United States Patent [19]

Freeman et al.

[11] **3,718,040**

[45] **Feb. 27, 1973**

[54] **METHOD AND APPARATUS FOR EVALUATING RAILROAD TRACK STRUCTURE AND CAR PERFORMANCE**

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[73] Assignee: **Bessemer and Lake Erie Railroad Company by said Peterson; Quebec Cartier Mining Company by said Freeman; United States Steel Corporation by said Wandrisco.**

[22] Filed: **Sept. 7, 1971**

[21] Appl. No.: **178,131**

[52] **U.S. Cl.** 73/146

[51] **Int. Cl.** G01m 19/00

[58] **Field of Search** 73/146; 33/144, 146

[56] **References Cited**

UNITED STATES PATENTS

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Primary Examiner—Donald O. Woodiel

Attorney—Rea C. Helm

[57] **ABSTRACT**

Method and apparatus for determining dynamic lateral and vertical wheel-rail forces. Axle bending sensors and axle load cells provide signals to a computer programmed to calculate lateral and vertical wheel-rail forces. These forces are used as the basis for comparing the effects of a variety of car truck design criteria and track conditions. Comparison of forces developed by the same equipment on different runs over the same trackage discloses track condition changes between runs.

43 Claims, 21 Drawing Figures

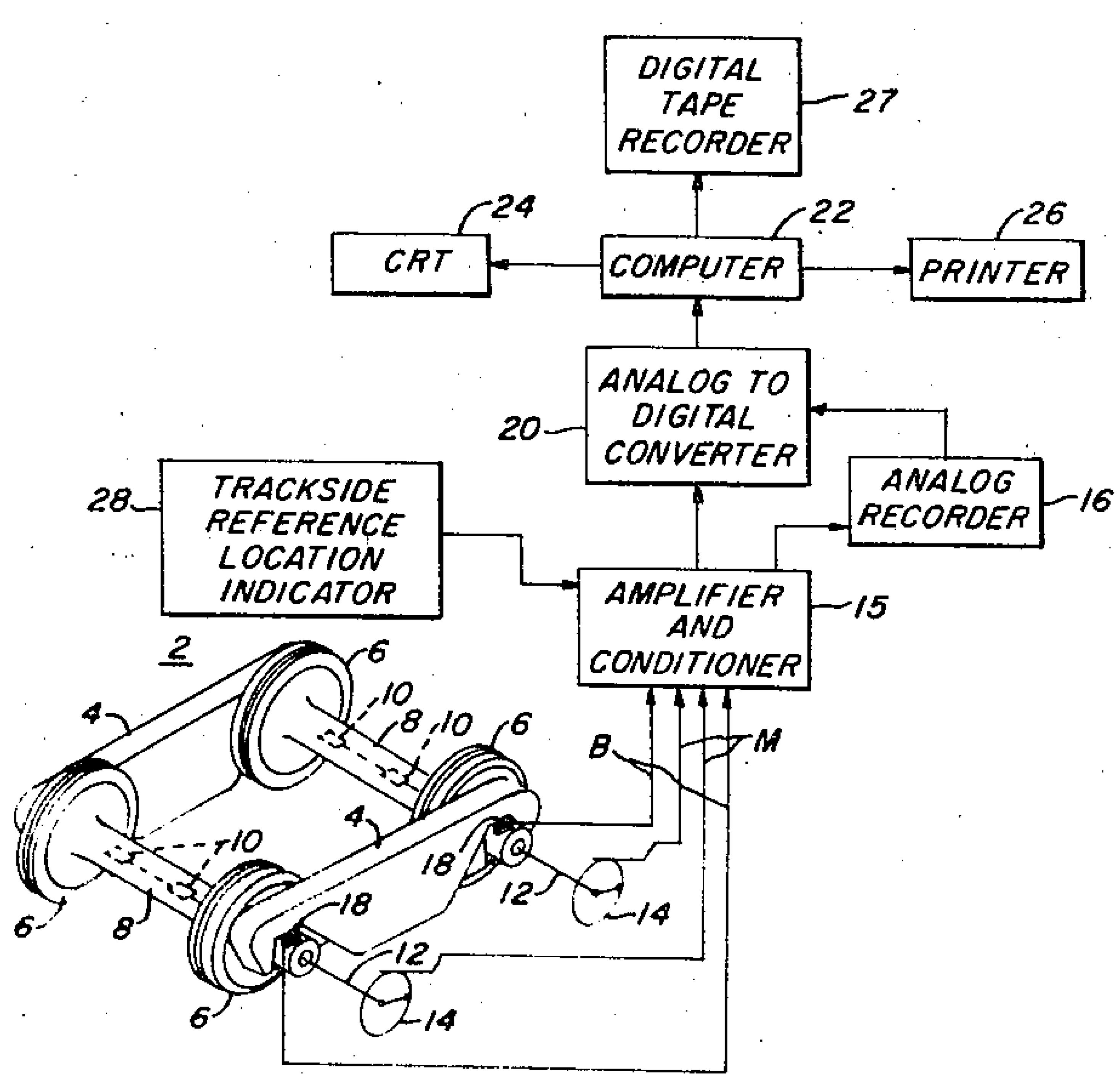


FIG. 1

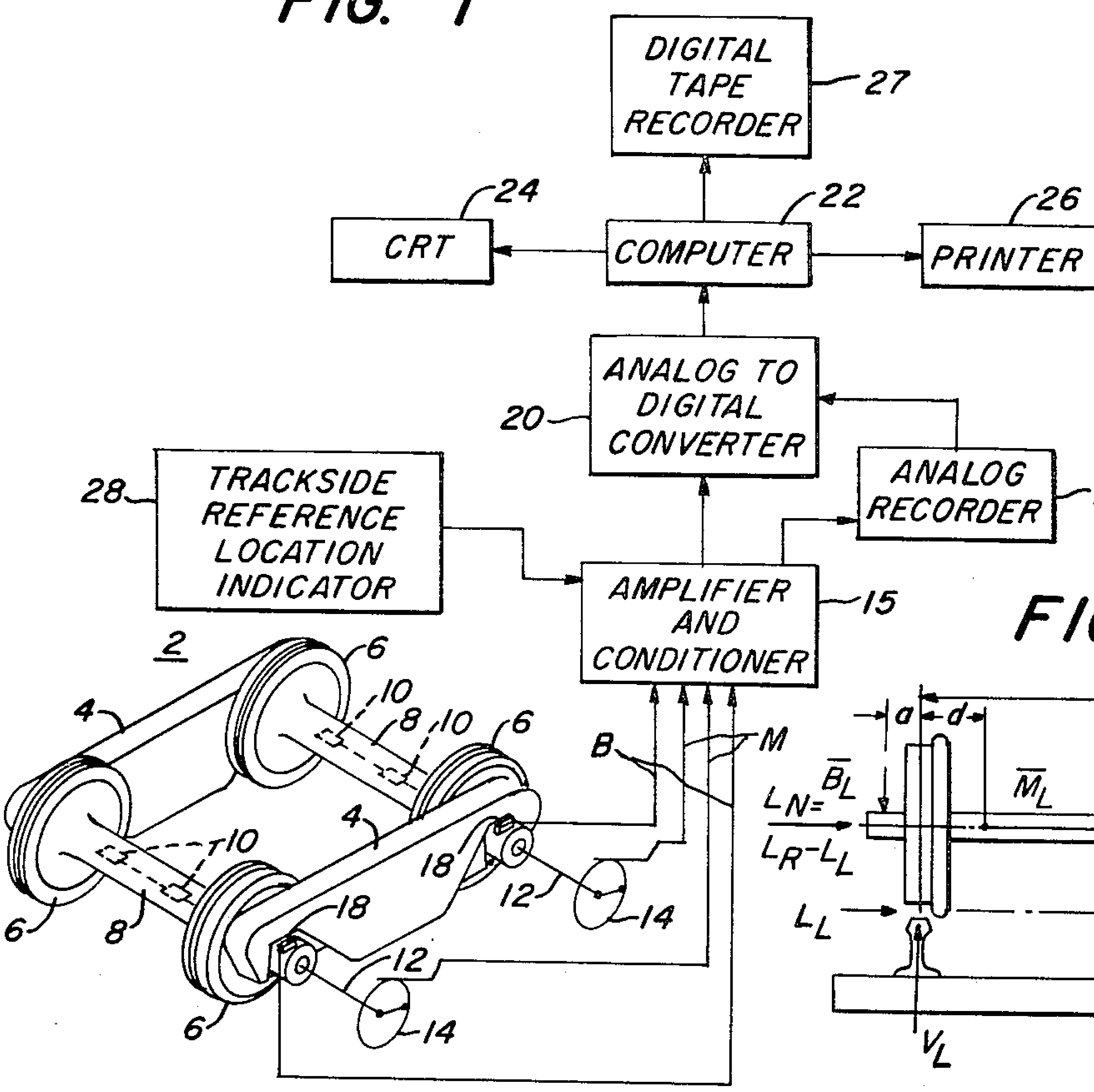


FIG. 2

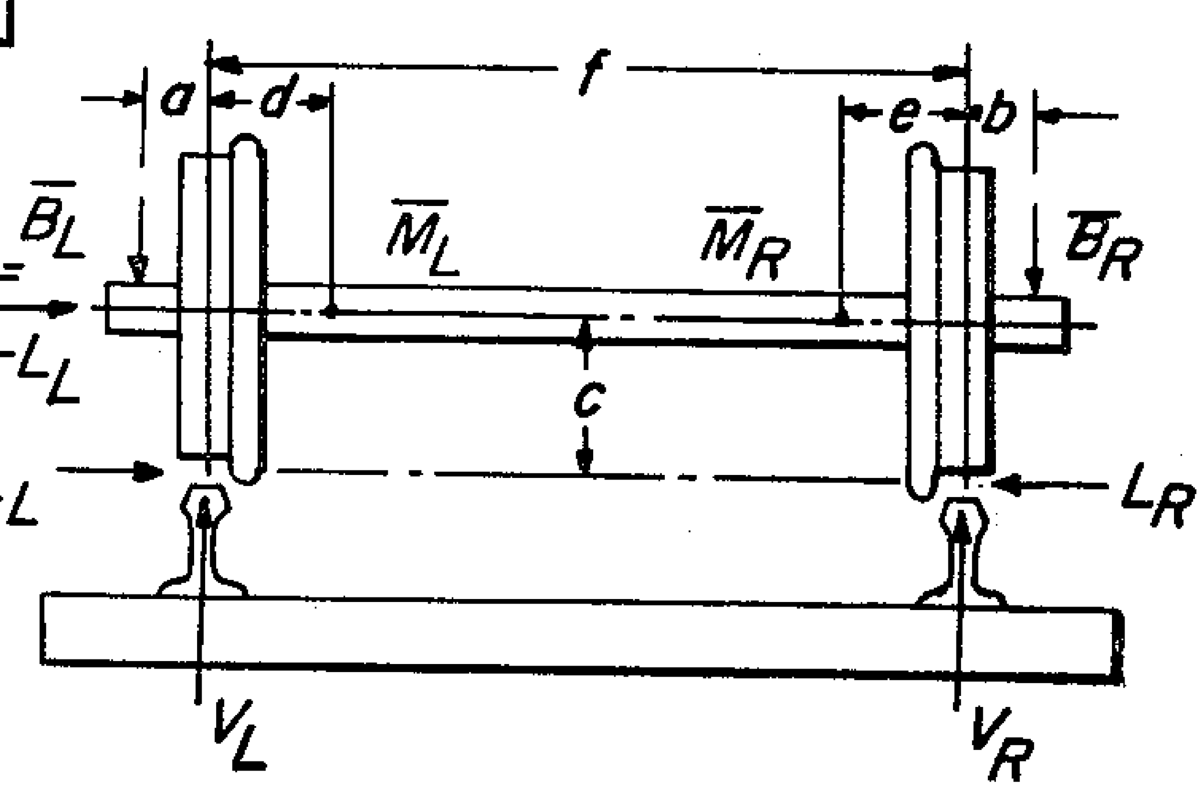


FIG. 3

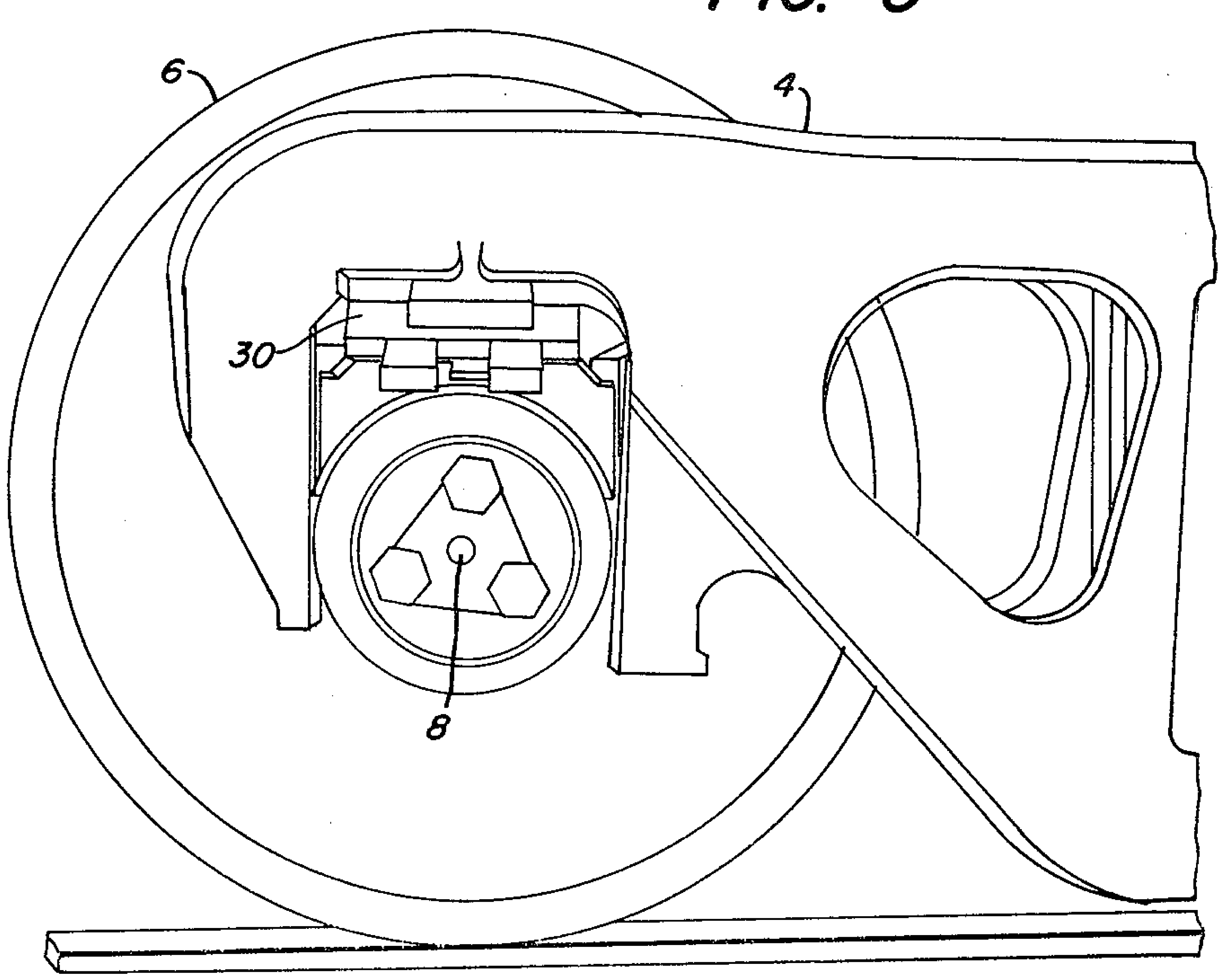
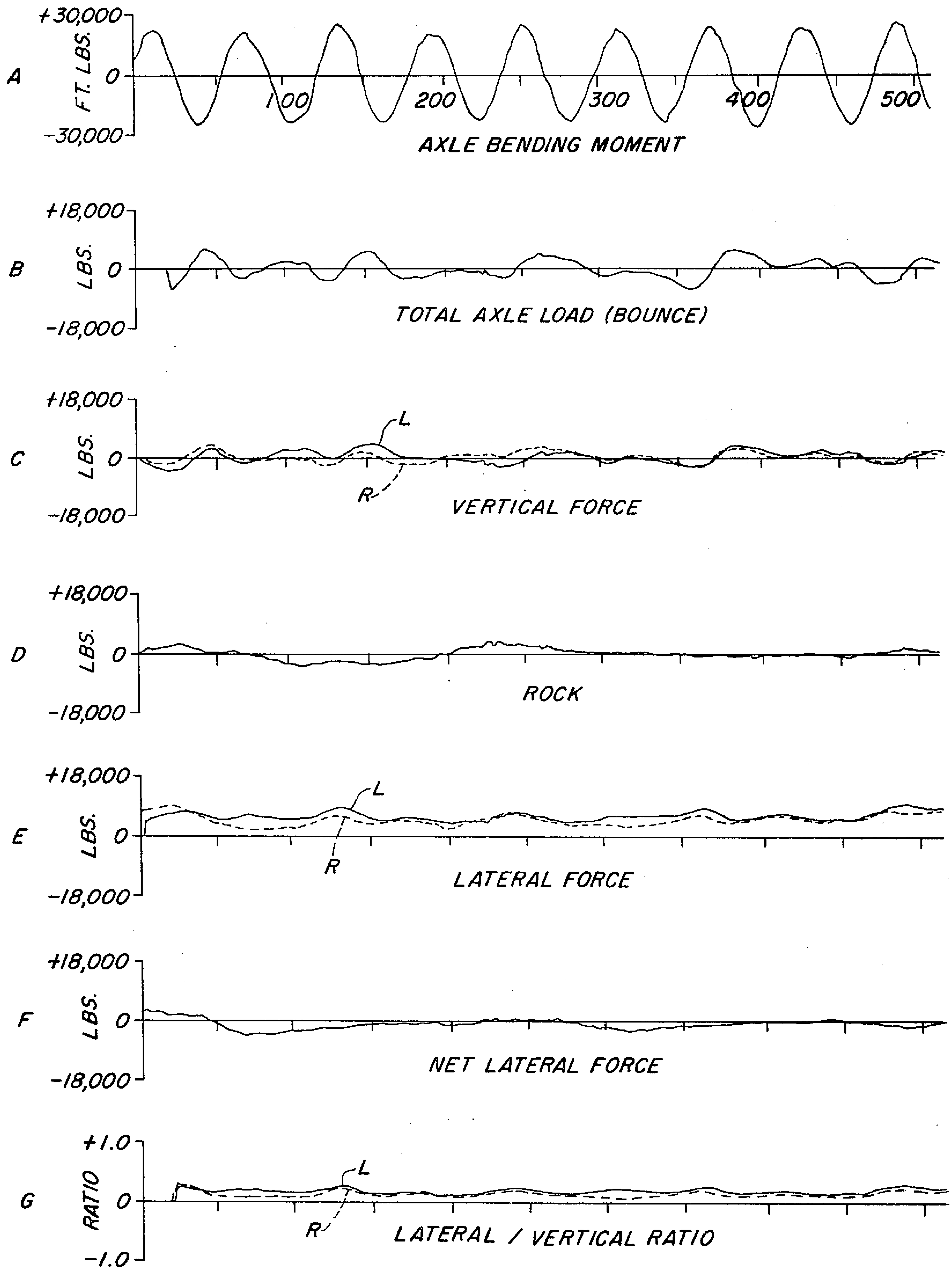


FIG. 4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200
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AXLE BENDING MOMENT
B
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LBS.
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TOTAL AXLE LOAD (BOUNCE)
C
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LBS.
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-18,000
L
R
VERTICAL FORCE
D
+18,000
LBS.
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ROCK
E
+18,000
LBS.
0
-18,000
L
R
LATERAL FORCE
F
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LBS.
0
-18,000
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G
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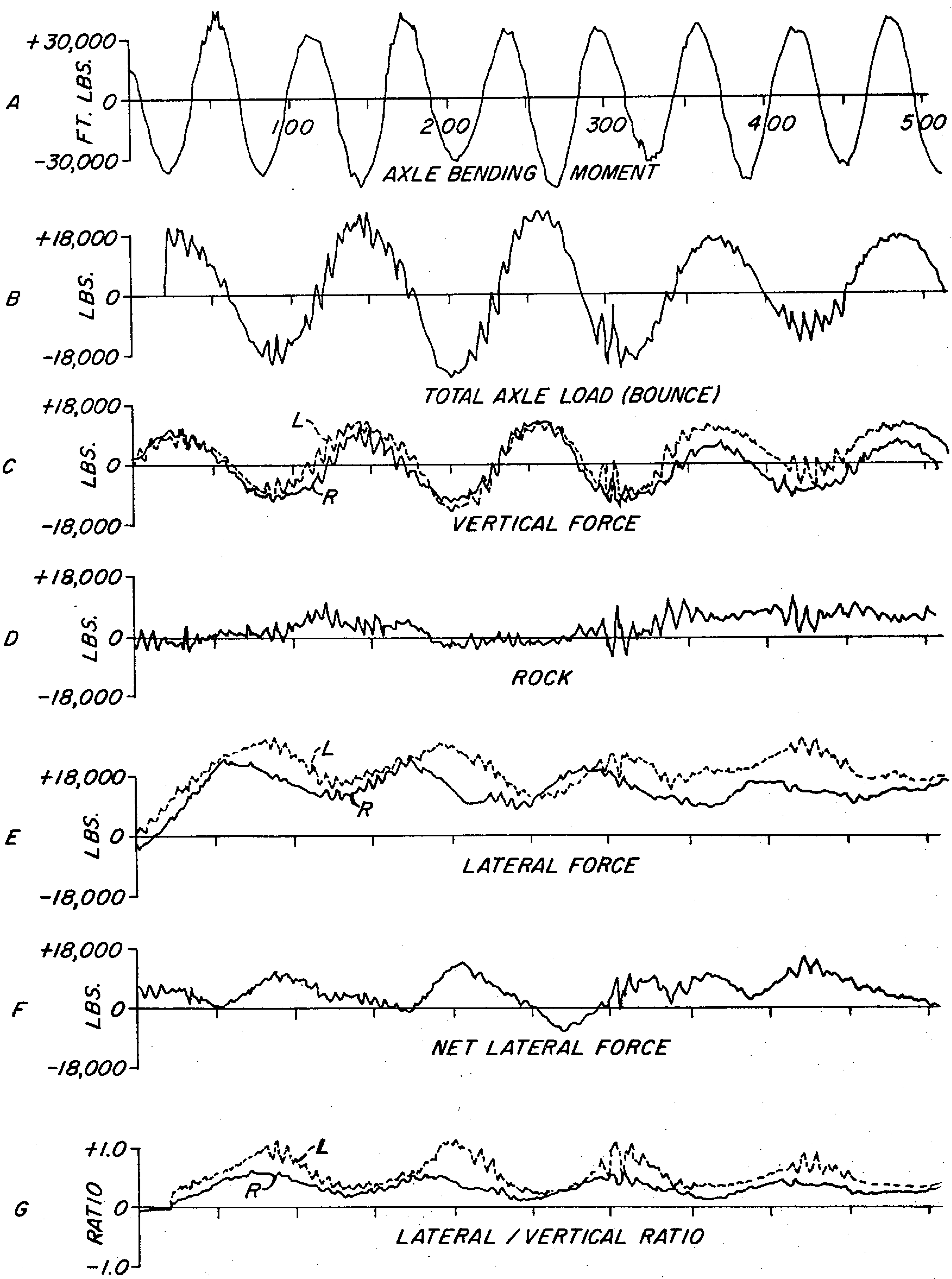
RATIO
0
-1.0
L
R
LATERAL / VERTICAL RATIO

FIG. 5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TOTAL AXLE LOAD (BOUNCE)
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LBS.
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VERTICAL FORCE
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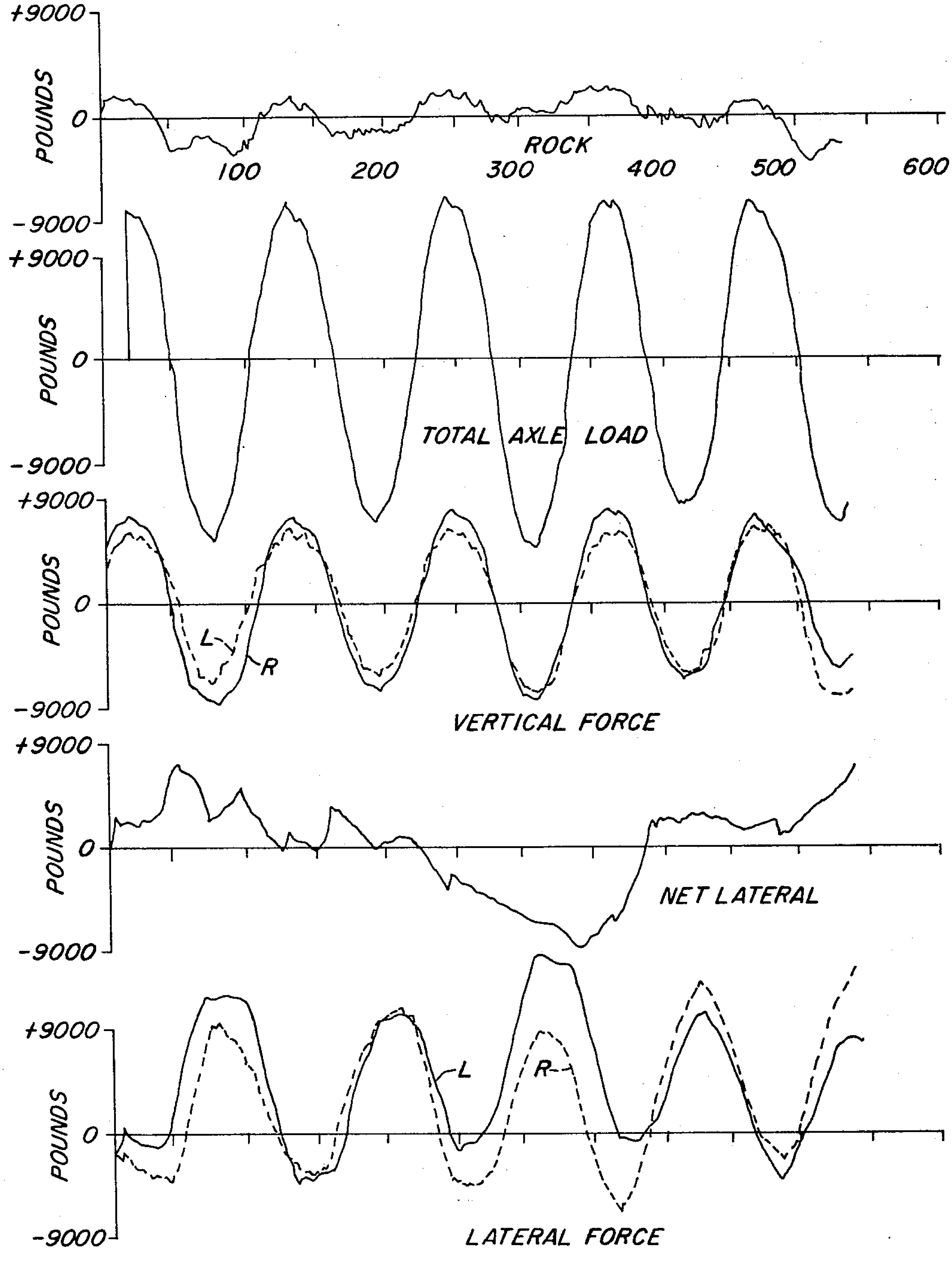
LATERAL / VERTICAL RATIO

FIG. 6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VERTICAL FORCE
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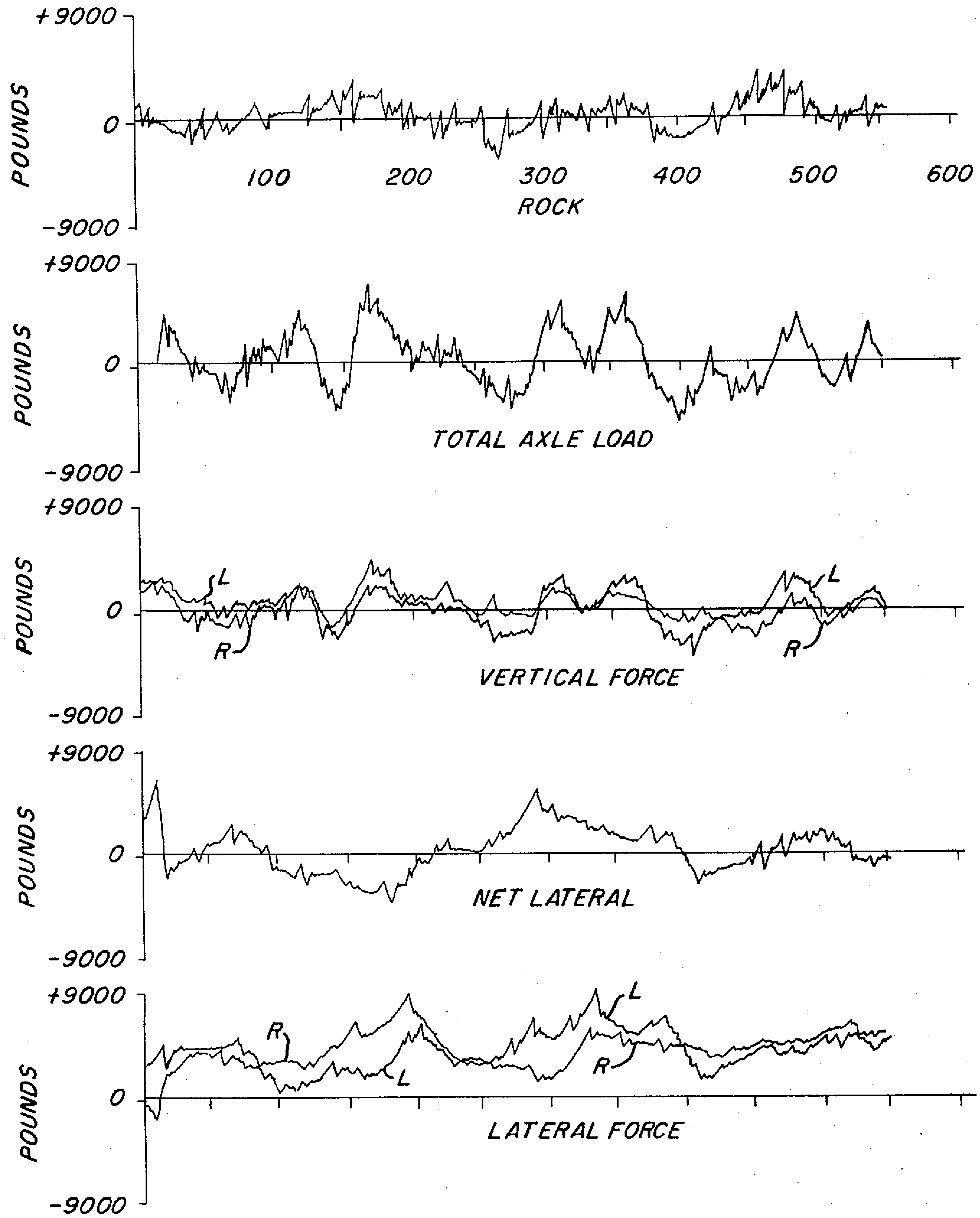
-9000
LATERAL FORCE

FIG. 7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-9000
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R
VERTICAL FORCE
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NET LATERAL
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L
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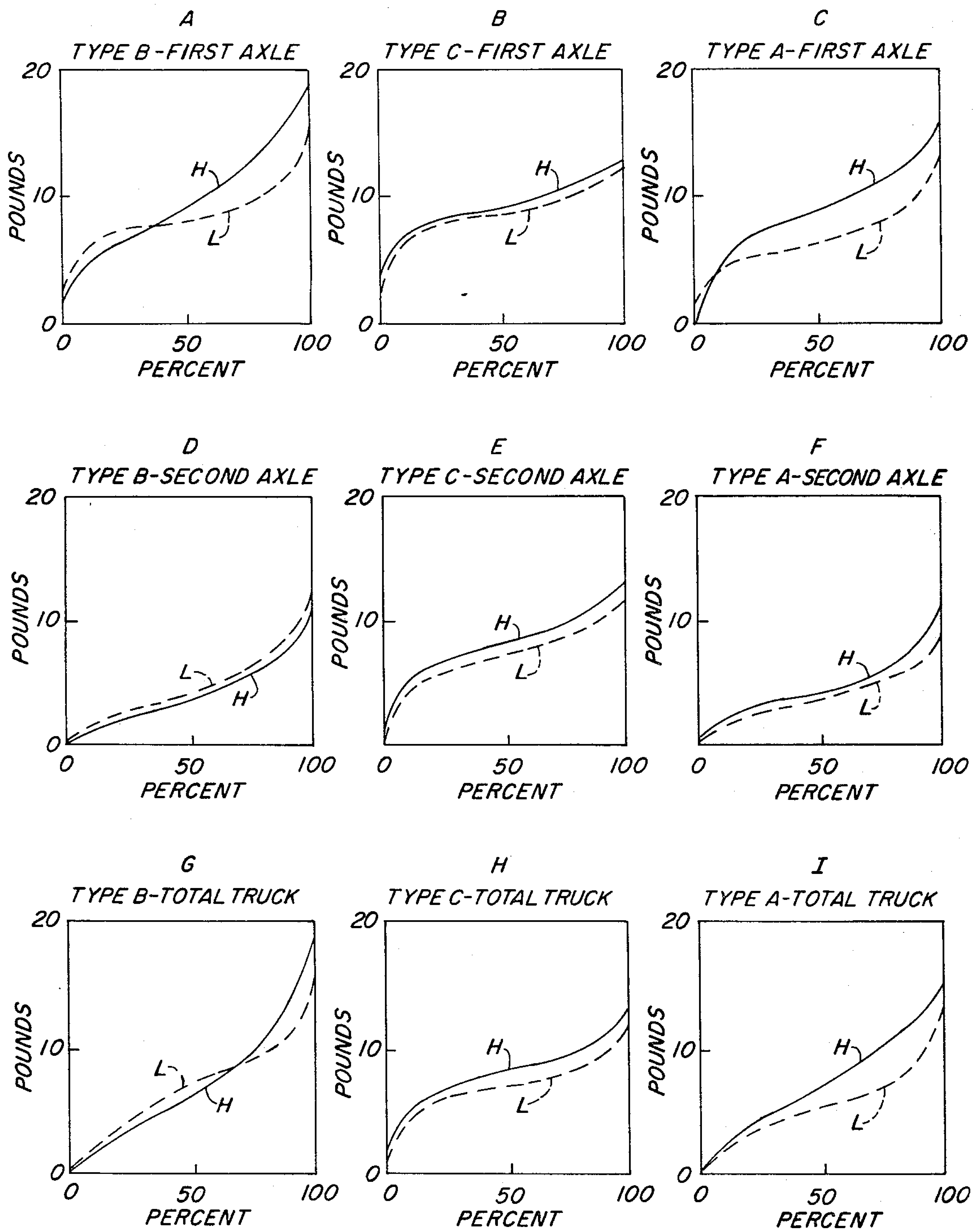
LATERAL FORCE

FIG. 8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B
TYPE C-FIRST AXLE
C
TYPE A-FIRST AXLE
D
TYPE B-SECOND AXLE
E
TYPE C-SECOND AXLE
F
TYPE A-SECOND AXLE
G
TYPE B-TOTAL TRUCK
H
TYPE C-TOTAL TRUCK
I
TYPE A-TOTAL TRUCK
POUNDS
PERCENT
H
L
0
10
20
50
100

FIG. 9

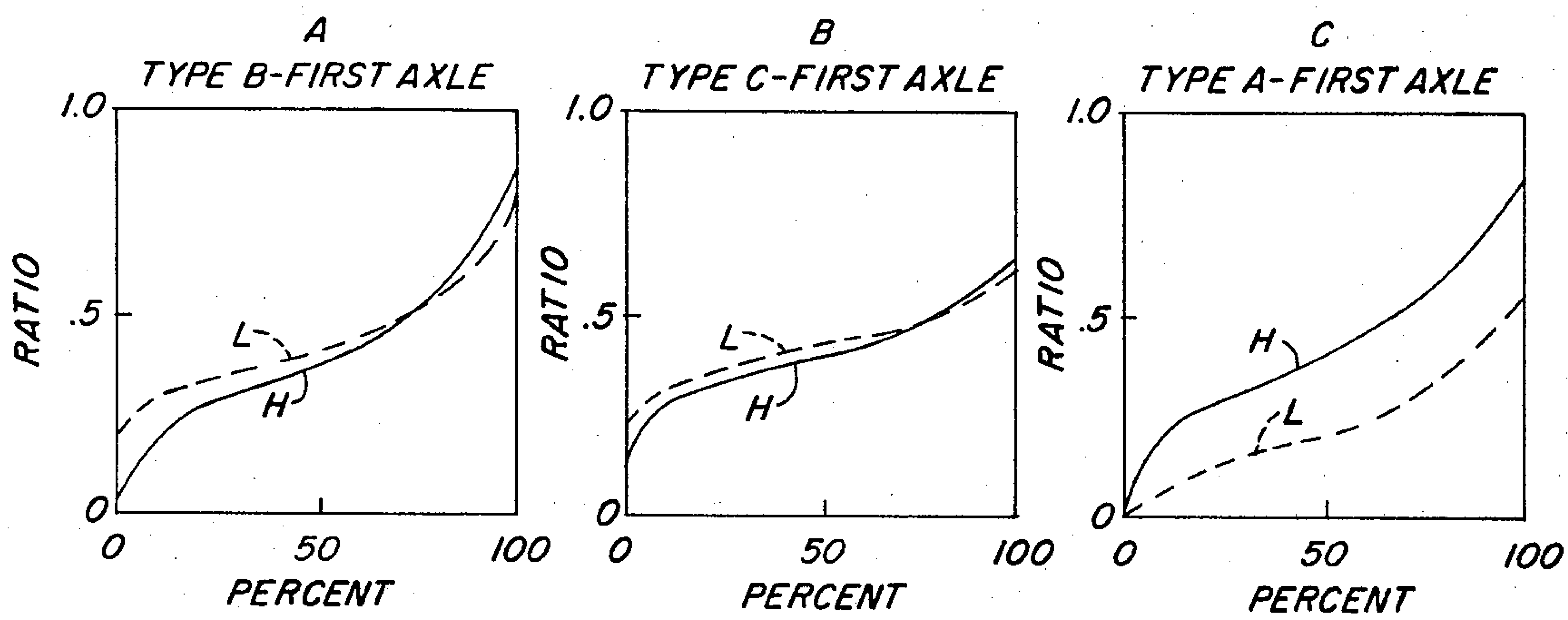


FIG. 10

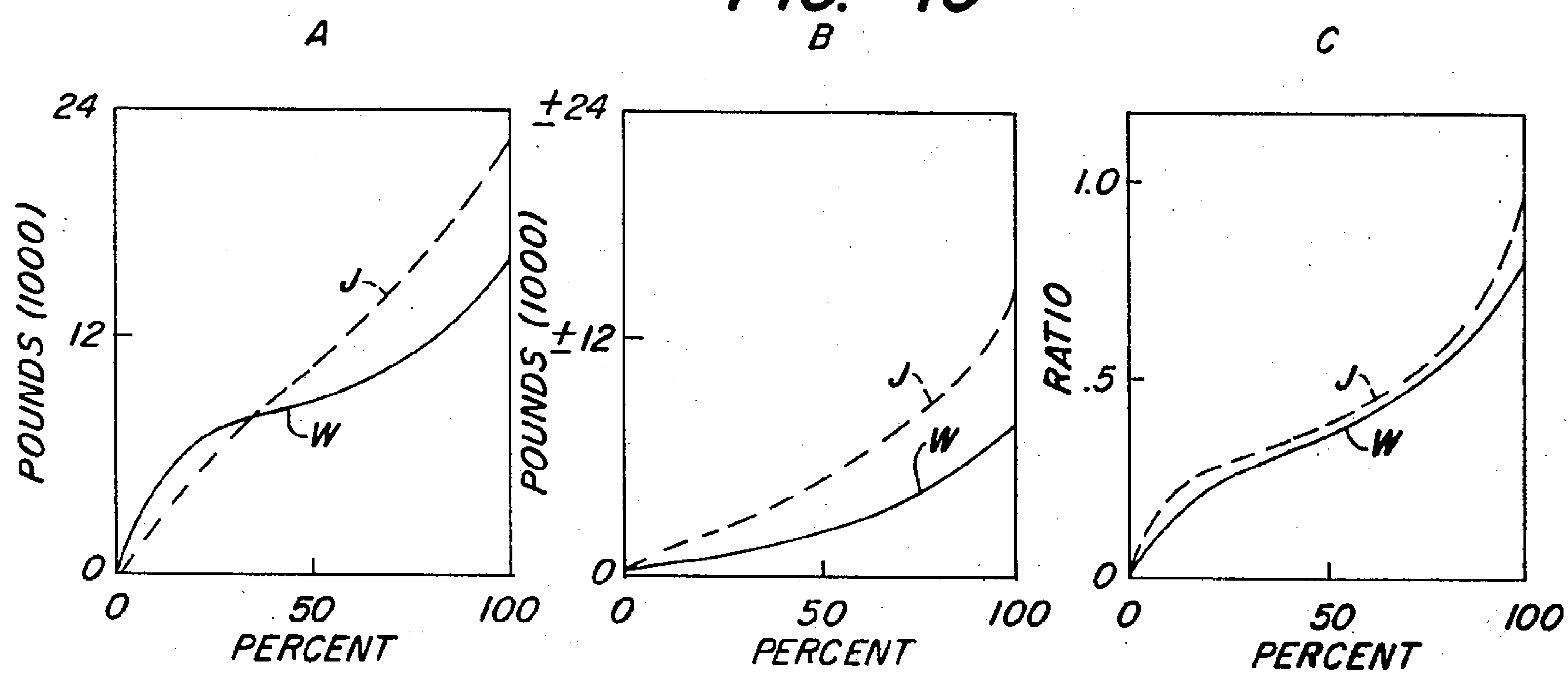


FIG. 11

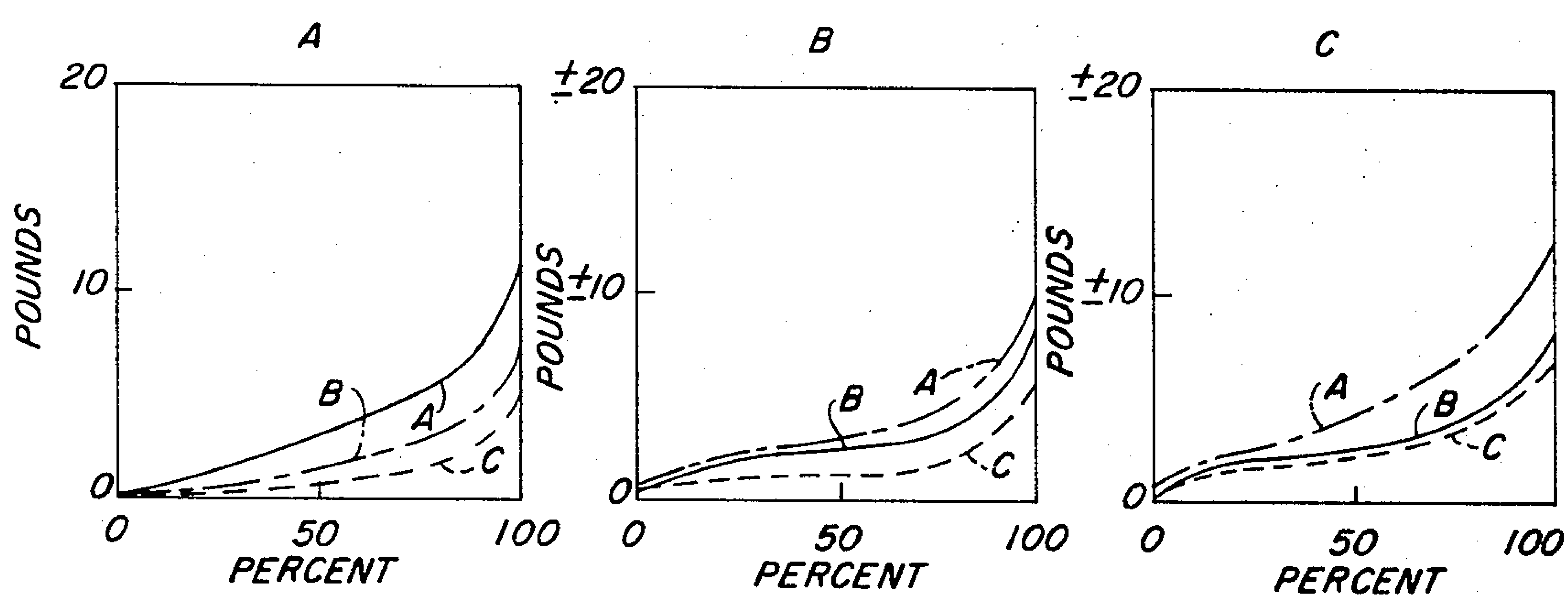


FIG. 12

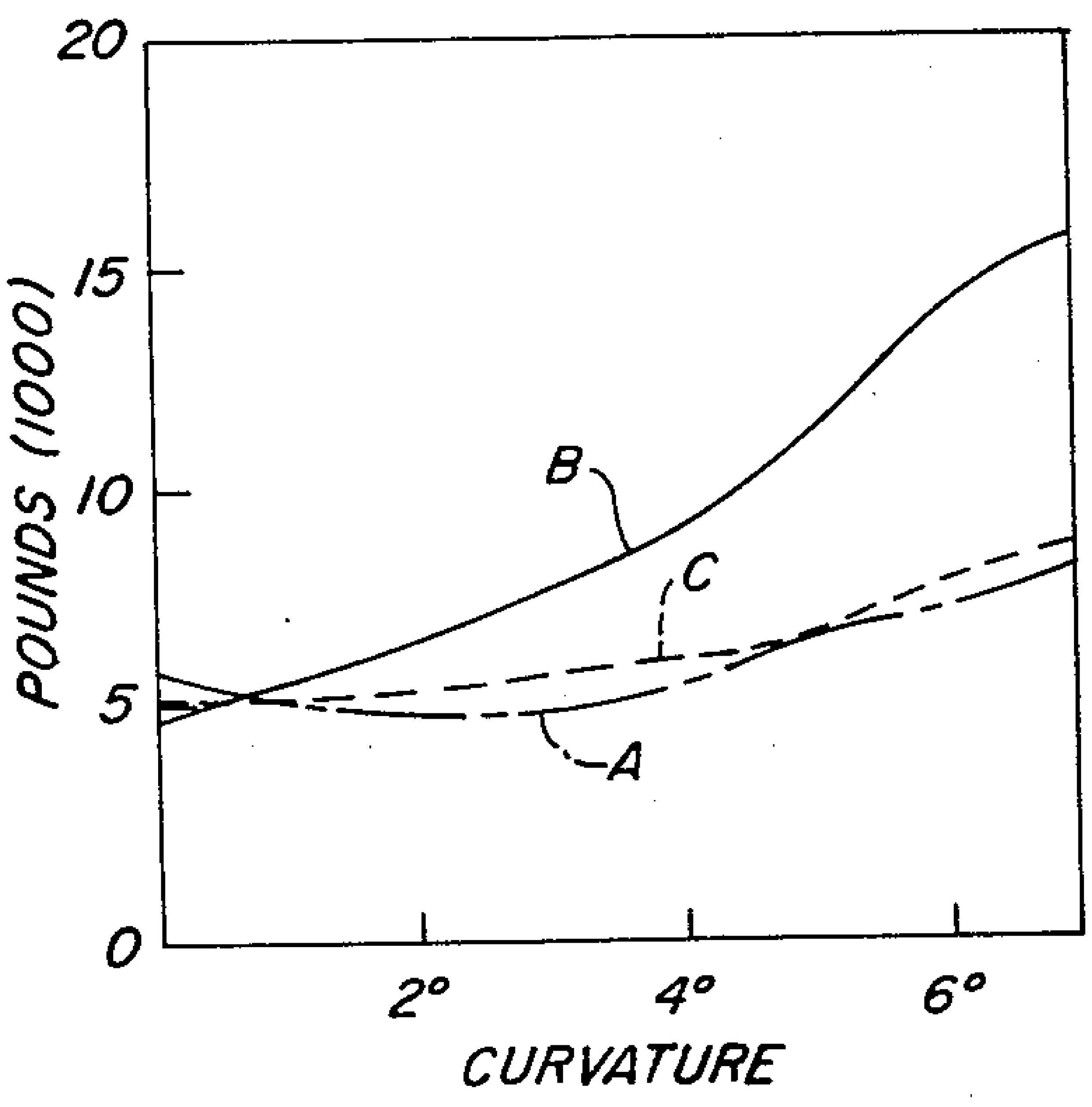


FIG. 13

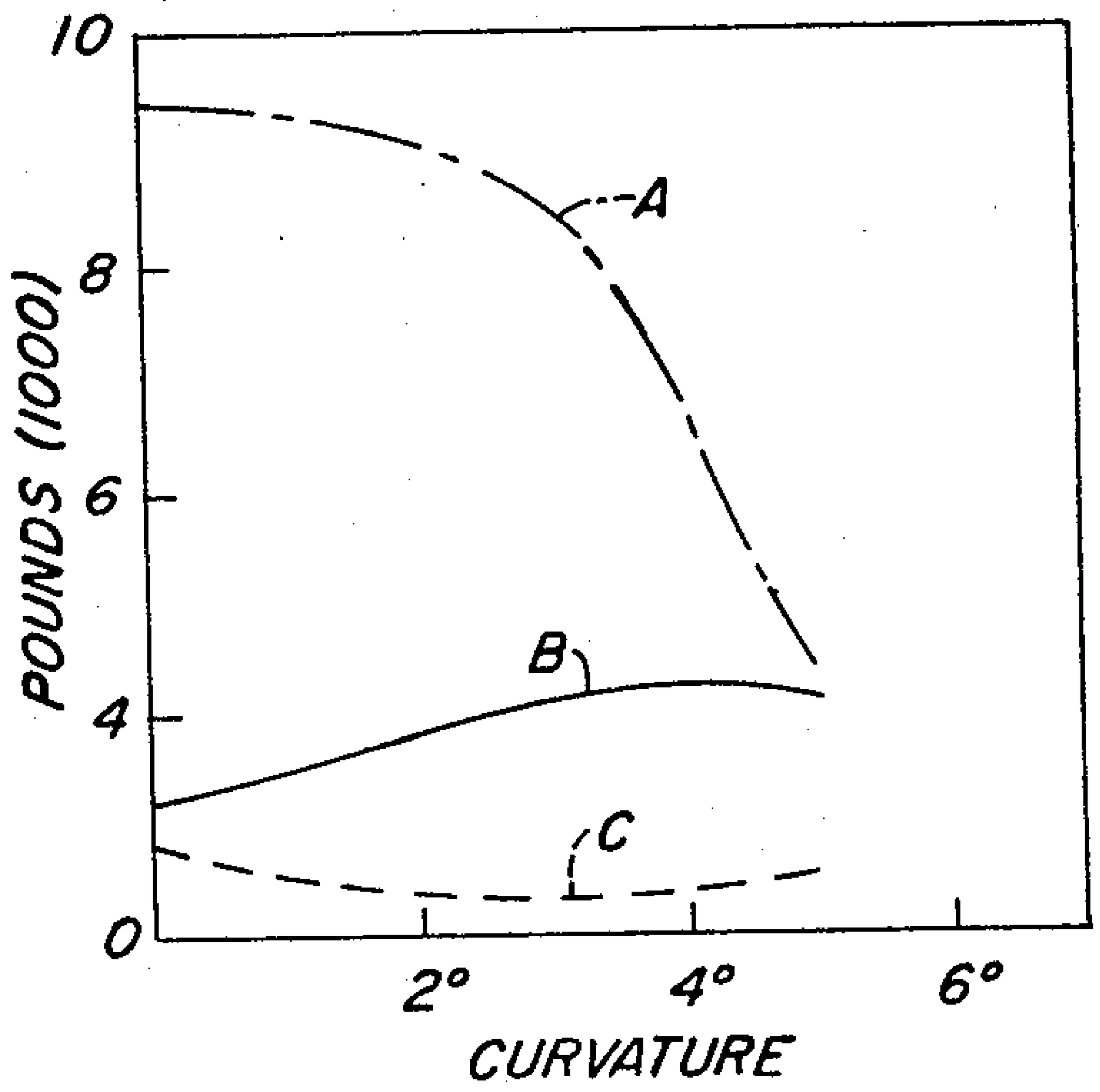


FIG. 14

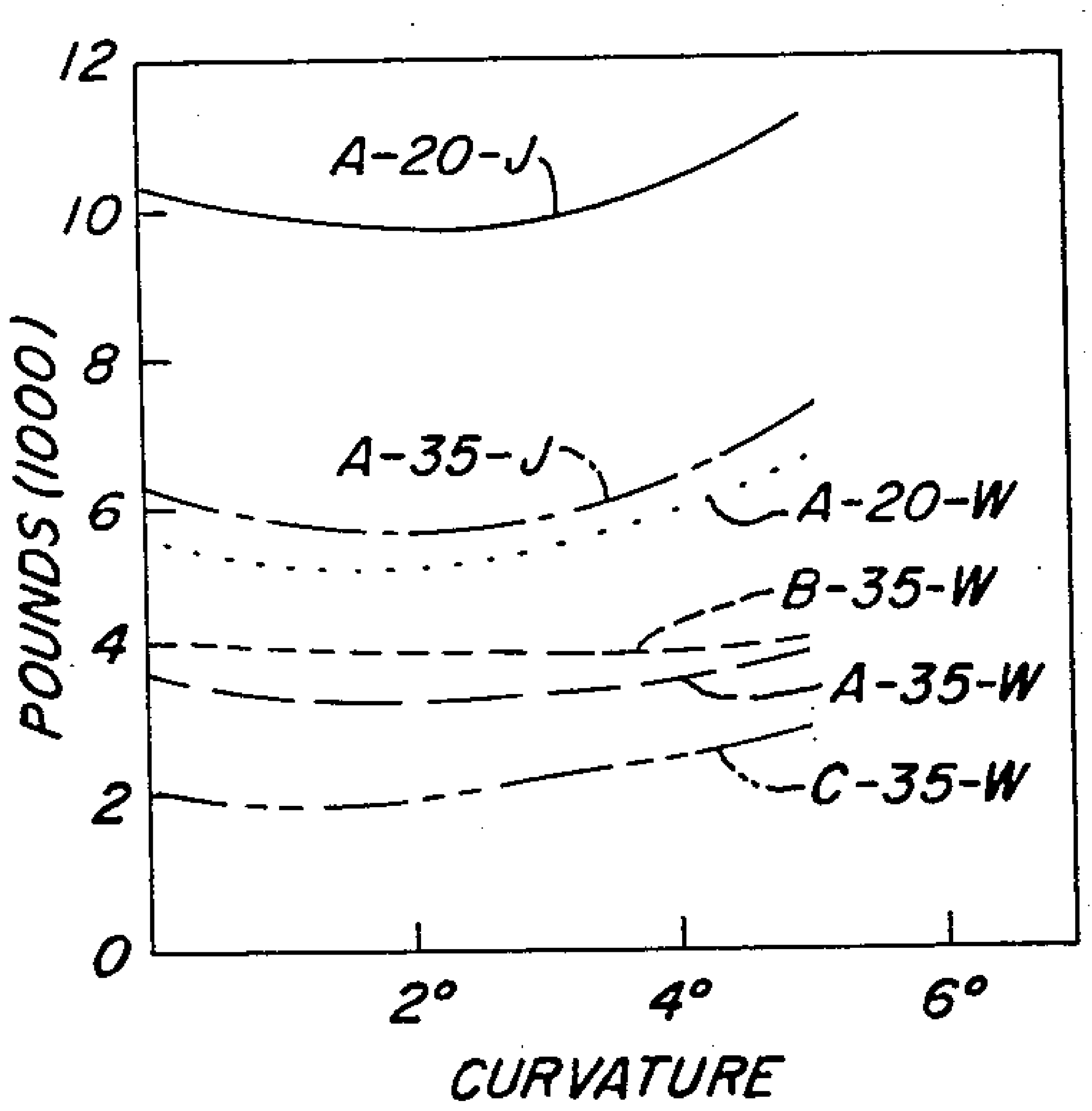


FIG. 15A

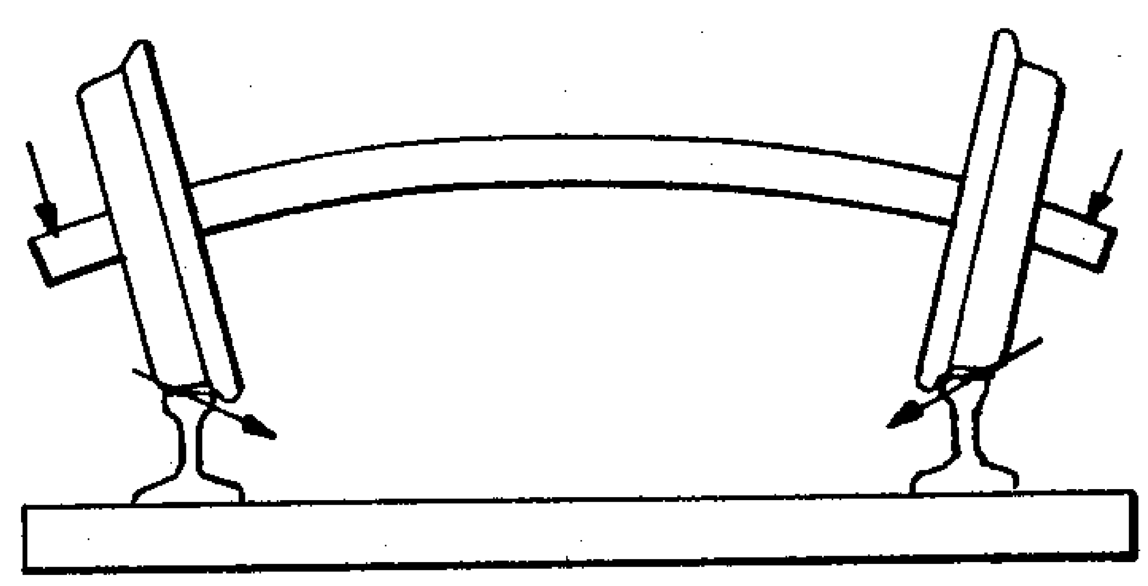


FIG. 15B

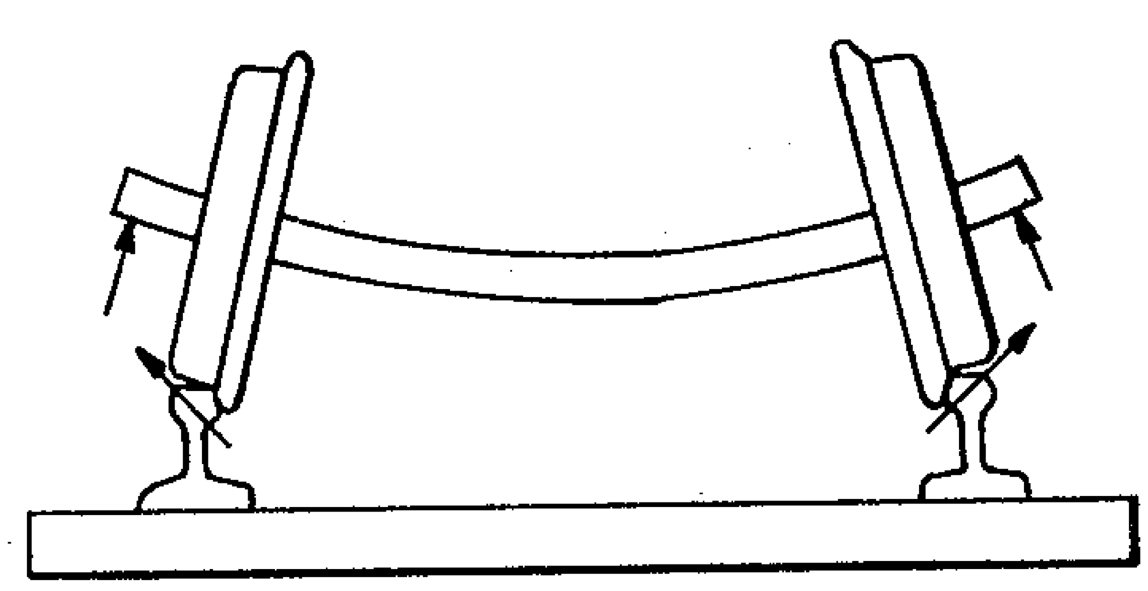


FIG. 16

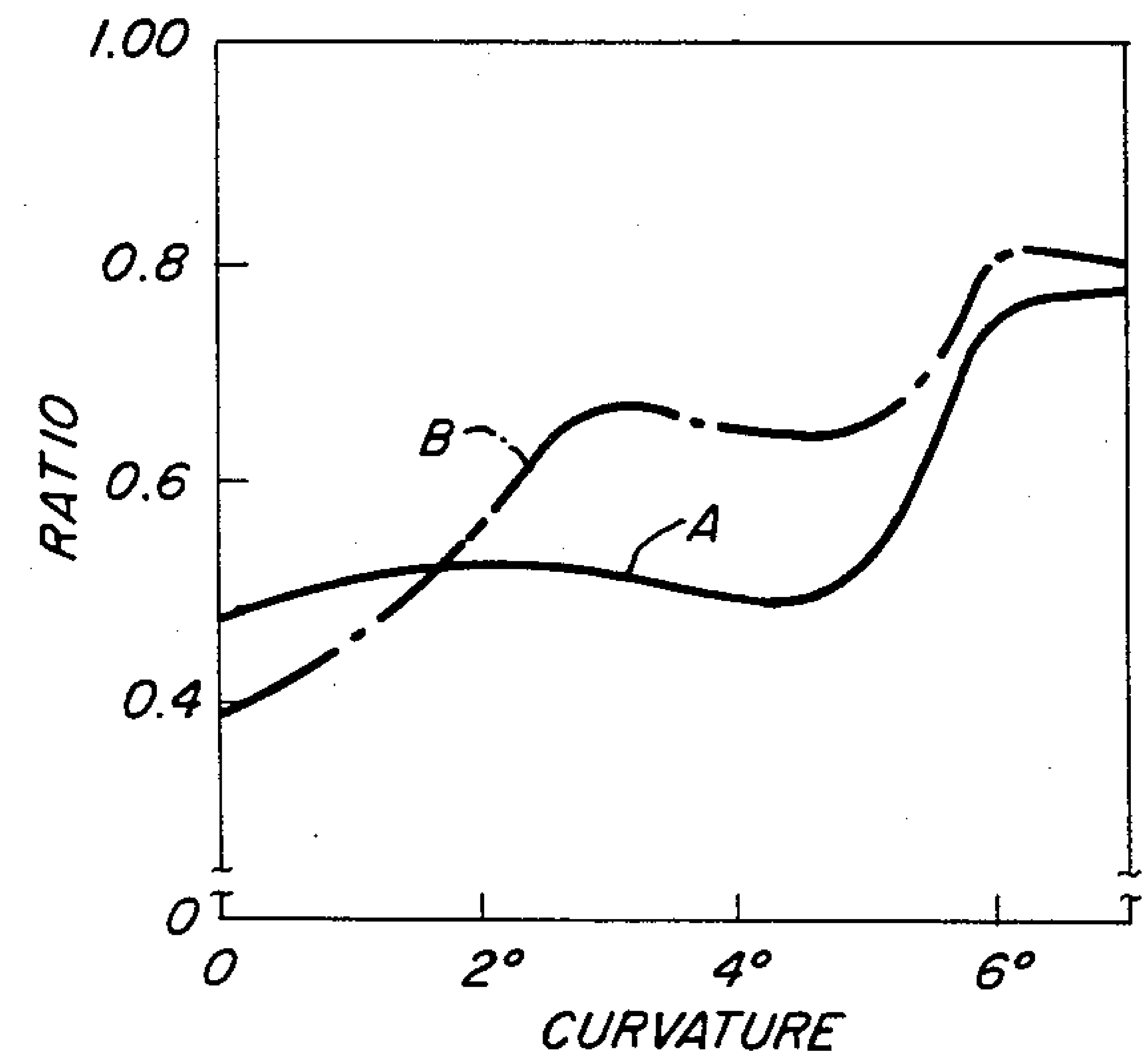


FIG. 17

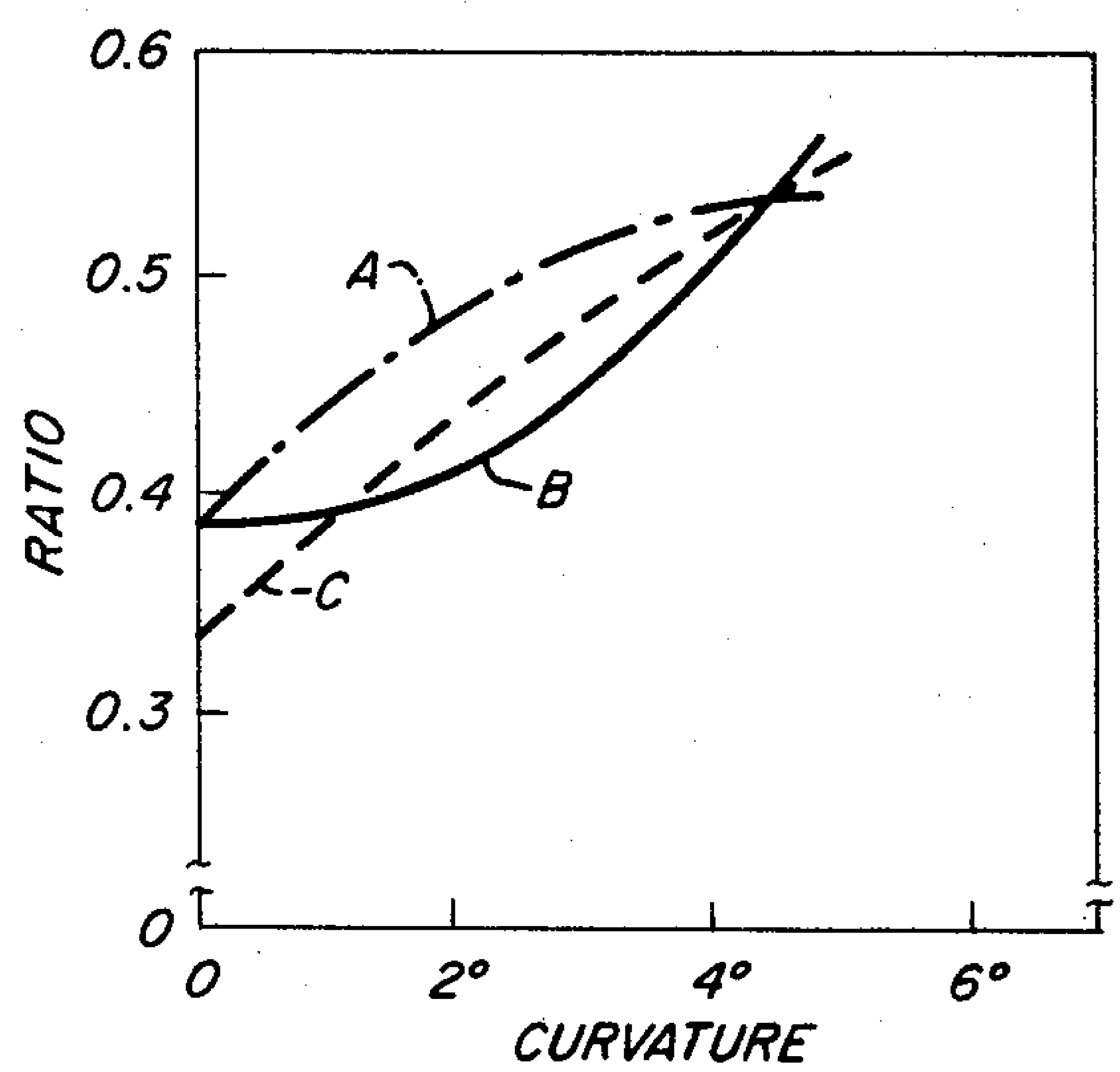


FIG. 18

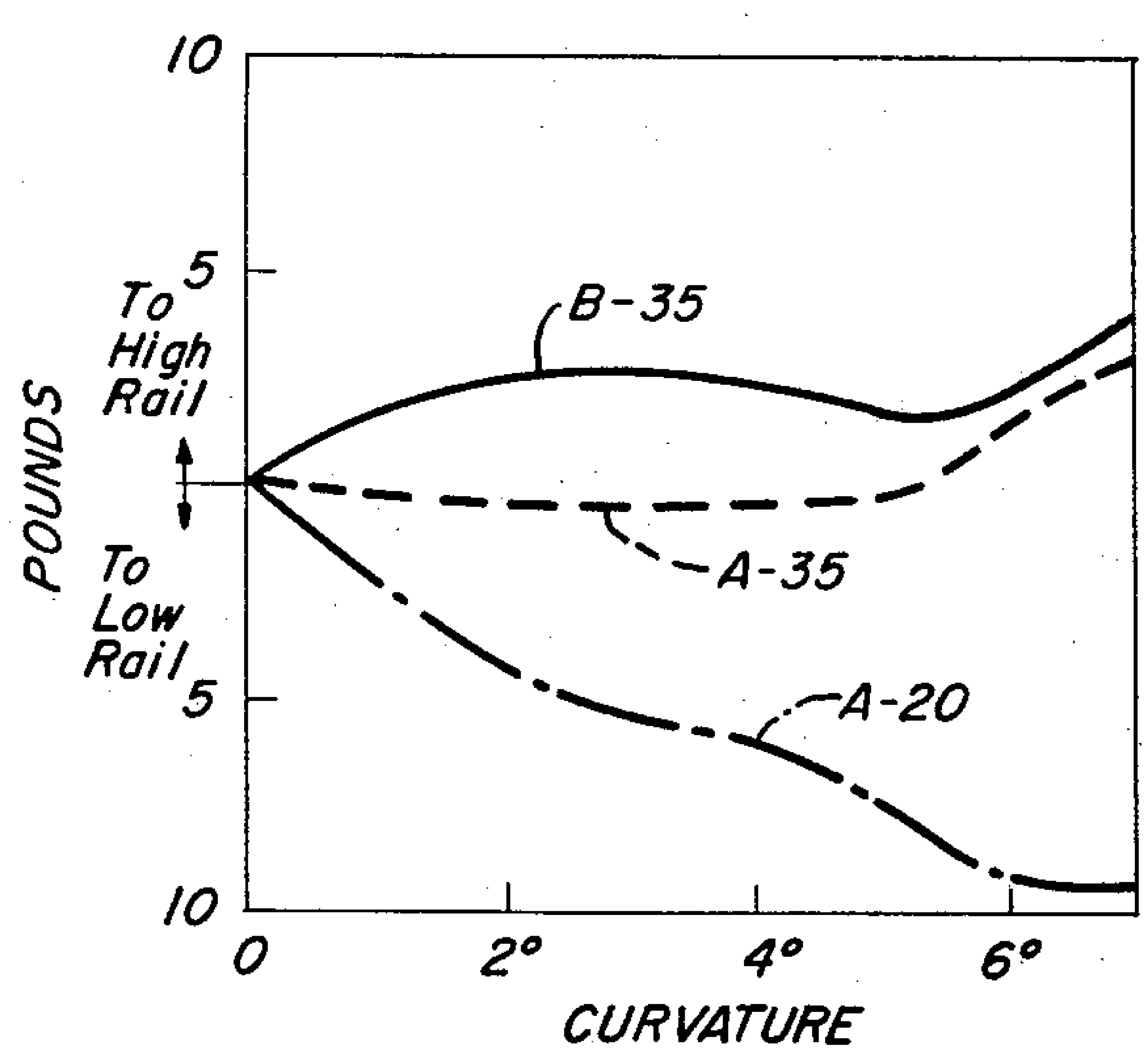
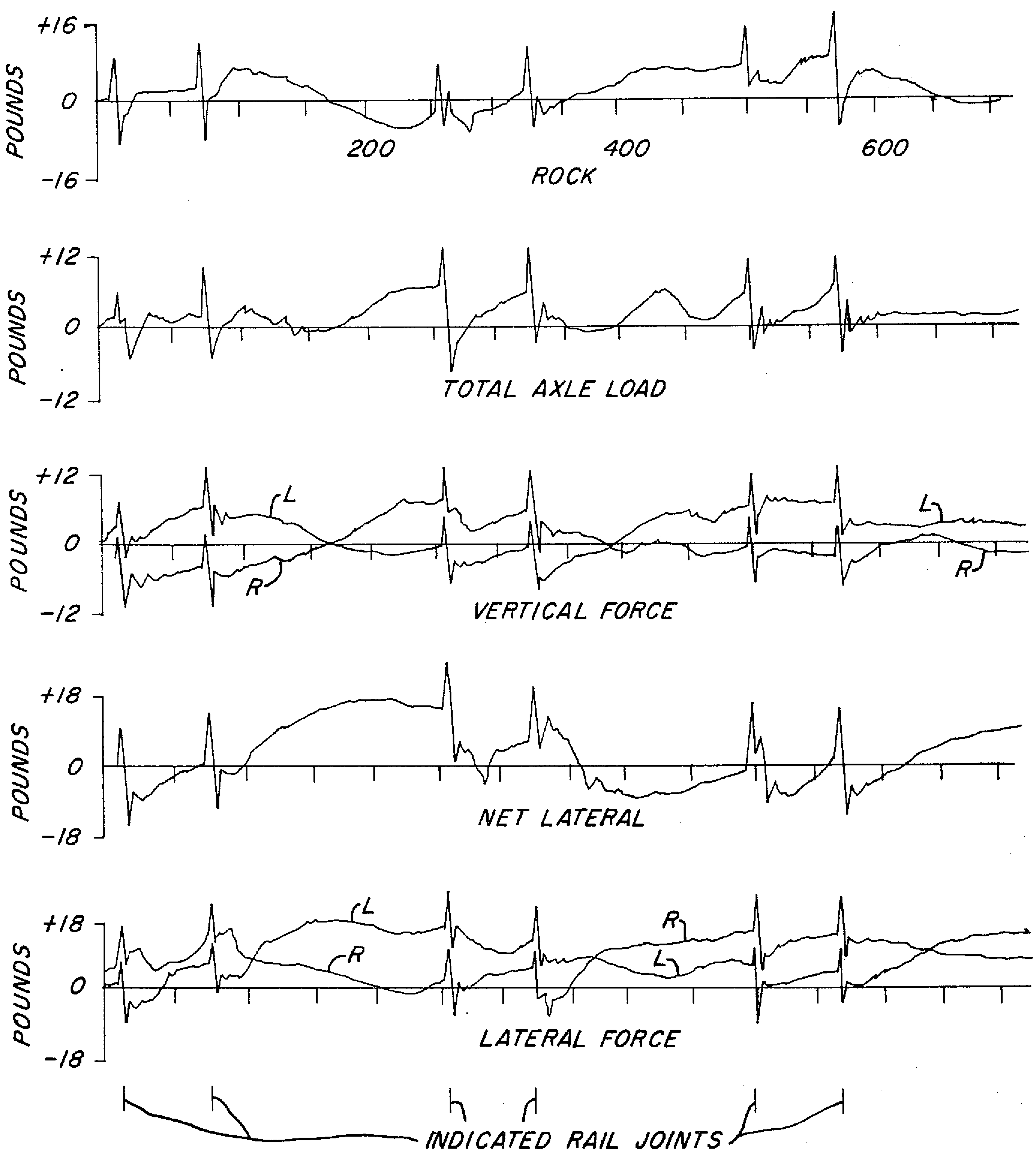


FIG. 19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POUNDS
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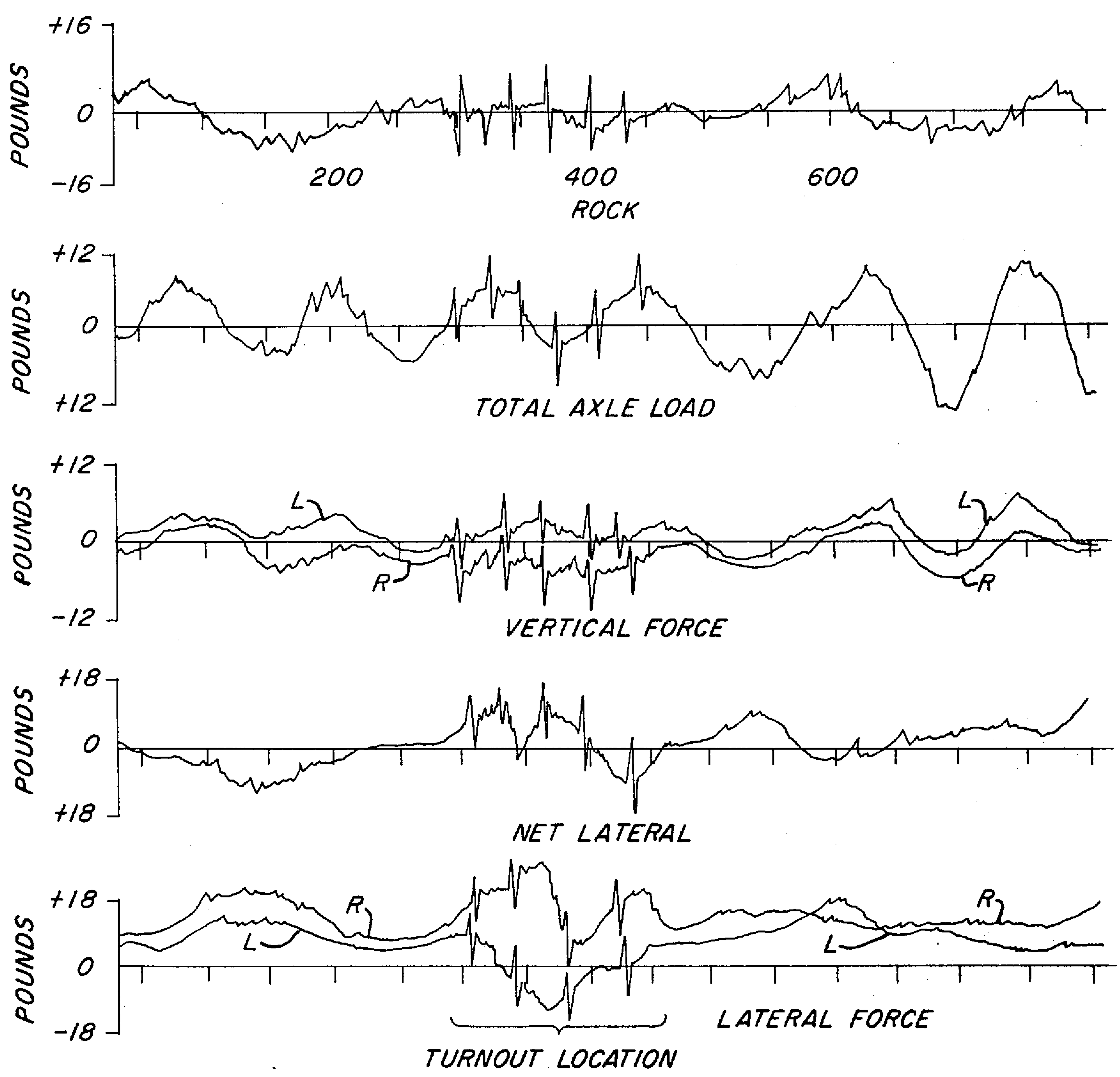
LATERAL FORCE
INDICATED RAIL JOINTS

FIG. 20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TURNOUT LOCATION

METHOD AND APPARATUS FOR EVALUATING RAILROAD TRACK STRUCTURE AND CAR PERFORMANCE

This invention relates to a method and apparatus for evaluating railroad track structure and car performance and more particularly, to a method and apparatus for determining the dynamic forces between wheels and rails, analyzing the results and applying the results of the analysis to evaluation of truck design and track conditions.

There is a continuous and steady effort within the railroad industry to carry heavier loads, to achieve higher speeds and to reduce derailments while at the same time to decrease car and track maintenance. The ability to set track standards to meet these requirements, as for example, for a given unit train hauling operation or for a defined high speed passenger run in order to prevent severe and accelerated rail deterioration or a relatively high ratio of derailments, is highly depended upon knowledge of the dynamic response of components of cars to track input. The dynamic response is significantly affected by car conditions, such as the dimensional relationships and characteristics of the bearings, springing and snubbing in the suspension system of the cars and track conditions, such as alignment, vertical profile, lateral profile, gauge, cross-level and ballast characteristics.

There have been studies made of wheel-rail interaction. Two articles, one entitled "Lateral Forces Between Wheels and Rails" by P. E. Olson and S. Johnsson, A.S.M.E. Publication 60–RR–6 dated Apr. 20, 1960 and the other entitled "Lateral Loading Between Locomotive Truck Wheels and Rail Due To Curve Negotiation" by L. F. Koci and H. A. Marta, A.S.M.E. Publication 65–WA/RR–4 dated Nov. 11, 1965, describe limited studies on lateral forces. Both studies use the technique of placing strain gauges on the plate of a wheel at a radial distance supposedly insensitive to change in vertical load to measure lateral loads. Vertical loading was determined from static measurements. Interpretation of the data was inconclusive and apparently had limited usefulness. An article entitled "Research on the Operation Stresses in PATH Rail Car Axles, Drive Systems, Wheels and Rail Benders" by M. Youtar, A.S.M.E. Publication 66–RR–6 dated May 4, 1966, describes a study directed to identify causes of premature cracking of axles with inboard bearings and based on axle strain measurements. Since the study was oriented towards axle cracking, the disclosure on the dynamics of wheel-rail relationships was incidental and not complete.

According to our invention, railroad car trucks were equipped with axle bending sensors and axle load cells for different truck configurations. Test equipment including signal conditioning equipment, a tape recorder, an analog to digital converter, a computer and readout equipment was located on an equipment car and connected to the car truck undergoing test. Test runs were made over a section of track having a wide variety of curve radii for each type of truck configuration under study. The computer was programmed to develop dynamic vertical and lateral forces throughout the run and then to provide statistical comparison of the data of car behavior for different track inputs.

The data is then presented in form to evaluate such items as the desirability of making a vehicle suspension

system more resilient, the desirability of reducing the unsprung weight of a car, altering car design characteristics which might influence natural periods of oscillation, reduction of static and dynamic axle loading, adjustment of track condition parameters to match operations and decreasing the frequency and level of lateral wheel-rail forces in curves. The data is then used for altering track conditions and for car design criteria to minimize lateral wheel-rail forces and to achieve the objective of moving cars along the track centered on the rail with uniform loading on all wheels on curves and tangent track.

It is therefore an object of our invention to provide apparatus for continuously measuring dynamic vertical and lateral forces between railroad wheels and rails.

Another object is to provide apparatus for analyzing the dynamic and lateral forces between railroad wheels and rails.

A further object is to provide a method for determining the dynamic vertical and lateral forces between railroad wheels and rails.

A still further object is to provide a method of evaluating design criteria for railroad car trucks.

Still another object is to provide a method for evaluating railroad track conditions.

Yet another object is to provide a method for evaluating track maintenance and detecting track defects.

These and other objects will become more apparent after referring to the following drawings and specification in which:

FIG. 1 is a schematic drawing of the testing apparatus of our invention.

FIG. 2 is a force diagram illustrating the force calculations of our method.

FIG. 3 is a partial side view of a truck showing a pad which will absorb part of the lateral energy transmitted to the wheel.

FIG. 4 is a partial computer produced graph for a type C truck, first axle, trailing truck, tangent track, welded rail at 35 miles per hour.

FIG. 5 is a partial computer produced graph for a type A truck, first axle, lead truck, 7° curve, jointed rail at 35 miles per hour.

FIG. 6 is a partial computer produced graph showing various forces for a type A truck, first axle, trailing truck, tangent track, welded rail at 35 miles per hour.

FIG. 7 is a partial computer produced graph showing various forces for a type B truck, first axle, trailing truck, tangent track, welded rail at 35 miles per hour.

FIG. 8 is a series of charts showing the cumulative frequency distribution of lateral wheel forces in thousands of pounds for three types of trucks as percent of time less than ordinate value.

FIG. 9 is a series of charts showing the cumulative frequency distribution of the lateral to vertical force ratio for three types of trucks on a 5° curve, welded rail, at 35 miles per hour as percent of time less than ordinate value.

FIG. 10 is a series of charts showing the cumulative frequency distribution of the lateral wheel force (A), dynamic vertical wheel force (B) and lateral to vertical force ratio (C) on the high rail side for a type A truck on a 5° curve at 35 miles per hour for welded rail and jointed rail as percent of time less than ordinate value.

FIG. **11** is a series of charts showing the cumulative frequency distribution of the net lateral force (A), dynamic axle rock (B) and dynamic axle loading (C) in thousands of pounds of type A, B, and C trucks on a 5° curve, welded rail at 35 miles per hour as percent of time less than ordinate value.

FIG. **12** is a chart showing the net lateral force, at one standard deviation, 1σ, above the mean of the frequency distribution of the measured values for three trucks at different curvatures on jointed rails at 35 miles per hour.

FIG. **13** is a chart showing the total axle load, at one standard deviation, 1σ, above the mean of the frequency distribution of the measured values for three type trucks of different curvatures on welded rails at 35 miles per hour.

FIG. **14** is a chart showing the load transfer across an axle at one standard deviation, 1σ, above the mean of the frequency distribution of the measured values on various degrees of track curvatures for different type trucks at different speeds for both jointed and welded rail.

FIGS. **15A** and B are grossly exaggerated depictions of the forces on rails and wheels due to alternate axle loading and unloading.

FIG. **16** is a chart showing the maximum wheel lateral to vertical ratio at one standard deviation, 1σ, above mean of the frequency distribution of the calculated values on different track curvatures at 35 miles per hour on jointed rails for two types of trucks.

FIG. **17** is a chart showing the maximum wheel lateral to vertical ratio at one standard deviation, 1σ, above mean of the frequency distribution of the calculated values on different track curvatures at 35 miles per hour on welded rails for three types of trucks.

FIG. **18** is a chart showing the average vertical truck load shifting in thousands of pounds to high and low rails on different curves for different trucks at 20 and 35 miles per hour.

FIG. **19** is a partial computer produced graph showing various forces in thousands of pounds for a type A truck, second axle, lead truck, tangent track, jointed rail at 20 miles per hour correlated with rail joints.

FIG. **20** is a partial computer produced graph showing various forces in thousands of pounds for a type A truck, first axle, lead truck, tangent track, jointed rail at 35 miles per hour correlated with a turn out.

Referring now to FIG. **1**, reference numeral **2** generally indicates the truck of a railroad car with side frames **4**, wheels **6**, and axles **8** to be tested. Two conventional electrical resistance strain gauges **10** are mounted on each axle on a common axis parallel to the axis of rotation of the axle and spaced apart from the wheels as shown by distances d and e in FIG. **2**. Leads **12** from strain gauges **10** are connected to slip rings **14** through holes in the axle. The slip rings **14** are connected through leads M to a car mounted signal amplifier and conditioner module **15**. Four conventional compression load cells using electrical resistance strain gauges **18** (only two are shown in FIG. **1**) are located between side frames **4** and axles **8** and are connected through lines B to module **15**. Module **15** is connected to an analog signal recorder **16** and to a conventional analog to digital converter **20** which is in turn connected to a general purpose digital computer **22**. Computer **22** may include a conventional cathode ray tube display **24** and a conventional teleprinter **26**. A digital tape recorder **27** is connected to computer **22**. A track side reference location detector **28**, such as a mile post detector, is connected to module **15**. Detector **28** may be either the metal detector type in which detecting the presence of a metal plate attached to a tie at the desired location provides a signal to the recorder **16**, a mechanical trip type which has a track side arm at each desired location for tripping a car mounted switch to provide a signal to the recorder or light source and photocell receiver. Recorder **16** is connected to converter **20**.

Determination of the dynamic vertical and lateral force reactions at each wheel is a particularly critical part of the measurements. While the load cells **18** provide a measure of the vertical force transmitted from car to axle, it is not a measure of the forces transmitted from wheel to rail. In order to determine the vertical and lateral wheel-rail forces, calibrated strain gauges **10** are used at known locations and positions on the axle to measure instantaneous bending moments.

Continuous measuring of bending strains at any known point in the revolving axle **8** is made by strain gauges **10**. The strain gauge will provide a sine wave output as the location undergoes bending with axle rotation. This output is due to gauge position and is not a direct measurement of the maximum bending strain except at the extreme points twice each revolution. More than one strain gauge may be used at each location to improve the accuracy of the determination of bending moments. Once the axle strain gauges are calibrated by well known static methods, the vertical and lateral forces are calculated using the following free-body based equations based on the force diagram of FIG. **2**. Under conditions of equilibrium using the forces and moment arms shown in FIG. **2**

$$\overline{M}_L = L_L c + \overline{B}_L (a+d) - V_L d \tag{1}$$

where a, c, and d are distances shown in FIG. **2**, $\overline{M}_L$ is the bending moment of the axle at the left point, $\overline{M}_L$ where a strain gauge **10** is located, L_L is the left lateral force, $\overline{B}_L$ is the left bearing force as measured by a load cell **18**, and V_L is the left vertical force. In addition,

$$\overline{M}_R = L_L c + \overline{B}_L (a+(f-e)) - V_L (f-e) \tag{2}$$

where a, c, e, and f are distances shown in FIG. **2**, $\overline{M}_R$ is the bending moment of the axle at the right point, $\overline{M}_R$ where a strain gauge **10** is located and $\overline{B}_L$ and V_L are as previously described. And,

$$\overline{M}_R = L_R c + \overline{B}_R (b+e) - V_R e \tag{3}$$

where b, c, and e are distances shown in FIG. **2**, L_R is the right lateral force, $\overline{B}_R$ is the right bearing force as measured by a load cell **18**, V_R is the right vertical force and M_R is as previously described. And,

$$\overline{M}_L - L_R c + \overline{B}_R (b+(f-d)) - V_R (f-d) \tag{4}$$

where b, c, d, and f are distances shown in FIG. **2** and $\overline{M}_L$, L_r, $\overline{B}_R$, and V_R are as previously described.

The equations are solved simultaneously R*pair to provide* V_L, V_R, L_L, and L_R as follows:

$$V_L = \frac{\bar{M}_L - \bar{M}_R}{f-d-e} + \bar{B}_L \quad (5)$$

$$V_R = \frac{\bar{M}_R - \bar{M}_L}{f-d-e} + \bar{B}_R \quad (6)$$

$$L_L = \frac{\bar{M}_L - \bar{M}_R}{f-d-e}\left(\frac{b-e}{c}\right) + \frac{\bar{M}_R}{c} - \bar{B}_L\left(\frac{a}{c}\right) \quad (7)$$

$$L_R = \frac{\bar{M}_R - \bar{M}_L}{f-d-e}\left(\frac{b-d}{c}\right) + \frac{\bar{M}_L}{c} - \bar{B}_R\left(\frac{b}{c}\right) \quad (8)$$

In addition by way of definition,

$$L_N = L_R - L_L \quad (9)$$

where L_R is the right lateral force, L_L is the left lateral force and L_N is the net lateral force which will have a sign indicating its direction.

The desired vertical and lateral forces are obtained by substituting the known dimensions and measured quantities for the appropriate terms. A resultant computed value having a negative sign indicates that the force vector is in the opposite direction to that shown in FIG. 2.

Although the physical dimensions *a, b, c, d, e,* and *f* represent known measured constants, it is recognized that force application loci, such as the vertical rail-wheel locations, actually do shift on the wheel tread as the car rolls along the rails and that the force vectors in general are not always oriented on or acting in a simple manner as illustrated in the diagram. However, the magnitude of error introduced by treating the moment arms as constant appears to be well within acceptable limits.

For the determination of truck design and performance characteristics, a number of trucks are equipped with strain gauges and load cells, at least one truck for each type of truck for which the particular design and performance characteristics are to be compared. The instrumented trucks are used on loaded or empty freight cars and the connections B and M of FIG. 1 are made by cable to a test car in which the remainder of the components of FIG. 1 are installed. The test car may be conveniently located in a train consist of loaded and/or empty freight cars and the test car.

The test train is then run along a section of track at various speeds. The section of track preferably includes an adequate variety of curvatures and other track conditions to insure the accumulation of sufficient representative data. For example, a 15-mile section of track which included 6 miles of tangent track, curves from 1° to 8° and grades up to 0.8 percent gave very satisfactory results. Speeds should be relatively constant, such as a run at 20 miles per hour and a run at 35 miles per hour, with other runs over specific track sections at 25, 30, 40, and 45 miles per hour. Instrumented trucks should be placed in both leading and trailing positions.

The strain gauges and load cells are calibrated statically before a run. During the run the data is continuously fed into the module **15**. The track side reference location detector **28** simultaneously records location signals for later synchronization. Module **15** includes amplifiers for altering all the signals recorded to be compatible with the inputs to the analog to digital converter. The signals are then processed by the computer in specific ways for specific purposes.

The data shown hereafter includes measurements from three different types of railroad trucks designated as a type A car equipped with standard plain bearings, a type B car equipped with standard roller bearings and a type C car equipped with standard roller bearings and lateral energy absorbing pads. FIG. 3 shows a lateral energy absorbing pad **30** placed between axle **8** and side frame **4**. Four lateral pads are provided on each truck.

FIGS. 4 and 5 are computer produced graphs illustrating one method of displaying the data acquired on test runs and processed by the computer. FIG. 4 is for a type C car, first axle, trailing truck on tangent track with welded rails at 35 miles per hour. FIG. 5 is for a type A car, first axle, lead truck on a 7° curve with jointed rails at 35 miles per hour speed. The abscissae are shown marked in units, each **357** units represents one second as shown in FIG. 4A. The same scale is used in the remainder of FIG. 4 and FIGS. 5, 6, 7, 19 and 20. Only a small fragment of the total run is shown.

FIG. 4A shows the output of one of the strain gauges **10** expressed as a bending moment in foot pounds.

FIG. 4B shows the total axle load in pounds as determined from gauges **18** in FIG. 1. The total axle load may be expressed as

$$\bar{B}' = \bar{B}_R' + \bar{B}_L' \quad (10)$$

where $\bar{B}_R'$ and $\bar{B}_L'$ are the algebraic deviations from static values. $\bar{B}'$ is a measure of car bounce.

FIG. 4C shows the calculated vertical forces V_R' and V_L' as deviations from static values.

FIG. 4D shows a measure of car rock in pounds and may be expressed as

$$R_R' = \bar{B}_R' - \bar{B}_L' \quad (11)$$

where R_R' is a measure of rock based on the right wheel and $\bar{B}_R'$ and $\bar{B}_L'$ are deviations from static values.

FIG. 4E shows the dynamic lateral forces L_L and L_R where a positive value indicates an inward force applied to the wheel and a negative value indicates an outward force applied to the wheel.

FIG. 4F shows the dynamic net lateral force expressed as

$$L_N = L_R - L_L \quad (9)$$

where L_N is the dynamic net lateral force based on the right wheel and L_R and L_L are the dynamic lateral forces.

FIG. 4G shows the ratio of the lateral to vertical forces for each wheel expressed as

$$\text{Ratio}_R = L_R/V_R \quad (12)$$

for the right wheel, and

$$\text{Ratio}_L = L_L/V_L \quad (13)$$

for the left wheel. The ratios are a measure of potential derailing tendencies.

FIG. 5 shows the same data as FIG. 4, but for a type A car, first axle, lead truck on a 7° curve, jointed rail at 35 miles per hour.

By comparing many of the instantaneous lateral and vertical forces and through appropriate combinations and calculations instantaneous values of other forces and force relationships, such as ratios, rack, bounce, and other factors, it is possible to determine many design criteria for car trucks and to evaluate track conditions.

However, we have found that merely comparing average values of force did not disclose the true details of the relationships between car and track. Accordingly, the data was rearranged to distribute instantaneous force measurements into various range groupings and to determine the percent of time the forces were at each of these designated levels. This analysis was programmed into the computer.

Such range groupings of forces are the true measure of car behavior on a given section of track. Table 1 shows the data of FIG. 4 separated into frequency distribution groups; averaged, and the value of first standard deviation from the mean determined by conventional statistical methods. When these techniques were applied to the accumulated data, we found that each car, and each minor modification to car characteristics, has a unique and identifiable response to a given set of track input conditions which would repeat under the same set of circumstances.

TABLE 2

Vertical Axle Loading (Thousands of Lbs.)

Truck Type	Range of Total Axle Load During Long Term Bounce Cycles Tangent	5° Curve	Characteristic Dynamic Peak to Peak "Shocks" Tangent	5° Curve
A	44 – 84	56 – 72	0.5	5
B	58 – 70	59 – 69	1.5	5
C	58 – 70	60 – 68	1.0	3

The type A truck was equipped with a free spring travel system and the type B truck was equipped with friction snubbing in the spring group. The total axle load as shown in FIG. 6 shows a relatively smooth sinusoidal curve of large magnitude whereas the snubbed car in FIG. 7 shows the effects of snubbing by reducing the magnitude but creating high frequency vibrations, about 60–80 Hz.

TABLE 1.—FREQUENCY DISTRIBUTIONS FOR SELECTED MEASUREMENTS

[Type C car on tangent track—1st axle, trailing truck 35 m.p.h., welded rail]

Freq. dist.	Measurement	Force level ranges (100 lbs.) −51 to −75	−26 to −50	−0 to −25	0 to 25	26 to 50	51 to 75	76 to 100	Average (100 lbs.)	Std. dev. (σ) (100 lbs.)
1	Dynamic part of vertical force (V_L') (Fig. 4C), percent		6	41	48	5			−1	17
2	Dynamic part of vertical force (V_R') (Fig. 4C), percent		6	45	46	3			1	17
3	Dynamic total axle force (Fig. 4B), percent	1	15	33	36	13	2		0	25
4	Dynamic axle rock ($Rock_R$) (Fig. 4D), percent		4	31	59	7			3	17
5	Lateral force (L_L) (Fig. 4E), percent				9	75	16		39	13
6	Lateral force (L_R) (Fig. 4E), percent				21	53	25	1	39	17
7	Net axle lateral force (L_N) (Fig. 4F), percent		2	47	46	5			1	17
					L/V Ratios 0 to .25	.26 to .50				
8	L_L/V_L Ratio (Fig. 4G), percent				91	9			.16	.07
9	L_R/V_R Ratio (Fig. 4G), percent				79	21			.18	.10

VERTICAL FORCE COMPARISONS

Transmittal of the load vertically to the axle bearings and eventually to the wheel-rail contact area is a much more complex occurrence when a car is moving as compared to when a car is standing still. The dynamic pattern of load transmittal as a car traverses a given section of track is determined chiefly by the suspension system of the truck, such as bolsters, springs, bearings, wedges or snubbers, and the magnitude may vary considerably depending upon the characteristics and clearance between these components.

Dynamic load transfer from one side of an axle to the other is termed "rock," equation 11, and has been long recognized as a potential cause for derailment. Rock is associated with rail joint spacing and truck type and the relation with truck type is shown in FIG. 11B. FIGS. 4D and 5D illustrate rock by the computer produced graph method. FIG. 14 also shows the nature of rock for different type cars at different speeds for both jointed and welded rail, the type C car exhibits the highest capability for absorbing shocks.

Bounce, equation 10, is the simultaneous and synchronous loading of all bearings of the trucks of a car. Car bounce is shown in FIGS. 4B and 5B, and FIG. 13 shows the bounce comparison of the three type trucks, type C exhibits the smallest bounce tendency. Table 2 shows a summary of vertical loadings and high frequency peak to peak shocks.

In-depth examination of the dynamic vertical forces in relationship to the simultaneous lateral forces showed that the outboard bearing on axles transforms pure vertical force input at the bearing into well defined lateral as well as vertical force elements at the wheel-rail contact area. Vertical loading and unloading of the axle, such as rock or bounce, will produce a reciprocating lateral reaction at the rail tending to force the rail inward and outward respectively. This effect is grossly exaggerated diagrammatically in FIG. 15A for loading and FIG. 15B for unloading. An axle will not, of course, bend as shown in FIG. 15B, but the axle has a tendency towards being, thus creating the wheel-rail forces as shown.

LATERAL FORCE COMPARISONS

The type B truck, equipped with roller bearings, allows only a small amount of lateral movement between the side frames and axles. Such truck is commonly called a "rigid" truck as compared to the "flexible" plain bearing truck, type A, which has considerable more lateral movement tolerance, or the type C which has the lateral pads to facilitate lateral movement. The differences are shown in FIGS. 8, 11A, and 12. These comparisons may also be made for any particular change in dimension to optimize the lateral force effects.

The dynamic lateral force frequencies correlates with the distances (7 to 12 inches) between rail corrugations at the predominant train speed.

This type of analysis may also be used to determine the effectiveness of lateral energy absorbing pads by comparing different dimensions, different materials, and different compressibilities.

LATERAL TO VERTICAL RATIOS

A potential derailing tendency for each type of truck was indicated by the ratio of dynamic lateral to vertical forces for each wheel. This is shown in computer produced graph form in FIG. 4G and FIG. 5G. FIGS. 9 and 10C compare the ratio on two different types of track and on high and low side of curves. FIG. 16 compares the ratio for types A and B trucks over jointed rail at 35 miles per hour over several curves, and FIG. 17 is a similar comparison, but on welded rails. Obviously, the lateral to vertical ratio could be calculated using net lateral force on each axle, or total net truck lateral force as well as different vertical forces could be used to develop potential derailment tendencies. FIGS. 16 and 17 show that under certain conditions the type A truck has a much higher probability of derailment on curves than the type B truck because it develops higher ratios of instantaneous lateral to vertical forces at each wheel, which means that for a given lateral force attempting to push one of its wheels off the rail, it has a lower vertical force tending to keep the wheel seated on the rail, for a greater portion of time. Car bounce, as determined by equation 10, significantly contributes to potential derailment tendencies and effects the lateral to vertical ratio as shown in FIG. 5.

TRUCK DESIGN CRITERIA

The car equipped with type B truck as compared to the cars equipped with type A truck or the type C truck exhibited a definite pitching action on curves, i.e., the weight was being transferred back and forth between the rear and the front truck. This tendency is determined by comparing the instantaneous vertical forces on all four axles. In addition, the type B truck had a tendency to rotate, as through it were a rigid rectangle, about the lead wheel on the high rail side of the curve. On the other hand, the type A and C trucks allow lateral movement of the axles relative to the side frame and truck bolsters have the capability of absorbing the minor lateral profile changes of the rail by a back and forth hunting movement of the wheel and axle units only, without the heavier mass of the car and lading. This independent lateral freedom allows such trucks to accommodate to track inputs, such as lateral irregularities, curve alignment and gauge variations, without the higher mass reinforced pivoting and pitching action of more rigid designs. The relative effect of pivoting action may be determined by comparing the charts of FIG. 8 and is further shown in FIG. 11A.

Other car truck design criteria that may be determined in a similar manner include the effectiveness of suspension systems and damping devices, optimal wheelbase, spacing between wheels on an axle, allowable lateral motion of truck components, clearances between truck components and wheel tread contours. This technique can also be used to determine that best compromise in two or more car truck design features. For example, referring to FIGS. 6 and 7, the type A car has a higher dynamic axle load than the type C car and therefore more truck wear and damage potential, but the type C car, because of the snubbing effect produces lower axle forces at high frequency levels which can result in a corrugated rail wear pattern.

TRACK CONDITIONS

The rails, roadbed, and general track structure conditions act as stimuli for inducing various reactions in passing cars. Track geometry, such as alignment, vertical profile, lateral profile, gauge, cross-level rail joint conditions and ballast characteristics very noticeably affect car behavior. Statically measured track dimensions do not necessarily yield on accurate index as to what a car will sense under moving or dynamic conditions because physical measurements of gauge, cross-level, alignment and the like under no-load conditions do not necessarily reflect the acceptability of the track under operation conditions. This is particularly true of roadbed resiliency. The methods already described for evaluating truck features are also used in evaluating track conditions by comparing truck reactions with different track inputs.

As an example, FIG. 13 clearly identifies the effect of track curvature on total axle load for the three type trucks. FIGS. 14, 16, 17, and 18 also illustrate the effect of curves. Table 3 shows a typical lateral force for different degrees of curvature for a type b truck, first axle, high rail, welded rail at 35 miles per hour.

TABLE 3

Lateral Force, (Lbs.)

	average	maximum	% of time incurred
Tangent Track	4000	10000	1.4
3° Curve	6000	16000	0.2
5° Curve	11500	17000	19.4

This table indicated that curves over 3° generally result in much higher forces, further indicated in FIG. 12. This information is useful in car truck design because, if there are little or no curves greater than 3°, the design could include smaller axles and other components need not be designed to handle large stresses.

While the railroad industry has known that welded rail has advantages over jointed rail, there is no known complete evaluation of the advantages of welded rail. The method of our invention provides such an evaluation. For example, Table 4 compares maximum values for various measurements for jointed and welded rails.

TABLE 4

Comparison of Maximum Values For Various Measurements Jointed Vs. Welded Rail Type A Truck on 5° Curves at 35 MPH

Measurement	Rail jointed	Rail welded	% reduction
1. Vertical Dynamic Wheel-Rail Force (1000 Lbs.)	14	8	43
2. Lateral Wheel-Rail Force (1000 lbs.)	21	16	24
3. Net Lateral Force Per Axle (1000 lbs.)	16	8	50
4. Rock Per Axle (1000 lbs.)	15	8	47
5. L/V Force Ratio per Wheel	1.10	0.85	23
6. Axle Bending Moment (1000 lb.–ft.)	46	38	17

In addition, FIG. 10 illustrates comparisons concerning lateral forces, vertical forces and lateral to vertical ratios for a type A truck.

While car rock has already been discussed, our method showed that repetitive car rocking occurred on jointed rail but only at widely separated locations on welded rail. Maximum rocking amplitudes on jointed track consistently correspond to joint half intervals of 19½ feet for staggered joints of 39 foot rail lengths during speed ranges of 18 to 25 miles per hour. Above 25 miles per hour, rail joint input produced very short duration shock impacts instead of the longer term resonant reactions at lower speeds. This is illustrated in FIGS. 19 and 20.

An analysis of the behavior of the three type trucks when travelling at 35 miles per hour over three separate but statically similar 5° curves showed that the true equilibrium speed is not solely a function of the degree of curvature and the amount of super-elevation of the curve. On the first curve, the type B and type C trucks had lateral and vertical forces indicating speeds above equilibrium. On the second curve, the type C truck appeared to be at equilibrium speed and the type A and type B trucks appeared to be below equilibrium speed. All of the trucks appeared to be below equilibrium speed on the third curve. This points out that there is a difference between static, no-load measured super-elevation and the elevation sensed by different types of moving cars. FIG. 18 shows one method of comparing equilibrium behavior based on average vertical truck load shifts between high and low rails on selected curves at constant speeds of 20 and 35 miles per hour.

There are a number of other track conditions that affect car behavior that may be evaluated by our method. For example, as shown in FIG. 19, the location of the rail joints is shown when car rock is plotted on a computer production graph. FIG. 20 shows the location of a turnout on a computer produced graph display. It is also possible to distinguish between trailing and facing movements in turnouts. Other track conditions that may be evaluated by this method include gauge-curve relationship and gauge control.

Track condition in relation to a standard may be established by using a test car equipped with a truck of known dynamic characteristics. When the test car is run over track, the computer may be programmed to detect any measured or calculated force or force relationship that deviate from an established standard.

TRACK MAINTENANCE

As an aid to track maintenance track conditions may be monitored by using the same method. A measuring car equipped with trucks having load cells, axle strain gauges and the necessary recording equipment is run over the section of track for which conditions are to be monitored. For this run a characteristic base is developed which may take the form, for example, of the information shown in FIG. 4 converted to a cumulative distribution. The results of another test run performed at a suitable monitor test interval, for example monthly, is then compared with the base run and any significant deviation in the information from the base run, such as magnitudes, frequencies, or occurrences of forces or ratios can be detected. By use of the track side reference location detector signal, the location of the deviation may be determined, or by use of the CRT display 24, the computer 22 may be programmed to display the deviation as it occurs. This method could be used to monitor the condition of track joints, turnouts, ballast movement, ballast resiliency and localized track defects, such as shelled spots, engine burns, spalling of rail head and partial fracture of head. Where defects create their particular pattern on strip charts, the computer may be programmed to identify the defect.

We claim:

1. Apparatus for evaluating railroad track structure and car performance comprising a railroad car truck including two side frames, an axle and a pair of wheels mounted on said axle, a pair of load cells mounted between said axle and each side frame for providing a first signal and a second signal responsive to the vertical load of each truck side frame on said axle, a pair of electrical resistance strain gauges mounted on said axle for providing a third signal and a fourth signal responsive to the bending moments in said axle as it rotates under load, and means connected to said load cells and said strain gauges for combining said signals and calculating the vertical and lateral wheel-rail forces at each wheel.

2. Apparatus according to claim 1 in which said pair of strain gauges are mounted on a common axis on said axle parallel to the axis of said axle and spaced apart from, and in near proximity to, each wheel.

3. Apparatus according to claim 1 in which said means for calculating includes a signal amplifier and conditioner module, an analog to digital converter connected to said module and a general purpose digital process computer connected to said converter.

4. Apparatus according to claim 3 which includes a trackside reference location detector connected to said module.

5. Apparatus according to claim 3 which includes a printer connected to said computer, a cathode ray tube display connected to said computer, a digital tape recorder connected to said computer, and an analog recorder connected to said module and said converter.

6. A method for evaluating railroad track structure and car performance comprising the steps of continuously determining axle bending moments of a truck axle under load at two locations on the axle between the wheels of said axle as the wheels roll over a desired section of track, continuously determining the bearing force each truck side frame exerts on the car axle under load as the wheels roll over said section of track, and calculating the vertical and lateral wheel-rail forces for each wheel from said axle bending moments and said bearing forces.

7. A method according to claim 6 in which the bending moments are determined by strain gauges and the bearing forces are determined by load cells and which includes statically calibrating the strain gauges and load cells, measuring the distance between the axis of the axle and the surface of the tread of the wheel, measuring the distance between each load cell and the adjacent wheel, measuring the distance between each strain gauge and the adjacent wheel, measuring the distance between the wheel treads, and said calculating of vertical and lateral forces are performed by combining an output from each strain gauge, an output from each load cell and said measurements.

8. A method according to claim 7 in which the step of calculating the vertical and lateral forces comprises

the steps of simultaneously solving the four free-body equations

$$\overline{M}_L = L_L c + \overline{B}_L (a+d) - V_L d$$

$$\overline{M}_R = L_L c + \overline{B}_L (a+(f-e)) - V_L (f-e)$$

$$\overline{M}_R = L_R c + \overline{B}_R (b+e) - V_R e$$

$$\overline{M}_L = L_R c + \overline{B}_R (b+(f-d)) - V_R (f-d)$$

for L_L, L_R, V_L, and V_R where L_L is the lateral force on the left wheel, L_R is the lateral force on the right wheel, V_L is the vertical force on the left wheel, and V_R is the vertical force on the right wheel and in which equations $\overline{M}_L$ is the calibrated strain gauge output adjacent the left wheel, $\overline{M}_R$ is the calibrated strain gauge output adjacent the right wheel, $\overline{B}_L$ is the calibrated load cell output adjacent the left wheel, $\overline{B}_R$ is the calibrated load cell output adjacent the right wheel, a is the measured distance between the left wheel and the adjacent load cell, b is the measured distance between the right wheel and the adjacent load cell, c is the measured distance between the axis of the axle and the surface of the tread of the wheel, d is the measured distance between the left wheel and the adjacent strain gauge, e is the measured distance between the right wheel and the adjacent strain gauge, and f is the measured distance between the wheel treads.

9. A method according to claim 8 which includes the steps of converting the outputs of the strain gauges and load cells from analog to digital form and calculating force relationships and in which said calculations are made by a general purpose digital computer.

10. A method according to claim 9 which includes recording the outputs of the strain gauges and the load cells in digital form.

11. A method according to claim 10 which includes the steps of simultaneously recording with said outputs periodic trackside reference location indicator signals.

12. A method according to claim 9 which includes the steps of determining the vertical and lateral forces for all axles of a truck.

13. A method according to claim 9 which includes the steps of determining the lateral and vertical forces for all axles of a car.

14. A method according to claim 8 which includes calculating the net lateral force on the two wheels of a single axle according to the equation

$$L_N = L_R - L_L$$

where L_R is the lateral force on the right wheel, L_L is the lateral force on the left wheel, and L_N is the net lateral force which will have a positive sign if toward the right wheel and a negative sign if toward the left wheel.

15. A method according to claim 8 including calculating the total dynamic axle load $\overline{B}'$, according to the equation

$$\overline{B}' = \overline{B}_R' + \overline{B}_L'$$

where $\overline{B}_R'$ is the deviation of the right load cell output from static value expressed as a force and $\overline{B}_L'$ is the deviation of the left load cell output from static value expressed as a force.

16. A method according to claim 8 including the steps of determining dynamic vertical forces V_L' and V_R' where V_L' is the deviation from static value of V_L and V_R' is the deviation from static value of V_R.

17. A method according to claim 8 including calculating dynamic axle rock according to the equation

$$R_R' = \overline{B}_R' - \overline{B}_L'$$

where $\overline{B}_R'$ is the deviation of the right load cell output from static value expressed as a force, $\overline{B}_L'$ is the deviation of the left load cell output from static value expressed as a force, and R_R' is the dynamic axle rock base on the right wheel expressed as a force.

18. A method according to claim 8 including calculating the lateral to vertical force ratio, R, according to the equation

$$R = L/V$$

where L and V are the calculated lateral and vertical forces.

19. A method according to claim 13 including calculating the transfer of vertical forces to and from leading and trailing axles.

20. A method according to claim 8 including determining the frequencies of occurrence of repetitive deviations from static values of lateral and vertical forces.

21. A method according to claim 9 in which said calculations are performed for increments of travel over said sections of track thereby providing a series of values for vertical forces, a series of values for lateral forces and a series of values for each force relationship and which includes placing each value of a series in a range group, determining the frequency of occurrence of values in said range group, determining the mean of values of said group and determining one standard statistical deviation from the mean of values of said group.

22. A method according to claim 9 in which said calculations are performed for increments of travel over said section of track thereby providing a series of lateral and vertical force values, and which includes measuring the transfer of vertical forces from one rail to the other rail on curved track.

23. A method according to claim 9 in which said calculations are performed for increments of travel over said section of track thereby providing a series of lateral and vertical force values and which includes grouping said values according to track curvature.

24. A method according to claim 9 in which the vertical forces, lateral forces, and force relationships are determined for a first car truck design criterion and a second car truck design criterion and which includes the steps of comparing said forces and force relationships for the first criterion and the second criterion.

25. A method according to claim 24 in which the design criterion is the size of the suspension springs and the damping of the suspension system.

26. A method according to claim 24 in which the design criterion is the effectiveness of devices controlling movement between truck side frames and axles.

27. A method according to claim 24 in which the design criterion is the lateral clearance in the car axle assembly.

28. A method according to claim 24 in which the design criterion is the lateral clearance between truck bolster and side frames.

29. A method according to claim 24 in which the design criterion is the wheel base of the car truck.

30. A method according to claim 24 in which the design criterion is the wheel tread contour.

31. A method according to claim 24 in which the design criterion is the derailment tendency of the axle.

32. A method according to claim 24 in which the design criterion is the spacing between wheels on an axle.

33. A method according to claim 9 in which the vertical forces, lateral forces, and force relationships are determined for a car passing over a first section of track having a first condition and for said car passing over a second section of track having a second condition and which includes the steps of comparing said forces and force relationship for the first track condition and the second track condition.

34. A method according to claim 33 in which the track conditions are different track gauges on tangent track.

35. A method according to claim 33 in which the track conditions are different track gauges on curved track.

36. A method according to claim 33 in which the track conditions are the amount of super-elevation on curved track.

37. A method according to claim 33 in which the track conditions are track having welded joints and tracks having spliced joints.

38. A method according to claim 33 in which the track conditions are the amount of lateral profile.

39. A method according to claim 33 in which the track conditions are the amount of vertical profile.

40. A method according to claim 33 in which the track conditions are ballast resiliency of a first consistency and ballast resiliency of a second consistency.

41. A method according to claim 33 in which the track conditions are the amount of curvature in the track.

42. A method according to claim 9 which includes the steps of recording the values of the vertical forces, lateral forces, and force relationships for said section of track, determining a second set of vertical and lateral force values with the same rolling equipment, and comparing the recorded values with the second set of values.

43. A method according to claim 9 which includes the steps of establishing acceptable standards for values of said forces and force relationships for a test car, operating said test car on a second section of track, calculating said forces and force relationships for said test car for said second section of track and comparing said calculated values of said forces and force relationships for said second section of track with said standard values.

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