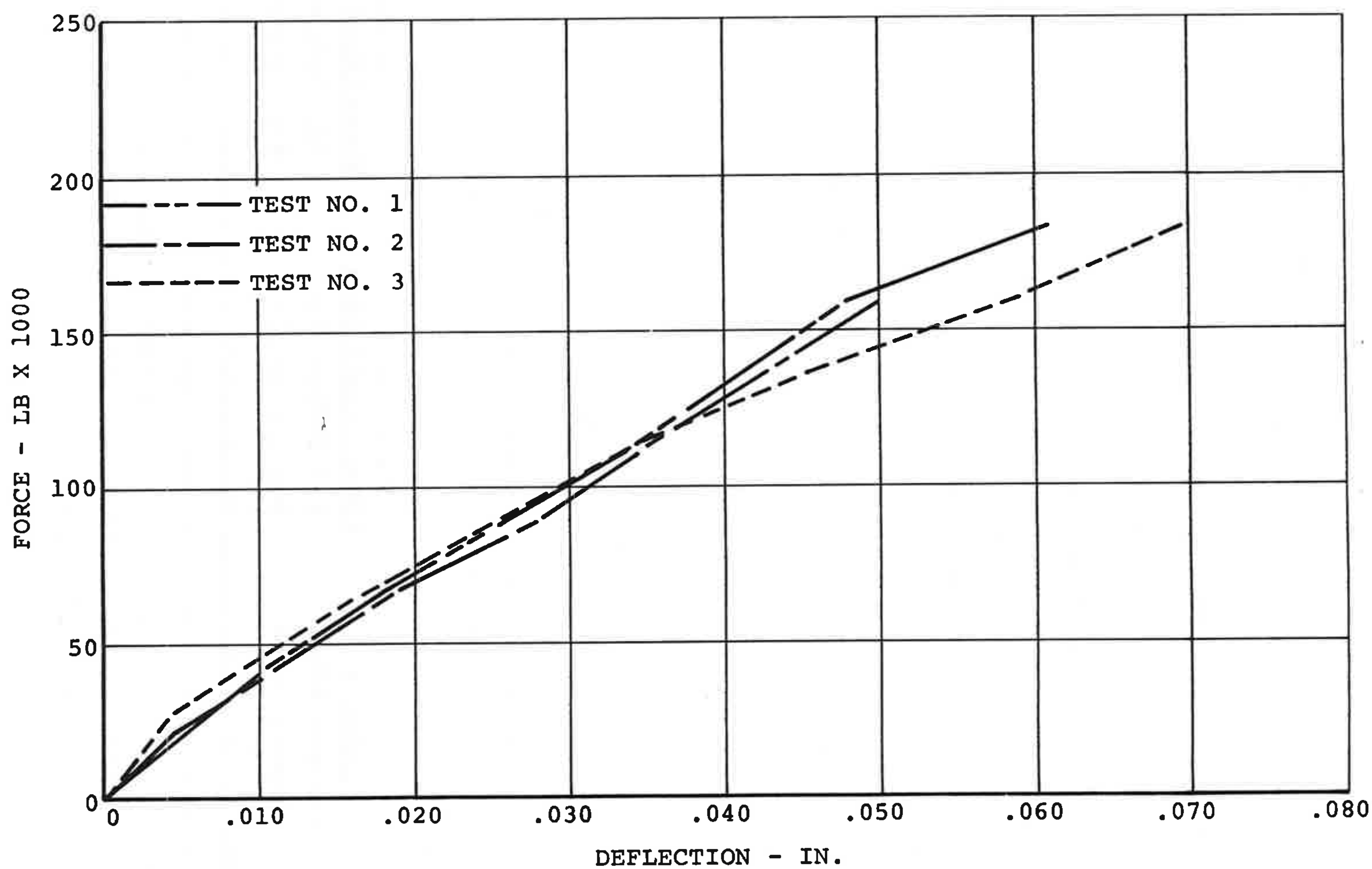


FIGURE 41. CABOOSE MP918 CENTER SILL LONGITUDINAL FORCE DEFLECTION.

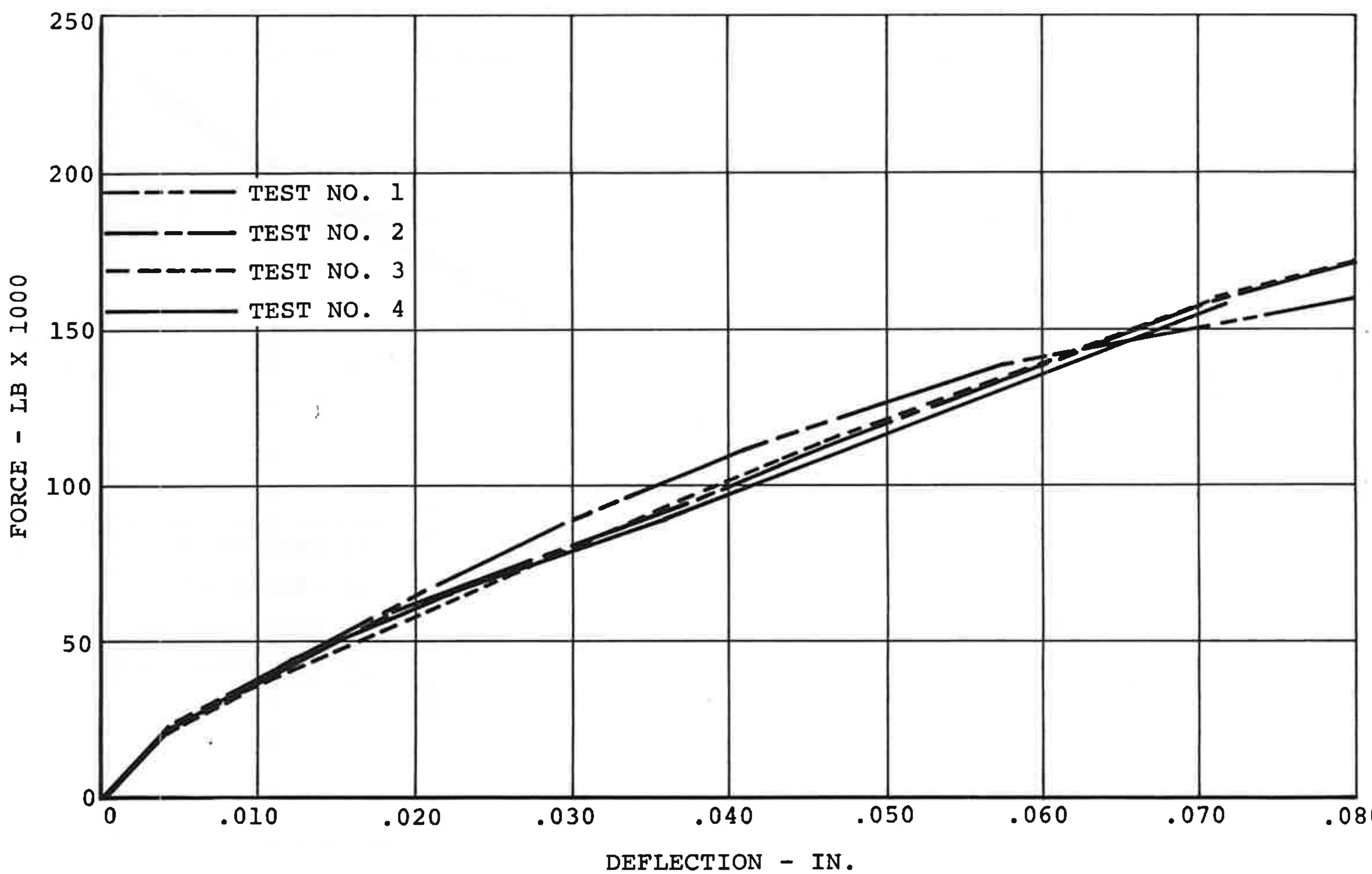


FIGURE 42. CABOOSE MP912 CENTER SILL LONGITUDINAL FORCE DEFLECTION.

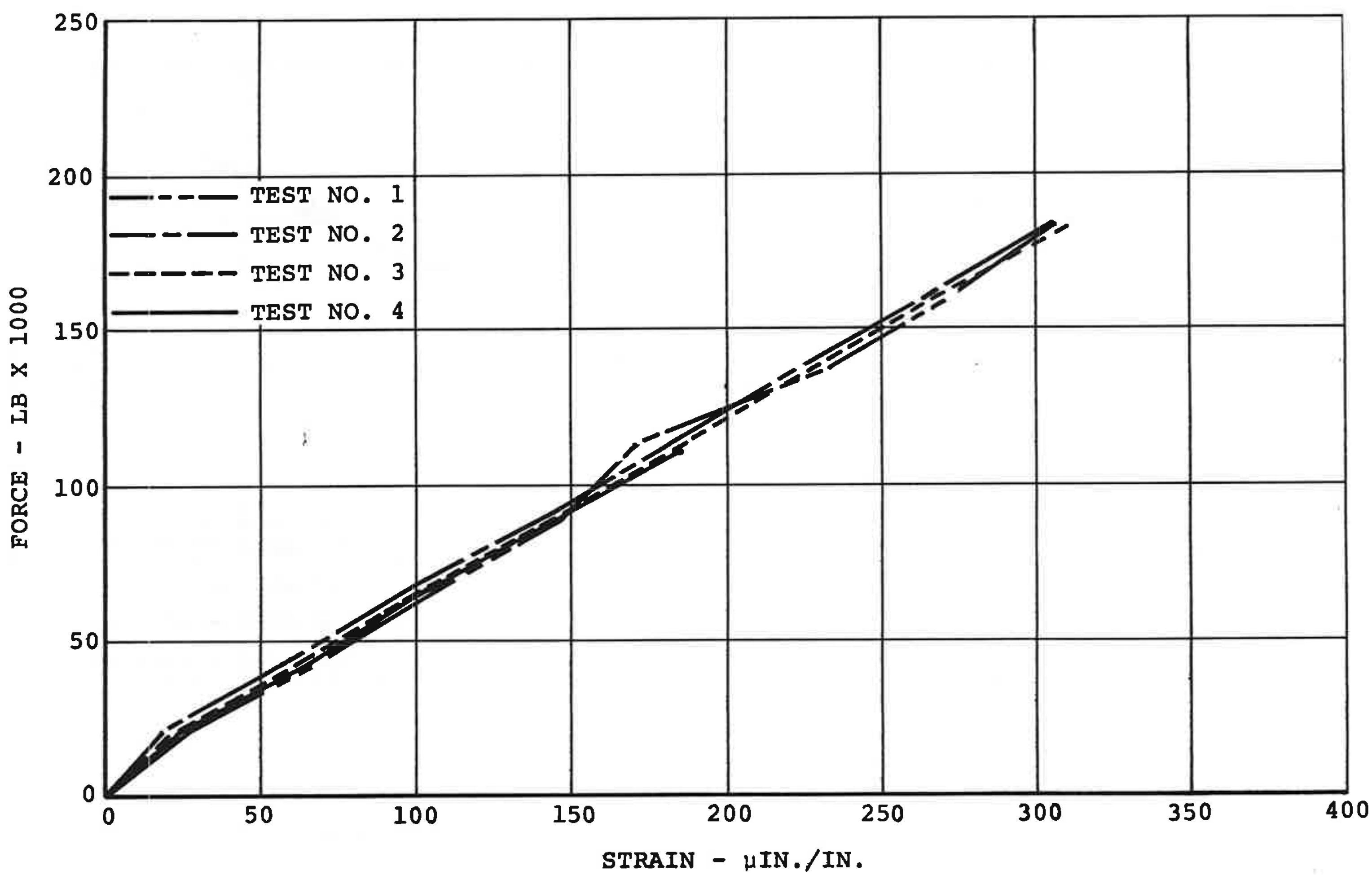


FIGURE 43. CABOOSE MP912 CENTER SILL STRAIN.

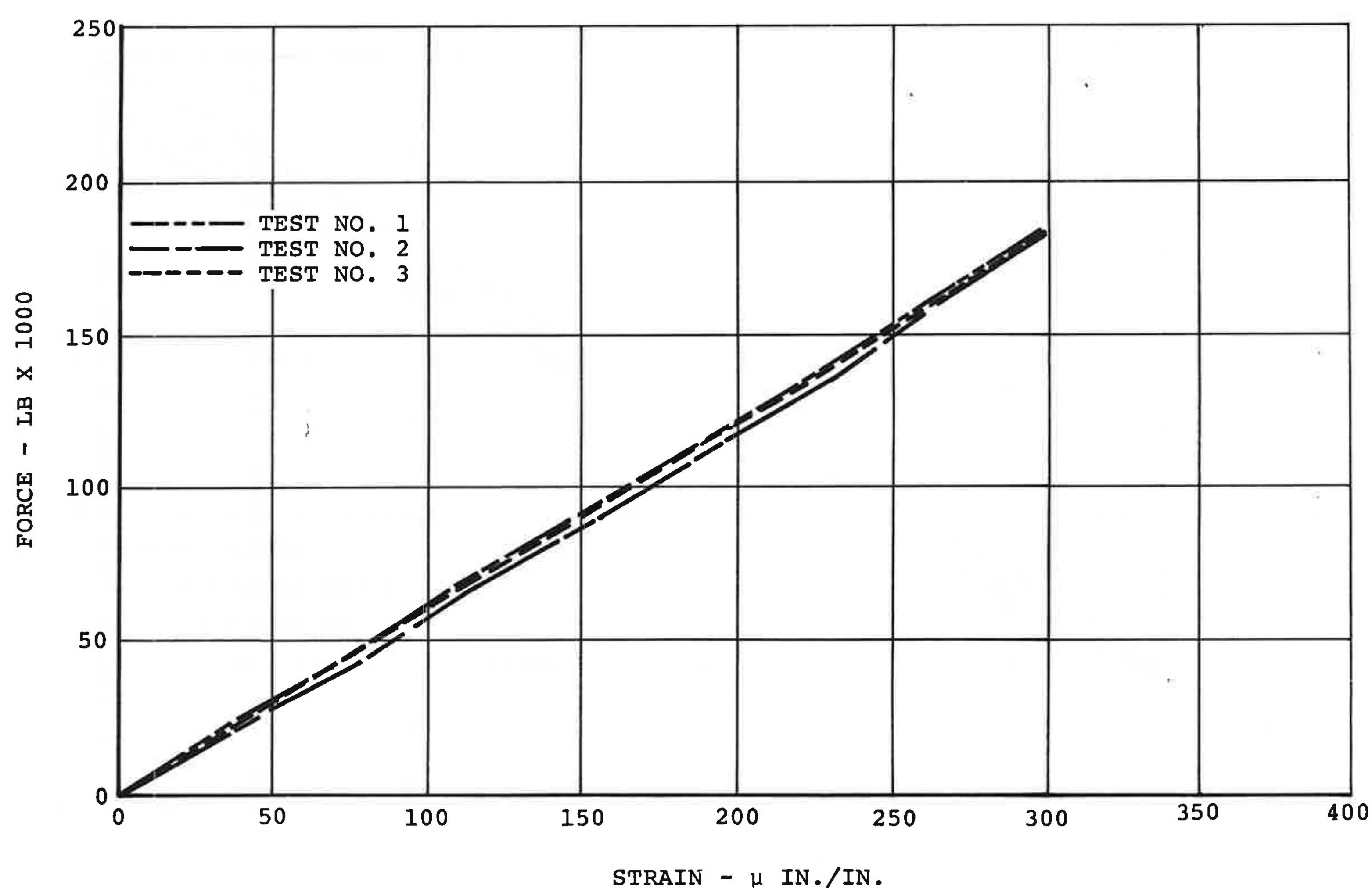


FIGURE 44. CABOOSE MP918 CENTER SILL STRAIN.

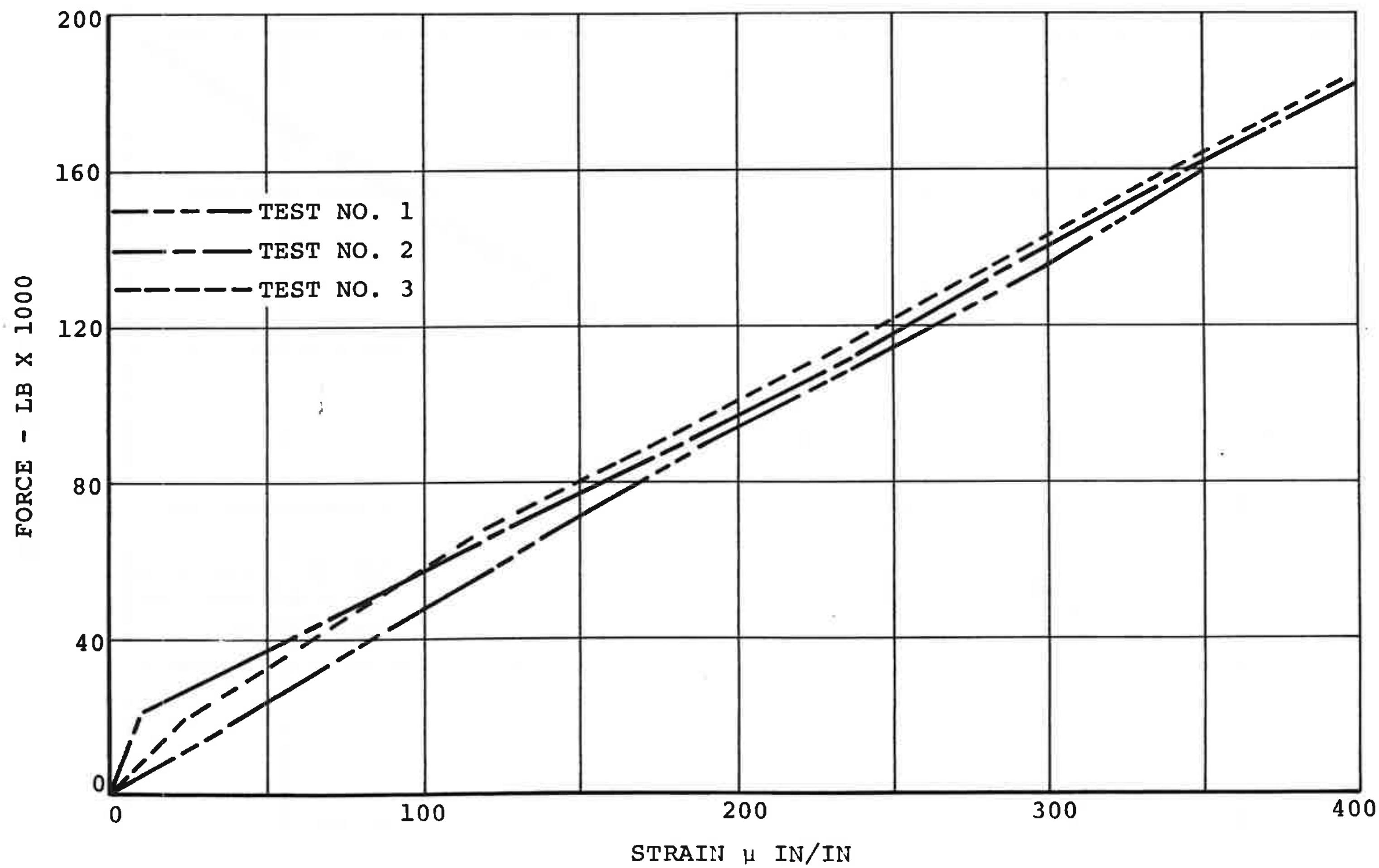


FIGURE 45. HOPPER CENTER SILL STRAIN.

3.5.3 Longitudinal Stiffness of Coupler

Prior to any impact testing, two couplers from Caboose MP918 and one coupler from Locomotive 8003 were sent to National Castings Divisions, Midland-Ross Corporation, Cleveland, Ohio, to be calibrated. Strain gauges were installed on the shank and the couplers were longitudinally compressed to a point where yield was indicated. The result was a curve of force versus strain, thus calibrating the strain gauges for force readings during impact tests. The approximate cross sectional area for the couplers in the strain gauged area is 20 square inches. The three curves for the couplers, Figures 46, 47, and 48 are linear over the range tested. Their slopes vary because the position of the strain gauge on the shank was not exactly the same for each coupler causing the strain to be measured at points with different cross sectional areas. The couplers from Caboose MP912 and a second coupler from Locomotive 8003 which were used for impact Test 9 are also included on these graphs. Table 8 is a summary of the force-strain characteristic for each coupler.

3.6 CENTER OF GRAVITY

3.6.1 Requirements

The location of the center of gravity is important in the definition of the equations of motion of an object. Since the computer simulation used is two dimensional, the longitudinal and vertical location of the c.g. are required.

3.6.2 Test Procedure

The longitudinal center of gravity was obtained by using the weight reaction method. Knowing the weight of one end of the vehicle, total weight, and the distance between support points (bolster centers), the longitudinal center of gravity can be calculated from summing the moments about a point. For example, Figure 49 is a free body diagram of a car showing the forces acting on the car along with the appropriate equation obtained by summing moments about the right support. For the hopper and

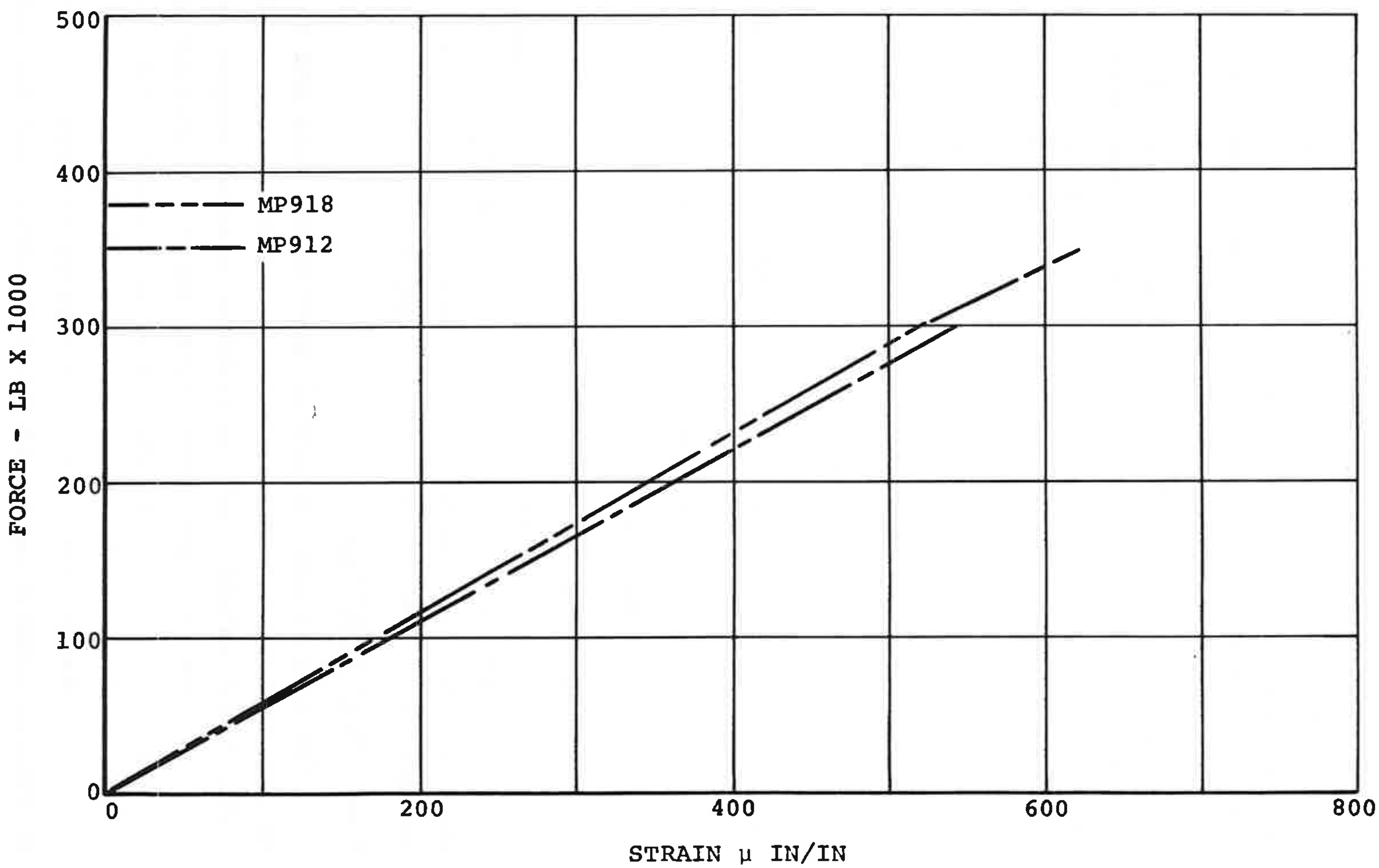


FIGURE 46. CABOOSE COUPLER CALIBRATION - A END.

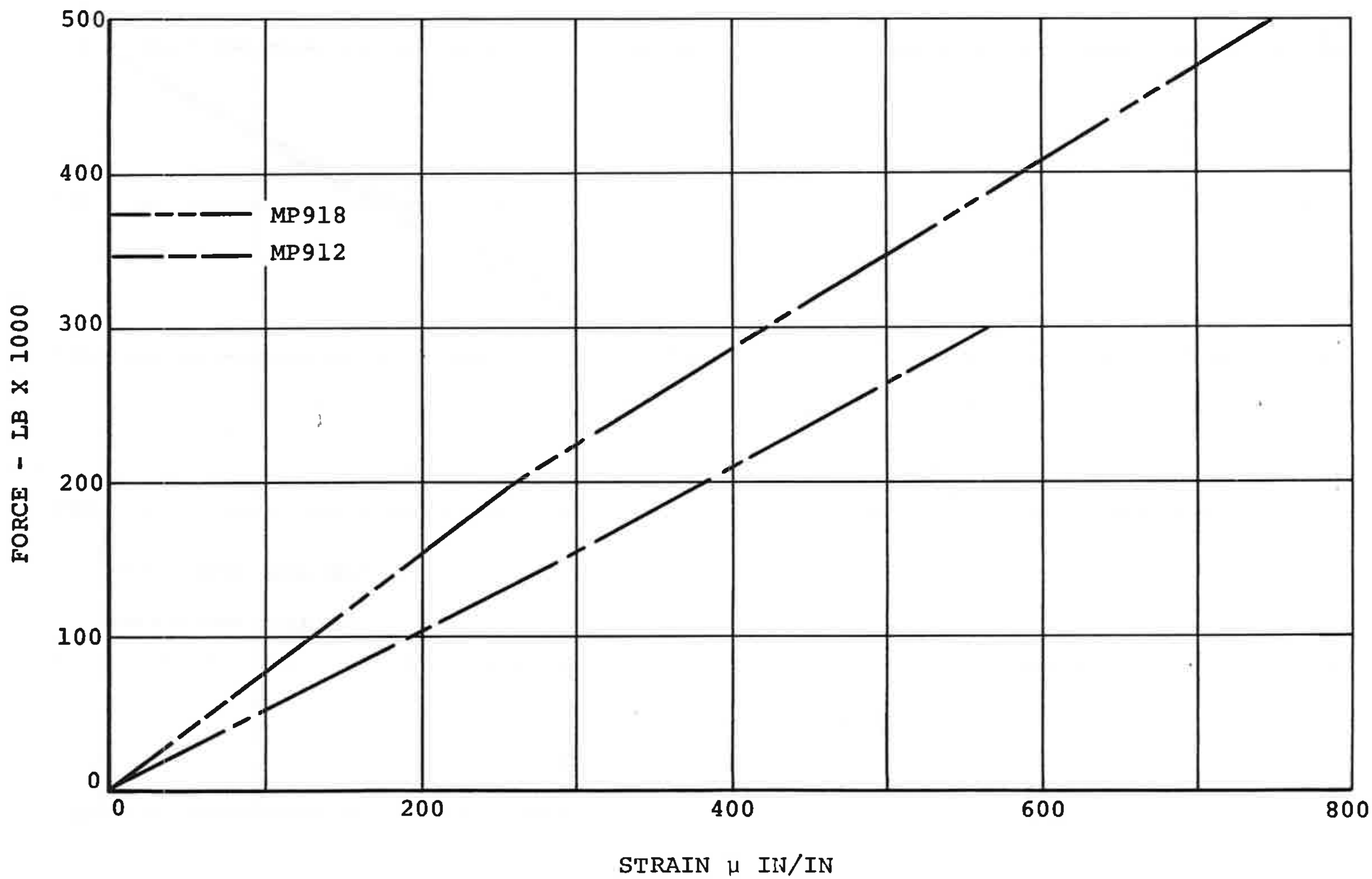


FIGURE 47. CABOOSE COUPLER CALIBRATION - B END.

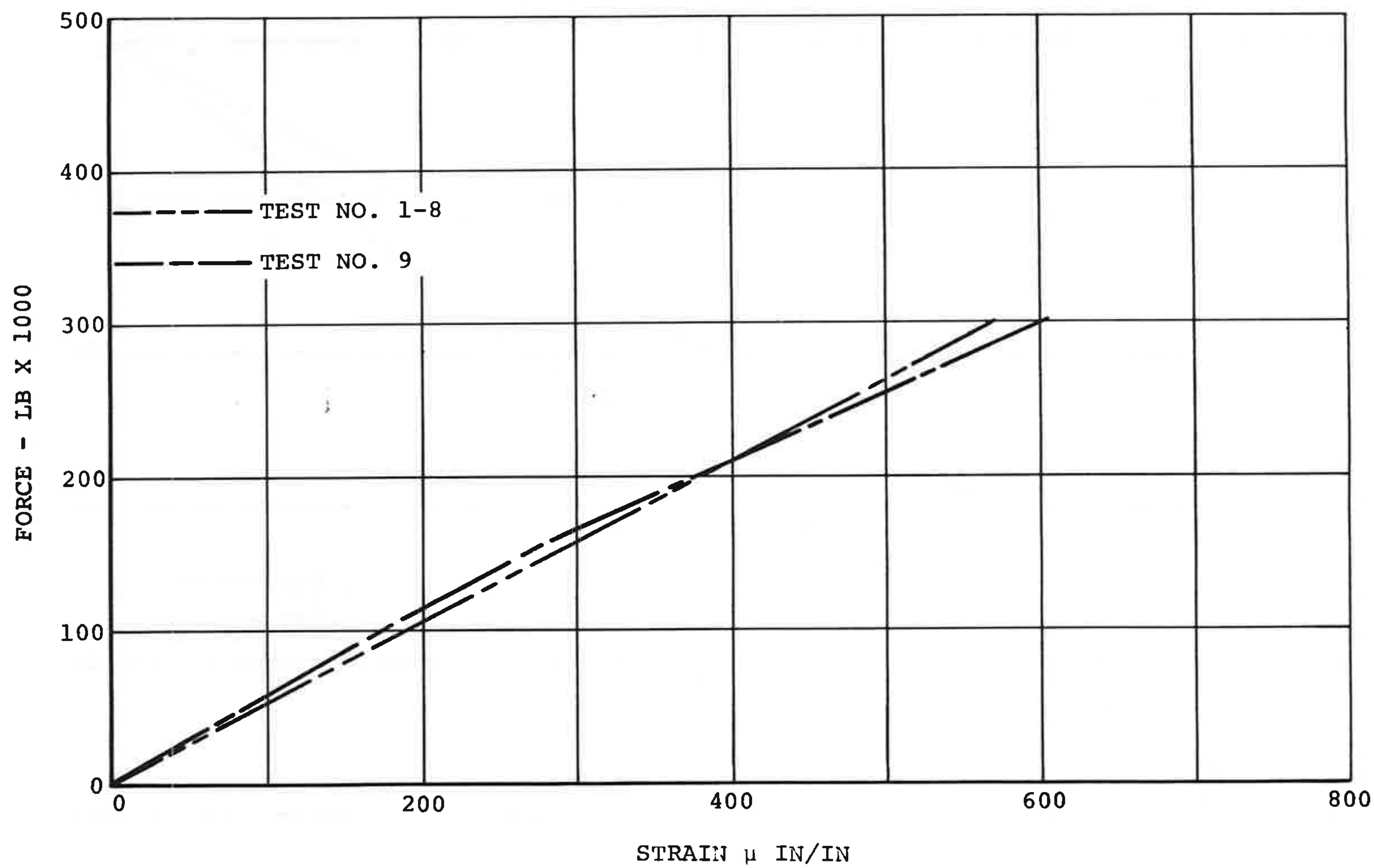
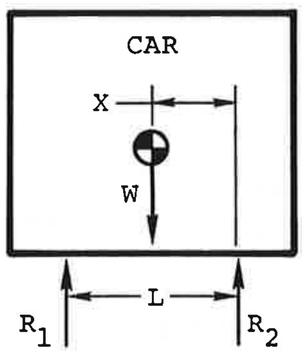


FIGURE 48. LOCOMOTIVE COUPLER CALIBRATION.

TABLE 8. COUPLER FORCE-STRAIN RATE		
Vehicle	End of Vehicle	Force-Strain Rate (lb/μin./in.)
Caboose MP918	"A" End	568
	"B" End	680
Caboose MP912	"A" End	522
	"B" End	532
Locomotive 8003	Front (Test 1-8)	520
	Front (Test 9)	504



ΣM about R_2

$$wx - R_1 L = 0$$

or

$$X = \frac{R_1 L}{W}$$

FIGURE 49. FREE BODY DIAGRAM OF CAR FOR CENTER OF GRAVITY CALCULATION.

caboose, the longitudinal centers of gravity are shown in Table 9.

The vertical center of gravity was calculated by the weight reaction method when the vehicle is rotated (see Figure 50). A special fixture was fabricated to allow the car to pivot about a fixed point. Figure 51 shows how a pivot bar was attached to the buffer casting of the car. Cables were attached to the side rails at the opposite end of the car and connected to a spreader bar. The spreader bar was then connected to an overhead crane with a

TABLE 9. LONGITUDINAL CENTER OF GRAVITY				
Vehicle	Body Weight (lb)	Weight,* Each End (lb)	Distance Between Bolsters (in.)	Longitudinal** Center of Gravity (in.)
Caboose MP912	28,828	"A" End: 14,860 "B" End: 13,968	250.0	212.8
Caboose MP918	28,020	"A" End: 14,240 "B" End: 13,780	248.0	210.0
Hopper 538021	30,920	"A" End: 14,700 "B" End: 16,220	381.0	259.5
Locomotive 8003	248,173 (Dry)	Rear End: 120,633 Front End: 127,540	375.0	334.7***

*Weight of vehicle less trucks.
**Distance from "B" end coupler face.
***Distance from impact end coupler face.

load cell between the bar and the crane hook. As the end of the vehicle was lifted, the change in force was recorded for several angles up to 30 degrees. To measure the different angles, a plumb bob was suspended from the side rail and used to point at a tape measure laying on the ground directly under the car. Figure 52 defines how the angle of the car is calculated by measuring distances. Once the angle is obtained, the height of the center of gravity can be calculated using the equation in Figure 53.

The vertical heights of the center of gravity of the test cars as calculated are:

Vehicle	Center of Gravity (Inches Above Coupler Centerline)
Caboose MP918	24.2
Hopper 536021	20.5

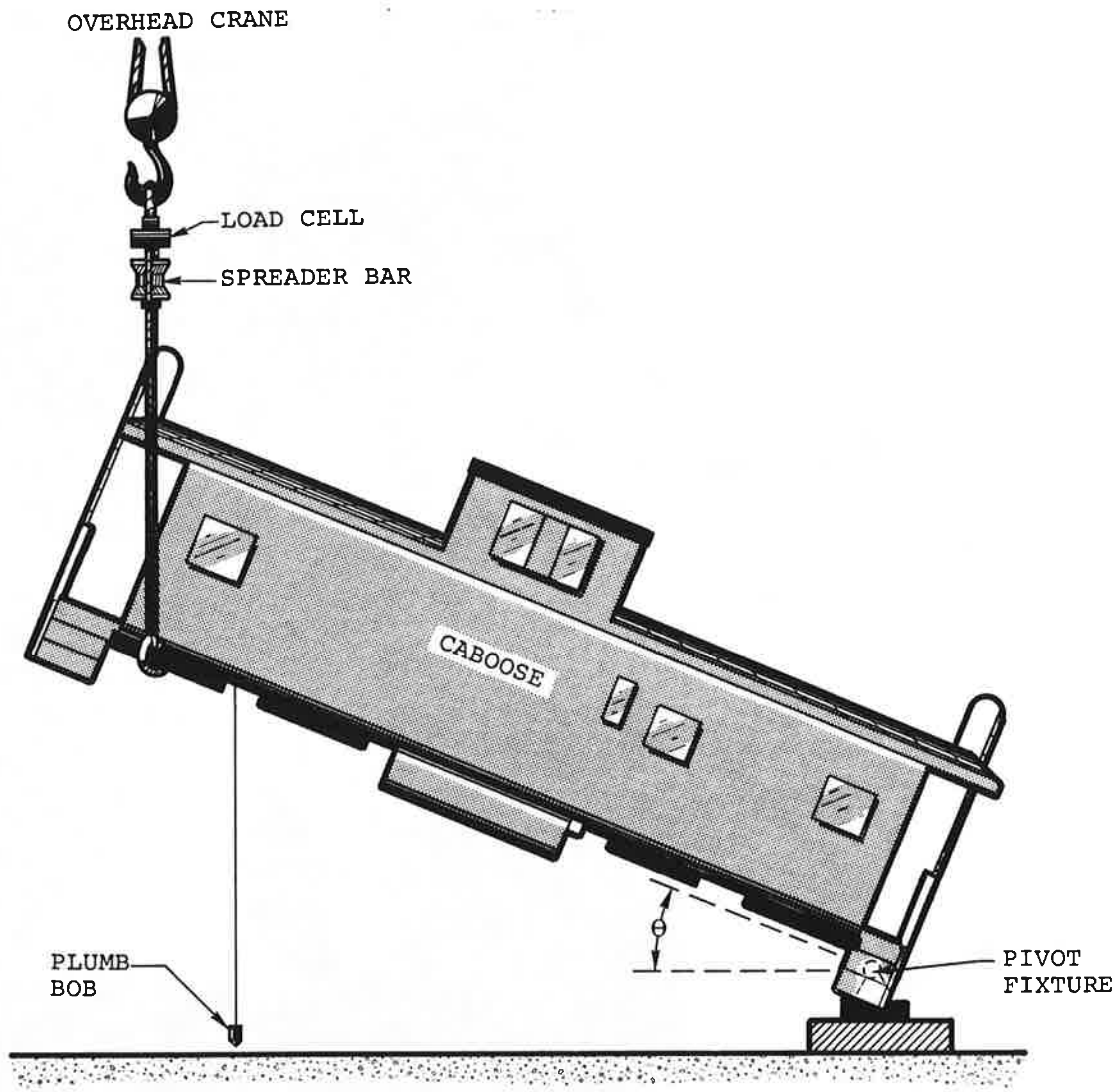


FIGURE 50. VERTICAL CENTER OF GRAVITY MEASUREMENT.

3.7 MOMENT OF INERTIA

3.7.1 Requirements

The cars near the impact were expected to rotate about the pitch axis. Since the mass moment of inertia is a measure of the resistance of an object to angular acceleration, the pitch moment of inertia was obtained for a caboose and an empty hopper

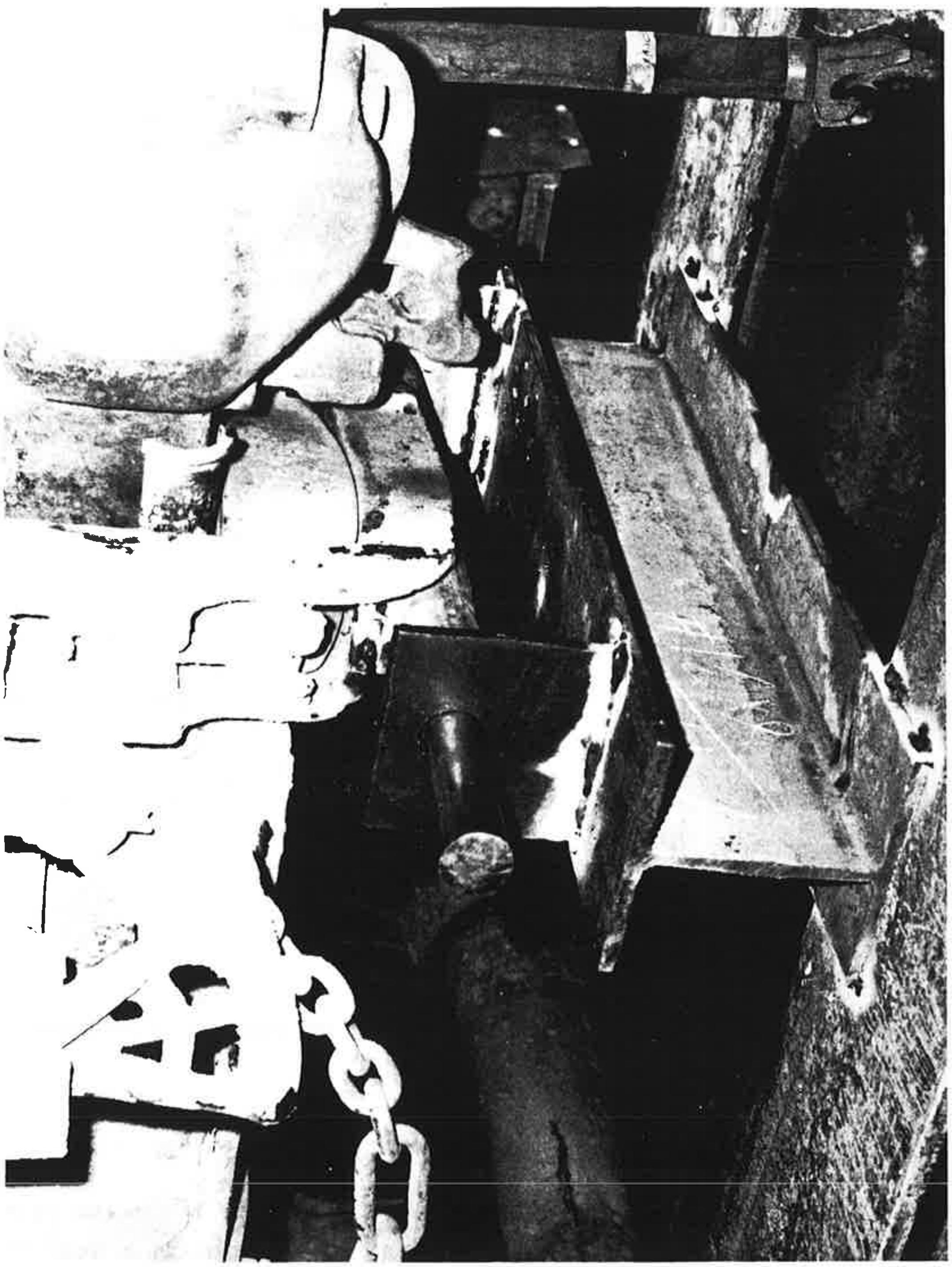
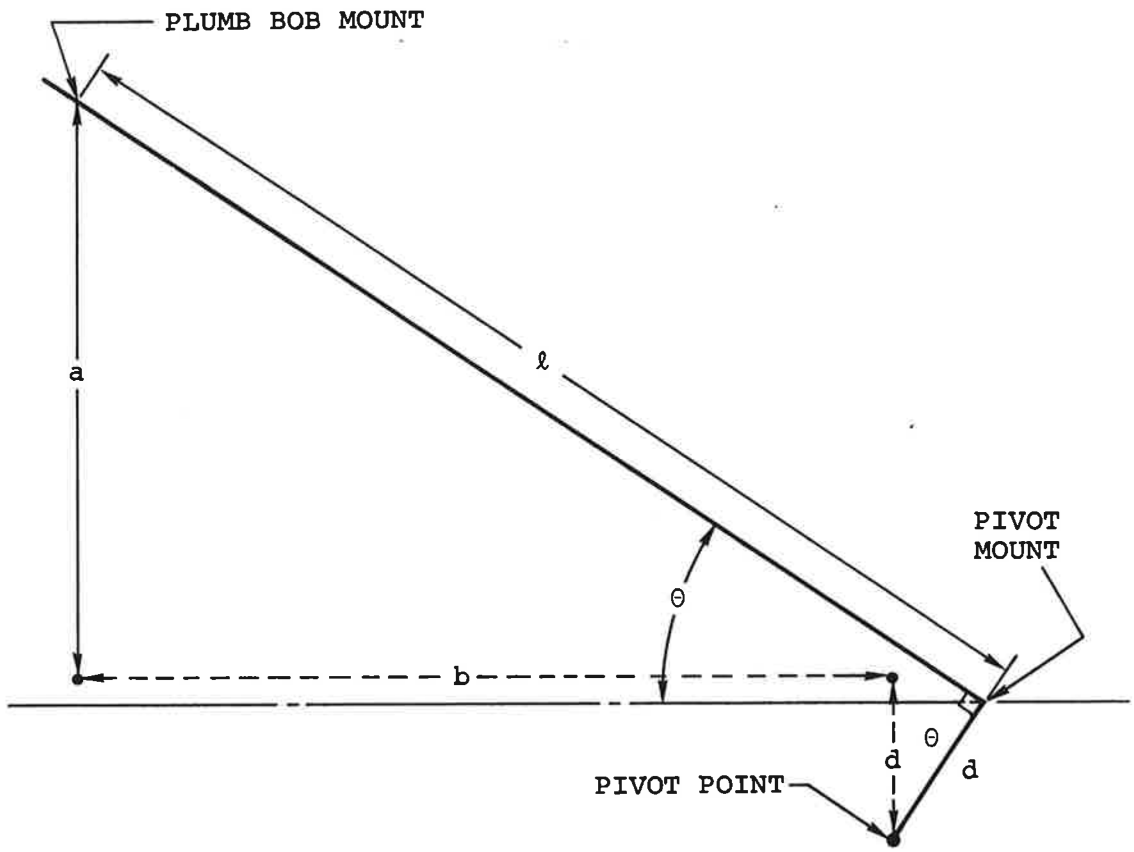


FIGURE 51. PIVOT FIXTURE FOR CENTER OF GRAVITY TEST.



FROM TRIGONOMETRY:

$$a = \ell \text{ SIN } \Theta - (d - d \cos\Theta)$$

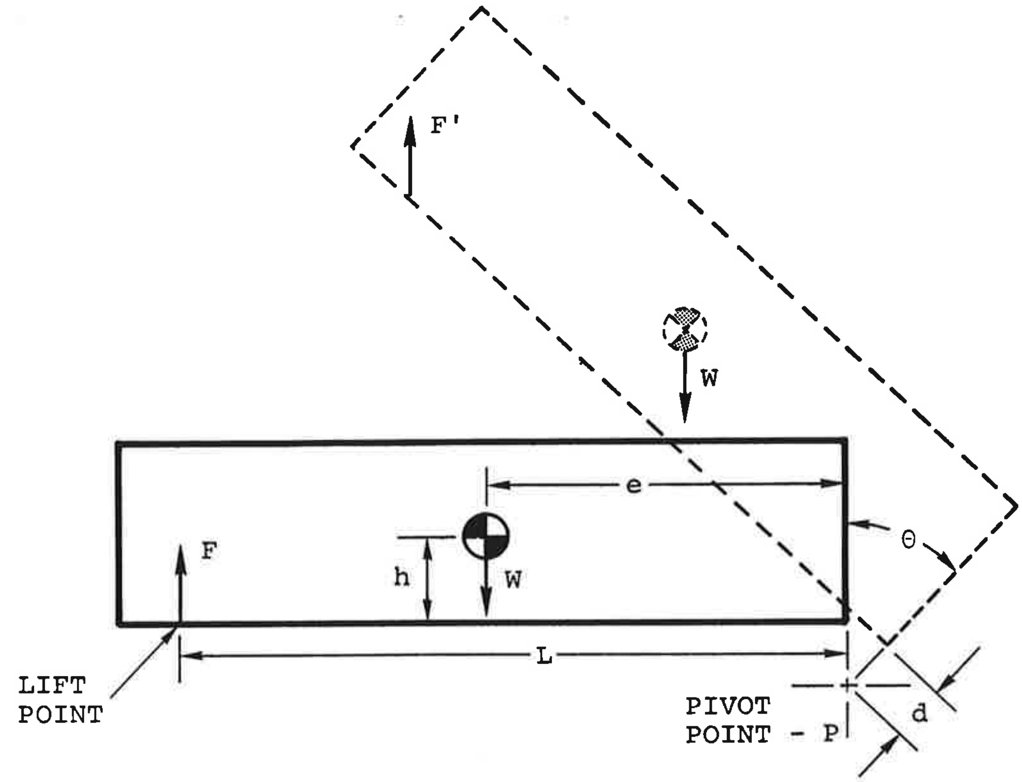
and $b = \ell \text{ COS}\Theta - d \sin\Theta$

OR

$$\text{SIN } \Theta = \frac{a \ell + d(\ell - b)}{\ell^2 + d^2}$$

Where: d = Distance from Pivot Point to Horizontal Plane through Plumb Bob Mount
ℓ = Distance from Plumb Bob Mount to Pivot Mount
a = Plumb Bob Length Minus d
b = Distance from Pivot Point to Plumb Bob

FIGURE 52. EQUATION FOR ANGLE OF ELEVATION.



$\Sigma M_p = 0$ MOMENT SIGN CONVENTION: (+ (-

$$F'(L\cos\Theta - d\sin\Theta) - W(e\cos\Theta - (d+h)\sin\Theta) = 0$$

or

$$h = \cot\Theta\left(e - \frac{F'L}{W}\right) + d\left(\frac{F'}{W} - 1\right)$$

Where:
- W = Vehicle Weight
- F' = Force to Lift Car to Angle Θ
- L = Distance from Pivot Point to Lift Point
- e = Distance from Pivot Point to Longitudinal Center of Gravity
- d = Distance from Pivot Point to Horizontal Plane through Lift Point
- h = Vertical Distance from Horizontal Plane through Lift Point to Center of Gravity

FIGURE 53. EQUATION FOR VERTICAL CENTER OF GRAVITY.

car. The mass moment of inertia is required as a parameter in the computer model if angular rotation is to be simulated. The pitch mass moment of inertia was estimated for the locomotive. The locomotive was not expected to have large angular motion.

3.7.2 Test Procedure

The procedure used to measure the mass moment of inertia of a test vehicle about its c.g. was to suspend it from a pivot point and allow it to oscillate about the pivot, similar to a pendulum. For small angles of oscillation, the period of a simple pendulum is proportional to its mass moment of inertia. For these tests, an "A-Frame" fixture was fabricated at TTC, and, using the overhead crane in the RDL. the test cars were suspended without trucks from the pivot bars at the top of the "A-Frame." Cables were attached to the pivot point at one end and hooked directly to the test vehicle at the other end. Figure 54 shows the Caboose MP918 suspended in the "A-Frame" fixture ready for test. The vehicle was then put into an oscillating motion by pushing gently on the end of the car. This "forcing function" was applied at the center of the coupler to eliminate any yaw motion. The amplitude of motion was kept small to keep consistent with the small angle assumption. A stop watch was the only instrument used during the test, and the number of oscillations counted was large (100 oscillations/test) to improve accuracy of the time per oscillation. The equation used to compute mass moment of inertia is:

$$I = \frac{W\ L\ T^2}{4\pi^2} - \frac{W\ L^2}{g}$$

where W = weight of vehicle

L = distance between axis of oscillation and center of gravity of the vehicle

T = period of oscillation

g = acceleration of gravity

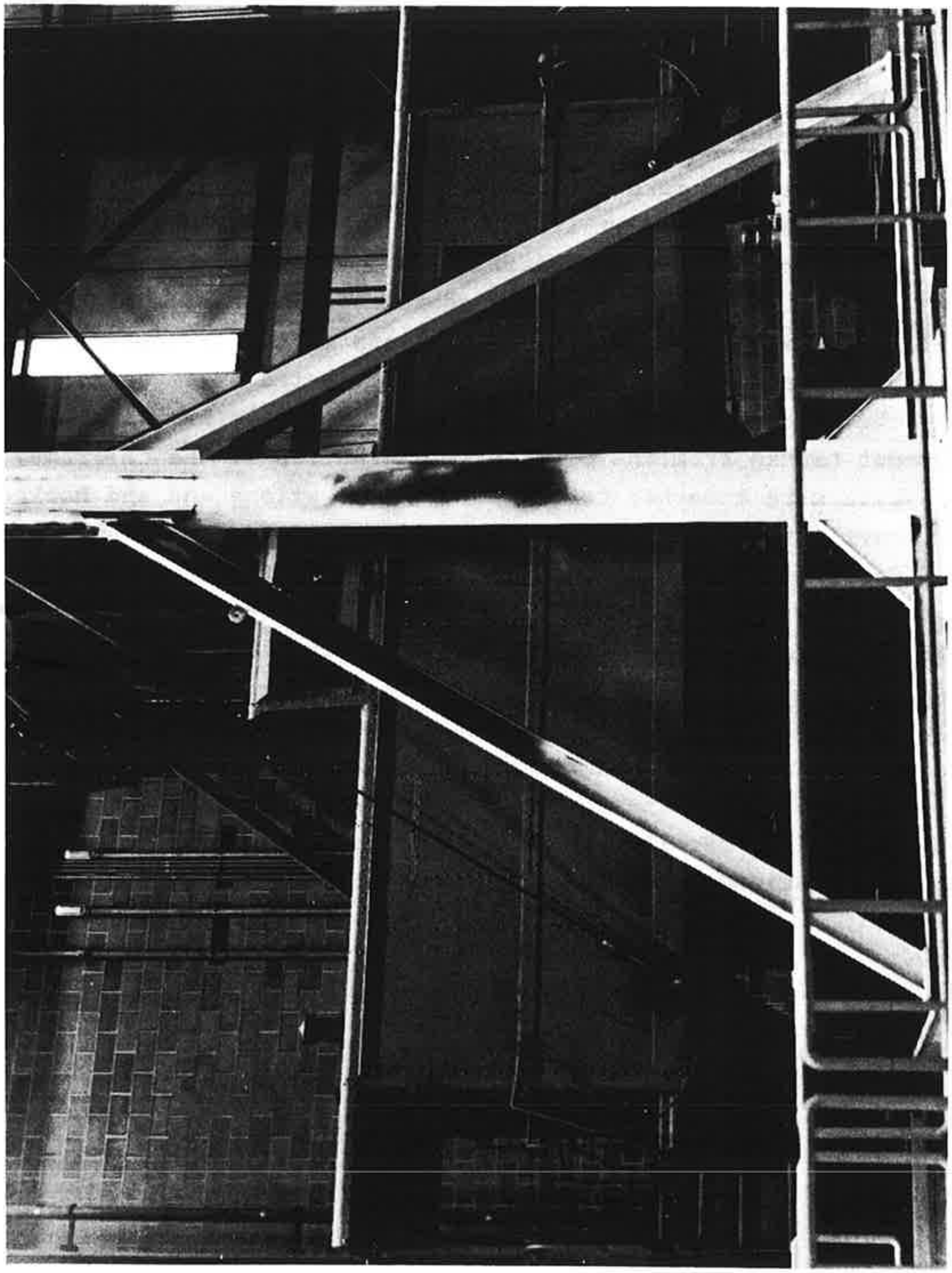


FIGURE 54. CABOOSE MOMENT OF INERTIA TEST CONFIGURATION.

Table 10 lists the time/oscillation as an average of three 100-cycle tests.

TABLE 10. MOMENT OF INERTIA				
Vehicle	Weight (lb)	Pendulum Length (in.)	Period (sec)	Moment of Inertia (in.-lb-sec^2)
Caboose MP918	28,020	185.3	4.96	745,643
Hopper 538021	30,920	193.0	5.63	1,810,614
Locomotive 8003	259,849	-	-	27.5×10^6
Note: The value for the locomotive was calculated by assuming it to be a uniform rod and using the equation $I = 1/12\ ML^2$ where M is the weight/g and L is the total length in inches.				

3.8 RAIL DEFLECTION TEST

3.8.1 Requirements

The cars in the train are supported by the track. As the load from the wheels on the track changes, the track deflects. In order to define the forces on the cars, which is required in order to simulate the behavior of the cars in the train on the computer analysis, it is important to define the vertical force deflection characteristic of the track.

A test was performed to determine the vertical force deflection of the track near the impact point directly below the rearmost wheel of the caboose. The rail was 136 lb/yard with crushed slag ballast.

3.8.2 Test Procedure

The deflection of the rail was measured under various loads. The procedure used to apply the different vertical loads at a given point was to move rail cars of known axle weights over the test point. Table 11 summarizes the vertical weights and the force applied by one wheel of each vehicle. The weights were obtained at the TTC track scales.

TABLE 11. RAIL DEFLECTION WEIGHT

Vehicle	Total Weight (lb)	Weight of Test Wheel (lb)
Locomotive 8003	248,620	20,068
Boxcar 147317	143,480	14,790
Boxcar 276567	46,280	5,855

Each vehicle was pulled over the test area and stopped with the test wheel directly over the deflection measuring device. To measure the deflection, a dial indicator was mounted onto a horizontal beam directly under the rail between two ties (see Figure 55). The beam was attached to a vertical pipe driven into the ground at a point where movement of the ballast due to vehicle weight would not disturb the horizontal beam (see Figure 56). The change in dial indicator readings versus weight of the wheel is shown in Figure 57. The slope of the lower part of the curve (0 to 6,000 pounds) is 172,205 lb/in. and includes settling of the ties into the ballast. The upper part of the curve (6,000 pounds to 20,000 pounds) has a spring rate of 363,636 lb/in.

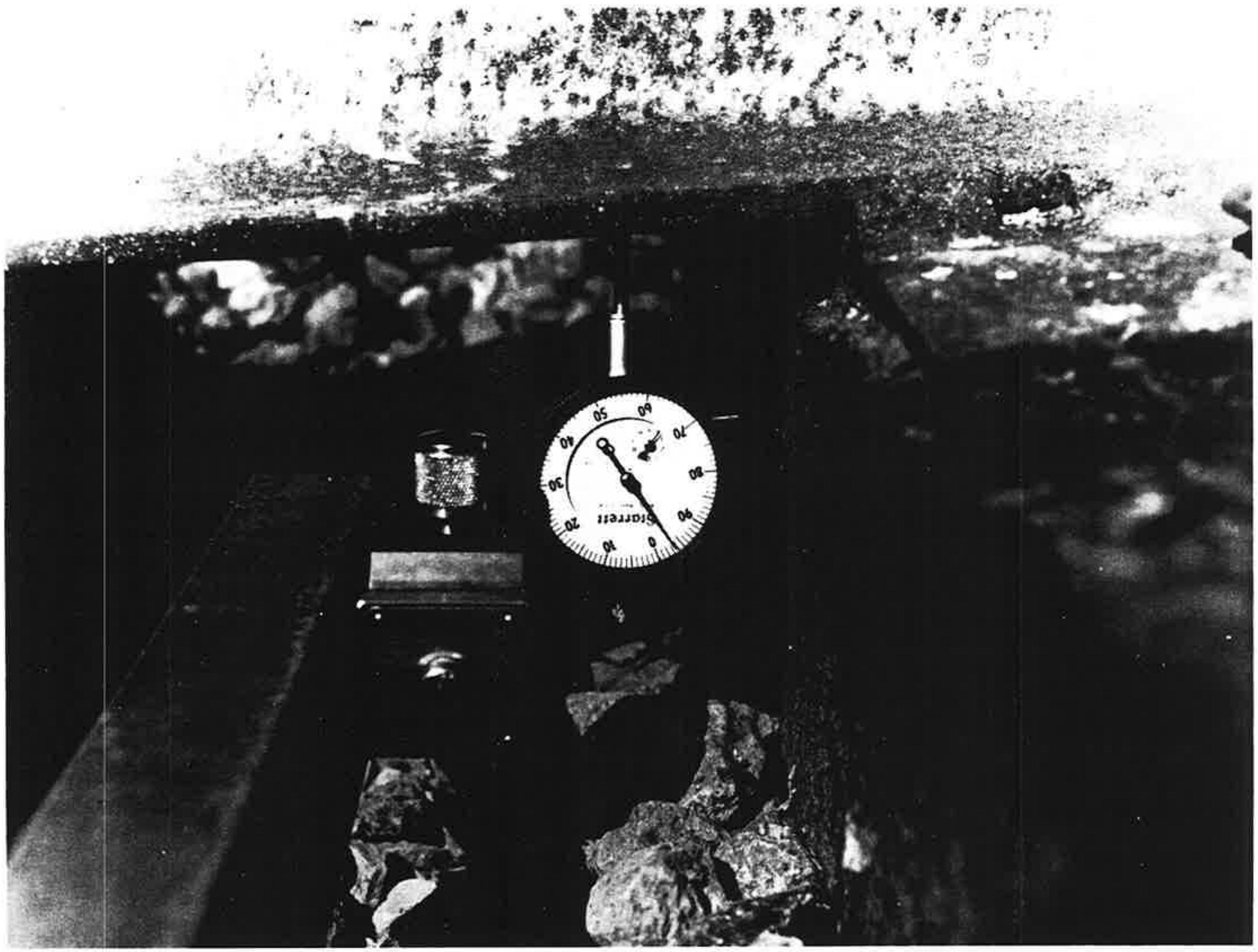


FIGURE 55. DIAL INDICATOR FOR RAIL DEFLECTION MEASUREMENT.

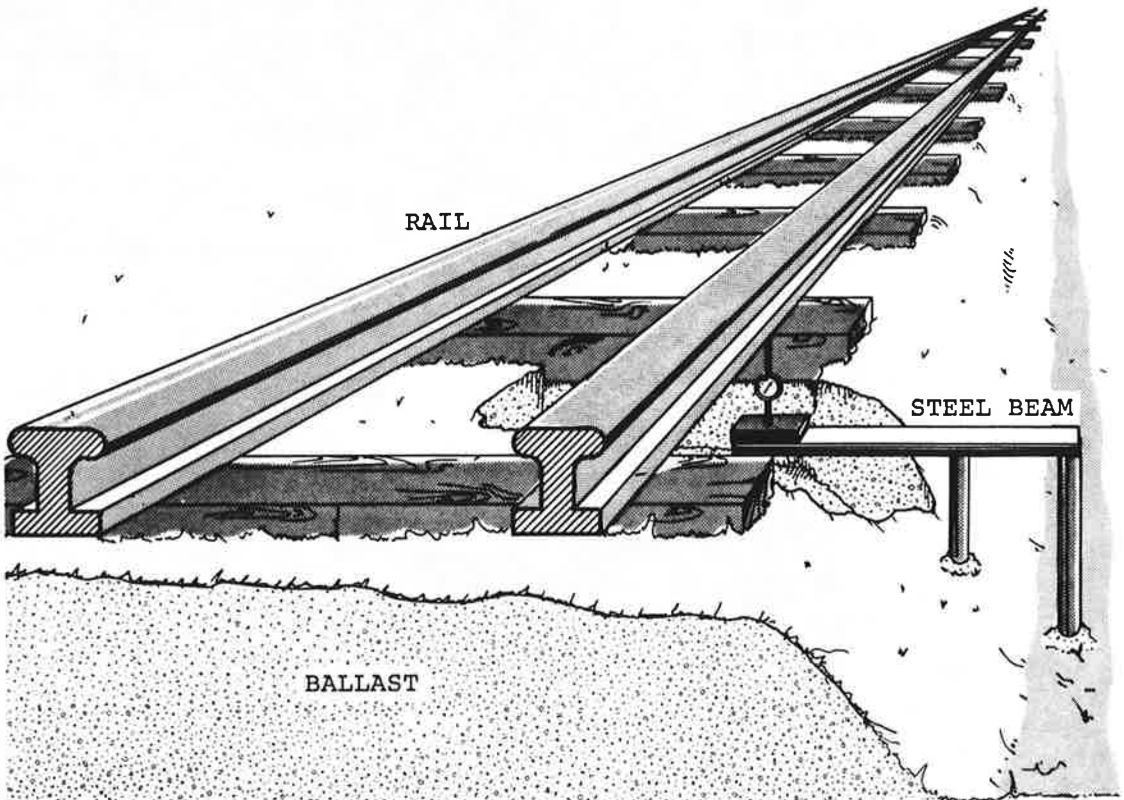


FIGURE 56. BEAM FOR THE RAIL-DEFLECTION TEST.

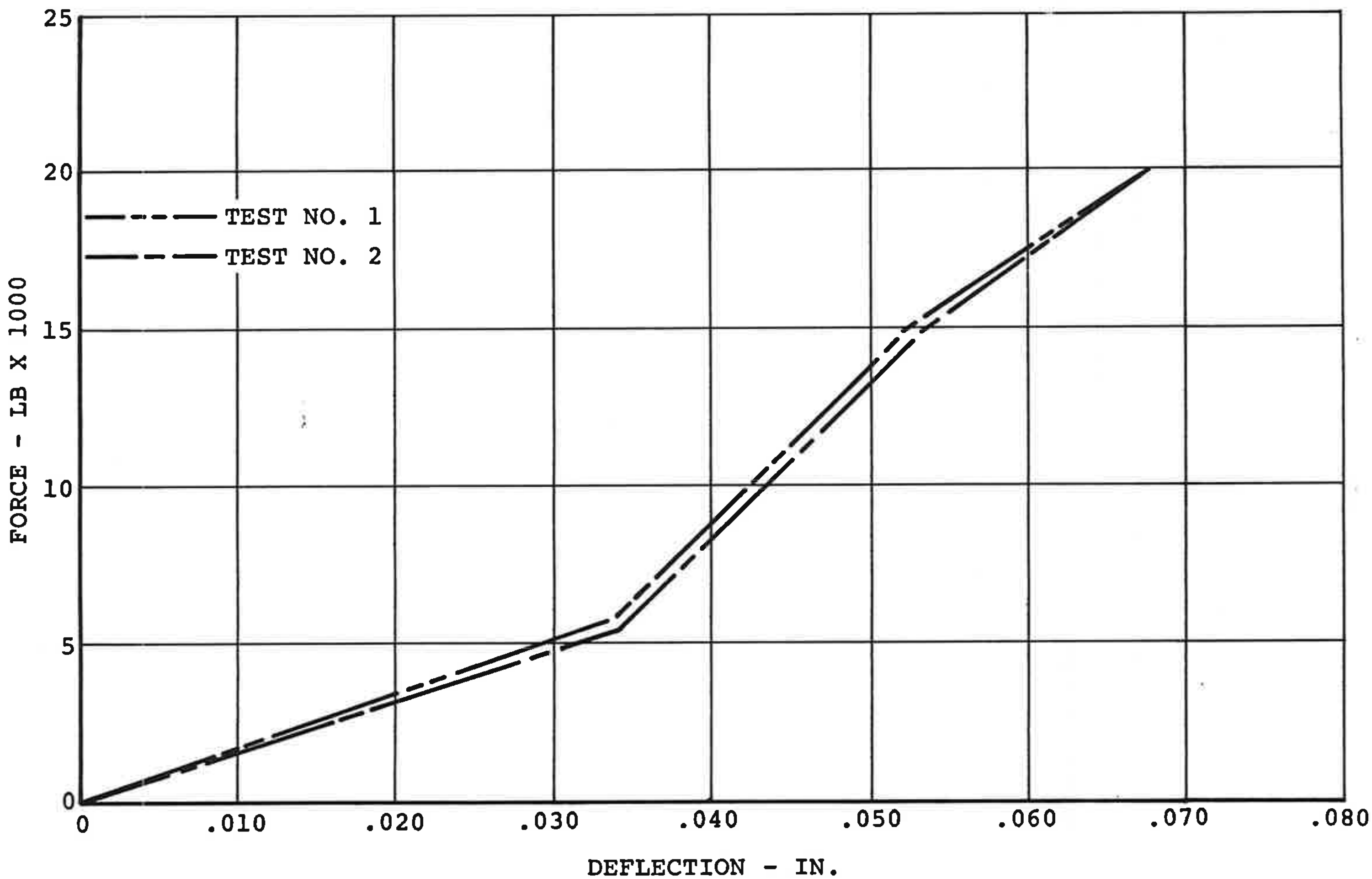


FIGURE 57. RAIL FORCE-DEFLECTION CURVE.

4. SUMMARY

Summaries of all of the pre-test measurements are shown in Tables 12 through 15.

Several values measured on the Caboose MP918 were not measured on Caboose MP912. For the summary table the unmeasured values for MP918 were taken as the same as those for MP912 even though disparities were noticed between some parameters actually measured on both cabooses. An example of a parameter measured only on one caboose is the vertical spring rate of the center sill (41,700 lb/in.). As mentioned in the text, several values were estimated for the locomotive (for example, height of center of gravity).

Figure 58 shows the three types of vehicles tested and used for the impact tests.

TABLE 12. SUMMARY OF DATA FOR CABOOSE MP912

Measurement		Value
1. Body Weight		28,828 lb
2. Truck Weight		7,060 lb
3. Total Vehicle Weight		42,948 lb
4. Distance Between Coupler Faces		418.0 in.
5. Length of Center Sill		375.0 in.
6. Thickness of Buffer Casting (striker plate)		5.0 in.
7. Distance Between Truck Centers		250.0 in.
8. Longitudinal Coupler Slack (knuckles + draft gear)		2.25 in.
9. Draft Gear Spring Travel		2.25 in.
10. Coupler Height Above Rail:	"A" End	32.0 in.
	"B" End	34.0 in.
11. Bolster Height Above Rail:	"A" End	25.5 in.
	"B" End	27.1 in.
12. Sill Height Above Rail:	"A" End	27.25 in.
	"B" End	29.25 in.
13. Coupler Height Above Bolster:	"A" End	6.5 in.
	"B" End	6.9 in.
14. Vertical Coupler Slack:	"A" End	3.5 in.
	"B" End	2.9 in.
15. Center Pin Height Above Bolster Surface:	"A" End	N/A
	"B" End	N/A
16. Truck Spring Compliance		8,004 lb/in.
17. Truck Bolster and Side Frame Compliance		370,786 lb/in.
18. Truck Spring Travel to Top		1.5 in.
19. Truck Spring Travel to Bottom		3.6 in.
20. Inside Diameter of Bolster Dish		12.0 in.
21. Depth of Truck Bolster Dish		1.25 in.
22. Vertical Spring Rate of Vehicle (Location B)		41,700 lb/in.
23. Draft Gear Spring Rate		254,500 lb/in.
24. Total Underframe Spring Constant		612,394 lb/in.
25. Longitudinal Spring Rate of Center Sill		1,720,000 lb/in.
26. Height of Center of Gravity to Coupler Centerline		24.2 in.
27. Distance of Center of Gravity to Coupler Face*		212.8 in.
28. Moment of Inertia of Body		745,643 in.-lb-sec^2
29. Area Moment of Inertia of Center Sill		597.0 in.^{4}
Note: The "B" End was the impact end.		
*Distance from "B" End coupler.		

TABLE 13. SUMMARY OF DATA FOR CABOOSE MP918

Measurement			Value
1.	Body Weight		28,020 lb
2.	Truck Weight		7,060 lb
3.	Total Vehicle Weight		42,140 lb
4.	Distance Between Coupler Faces		416.0 in.
5.	Length of Center Sill		372.0 in.
6.	Thickness of Buffer Casting (striker plate)		5.0 in.
7.	Distance Between Truck Centers		248.0 in.
8.	Longitudinal Coupler Slack (knuckles + draft gear)		2.5 in.
9.	Draft Gear Spring Travel		2.5 in.
10.	Coupler Height Above Rail:	"A" End	31.5 in.
		"B" End	34.5 in.
11.	Bolster Height Above Rail:	"A" End	25.5 in.
		"B" End	27.4 in.
12.	Sill Height Above Rail:	"A" End	27.25 in.
		"B" End	29.9 in.
13.	Coupler Height Above Bolster:	"A" End	6.0 in.
		"B" End	7.1 in.
14.	Vertical Coupler Slack:	"A" End	3.25 in.
		"B" End	3.9 in.
15.	Center Pin Height Above Bolster Surface:	"A" End	6.5 in.
		"B" End	7.25 in.
16.	Truck Spring Compliance		8,004 lb/in.
17.	Truck Bolster and Side Frame Compliance		370,786 lb/in.
18.	Truck Spring Travel to Top		1.5 in.
19.	Truck Spring Travel to Bottom		3.6 in.
20.	Inside Diameter of Bolster Dish		12.0 in.
21.	Depth of Truck Bolster Dish		1.25 in.
22.	Vertical Spring Rate of Vehicle (Location B)		41,700 lb/in.
23.	Draft Gear Spring Rate		254,500 lb/in.
24.	Total Underframe Spring Constant		612,394 lb/in.
25.	Longitudinal Spring Rate of Center Sill		2,610,000 lb/in.
26.	Height of Center of Gravity to Coupler Centerline		24.2 in.
27.	Distance of Center of Gravity to Coupler Face*		210.0 in.
28.	Moment of Inertia of Body		754,643 in.-lb-sec^2
29.	Area Moment of Inertia of Center Sill		597.0 in.^{4}

Note: The "B" End was the impact end.

*Distance from "B" End coupler.

TABLE 14. SUMMARY OF DATA FOR HOPPER 538021

Measurement		Value
1. Body Weight		30,920 lb
2. Truck Weight		7,940 lb
3. Total Vehicle Weight (Empty)		46,800 lb
4. Distance Between Coupler Faces		538.0 in.
5. Length of Center Sill		491.0 in.
6. Thickness of Buffer Casting (striker plate)		5.0 in.
7. Distance Between Truck Centers		381.0 in.
8. Longitudinal Coupler Slack (knuckles + draft gear)		2.75 in.
9. Draft Gear Spring Travel		3.0 in.
10. Coupler Height Above Rail:	"A" End	32.0 in.
	"B" End	33.75 in.
11. Bolster Height Above Rail:	"A" End	25.9 in.
	"B" End	26.1 in.
12. Sill Height Above Rail:	"A" End	28.1 in.
	"B" End	28.4 in.
13. Coupler Height Above Bolster:	"A" End	6.1 in.
	"B" End	7.6 in.
14. Vertical Coupler Slack:	"A" End	3.25 in.
	"B" End	2.75 in.
15. Center Pin Height Above Bolster Surface:	"A" End	8.25 in.
	"B" End	5.5 in.
16. Truck Spring Compliance		50,078 lb/in.
17. Truck Bolster and Side Frame Compliance		809,735 lb/in.
18. Truck Spring Travel to Top		3.25 in.
19. Truck Spring Travel to Bottom		2.2 in.
20. Inside Diameter of Bolster Dish		14.0 in.
21. Depth of Truck Bolster Dish		1.25 in.
22. Vertical Spring Rate of Vehicle (Location B)		62,500 lb/in.
23. Draft Gear Spring Rate		493,800 lb/in.
24. Total Underframe Spring Constant		598,151 lb/in.
25. Longitudinal Spring Rate of Center Sill		1,834,000 lb/in.*
26. Height of Center of Gravity to Coupler Centerline		20.5 in.
27. Distance of Center of Gravity to Coupler Face**		295.5 in.
28. Moment of Inertia of Body		1,810,614 in.-lb-sec^2
29. Area Moment of Inertia of Center Sill		616.0 $in.^4$

Note: The "A" End was the impact end.

*Calculated value.
**Distance from "B" End coupler.

TABLE 15. SUMMARY OF DATA FOR LOCOMOTIVE 8003

Measurement	Value
1. Body Weight	203,849 lb
2. Truck Weight (estimated)	28,000 lb
3. Total Vehicle Weight (wet)	259,849 lb
4. Distance Between Coupler Faces	654.0 in.
5. Length of Center Sill	600.0 in.
6. Thickness of Buffer Casting (striker plate)	9.5 in.
7. Distance Between Truck Centers	354.0 in.
8. Longitudinal Coupler Slack (knuckles + draft gear)	1.1 in.
9. Draft Gear Spring Travel (estimated)	2.5 in.
10. Coupler Height Above Rail - Impact End	32.0 in.
11. Bolster Height Above Rail	40.0 in.
12. Sill Height Above Rail	43.0 in.
13. Coupler Height Below Bolster - Impact End	8.0 in.
14. Vertical Coupler Slack	1.0 in.
15. Truck Spring Compliance (estimated)	80,000 lb/in.
16. Bolster and Side Frame Compliance (estimated)	1×10^6 lb/in.
17. Truck Spring Travel to Top	2.0 in.
18. Truck Spring Travel to Bottom	3.0 in.
19. Height of Center of Gravity to Coupler Centerline (estimated)	36.0 in.
20. Distance of Center of Gravity to Coupler Face*	335.0 in.
21. Moment of Inertia of Complete Vehicle	27.5×10^6 in.-lb-sec^2
22. Distance From Impact Coupler to Rear Bolster	142.0 in.

*Distance from impact end.

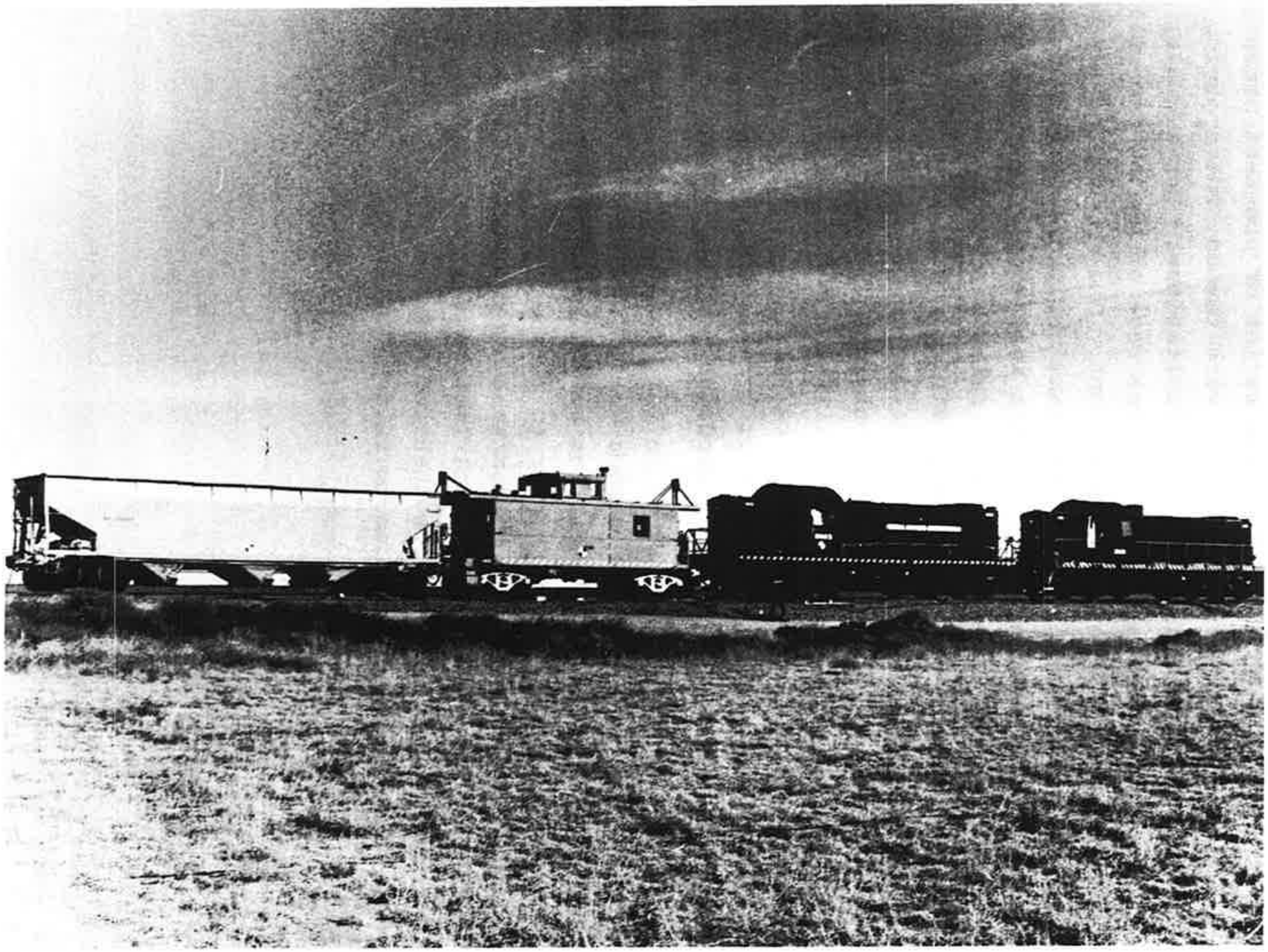


FIGURE 58. HOPPER, CABOOSE, AND LOCOMOTIVE USED FOR PRE-TEST MEASUREMENTS.

5. CONCLUSIONS AND RECOMMENDATIONS

The data obtained from the series of pre-test measurements are valid for the vehicles that were tested, but applying the values to all similar cars has limitations. For example, it was noted that the longitudinal spring rate of the center sill of the two cabooses differed by 34 percent. The tests were performed as exactly as possible; using the same equipment, same test procedure, etc. The measuring devices were checked after each test and found to provide repeatable data within one percent of previous values. Thus, the major difference was due to the physical differences in the cars. The cars were between 30 and 40 years old and their structural strengths could be quite different even though the models were similar. For example, the rivets on the cross braces on one car might be worn, cracked, or rusted, changing the structural support of the brace. One caboose had several cracks in the body structure.

The data values listed in the summary are good values for the cars tested, but a recommendation would be to use newer cabooses and hopper cars for the actual impact tests. Using newer vehicles would make it easier to obtain pre-test measurements and measurements could be compared with manufacturers design information. During the actual impacts the weak points of new cars are more predictable than for old vehicles which could have developed weak points due to rust and fatigue.

APPENDIX

CALIBRATION OF HYDRAULIC CYLINDERS

The hydraulic cylinders were calibrated by installing a load cell in series with a cylinder attached to a rigid fixture. A series of tests was then performed by applying pressure to the cylinder and recording the load cell reading. Several curves were attained for pressure versus force for two cylinders in the push condition and pull condition. This information was then used as input to a curve fitting computer program to give the optimum force versus pressure conversion factor. Figures 59 through 62 are the results of this process.

In addition, the load cell and strain meter were calibrated by an outside supplier prior to the above tests.

The dial indicators were calibrated against a standard one-inch bar. The linear potentiometers were calibrated against the above dial indicators directly after their use on a test.

Table 16 lists the manufacturers and data on the instruments used during the tests.

NO. OF POINTS = 10
MEAN VALUE OF X = .2414463E+04
MEAN VALUE OF Y = .5500000E+05
STD ERROR OF Y = .3027650E+05

POLYFIT OF DEGREE 1
INDEX OF DETERMINATION .9999706E+00
TERM COEFFICIENT

0	-.15454609E+04
1	.23419472E+02

X-ACTUAL	Y-ACTUAL	Y-CALC	DIFF	PCT-DIFF
.486600E+03	1.000000E+04	.985045E+04	.149547E+03	.151817E+01
.913300E+03	.200000E+05	.198435E+05	.156457E+03	.788453E+00
.135500E+04	.300000E+05	.301879E+05	-.187922E+03	-.622507E+00
.178660E+04	.400000E+05	.402958E+05	-.295766E+03	-.733987E+00
.219660E+04	.500000E+05	.498977E+05	.102250E+03	.204919E+00
.262500E+04	.600000E+05	.599307E+05	.693437E+02	.115707E+00
.306330E+04	.700000E+05	.701954E+05	-.195406E+03	-.278375E+00
.347660E+04	.800000E+05	.798747E+05	.125312E+03	.156886E+00
.390830E+04	.900000E+05	.899849E+05	.151406E+02	.168257E-01
.433333E+04	1.000000E+05	.999388E+05	.611562E+02	.611937E-01

STD ERROR OF ESTIMATE FOR Y = .174196E+03

CYLINDER NO. 1 - PUSHING

FORCE = 23.419472 PSI - 1545.4609 LB_F

FIGURE 59. COMPUTER PRINTOUT OF FORCE VERSUS PRESSURE CYLINDER 1 PUSHING.

NO. OF POINTS = 9
MEAN VALUE OF X = .1073889E+04
MEAN VALUE OF Y = .3000000E+05
STD ERROR OF Y = .1369307E+05

POLYFIT OF DEGREE 1
INDEX OF DETERMINATION .9998883E+00
TERM COEFFICIENT

0 -.46728515E+03
1 .28370983E+02

X-ACTUAL	Y-ACTUAL	Y-CALC	DIFF	PCT-DIFF
.373000E+03	1.000000E+04	.101151E+05	-.115092E+03	-.113782E+01
.543000E+03	.150000E+05	.149382E+05	.618418E+02	.413985E+00
.718000E+03	.200000E+05	.199031E+05	.969180E+02	.486950E+00
.900000E+03	.250000E+05	.250666E+05	-.666016E+02	-.265698E+00
.106600E+04	.300000E+05	.297762E+05	.223816E+03	.751662E+00
.126000E+04	.350000E+05	.352802E+05	-.280156E+03	-.794090E+00
.142500E+04	.400000E+05	.399614E+05	.386328E+02	.966754E-01
.160000E+04	.450000E+05	.449263E+05	.737109E+02	.164071E+00
.178000E+04	.500000E+05	.500331E+05	-.330703E+02	-.660969E-01

STD ERROR OF ESTIMATE FOR Y = .154689E+03

CYLINDER NO. 1 - PULLING

FORCE = 28.370983 PSI - 467.28515 LB_F

FIGURE 60. COMPUTER PRINTOUT OF FORCE VERSUS PRESSURE CYLINDER 1 PULLING.

NO. OF POINTS = 10
MEAN VALUE OF X = .2435000E+04
MEAN VALUE OF Y = .5500000E+05
STD ERROR OF Y = .3027650E+05

POLYFIT OF DEGREE 1
INDEX OF DETERMINATION .9998717E+00
TERM COEFFICIENT

0 -.13032187E+04
1 .23122471E+02

X-ACTUAL	Y-ACTUAL	Y-CALC	DIFF	PCT-DIFF
.497000E+03	1.000000E+04	.101886E+05	-.188648E+03	-.185156E+01
.918000E+03	.200000E+05	.199232E+05	.767891E+02	.385425E+00
.136000E+04	.300000E+05	.301433E+05	-.143340E+03	-.475527E+00
.179000E+04	.400000E+05	.400860E+05	-.860078E+02	-.214558E+00
.221500E+04	.500000E+05	.499131E+05	.869453E+02	.174194E+00
.263300E+04	.600000E+05	.595782E+05	.421750E+03	.707893E+00
.308000E+04	.700000E+05	.699140E+05	.860156E+02	.123031E+00
.349700E+04	.800000E+05	.795561E+05	.443937E+03	.558018E+00
.398200E+04	.900000E+05	.907705E+05	-.770453E+03	-.848793E+00
.437800E+04	1.000000E+05	.999270E+05	.730469E+02	.731003E-01

STD ERROR OF ESTIMATE FOR Y = .363709E+03

CYLINDER NO. 2 - PUSHING

FORCE = 23.122471 PSI - 1303.2187 LB_F

FIGURE 61. COMPUTER PRINTOUT OF FORCE VERSUS PRESSURE CYLINDER 2 PUSHING.

NO. OF POINTS = 9
MEAN VALUE OF X = .1064556E+04
MEAN VALUE OF Y = .3000000E+05
STD ERROR OF Y = .1369307E+05

POLYFIT OF DEGREE 1
INDEX OF DETERMINATION .9998739E+00
TERM COEFFICIENT

0 -.31962500E+03
1 .28481014E+02

X-ACTUAL	Y-ACTUAL	Y-CALC	DIFF	PCT-DIFF
.360000E+03	1.000000E+04	.993354E+04	.664590E+02	.669036E+00
.540000E+03	.150000E+05	.150601E+05	-.601230E+02	-.399220E+00
.716000E+03	.200000E+05	.200728E+05	-.727812E+02	-.362587E+00
.890000E+03	.250000E+05	.250285E+05	-.284766E+02	-.113777E+00
.105600E+04	.300000E+05	.297563E+05	.243676E+03	.818904E+00
.125000E+04	.350000E+05	.352816E+05	-.281641E+03	-.798264E+00
.141300E+04	.400000E+05	.399240E+05	.759531E+02	.190244E+00
.158600E+04	.450000E+05	.448513E+05	.148734E+03	.331617E+00
.177000E+04	.500000E+05	.500918E+05	-.917734E+02	-.183211E+00

STD ERROR OF ESTIMATE FOR Y = .164383E+03

CYLINDER NO. 2 - PULLING

FORCE = 28.481014 PSI - 319.625 LB_F

FIGURE 62. COMPUTER PRINTOUT OF FORCE VERSUS PRESSURE CYLINDER 2 PULLING.

TABLE 16. INSTRUMENTATION LIST FOR TRAIN IMPACT TESTS

Instrument	Manufacturer	Model	Range	Output Nominal	Linearity	Response
Strain Indicator	BLH Electronics	120	±30,000 μ in./in.	N/A	5 μ in./in.	DC
Load Cell	Interface	1320-AF-50K	50K lb	4MV/V/Range	.2% F.S.	>10 KHz
Load Cell	Interface	1330-AF-100K	100K lb	4MV/V/Range	.2% F.S.	>10 KHz
Displacement Transducer	Celesco	PT-101-30	30 in.	8MV/V/Range	.1% F.S.	100G Acceleration of Cable
Pressure Gauge	Marsh Instrument Company	Type 100	0-6,000 psi	-	.25% F.S.	-
Dial Indicator	Starrett		0-1 in.	-	-	-

380 copies

☆U.S. GOVERNMENT PRINTING OFFICE: 1976—700-866/159