track-train dynamics

Harmonic Roll Series

VOLUME 4

Torsional and Flexural Car Stiffness Characteristics

An International Government—Industry Research Program on Track-Train Dynamics

AAR

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R-184

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Harmonic Roll Series Volume 4

An International Government—Industry Research Program on Track-Train Dynamics

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E.F. Lind

E.F.LIND,
Project Director - Phase I
Track Train Dynamics Program

FOREWORD

In recent years, service demands on the railroad industry have undergone significant changes. As a result, a number of important design innovations have been introduced to the freight car fleet. The resultant has been the construction of a large number of heavy, high volume freight cars. As the utilization of this type of equipment has increased, so have the operating problems. One important problem is that of harmonic roll, most commonly referred to as "Rock and Roll".

There are several synonymous terms for harmonic roll that may be used throughout the text. They are:

Harmonic Roll - Rock and Roll
Car Rocking
Lateral Instability

In general, the rock and roll problem is related to the operation of high capacity, high center of gravity freight cars over track that has an uneven surface. The problem is most predominant on track that has surface variation due to alternately staggered joints. When operating over rough track of this nature at speeds usually between 15-25 mph, excessive carbody roll may be developed. Energy is added to the moving system with each roll cycle, and, if the car suspension does not have adequate damping, extreme carbody roll will develop, resulting in wheel lift and probably derailment.

A general solution to the rock and roll problem involves an extensive study of the entire system consisting of track, suspension, and carbody design. A study of this scope results in a better understanding of the mechanics and dynamic interactions of a freight car in its operating environment. The result can be a freight car that not only meets marketing requirements, but one that can be operated safely with minimal problems.

In order to begin investigation of track-train dynamics problems, the Southern Pacific under contract to the AAR sent a questionnaire to sixteen selected railroads in mid-1971. From the responses received, it was realized that two serious problems faced the railroads. These problems were

(a) Rock and Roll
(b) Sudden gage widening and rail roll-over

Keeping this in view and realizing work was needed in these areas, the planners of Phase I of the track-train dynamics research program designated Task 13, Special Projects, to handle these areas. With respect to rock and roll, the primary objectives of Task 13 were to

(1) Develop guidelines which could be used by individual railroads to assist them in minimizing rock and roll problems
(2) Develop a document to be used as a reference in solving rock and roll problems
(3) Develop a model for computer simulation of freight car dynamics
(4) Develop characteristics of common freight car trucks and their related components
(5) Develop a log of freight car characteristics critical in designing stable freight cars
(6) Develop comparisons of important parameters controlling rock and roll by using existing computer simulations

By using the information developed from the primary objectives, certain secondary goals could be accomplished during Phase II of the overall research program. These goals will be to

(1) Develop car design specifications
(2) Develop specifications for truck design
(3) Aid in developing laboratory procedures for evaluation of damping devices
(4) Define the function of the freight car truck within the dynamic system as related to transfer of energy from track to carbody
(5) Perform simulations to evaluate new car designs

To present the findings related to the primary objectives of Task 13 outlined above, it was decided to present a series of harmonic roll related documents to the industry to be used as reference material in a similar manner as the Track-Train Dynamics Bibliography. This document, Volume IV of the Harmonic Roll Series, presents experimental test data and calculated values characterizing flexural stiffness and torsional rigidity of various types of vehicles. This data in turn will be incorporated into our mathematical models to simulate the dynamic responses of freight cars to various operating conditions.

PART I

ACF Industries, Amcar Division Participation In
AAR-RPI-FRA Track Train Dynamics Program

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PART I

TORSIONAL STIFFNESS AND FLEXURAL RIGIDITY TESTS

ACF Industries, Amcar Division Participation
In AAR/RPI/FRA Track-Train Dynamics
Program

INTRODUCTION

A Track Train Dynamics Program has been instituted as a cooperative effort of the Association of American Railroads and the Railway Progress Institute (AAR/RPI) whereby computer simulation of dynamic freight train behavior may be obtained from the input of certain key vehicle parameters, e.g. torsional and flexural rigidity, center of gravity of loaded and unloaded cars and rotational inertias.

Dr. Greg Martin of the AAR Technical Center is responsible for the overall dynamic simulation, and Mr. George E. Reed, Director of Railroad Sales who is also Chairman of the RPI Technical Subcommittee on Rolling Stock, is coordinating Amcar's effort in the program.

The original request for determination of these key vehicle parameters was directed to Mr. R. H. Billingsley, Jr., Amcar's Director of Engineering and Research, by Mr. E. F. Lind, Project Director of the Track Train Dynamics Program, by letter dated February 27, 1973. As a result, a test program was initiated to experimentally determine the torsional and flexural rigidities for the following types of cars:

a. 40-ft. 50-ton box car.
b. 60-ft. 100-ton box car (cushioned).
c. 86-ft. 70-ton hi-cube box car.
d. 70-ton TOFC flat car.
e. Model 4600 100-ton covered hopper.

These cars were supplied through the efforts of the AAR who have agreed to accept calculated values for the center of gravity and rotational inertia parameters.

This report deals solely with the tests which were conducted to determine torsional and flexural rigidities of the five cars for the program.

OBJECT

To experimentally determine the torsional and flexural rigidity of three box cars, one covered hopper car and one flat car in conjunction with the AAR/RPI Track Dynamics Program.

CONCLUSIONS AND RECOMMENDATIONS

The shorter cars displayed a larger value of torsional rigidity than the longer ones. The ACF flat cars was the least rigid of all the cars tested. The Center Flow was stiffer in torsion than the 60 foot box car because of its unibody type construction. The torsional rigidity of the ACF 40 foot box car (171×10^7 lb.-in./radian) may not be precise since the amount of twist was very small. The Thrall Hi-Cube and the ACF 60 foot box car had torsional rigidities of the same magnitude. These two cars also exhibited approximately the same flexural stiffness. The ACF Center Flow showed the highest flexural stiffness of the five cars tested (918,800 lb./in.), although this value may be slightly low due to deflection of the rails on the transfer table.

It is recommended for future tests of this type that car body deflections be measured at the center of the truck bolsters, at the jacking pads, and at the center plus two intermediate points along the longitudinal centerline of the car. If additional torsion testing is also performed, a technique for measuring car body twist should be developed to determine the torsional rigidity constant with greater accuracy.

DESCRIPTION OF TEST ITEM

The five cars used for determining torsional rigidity and flexural stiffness in conjunction with the AAR/RPI Track Train Dynamics Program were furnished through the efforts of the AAR. All except one were ACF built. The exception was the 86 foot hi-cube box car which was built by the Thrall Car Manufacturing Company. The cars, in order of test, are listed below along with a brief description and any pertinent comments relative to their condition at the time of testing.

I. Flat car, 70 ton, 89 foot, ACF built, TOFC-COFC, STOX 763, new 5-73. This unit had a capacity of 149,400 pounds and a lightweight of 70,600 pounds. A photograph of the car is included as Figure 1. The car body was 108.25 inches wide and measured 1068.42 inches over the end sills. Distance between truck centers was 793 inches. The car was in excellent condition and appeared to have seen very little service.

II. Box car, 100 ton, 60 foot, ACF built, N&W 600976, new 12-69. This unit had a capacity of 184,000 pounds, a load limit of 185,000 pounds and a lightweight of 77,100 pounds. The serial number stamped on the 'A" end of the sliding center sill was 11-06354. It was of cushion underframe construction equipped with an ACF 20B Freight-Saver. It had double sliding doors and a volume of 6460 cubic feet. The car length over end sills was 731 inches, a width of 119.8 inches and measured 555 inches between truck centers. Both sides of the car were damaged to the extent that operation of the doors was very difficult. The car was loaded with 17 tiers of automobile transmission racks (5 per tier) as shown in Figure 2. Each of the eighty-five racks weighed approximately 555 pounds for a total weight of 47,175 pounds. The car was tested with the three center tiers removed to gain access to the floor area at the center of the car. The air brake hose bracket which is fastened to the underside of the coupler carrier was broken loose at both ends of the car - this was later repaired by welding. A side sheet seam weld, approximately ten inches long, had failed near the A-right corner of the car. This was also repaired by welding. When the trucks were removed from the car, the body bolster center plates on both ends were dye-checked. No cracks were detected on either center plate.

III. Box car, 70 ton, 86 foot, Thrall built, N&W 355230, new 10-65. This was 10,000 cu. ft. hi-cube box car with a capacity of 100,000 pounds, a load limit of 106,900 lbs., and a lightweight of 113,000 pounds. A photograph of the car is shown in Figure 3. It was of cushion underframe construction and had double plug doors. The car length over the end sills was 1038 inches, a width of 112 inches and measured 769 inches between truck centers. The sides of this car were also damaged making door operation rather difficult. Both ends of the sliding center sill including the coupler carriers had been severely damaged due to by-passed couplers. No attempt was made to repair any of this damage.

IV. Box car, 50 ton, 40 foot, ACF built, W.A.B. 6335, new 5-56. This was a 3885 cubic foot car with a capacity of 110,000 pounds, a load limit of 126,900 lbs. and a lightweight of 50,100 pounds. The car length over the end sills was 489 inches, the width was 119 inches and it measured 371 inches between truck centers. A photograph of the car is included as Figure 4. It had single sliding doors which were 8 ft. wide and 10 ft. high. The car had a wood plank floor and was in very good condition for its age.

V. Covered hopper, 100 ton, 50 foot, ACF Center Flow, N&W 171108. This was a three compartment, 4650 cu. ft. car with a capacity of 200,000 lbs., a load limit of 201,600 lbs., and a lightweight of 61,400 lbs. A photograph of the car is included as Figure 5. The car was built in May 1966 and was stamped with a 'steel number' of 10776. It was of the continuous hatch design and equipped with three gravity outlets and measured 604.5 inches over the end sills, 121 inches over the side sills and 496 inches between truck centers. The car interior had a Polyclutch lining which was applied at the time of manufacture.

It was in excellent condition. Cracks in the roof panel were observed at the juncture of both interior bulkheads adjacent to the hatch opening. This would account for the water streaking which was present on both sides of each interior bulkhead. No repair work was performed in this area.

METHOD OF TEST

Each car in succession was first tested to determine its torsional rigidity and then tested to determine its flexural stiffness. The general method of torsional testing for each car was to place the car on a level track in the laboratory and disconnect any of the brake linkage mechanism which might interfere with raising the car body clear of the trucks. Load cells were then placed on two jacks which were located at diagonal jacking pads at the centerline of the trucks. A clinometer was located at each end of the car at the body bolster centerline (longitudinally to length of bolster) to measure car body twist. A zero reading was recorded for each clinometer and each load cell. The jacks were then extended raising the car body 2½ to 3 in. or until the body bolster center plate was clear of the truck bolster center bowl. Readings of each clinometer and load cell were recorded. The jacks were then retracted until the body was again supported by the trucks. From this data the jacking loads and corresponding car body twist was determined from which torsional rigidity values were calculated.

The cars were then prepared for the flexure test. The general method of flexure testing for the flat car and the box cars was to rigidize the trucks by replacing the springs with 9 inch lengths of 4 inch schedule 40 pipe. The cars were then positioned in the strain frame and a specially fabricated beam (Figure 6) was placed directly on the floor at the transverse centerline of the car. Two 1½ inch diameter rods five feet long, which have threaded ends, were inserted through the ends of the beam - through the floor of the car and into a short intermediate beam beneath the car. Nuts were threaded onto each end of the rods. Two additional rods were similarly attached to the intermediate beam from which they extended downward through a rigidly fixed cross member of the strain frame and through the two center hole hydraulic loading rams. This set-up is shown in Figure 7 as it was attached to the ACF TOFC-COFC flat car.

Dial indicating gages were located at each side sill and both center sill flanges at the transverse centerline of the car to measure car body deflection. The average car deflection at the centerline, for the maximum load was then used to calculate the flexural stiffness. Since the method of test for each car varied slightly, the following paragraphs explain in detail the exact procedure used.

I. Flat car, 70 ton, TOFC-COFC. The first attempts at jacking the flat car for the torsion test were unsuccessful. The car would not balance since the 'B' end was heavier. Balance was obtained by placing a 10 kip weight on the car deck and adjusting it along the longitudinal centerline. The final position for the weight was 65 inches from the transverse centerline toward the 'A' end. This weight is shown in the photograph included as Figure 1. Car body twist was first measured at the truck centerline along the longitudinal centerline of the car. Several runs were made and good repeatability was obtained. In order to determine if the body bolster was deflecting, which would produce unreliable angular measurement data, three clinometers were placed along the bolster centerline (transverse to length of car) at the 'A' end and the jacking test was repeated. The car body twist agreed within one minute of arc as recorded with the three clinometers. It was therefore assumed that the body bolster was only rotating and not deflecting during the torsion test.

The deflection measurements for the flexure test were obtained by placing the dial indicators on the car deck at each side sill and over the center sill on top of the short loading beam. The dimensions were reference to a 4 in. x 6 in. tubular steel beam which spanned the car deck and was anchored to the side rails of the strain frame. This setup is shown in the photo which is included as Figure 8. The 10 kip weight used for the torsion test, was removed before flexure testing began. After the flexure test was completed, the holes in the deck were repaired, the springs were replaced in the trucks and the car was prepared for shipping.

II. Box car, 100 ton, N&W 600976. The three center tiers of transmission racks were removed for both the torsion and flexure tests. Car body twist for the torsion test was measured by placing the clinometers on a bracket which was rigidly clamped to the body bolster on both sides of the center plate. A sketch of this bracket is included as Figure 9 and it can also be seen installed on the car in the photograph of Figure 10. At the 'B' end of the car, a 14 inch length of 2 inch angle was also welded to to the jack pad opposite to that of the lifting jack. A clinometer was attached to the angle and readings were compared after the car was raised clear of the center plates. Both readings were identical, which indicated the bolster was only rotating and not deflecting.

When the torsion test was complete, the truck springs were removed and the car was positioned in the strain frame for flexure testing. Holes were cut through the car floor and the fabricated beam was installed as shown in Figure 6. Dial gages were placed beneath each side sill and on each flange of the fixed center sill at the center of the car. Using the hydraulic rams, a 70 kip vertical down load was applied to the beam in four increments while dial gage readings were recorded. The load was cycled twice prior to relocating the dial gages to measure the sill deflections beneath the 'A' end door posts. The 70 kip vertical loading was again applied and deflection measurements recorded. The car was then prepared for shipping by repairing the floor, replacing the transmission racks and reinstalling all truck springs.

III. Box car, 70 ton, N&W 355230. The Thrall hi-cube car was tested to determine its torsional rigidity in the same manner as the previous two cars. Clinometers were attached to short lengths of angle which were welded to the body bolster jacking pads opposite the lifting jacks at each end of the car. The clinometer bracket was not attached to the body bolster for this or the following cars since it was previously determined that the body bolsters were not deflecting in this type of test.

Flexural rigidity was also determined for this car as it was for the first two cars. The springs were removed and lengths of pipe were inserted to rigidize the truck bolsters. Deflection measurements were recorded at 10 kip load increments to a maximum of 75 kip at the center of the car with the doors open and also with the doors closed. Two cycles were recorded with the doors open. One cycle at each door position was also recorded at the 'A' end doorpost. At the completion of the tests, the car was restored and prepared for shipping.

IV. Box car, 50 ton, W.A.B. 6335. The clinometers were attached to angles on the body bolsters to measure car body twist for this car also. For flexure testing the springs were removed from the trucks and the fabricated beam was installed on the floor at the center of the car. The car doors were closed for both torsion and flexure testing. Dial gages were located below the side and center sills at the center of the car. Two dials were also located to measure deflection of the body bolster relative to the strain frame member at the 'B' end of the car. Loading was applied in 10 kip increments to a maximum of 60 kip and deflections were recorded at each increment. After the first loading cycle, a gap was noticed between the bottom of the loading beam and the surface of the car floor. This was due to the floor being cupped. Therefore for the second cycle, shims were inserted to fill the gap. Deflections of the side and center sills was not measured at the door post location for this test. After testing the car was restored and made ready for shipment.

V. Covered hopper, 100 ton, N&W 171108. For torsion testing the clinometers were attached to the body bolsters as described previously. A 1650 pound block of concrete was placed on the shear panel at the 'A' end of the car (Figure 11). Without this additional weight, the car could not be made to balance with jacks located at the jacking pads. The continuous hatch covers were in place during the torsion test. The car was prepared for flexure testing by removing the truck springs and replacing them with 9 in. lengths of pipe, the outlets were removed and flat steel cover plates were installed in their place (Figure 14). The empty car was then moved to the track scale and the weight was recorded. The car was placed on the transfer table at the North end of the laboratory (Figure 5) and seven dial gages were positioned to measure car body deflection (Figures 12 and 13). Two gages were located at the center of the car, two at the jack pads at the 'A' end, and two midway between the center of the car and the 'A' end bolster centerline. The seventh gage was located between the truck bolster and the transfer table directly under the center plate pin. A zero reading was recorded for each gage and then all three compartments were filled with water to a height of 79½ in. below the underside of the hatch opening. All of the deflection gage readings were recorded and the car was moved to the track scale and reweighed to determine the amount of water contained in the three compartments. The water was then drained from all three compartments, the outlets and the truck springs were replaced and the car was made ready to be shipped. The water remained in the three compartments for a total elapsed time of five hours.

INSTRUMENTATION

In performing the torsional rigidity tests on all five cars, the suspended vertical car body forces were measured by using two 50,000 pound capacity bonded strain gage type load cells. One load cell was a Baldwin-Lima-Hamilton type "C" compression only, 120 ohm bridge, serial number 28001, having a full scale sensitivity of 2 millivolts per volt. The cell was monitored using a Vishay Instruments, Model P-350AK digital strain indicator, serial number 7776. The calibration chart for the above load cell and strain indicator is included as Table I. The other cell was a Baldwin-Lima-Hamilton type U-1, universal tension compression unit, 120 ohm single bridge, serial number 4991, with a full scale sensitivity of 2 millivolts per volt. This cell was monitored using a Baldwin-Lima-Hamilton, type 20 digital strain indicator, serial number 25-4917. A calibration chart for this cell and indicator combination is included as Table II.

Angular twist or rotation of the five cars tested was measured with a M.C. Clinometer, Model TB 108-1, manufactured by Hilger and Watts Limited, London, England. It is a pendulum type clinometer fitted with a loaded, divided drum and vernier from which readings to an accuracy of one minute of arc can be obtained. Repeatability on any one setting is to one minute of arc.

In performing the flexure tests of the three box cars and the one flat car, midpoint loads were applied using two 60 ton hydraulic center-hole rams, Simplex Model RC-6010, each having an effective ram area of 13.75 in^2 and a 10 in. lift. These rams were hydraulically connected in parallel and pressure calibrated using a 100,000 pound, Southwark-Emery Universal Testing Machine, Model 100 BTE, serial number 65365, which was certified against secondary standards traceable to the National Bureau of Standards on October 2, 1974. Hydraulic ram pressure was sensed with a Baldwin-Lima-Hamilton 10,000 psi, type GP, bonded strain gage pressure transducer having a 350 ohm bridge and a full scale sensitivity of 3 millivolts per volt of excitation. The pressure cell was read with a Vishay Instruments Model P-350AK Digital Strain Indicator, serial number 7776. A calibration chart of the dual ram load versus pressure cell output in micro-inches per inch is included as Table III.

Loading for the flexure test of the Center Flow car was determined by filling all three compartments with water. The magnitude of the load was determined by weighing each end of the car, both before the after the compartments were filled. The car was weighed on the Amcar Technical Center weigh scale which is located on the rail siding at the East side of the laboratory. The scale was calibrated on April 4, 1974 by the Missouri-Kansas-Texas Railroad Company using scale test car number MKT-77 having a gross rail weight of 80,000 pounds.

Car body deflections, measured on all five cars to determine flexural rigidity, were recorded using Federal dial indicating gages which are accurate to within plus or minus 0.001 inch.

RESULTS

The torsional rigidity for all cars tested was determined by dividing the torque applied to the car body, as a result of its own weight when supported at diagonal corners, by the measured amount of angular rotation under the same condition. Specifically, T/θ was calculated by multiplying the average car body support load by the distance from the longitudinal centerline to the support points and dividing the product by the total angular rotation in radians of both ends of the car body. The support load includes the car body plus the weight of any lading present or loading added to achieve balance.

The flexural stiffness for each car was calculated by dividing the magnitude of applied load by the measured car body deflection at that load.

The equivalent JG was obtained by using the basic relation for determining the angle of twist of circular members, i.e.,

$$\theta = \frac{TL}{JG};$$

where the torque (T) is the average of the two support loads multiplied by the distance from the longitudinal centerline of the car to the support points, L is the distance between truck centers where the jacks are located and θ is the total angle of twist in radians as measured by the inclinometers.

To obtain the equivalent EI for the flat car and box cars, the elementary formula for calculating the deflection of a simply supported beam of uniform cross section with a concentrated load at mid-span was used, e.g.,

$$\Delta = \frac{PL^3}{48EI}$$

The equivalent EI for the Center Flow car was calculated from the deflection formula for a uniformly loaded, symmetrically supported beam with both ends overhanging the supports, e.g.,

$$= \frac{WA^2}{96EIL} \left[\frac{5A^2}{4} - 6c^2 \right]$$

where W is the total load, L is the overall length, A is the distance between supports and c is the length of overhang.

The calculated torsional rigidity and flexural stiffness values for each car are included as Table IV. The load versus

versus deflection data for the 40 foot box car is presented in Tables V and VI. The flexural stiffness value was calculated using the data from Table VI (with shims between beam and floor). The torsional rigidity of 171 x 10^7 lb.-in. per radian for this car could be in considerable error since the measured angle of twist was only 2 minutes (0.033 degrees) and the clinometer is accurate to plus or minus 1 minute of a degree. If the error is assumed to be in the low direction and the angle of twist was in fact 4 minutes, the torsional rigidity would still be greater than 85 x 10^7 lb.-in. per radian. For future torsional tests of short cars, a method should be devised to obtain either greater and/or more accurate twist angles.

Table VII contains two cycles of load-deflection data for the 89 foot flat car. Load versus deflection data for the 60 foot box car is presented in Table VIII, while Tables IX anc X present data for the 86 foot hi-cube with and without the doors open at the center and at the door post respectively.

Table XI contains the measured deflections along with the deflection gage locations for the Center Flow car. The flexural stiffness of this car (918,800 lb./in.) is probably slightly low because of the deflection of the rails on the transfer table. If this type of test is ever repeated, the car should be placed at a more rigid location or additional measurements should be made to accurately determine the track deflections.

TABLE I

CALIBRATION DATA

50,000 Pound Compression Load Cell

LOAD (pounds)	CYCLE I (microinches/inch)	CYCLE II (microinches/inch)
-0-	0	0
10,000	998	995
20,000	1997	1994
30,000	2993	2994
40,000	4002	4001
50,000	5005	5001
-0-	3	2

BLH, Type C, Compression Load Cell, 50,000 pound Serial No. 28001, 120 ohm, 2 mV/Volt Full Scale

Vishay, Type P-350AK DSI, Serial No. 7776, G.F. 1.60 Zero Balance (-1592 microstrain)

TABLE II

CALIBRATION DATA

50,000 Pound Universal Load Cell

LOAD (pounds)	CYCLE I (microinches/inch)	CYCLE II (microinches/inch)
-0-	25-4754	25-4755
10,000	25-3950	25-3955
20,000	25-3160	25-3167
30,000	25-2369	25-2375
40,000	25-1609	25-1590
50,000	25-0804	25-0803
-0-	25-4755	25-4755

BLH, Type U-1, Tension-Compression Load Cell, 50,000 pounds
Serial No. 4991, 120 ohm, 2 mV/Volt Full Scale

BLH, Type 20 DSI, Serial No. 25-4917, G.F. 2.00

TABLE III

CALIBRATION DATA

Two 60-Ton Center Hole Rams in Parallel

LOAD (pounds)	CYCLE I (microinches/inch)	CYCLE II (microinches/inch)
-0-	- 146	- 146
10,000	+ 91	+ 90
20,000	+ 311	+ 310
30,000	+ 530	+ 532
40,000	+ 750	+ 750
50,000	+ 972	+ 971
60,000	+1189	+1192
70,000	+1411	+1409
80,000	+1625	+1628
90,000	+1845	+1848
100,000	+2069	+2070
-0-	- 147	- 147

BLH, Type GP, Pressure Cell, 10,000 PSI, Serial No. 20863, 350 ohm, 3.0 mV/Volt Full Scale

Vishay, Type P-350AK DSI, Serial No. 7776, G.F. 2.00

TABLE IV

TORSIONAL RIGIDITY AND FLEXURAL STIFFNESS AAR/RPI TRACK TRAIN DYNAMICS PROGRAM

TYPE OF CAR	CAR NUMBER	LENGTH BETWEEN TRUCK CENTERS (in.)	DISTANCE CENTERLINE OF CAR TO SUPPORT (in.)	SUPPORT LOADS (lbs.)		ANGLE OF TWIST (minutes)		TORSIONAL RIGIDITY T/θ (lb.-in./radian)	EQUIVALENT JG =TL/θ (lb.-in.^{2}/radian)	VERTICAL LOAD (lb.)	AVERAGE DEFLECTION (in.)	FLEXURAL STIFFNESS P/Δ (lb./in.)	EQUIVALENT EI (lb.-in.^{2})
				'A'END	'B'END	'A'END	'B'END						
ACF TOFC Flat Car	STOX 763	793	52	31,200	30,981	50	42	6.04×10^7	4.79×10^{10}	70,000	2.168 (1)	32,300	3.35×10^{11}
ACF 100-Ton Box Car (Cushioned)	N&W 600976	555	57.9	47,977	47,380	24	25	19.4×10^7	10.8×10^{10}	70,000	0.393	178,100	6.34×10^{11}
Thrall 70-Ton Box Car (Cushioned)	N&W 355230	769	54	47,726	47,680	16	28	20.1×10^7	15.5×10^{10}	75,000	0.454 (2)	165,200	15.7×10^{11}
ACF 50-Ton Box Car	WAB 6335	371	57.5	17,449	17,197	1	1	171×10^7 (3)	63.5×10^{10} (3)	60,000	0.101 (4)	594,000	6.32×10^{11}
ACF 100-Ton Center Flow	N&W 171108	496	58.5	21,629	21,870	4	5	48.6×10^7	24.1×10^{10}	105,663 (5)	0.115	918,800	11.3×10^{11}

(1) Average of two runs.
(2) Car doors closed.
(3) May be in error due to very small twist angle.
(4) Shims between loading beam and floor.
(5) Uniformly distributed load.

TABLE V

LOAD VERSUS DEFLECTION, RUN NO. 1
50-TON ACF BOX CAR, W.A.B. 6335
(Gap Between Loading Beam and Floor)

LOAD (pounds)	DEFLECTION (inch)					
	Gage 1	Gage 2	Gage 3	Gage 4	'B' Left	'B' Right
10,000	.023	.023	.023	.035	.022	.013
20,000	.053	.048	.048	.069	.042	.029
30,000	.089	.074	.075	.104	.061	.046
40,000	.121	.098	.098	.135	.079	.063
50,000	.154	.123	.124	.169	.098	.078
60,000	.184	.147	.149	.203	.115	.093
-0-	.006	.008	.007	.014	.013	.004

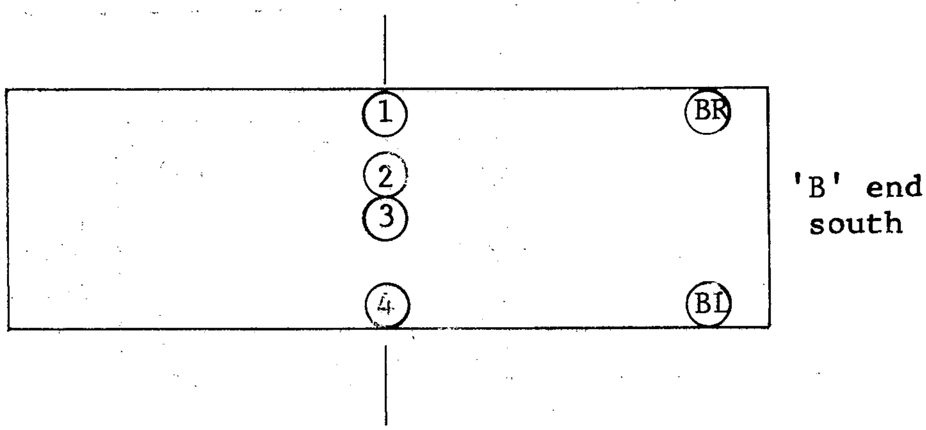


TABLE VI

LOAD VERSUS DEFLECTION, RUN NO. 2
50-TON ACF BOX CAR, W.A.B. 6335
(Shims Between Loading Beam and Floor)

LOAD (Pounds)	DEFLECTION (inch)					
	Gage 1	Gage 2	Gage 3	Gage 4	'B' Left	'B' Right
10,000	.024	.044	.051	.027	.018	.017
20,000	.055	.082	.089	.057	.035	.035
30,000	.087	.117	.126	.085	.051	.053
40,000	.117	.153	.162	.114	.067	.069
50,000	.146	.188	.196	.144	.082	.085
60,000	.175	.221	.230	.174	.098	.100
-0-	.012	.010	.008	.004	-.003	.011

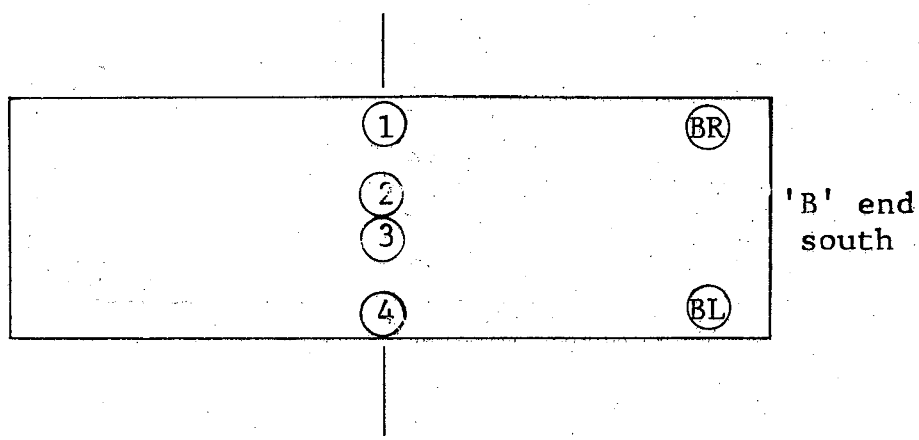


TABLE VII

LOAD VERSUS DEFLECTION
ACF, TOFC-COFC, STOX 763

LOAD	DEFLECTION (Run No. 1)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.284	.306	.307	.289
20K	.587	.608	.619	.591
30K	.891	.910	.935	.895
40K	1.195	1.220	1.244	1.199
50K	1.501	1.529	1.552	1.503
60K	1.820	1.847	1.872	1.825
70K	2.157	2.190	2.211	2.164
Zero	0.042	0.044	0.043	0.040

LOAD	DEFLECTION (Run No. 2)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.277	.277	.293	.271
20K	.580	.579	.604	.574
30K	.880	.882	.910	.877
40K	1.183	1.189	1.215	1.181
50K	1.489	1.495	1.521	1.486
60K	1.806	1.815	1.841	1.805
70K	2.142	2.156	2.180	2.142
Zero	0.050	0.052	0.052	0.049

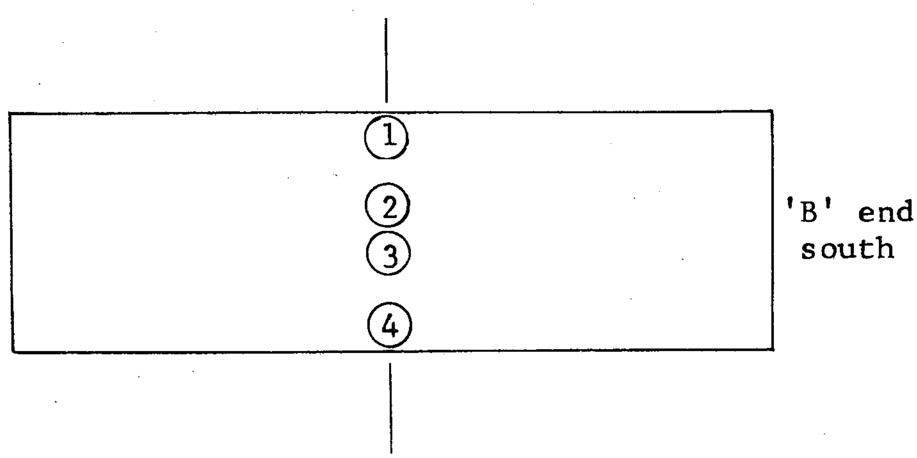
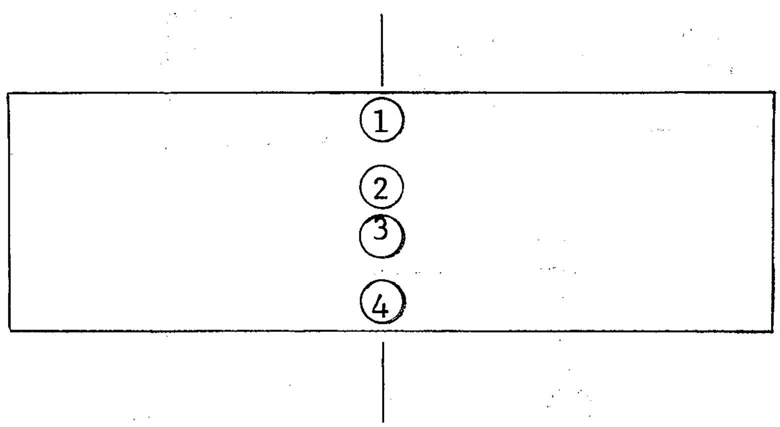


TABLE VIII

LOAD VERSUS DEFLECTION
ACF, 60 Foot, 100-Ton Box Car, N&W 600976

LOAD	DEFLECTION AT CENTER OF CAR			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.053	.046	.053	.055
30K	.170	.151	.164	.175
50K	.288	.261	.276	.298
70K	.402	.371	.384	.413
Zero	.022	.014	.013	.011

LOAD	DEFLECTION AT 'A' END DOOR POST			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.023	.032	.034	.030
30K	.072	.100	.105	.090
50K	.127	.172	.177	.148
70K	.182	.242	.249	.206
Zero	.001	.002	.001	.002



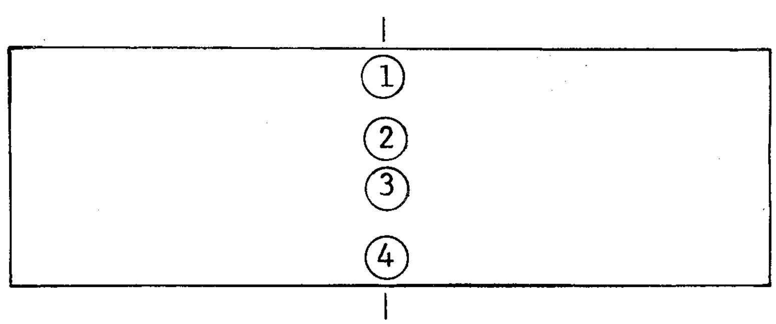
'B' end
south

TABLE IX

LOAD VERSUS DEFLECTION, CENTER OF CAR
Thrall - 86 Foot Box Car, N&W 355230

LOAD	DEFLECTION (Doors Closed)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.043	.052	.053	.063
20K	.089	.110	.113	.137
30K	.136	.168	.176	.217
40K	.185	.226	.238	.296
50K	.238	.286	.301	.372
60K	.290	.346	.363	.447
70K	.344	.407	.424	.519
75K	.371	.437	.453	.554
Zero	.013	.012	.013	.017

LOAD	DEFLECTION (Doors Open)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.040	.057	.058	.070
20K	.086	.115	.121	.153
30K	.134	.175	.186	.237
40K	.191	.235	.246	.309
50K	.250	.296	.306	.380
60K	.306	.357	.367	.451
70K	.357	.415	.425	.518
75K	.386	.445	.445	.552
Zero	-0-	.002	.002	.003



'B' end
south

TABLE X

LOAD VERSUS DEFLECTION, 'A' END DOOR POST
Thrall - 86 Foot, Box Car, N&W 355230

LOAD	DEFLECTION (Doors Closed)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.027	.036	.037	.035
20K	.051	.070	.074	.075
30K	.077	.105	.114	.115
40K	.105	.140	.150	.151
50K	.145	.176	.186	.184
60K	.166	.211	.222	.217
70K	.195	.245	.256	.248
75K	.210	.262	.274	.263
Zero	.002	.002	.002	.202

LOAD	DEFLECTION (Doors Open)			
	Gage 1	Gage 2	Gage 3	Gage 4
10K	.026	.035	.035	.032
20K	.050	.069	.073	.071
30K	.077	.104	.110	.110
40K	.107	.140	.146	.143
50K	.140	.176	.182	.175
60K	.172	.212	.218	.206
70K	.203	.247	.253	.237
75K	.220	.624	.271	.253
Zero	-.002	-.002	-.002	-0-

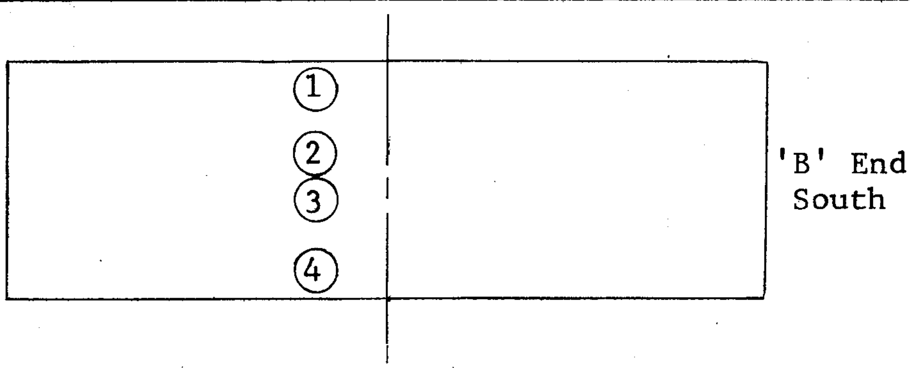
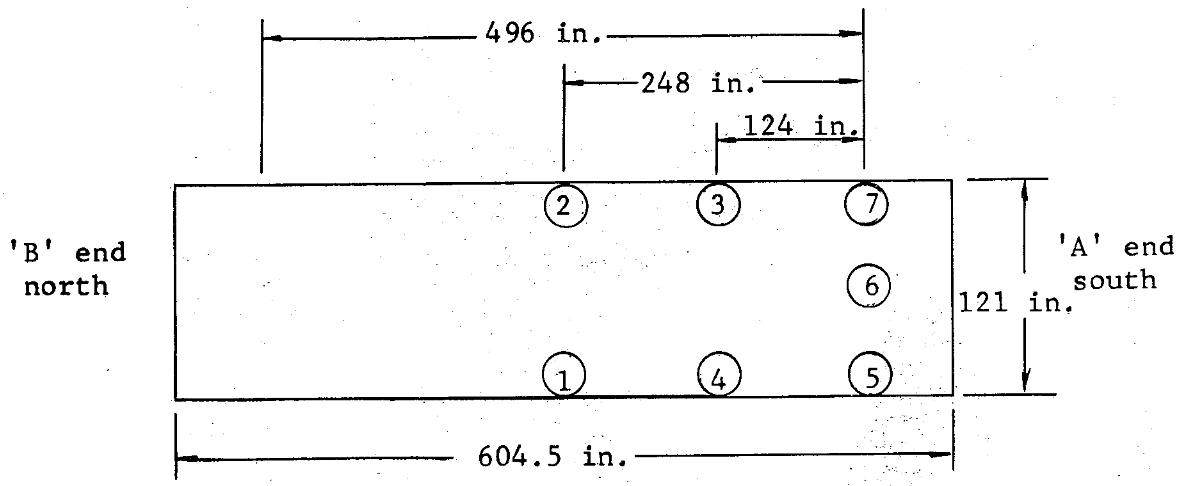


TABLE XI

FLEXURAL STIFFNESS DATA
100-Ton ACF Center Flow, N&W 171108



All three compartments filled with water to 79.5 inch of underside of hatch opening.

Scale weight of empty car - 62,163 pounds
Scale weight of loaded car - 167,826 pounds
Weight of water - 105,663 pounds (12,668 gallons)

GAGE NUMBER	1	2	3	4	5	6	7
Deflection (inches)	.162	.175	.120	.105	.065	.054	.079

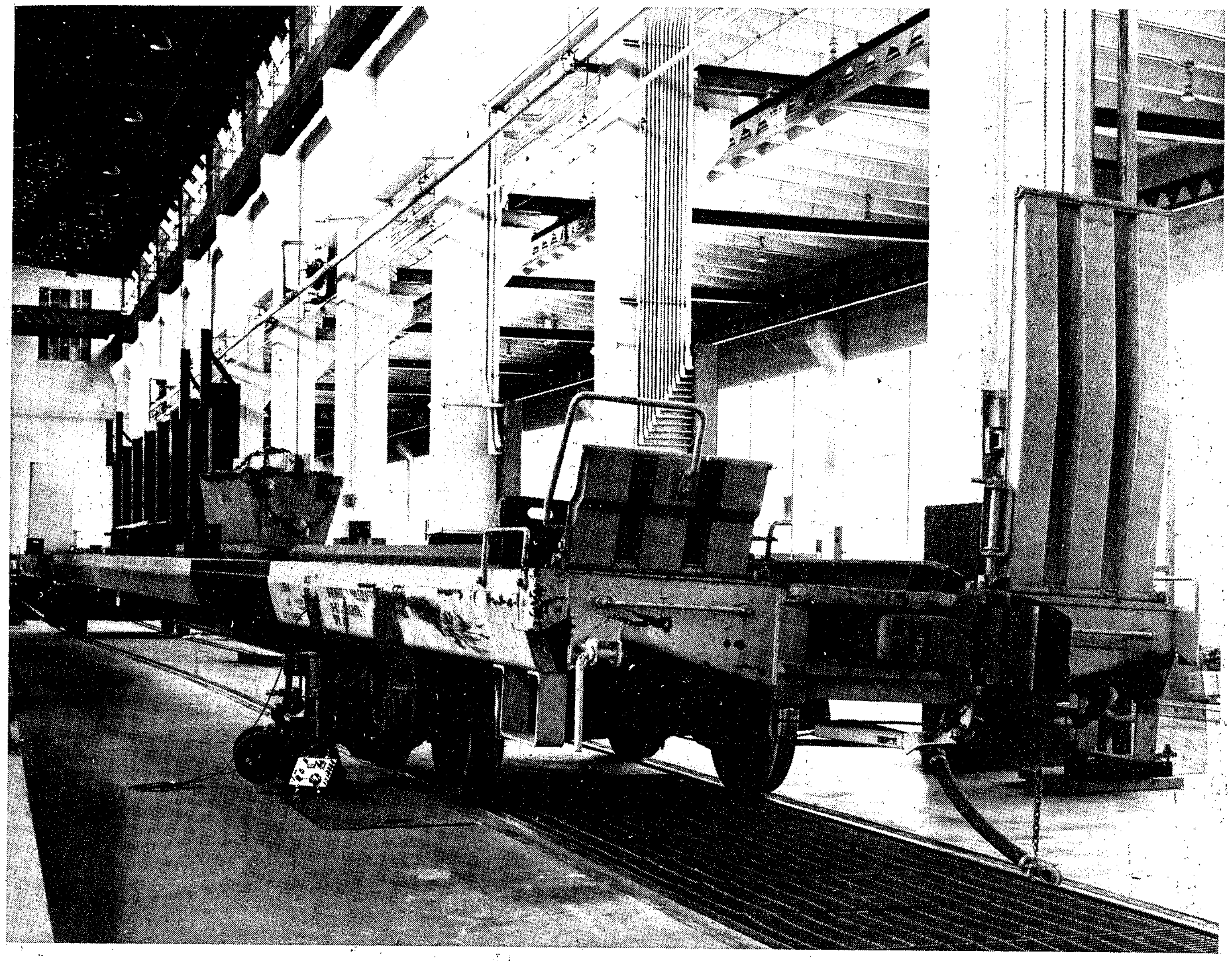
Average midpoint deflection minus bolster deflection =

$$\frac{(.162) + (.175)}{2} - .054 = .115 \text{ inch}$$

$$P/\Delta = \frac{105.663 \text{ lb.}}{0.115 \text{ in.}} = 918{,}800 \text{ lb./in.}$$

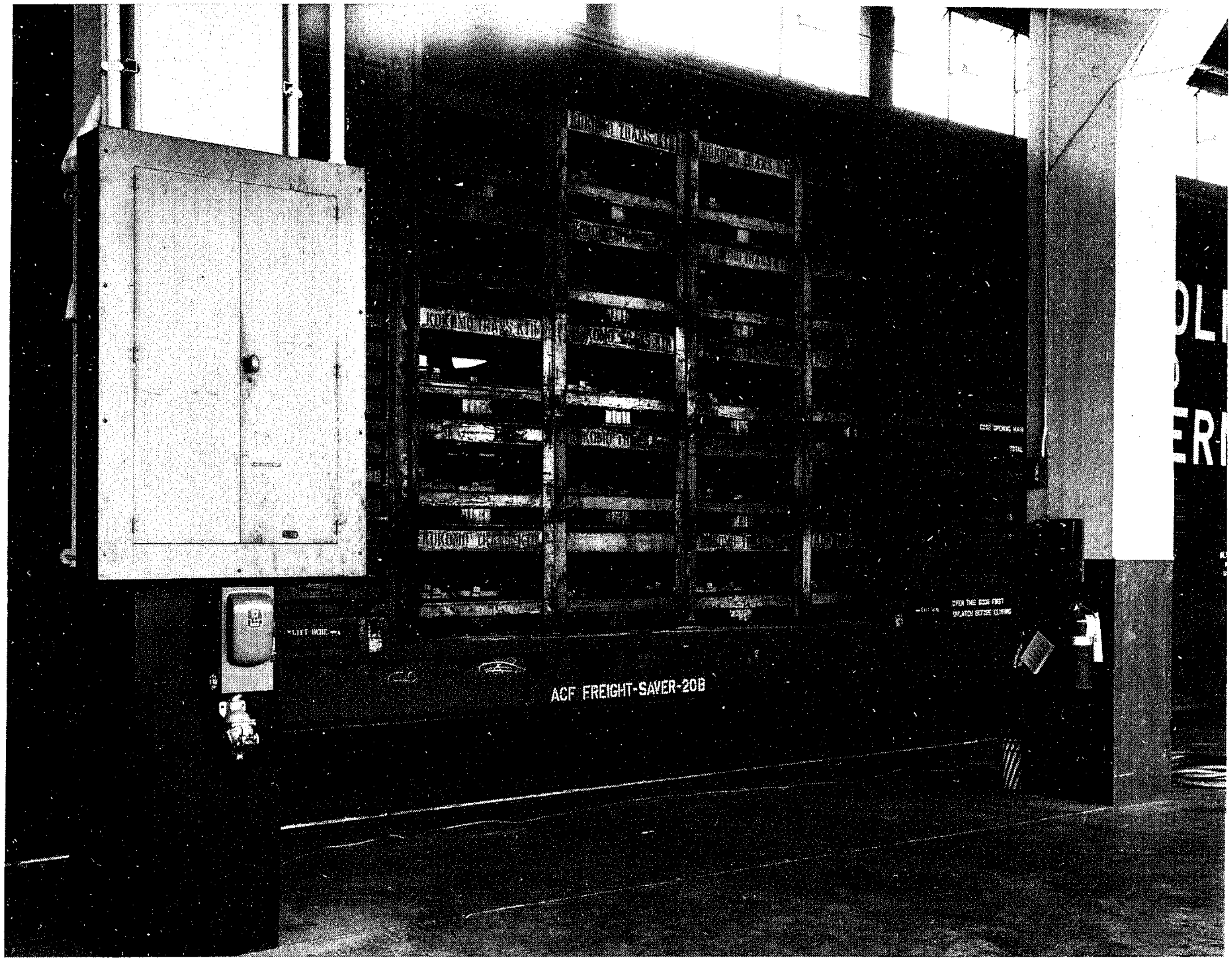
$$\text{Equivalent EI} = \frac{WA^2}{96L\Delta}\left[\frac{5A^2}{4} - 6C^2\right] = \frac{105{,}663 \text{ lb.}(496 \text{ in.})^2}{96(604.5 \text{ in.})0.115 \text{ in.}}$$

$$\left[\frac{5(496 \text{ in.})^2}{4} - 6(54.25 \text{ in.})^2\right] = 11.3 \times 10^{11} \text{ lb.-in.}^2$$



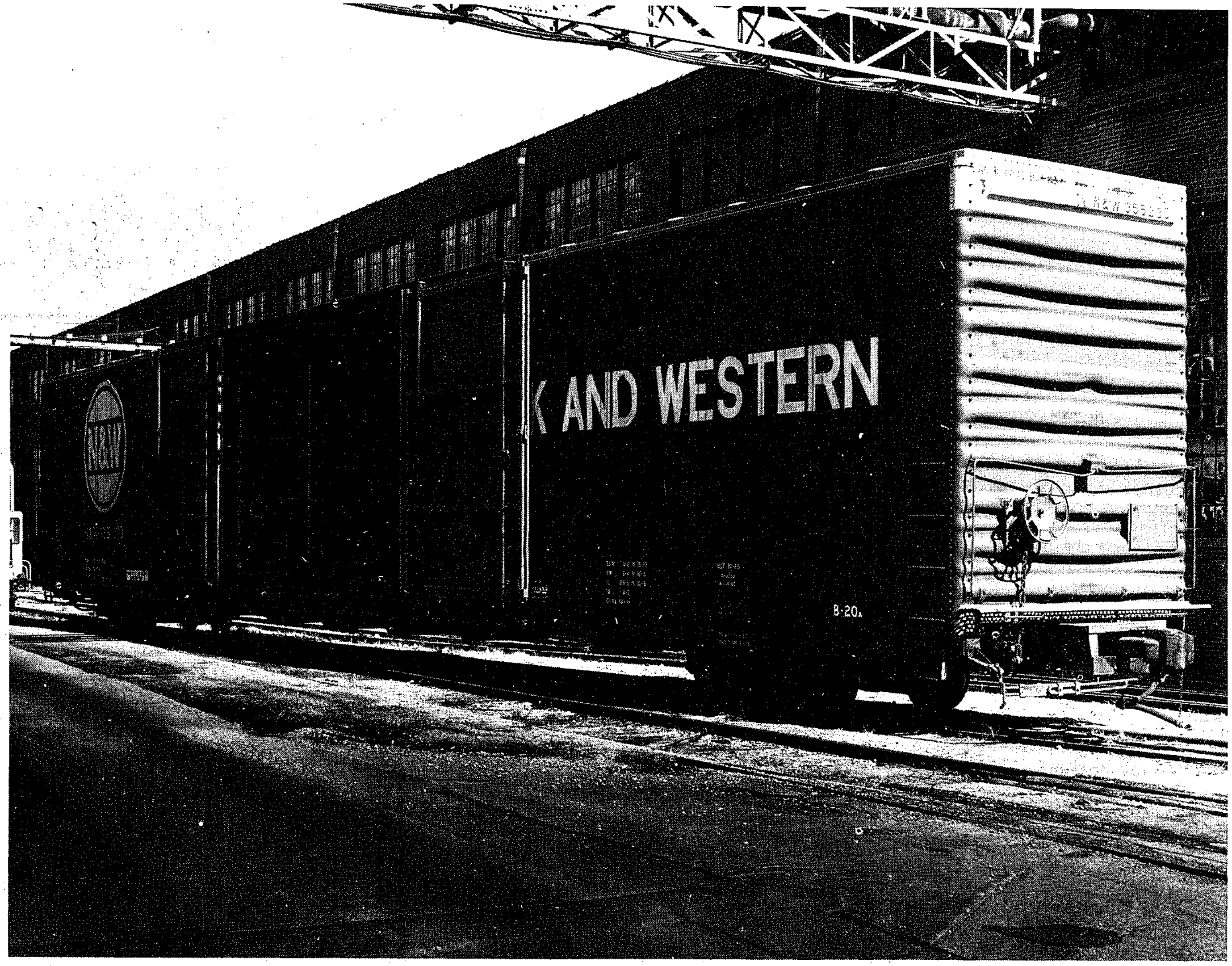
ACF, TOFC-COFC, STOX - 763

Figure 1



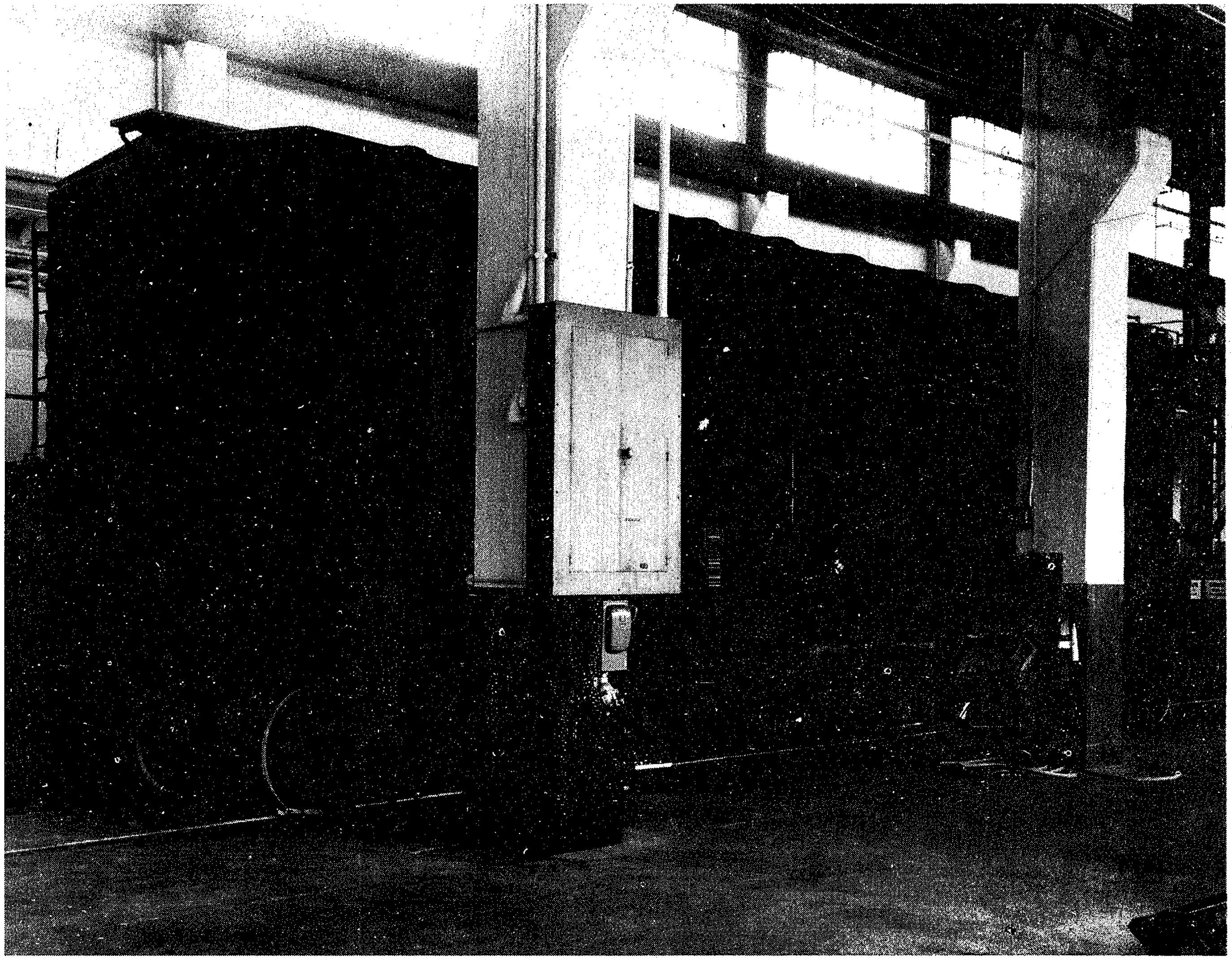
ACF, 60 FOOT, 100 TON, N&W 600976

Figure 2



THRALL, 85 FOOT, 70 TON, HI-CUBE, N&W 355230

Figure 3



ACF, 40 FOOT, 50 TON, W.A.B. 6336

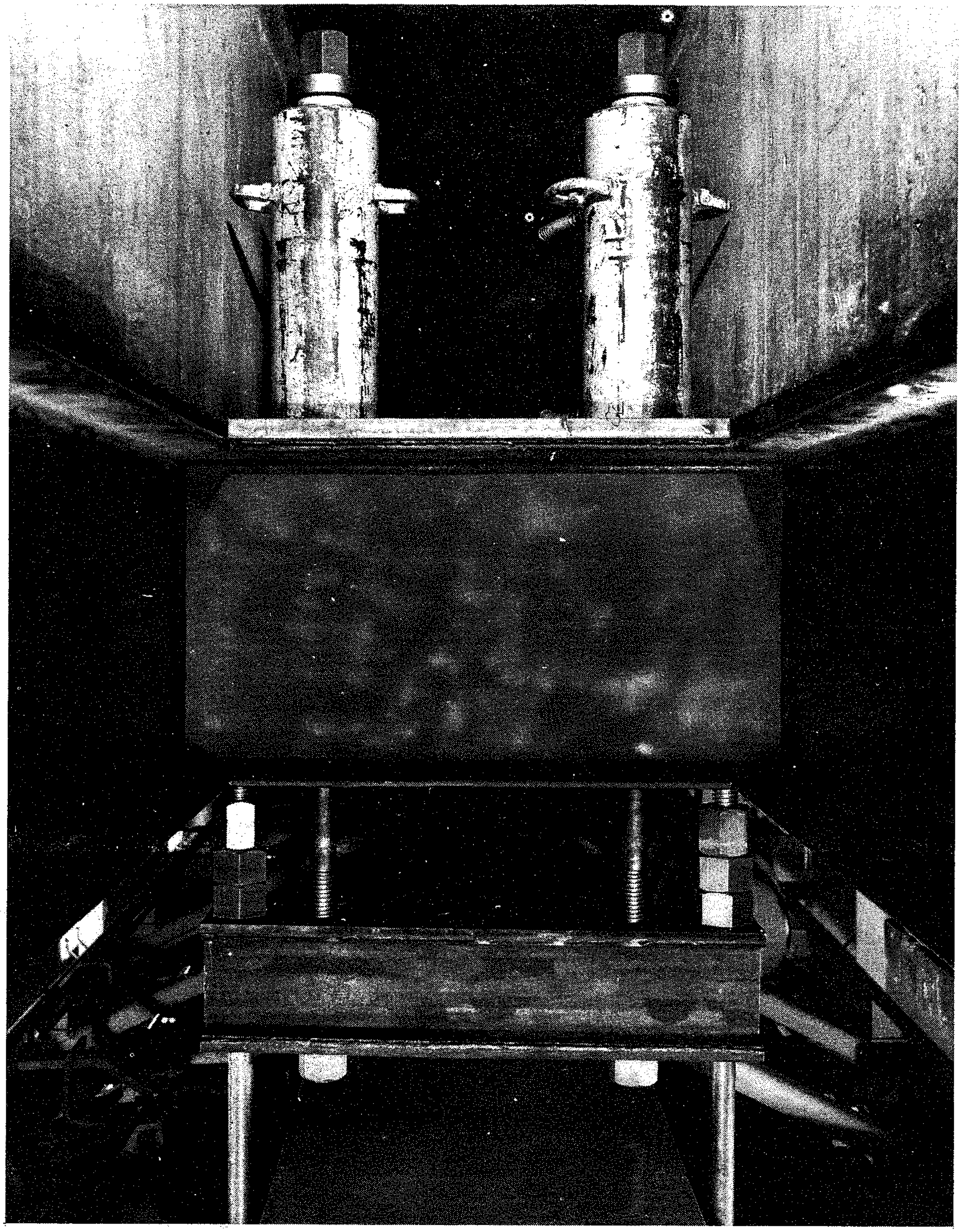
Figure 4

ACF CENTER FLOW, N&W 171108

Figure 5

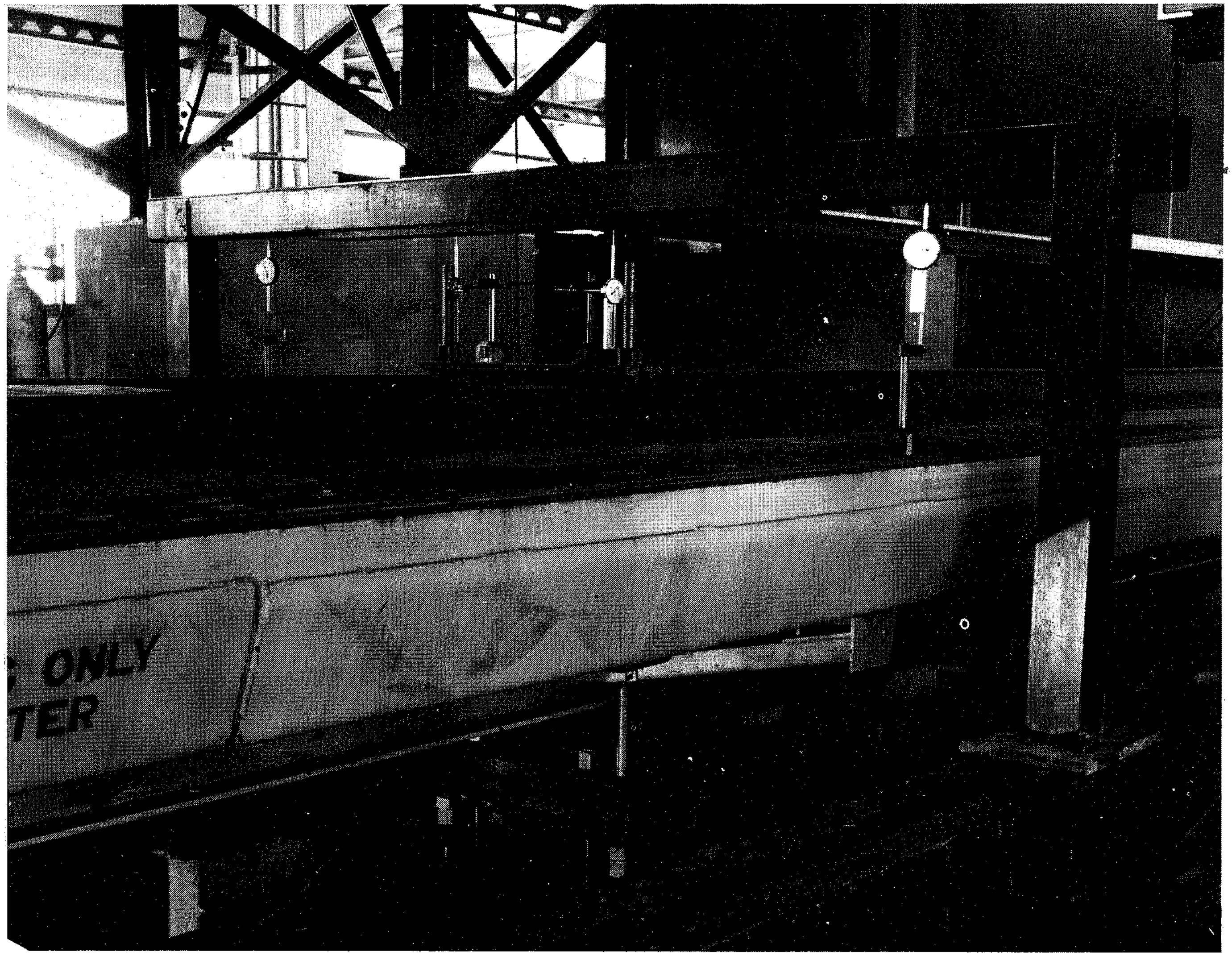
FLOOR LOADING BEAM INSTALLED IN N&W 600976

Figure 6



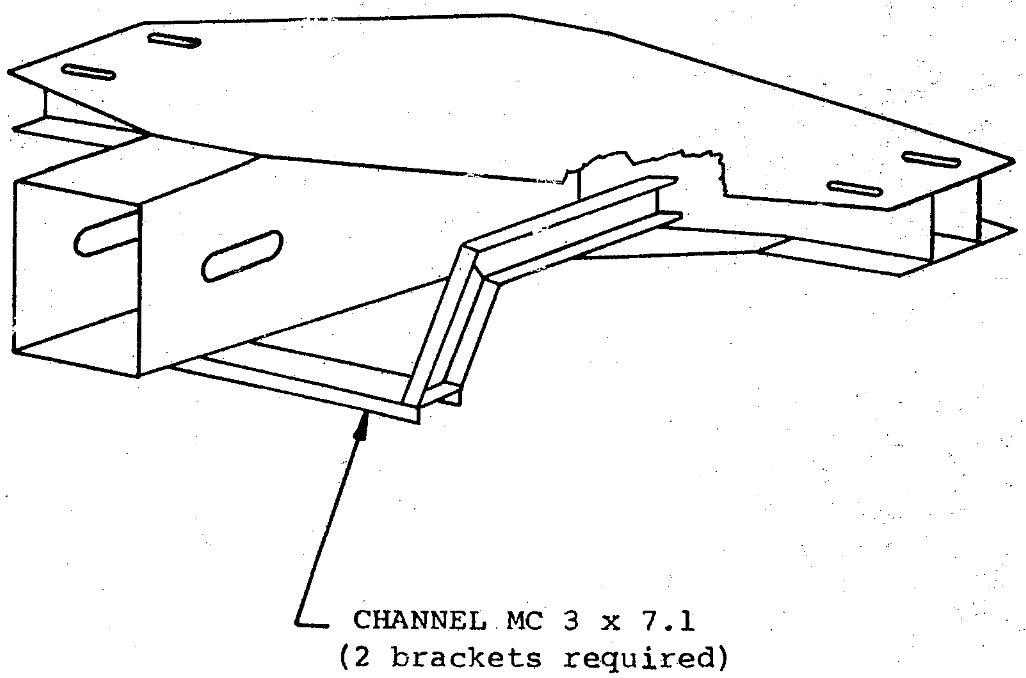
FLEXURAL RIGIDITY TEST LOADING ARRANGEMENT, STOX 763

Figure 7



FLEXURAL RIGIDITY TEST DEFLECTION MEASUREMENTS, STOX 763

Figure 8

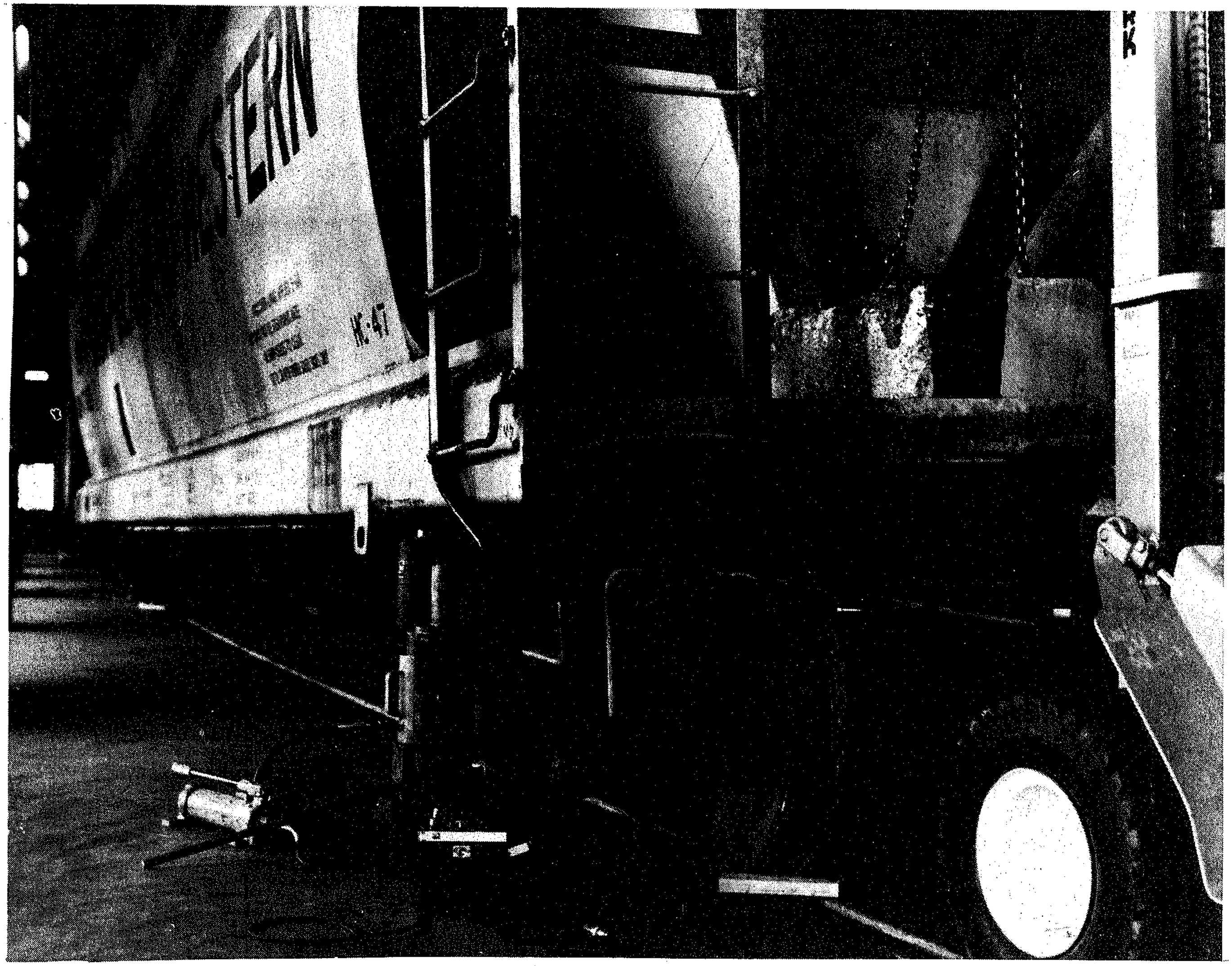


CLINOMETER SUPPORT BRACKET

Figure 9

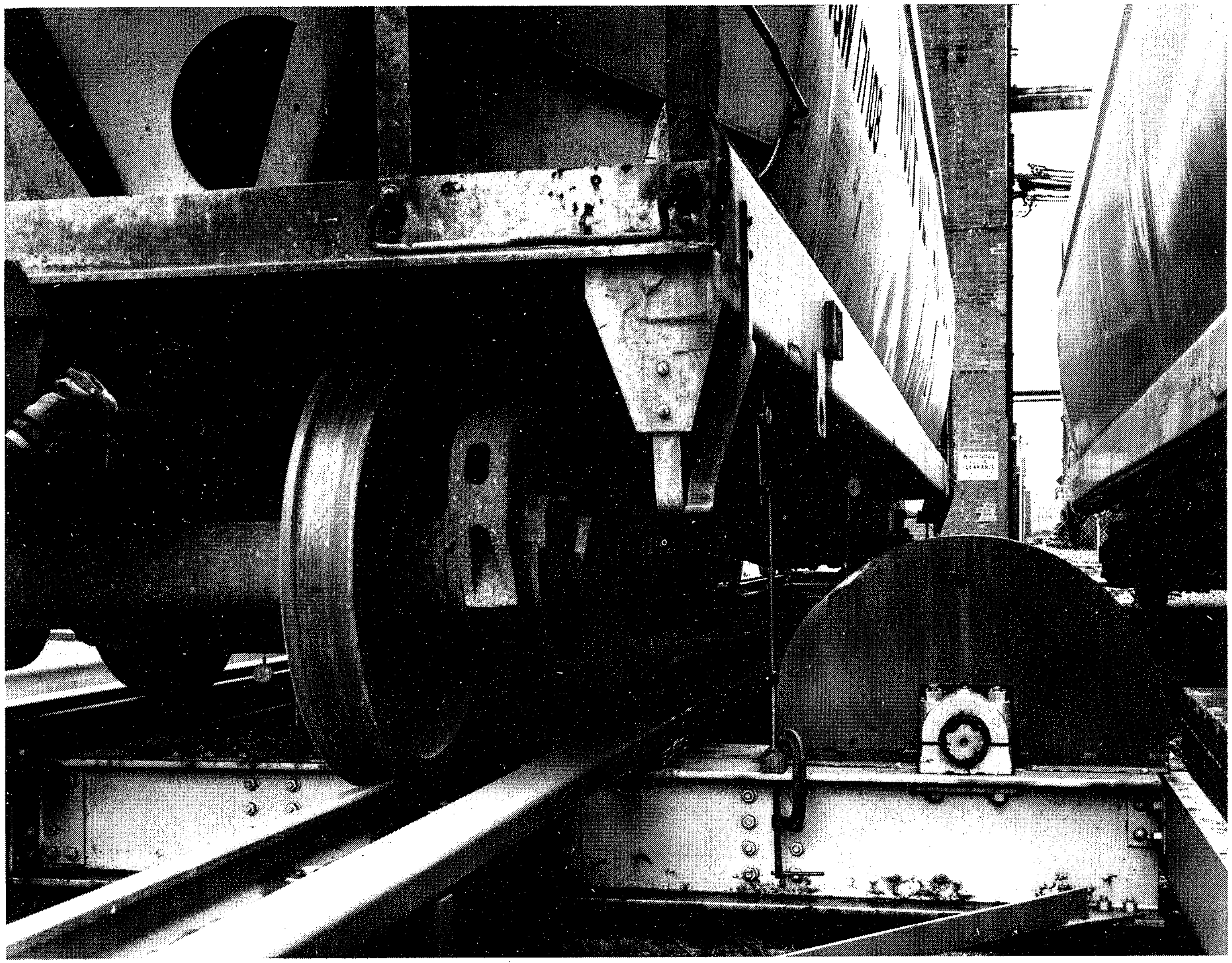
TORSIONAL RIGIDITY TEST, N&W 600976

Figure 10



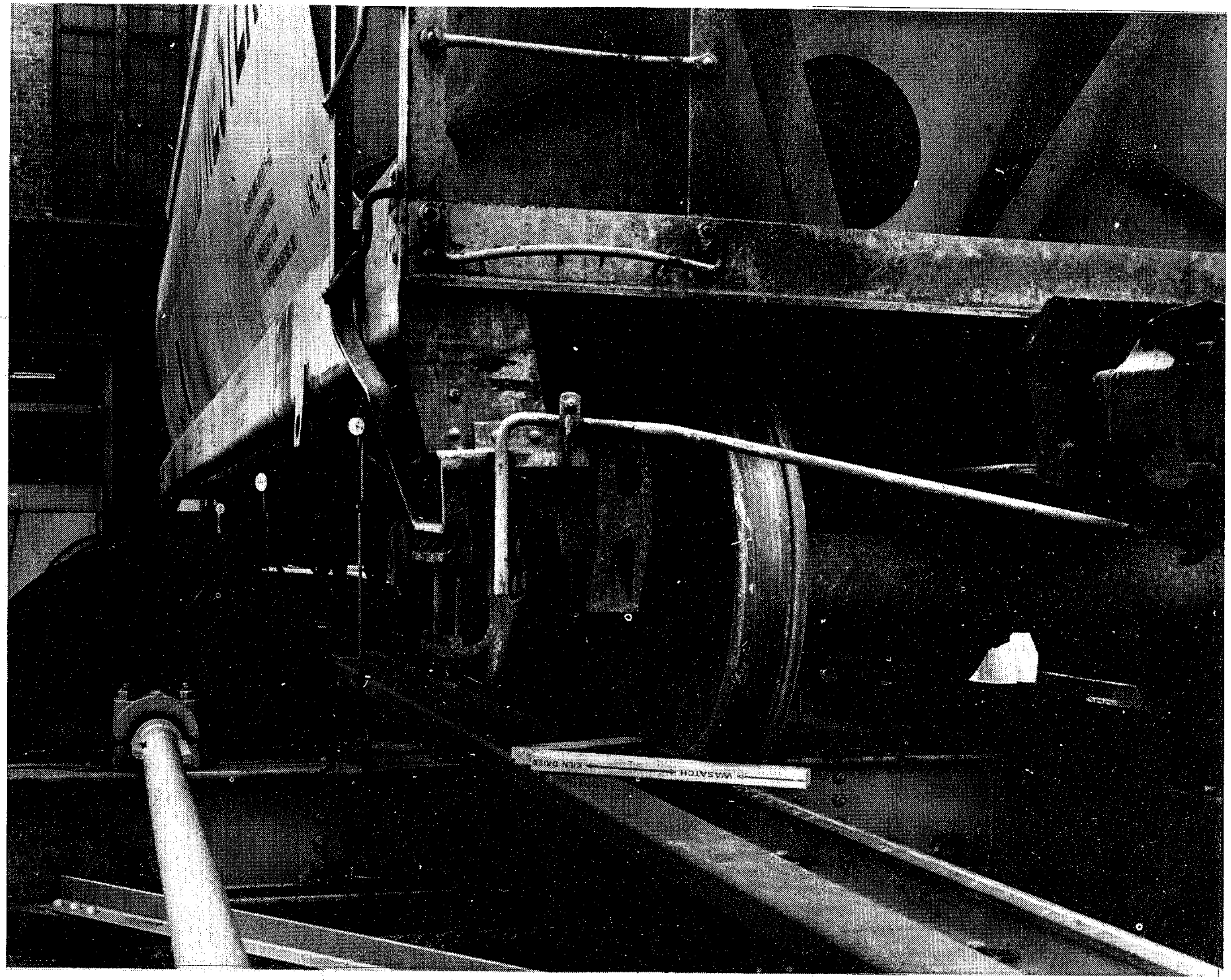
ACF CENTER FLOW TORSIONAL RIGIDITY TEST

Figure 11



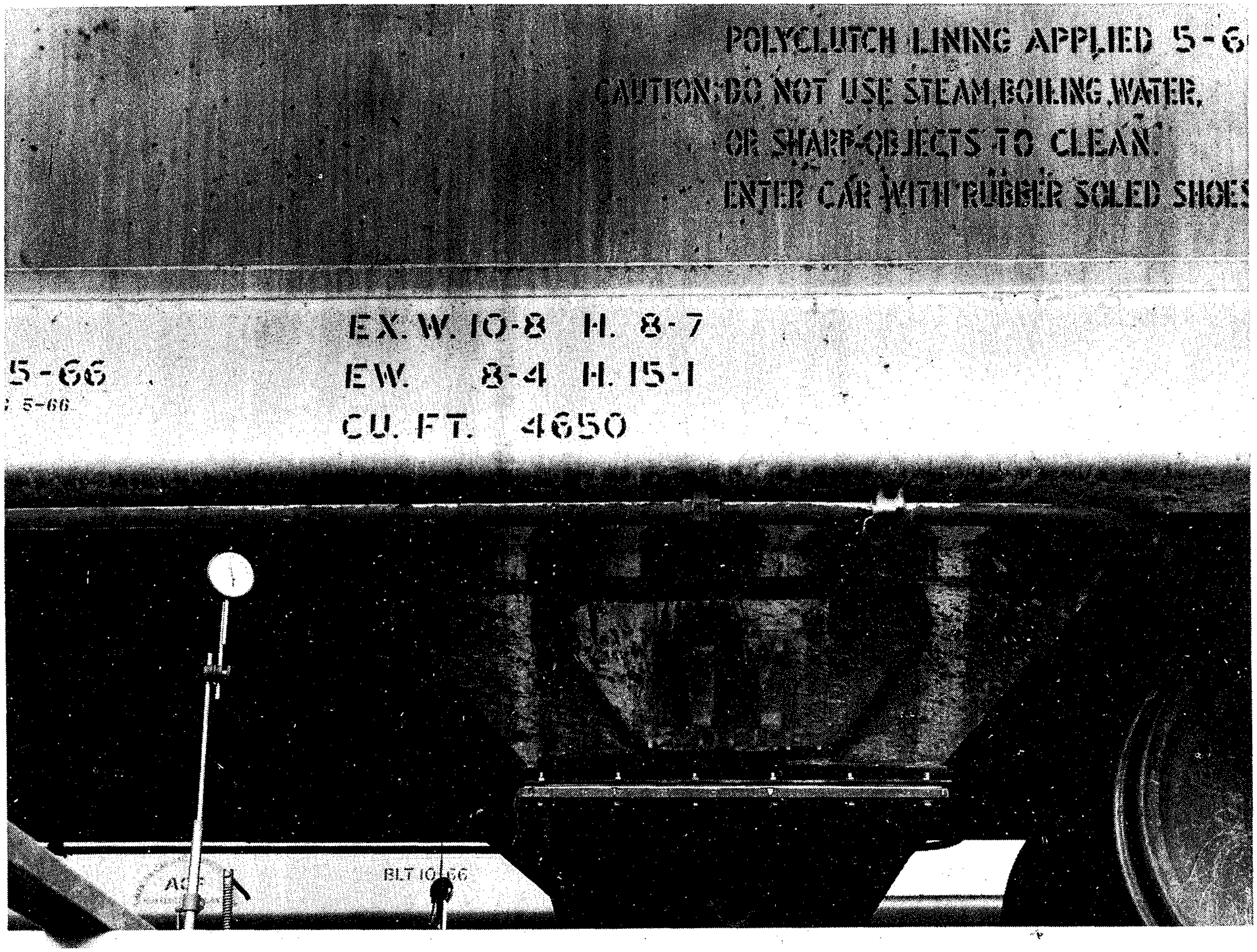
ACF CENTER FLOW FLEXURAL STIFFNESS TEST, N&W 171108

Figure 12



ACF CENTER FLOW FLEXURAL STIFFNESS TEST, N&W 171108

Figure 13



CENTER FLOW FLEXURAL TEST HOPPER CLOSURE PLATE, N&W 171108

Figure 14

PART II

TORSIONAL STIFFNESS, FLEXURAL RIGIDITY AND MASS INERTIA CHARACTERISTICS

Pullman-Standard Participation

In AAR/RPI/FRA Track-Train Dynamics

Program

INTRODUCTION

The following is a report of the work performed by Pullman-Standard, paralleling the efforts of American Steel Foundries, to develop torsional stiffness, flexural rigidity, and mass moments of inertia for additional types of freight cars. The following types of cars were used in the test program:

a. 100 ton covered hopper car
b. 70 ton box car
c. 100 ton open top hopper car

DESCRIPTION OF TEST CARS

1. The 100 ton covered hopper car, PS-2-CD, has a capacity of 4,750 cubic feet. Car measures 57 ft. 4 in. over strikers, 45 ft. 9 in. between truck centers, and has a lightweight of 60,200 lbs.

2. The 70 ton PS-1 box car was supplied by the L.&N. Railroad for the tests. Car measures 54 ft. 10 in. over strikers, 39 ft. 6 in. between truck centers and has 10 ft. sliding doors; the lightweight is 62,500 lbs.

3. The open top hopper car was a "New Family" PS-3-SD of 4,000 cubic foot capacity. The car is 50 ft. 5½ in. over strikers and 40 ft. 6 in. between truck centers; the lighweight is 60,200 lbs.

PROCEDURES

The following is a brief description of the procedures used by Pullman-Standard to obtain the flexural rigidity, center of gravity height for both empty and loaded car, torsional rigidity, and mass moments of inertia in pitch and roll for the cars tested.

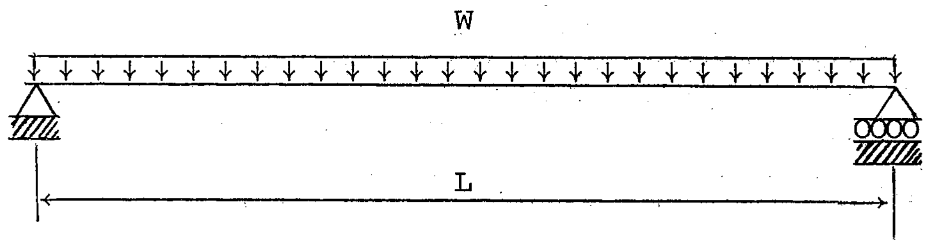
I. Flexural Rigidity

For the two hopper cars, the flexural rigidity is obtained analytically rather than experimentally since the moment of inertia of the side girders is readily calculated. The moment of inertia, the modulus of elasticity and the truck center distance of a given car are combined in the equation,

$$\text{Flexural Rigidity} = \frac{EI}{L}, \quad \text{in ft-lb radian.}$$

For the box car, due to the doorway, an equivalent flexural inertia was calculated as illustrated below.

Assume the carbody is a simply supported beam in bending:



where Length - L $= 473.376$ in.

Uniformly Distributed Load - W = 115.512 lb/in

Modulus of Elasticity - E $= 30\text{x}10^6$ psi

Deflection of the Centerline $-\Delta$ Max ℄

$$= \frac{5wL^4}{384\ E\ Ieq} = \frac{5(115.512^{\text{lb}}/\text{in})\ (473.376)^4\text{in}^4}{384\ (30\text{x}10^6\text{lb}/\text{in}^2)\ Ieq}$$

$$\Delta \text{ Max } ℄ = \frac{1}{Ieq}\ (1.5040625\frac{(473.376)^4}{30\text{x}10^6}) = \frac{1}{Ieq}\ (2517.5002)$$

The deflection at centerline from test Δ Max ℄ = .0491971"

Therefore, $.0491971'' = \frac{1}{Ieq}\ (2517.5002)$

Therefore, $Ieq = 51223.779\ \text{in}^4$

Then:

$$\text{Flexural Rigidity} = \frac{E\ Ieq}{L} = \frac{1.5367133 \text{ x } 10^{12}}{473.376}\ \frac{\text{lb-in}^2}{\text{rad}}$$

$$E\ Ieq = \frac{1.067162 \text{ x } 10^{10}}{39.448}\ \frac{\text{ft}^2\text{-lb}}{\text{rad}}$$

$$\frac{EIeq}{L} = 2.7052343 \text{ x } 10^8\ \frac{\text{ft-lb}}{\text{rad}}$$

II. Center of Gravity Height Test

The test was conducted by placing four (4) jacks at each jacking pad of the car with load cells in between. The car body was then jacked up until free of of trucks and set generally in level position. An attempt was made to adjust the load on the jacks at both sides of each car end to be equal. The car was then jacked up to tip the car with height increments of 2.1 in. until 10.5 in. maximum height was reached. The load cell readings were recorded at each position of tip.

The recorded data were then fed in the following equation to obtain the c.g. height of the car relative to the jacking pad.

$$h = \frac{\Delta W}{W} \cdot \frac{\ell^2}{S}$$

where h = c.g. height relative to jacking pad (in.)

ℓ = distance between two jacking pads (in.)

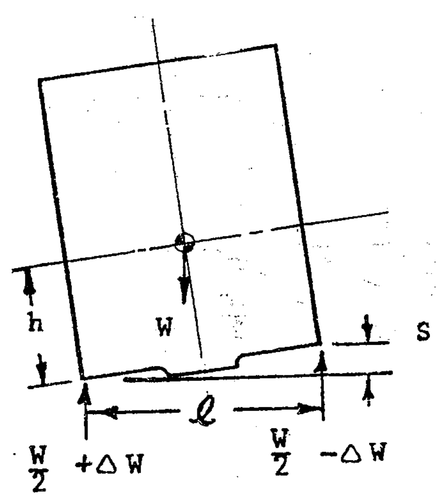
S = tip height (in.)

W = Total weight of empty car (lbs.)

ΔW = Weight change in tip position (lbs.)

By adding the height of the jacking pad to rail, the c.g. height of the car relative to the rail was then obtained.

The problem geometry is depicted as follows:



III. Torsional Rigidity Test

The preparation for the car body torsional rigidity test is identical to the c.g. height test. In this the test, the two diagonal corners were jacked up in an equal amount of height with ¼" increments until the other two corners were lifted off the jacks. The height of all four corners, along with their load cell readings, were recorded. This test procedure was repeated by jacking up the opposite diagonal corners.

The test data were then fed into the following equation to obtain the GJ values.

$$GJ = \frac{T + \frac{Wh}{2}\,\frac{S}{2\ell}}{\frac{S}{\ell}} \quad \frac{TC}{2}$$

where

GJ = Torsional Rigidity lb-ft^2/rad.

h = c.g. height relative to jacking pad (ft.)

ℓ = Distance between jacking pads (in.)

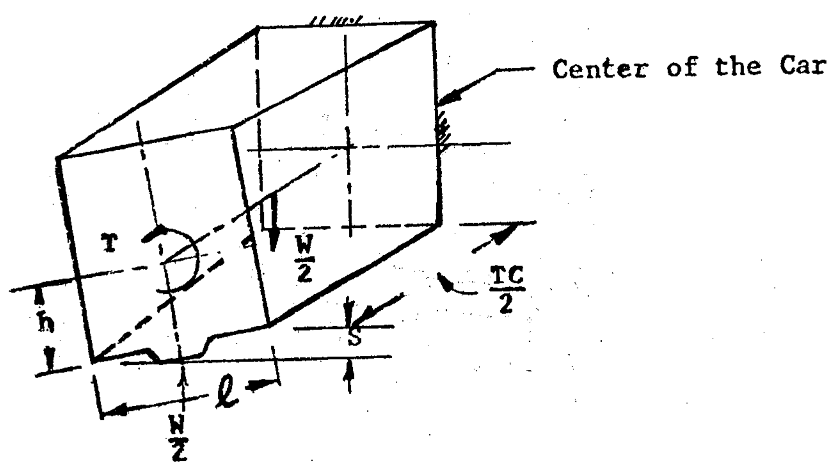
S = Average elevation of the jacked-up corners relative to the corner at the other side (in.)

T = Average of applied torque at two bolsters (ft-lbs)

TC = Truck center distance (ft.)

W = Empty car weight (lbs)

The problem geometry is depicted as follows:



IV. Mass Moment of Inertia

A. Pitch Inertia at car C.G.

The mass moment of inertia for the car in pitch is obtained by an actual dynamic test. In this test, the car was hinge-supported at the side sill directly beneath the c.g. of the car as a pivot point in pitch motion. Truck snubbers were inactive. The car was then excited in pitch at one end of the car. The response pitch motions were then recorded on the oscillograph. The resonant frequency was then fed in the following equation to obtain the pitch inertia of the empty car about its c.g.

$$I_p = \frac{K \frac{TC^2}{2} - Wh}{\omega_p^{\,2}} - \frac{W}{g} h^2$$

where

g = Gravitational acceleration (32.2 ft/sec^2)

h = Pivot point to c.g. (ft.)

I_p = Pitch inertia (ft-lb-sec^2)

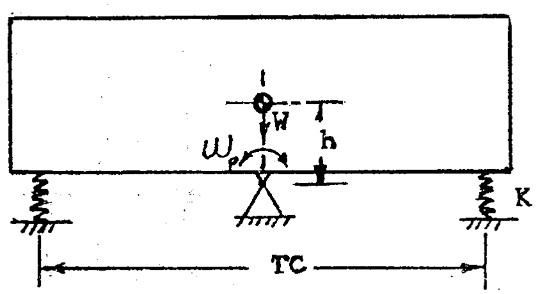
K = Spring constant per truck (lbs/ft)

TC = Truck center distance (ft)

W = Empty car weight (lbs)

ω_p = Measured resonant pitch frequency (rad./sec)

The problem geometry is depicted as follows:



V. Mass Moment of Inertia (Cont'd)

B. Roll Mass Inertia at car C.G.

The mass moment of inertia for the car in roll is also obtained by an actual roll test. In this test, the car was hinged at center sill ends as a pivot point in roll motion. Shims were provided at side bearings to restrict relative movement between car body and truck bolster; therefore, when the car rolls, the car body and truck bolster roll together. The car was excited at the middle of the side sill to determine resonance. The motion was recorded on an oscillograph. The resonant frequency was then fed into the following equation to obtain the roll mass inertia of the empty car at its c.g.

$$I_r = \frac{\frac{K}{2} d^2 - WR}{\omega_r^{\ 2}} - \frac{W}{g} R^2$$

where

d = Distance between truck spring group centers (ft.)

g = Gravitational acceleration (32.2 ft/sec^2)

I_r = Roll inertia (ft-lb-sec^2)

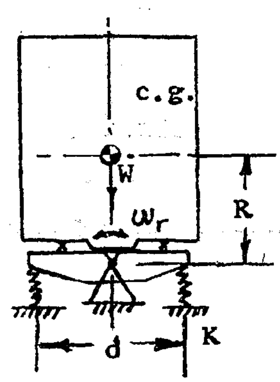
K = Spring constant (lbs/ft.)

R = Distance of c.g. to the pivot point (ft)

W = Empty car weight (lbs.)

ω_r = Measured resonant roll frequency (rad./sec)

The problem geometry is depicted as follows:



VI. Example Calculation for Loaded Car Moment of Inertia 50 Foot 70-Ton Box Car, L&N 450019

1). Mass Moment of Inertia - Lading

Assuming lading dimension as:

Parallelpiped 50'-6" x 9'-5-5/8" x 7'-6"

Therefore, moments of inertia for Lading (L_{r1}), Pitch (I_{p1}), Roll (I_{r1}), and Yaw (I_{y1}) are as follows:

$$\text{Pitch } I_{p1} = 1/12 \left(\frac{140,000}{32.2}\right) (50.5^2 + 7.5^2) = 944,384 \text{ lb-ft sec}^2$$

$$\text{Roll } I_{r1} = 1/12 \left(\frac{140,000}{32.2}\right) (7.5^2 + 9.46875^2) = 52,864.932 \text{ lb-ft-sec}^2$$

$$\text{Yaw } I_{y1} = 1/12 \left(\frac{140,000}{32.2}\right) (50.5^2 = 9.46875^2) = 956,488.08 \text{ lb-ft-sec}^2$$

where the center height is $\bar{y}$ = 45" from deck.

2). Mass Moment of Inertia - Car Body

Calculations for moments of inertia for Pitch (I_{p2}) and Roll (I_{r2}) of carbody only, are as follows:

$$\text{Pitch } I_{p2} = 510,000 \text{ ft-lb-sec}^2$$

$$\text{Roll } I_{r2} = 27,000 \text{ ft-lb-sec}^2$$

and the center is $\bar{y}$ = 36.425" from deck.

3). Moments of inertia for the lading and carbody combined are:

$$\bar{Y} = \frac{140,000\ (45) + 40,000\ (36,425)}{180,000} = 43.1\text{" from deck}$$

$$\text{Pitch } I_p = 944,384 + \frac{140,000}{32.2} (45-43.1)^2 + 510,000 + \frac{40,000}{32.2} (43.1-36.425)^2$$
$$= 1,525,428.2 \text{ ft-lb-sec}^2$$

$$\text{Roll } I_r = 52864.932 + \frac{140,000}{32.2} (45-43.1)^2 + 27,000 + \frac{40,000}{32.2} (43.1\ 36.425)^2$$
$$= 150,909.18 \text{ ft-lb-sec}^2$$

RESULTS

	100-Ton Covered Hopper	100-Ton Open Hopper	70-Ton 50-Foot Box Car
Empty Car Center of Gravity (Body only above rail)	82.92"	67.16"	80.3"
Flexural Rigidity	*4.28×10^8	*1.46×10^8	2.71×10^8
Torsional Rigidity	1.17×10^9	5.52×10^7	1.18×10^9
Roll Inertia **Empty Loaded	5.66×10^4 *1.93×10^5	8.8×10^4 *2.43×10^5	2.7×10^4 *1.5×10^5
Pitch Inertia Empty Loaded	6.14×10^5 *1.92×10^6	4.70×10^5 *1.60×10^6	5.1×10^5 *1.5×10^6

* Indicates calculated value

** Empty car roll inertia values appear inconsistent but are included to show approximate order of magnitude.

SUMMARY

MASS MOMENTS OF INERTIA

Type of Car	Light Weight	Loaded Weight	C.G. Above Rail	I_{Pitch}	I_{Yaw}	I_{Roll}
	LBS.	LBS.	IN.	LB-IN^2	LB-IN^2	LB-IN^2
40 FT 50T Box Car (Empty)	40,900	---	68.7	9.02×10^8	8.64×10^8	1.46×10^8
40 FT 50T Box Car (Loaded)	---	163,000	95.0	33.8×10^8	33.4×10^8	4.93×10^8
50 FT 100T Hi-Cube Box Car (Empty)	51,400	---	79.6	17.6×10^8	16.9×10^8	2.29×10^8
50 FT 100T Hi-Cube Box Car (Loaded)	---	242,500	108.9	78.0×10^8	77.4×10^8	9.24×10^8
60 FT 100T Box Car (Empty) Cushioned	62,700	---	76.6	30.6×10^8	29.4×10^8	2.58×10^8
60 FT 100T Box Car (Loaded) Cushioned	---	242,500	103.6	111.0×10^8	110.0×10^8	8.52×10^8
86 FT 70T Hi-Cube Box Car (Empty)	78,500	---	83.6	78.2×10^8	75.2×10^8	3.86×10^8
86 FT 70T Hi-Cube Box Car (Loaded)	---	203,300	108.7	189.0×10^8	188.0×10^8	8.52×10^8
CF 4600 Center Flow (Empty)	42,000	---	78.7	19.1×10^8	18.5×10^8	1.54×10^8
CF 4600 Center Flow (Loaded)	---	242,500	104.2	73.8×10^8	72.7×10^8	6.53×10^8
70 Ton F894 F/D Flat Car (Empty with no hitches)	47,800	---	35.7	40.0×10^8	39.2×10^8	$.41 \times 10^8$
70 Ton F894 F/D Flat Car (Loaded w/2 trailers)	---	182,500	89.8	170.7×10^8	170.0×10^8	3.19×10^8

Notes:
1. Trucks are not included in any of the above values.
2. Trailers are 40 ft., 65,000# each.