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AN INVESTIGATION OF WROUGHT STEEL RAILWAY CAR WHEELS

Part I

TESTS OF STRENGTH PROPERTIES OF WROUGHT STEEL CAR WHEELS

A REPORT OF AN INVESTIGATION
CONDUCTED BY
THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS
IN COOPERATION WITH
THE CARNEGIE-ILLINOIS STEEL CORPORATION

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AN INVESTIGATION OF WROUGHT STEEL RAILWAY CAR WHEELS

Part I

TESTS OF STRENGTH PROPERTIES OF WROUGHT STEEL CAR WHEELS

I. Introduction

1. *Purpose of Investigation.*—Steel wheels are being used rather widely on railway freight cars and almost entirely for passenger service. The recent development of the high-speed streamlined passenger trains and the use of greater wheel loads and higher speed freight trains indicate that car wheels are being subjected to greater service stresses than in the past. Insufficient information is available regarding the stresses and strains developed in wrought steel wheels by external loads. It seemed desirable, therefore, to study the action of steel wheels when subjected to laboratory tests using static loads applied approximately as the wheels are loaded in service.

The main purpose of the tests herein reported was to locate the most highly stressed portions of the wheel under several types of loading, and to calculate (from strain measurements) the magnitude of the larger stresses occurring in the wheel under a given set of loading conditions. This information may be used to estimate the stresses encountered under nominal design loads, and might indicate possible revisions in the shape of the wheel that would lead to greater strength. The investigation here described was confined to a study of the strength properties of wrought steel wheels of the standard A.A.R. design. Observations were made of the strains developed in wheels while being pressed on standard axles, and of the strains and deflections of wheel and axle when subjected to vertical loads and to horizontal flange loadings. Tests were made on medium carbon-steel wheels, and also on silico-manganese alloy-steel wheels. It was felt that if strength is the main item to consider, the silico-manganese alloy-steel wheel might replace the carbon-steel wheel because of the greater strength of the alloy steel.

A parallel investigation is being conducted in the Department of Railway Engineering to determine the resistance to heat checking, the coefficients of friction, and the thermal stresses caused by the application of brakes to steel wheels revolving at high speed. The results of these latter tests will be reported on at an early date, as Part II of this coöperative investigation.

2. *Outline of Tests.*—One pair of heat-treated, medium-carbon-steel wheels and one pair of silico-manganese-steel wheels were mounted on standard A.A.R. 5½ in. x 10 in. axles and subjected to loading tests simulating service conditions. In these tests an attempt was made to determine the distribution of strains that occurred in the wheels when they were being pressed (or mounted) on an axle, and when subjected to vertical or horizontal loads after they were fully mounted or assembled on an axle. The several types of test made might be outlined as follows:

(a) *Mounting Test.*—The first studies included the determination of the strains developed at various points in a wheel as it was being mounted, or pressed on to a steel axle (see Fig. 4). In these tests a "mounting tolerance" or "mounting allowance" (excess of axle diameter at wheel seat over the bore diameter of wheel) of about 0.008 in. was used. At increments of each quarter-inch movement of the wheel as the mounting progressed, observations were made of the mounting pressure. For every one-inch movement of the wheel, observations were made of the radial and circumferential strains in the wheel hub and plate. The results of these tests are discussed in Chapter IV.

(b) *Vertical Load Test.*—After two wheels had been mounted on an axle, the axle was supported on journals at each end, and vertical loads were applied to the treads of the wheels at standard gage distance, much as static loads would be applied in service (see Fig. 8). The increase or change in wheel strains over those produced in the mounting test was noted. Equal loads were applied to both wheels in increments, the maximum total load applied to the assembly being 300 000 lb. Strains and deflections of the axle were also measured, and the movements of the rims of each wheel relative to the plane of the hub were observed. These tests are discussed in Chapter V.

(c) *Flange Thrust Test.*—A third series of tests were made in which inward horizontal loads were applied to the wheel flanges employing the same general arrangement as that used in the vertical load test. A total vertical load of 60 000 lb. was placed on the wheel and axle assembly, and inward horizontal flange thrusts were applied to each wheel at the point of rail contact (see Fig. 12). The thrusts employed were 20 000, 40 000 and 60 000 lb., the vertical load in each case being maintained constant at 30 000 lb. per wheel. The increases in strains in wheel and axle, axle deflections, and wheel rim movements were noted for each increment of load. The description and results of these tests are given in Chapter VI.

(d) *Ultimate Load Test.*—A final test was made on one of the carbon-steel wheels to determine the extent of damage which might occur in a wheel due to a heavy static overload. The wheel and axle assembly was placed on the table of the testing machine with the wheel in a vertical plane and the bottom of the tread resting on a section of the head of a standard 130-lb. rail. A vertical load was applied by the movable head of the testing machine through a second rail head placed on the top of the wheel. This load was gradually increased to a maximum value of 1 100 000 lb. At increments of 250 000 lb. load, observations were made of the movements of the rim of the wheel relative to the hub both in the radial and lateral directions. The results of this test are given in Chapter VII.

(e) *Photo-Elastic Analysis.*—A photo-elastic analysis was made of a thin transparent bakelite model cut in the shape of the radial section of a standard A.A.R. wheel (see Figs. 19 and 20). The model was stressed in a field of polarized light by loading with vertical and lateral loads applied to the tread of the wheel. From the interference fringes produced in the model the distribution of the radial stresses along the edges (representing the wheel surface) of the model was obtained. Although the photo-elastic method employed is not directly applicable to the analysis of stresses in three dimensions such as exist in the wheel, the tests were made in the expectation that their results would give an indication of the relative values of radial stresses and stress concentrations that exist at the sharply curved fillets on the wheel surface where it was difficult to get strain readings on the actual wheel. These tests are described in Chapter VIII.

3. *Acknowledgment.*—This investigation has been conducted by members of the staff of the Department of Theoretical and Applied Mechanics as part of the work of the Engineering Experiment Station in coöperation with the Carnegie-Illinois Steel Corporation. The work has been under the administrative direction of DEAN M. L. ENGER, Director of the Engineering Experiment Station, and PROFESSOR F. B. SEELY, Head of the Department of Theoretical and Applied Mechanics. PROFESSOR F. E. RICHART gave valuable assistance in the planning of the tests.

The Carnegie-Illinois Steel Corporation has been represented in the planning and general arrangements for the tests by the following persons from its staff:

MR. R. K. BOWDEN, Manager, Metallurgical Division, Chicago District.

MR. C. T. RIPLEY, Chief Engineer of the Technical Board of the Wrought Steel Wheel Industry.

MR. A. D. BEERS, Chief Metallurgist, Gary Works.

MR. A. N. SWANSON, Assistant-Chief Metallurgist, Gary Works.

MR. R. B. LUCAS, Superintendent, Wheel & Axle Mill, Gary Works.

Mr. H. B. WISHART, Metallurgist, Gary Works.

II. INFORMATION REGARDING WHEELS TESTED

4. *Design of Wheels and Axles.*—All of the wheels tested were rolled steel wheels made at the Gary Works of the Carnegie-Illinois Steel Corporation. Four heat-treated (quenched and tempered) steel wheels and two normalized and tempered steel axles were used in the tests. One comparison pair of wheels was made of medium carbon steel (Wheels Nos. 3331 and 3332), heat 85325, and the second pair was made of silico-manganese steel (Wheels Nos. 4891 and 4892), heat 84282. The chemical compositions of these heats are:

PER CENT COMPOSITION

Heat No.	C	Mn	P	S	Si	Cr
85325	0.57	0.75	0.017	0.027	0.26	....
84282	0.58	0.76	0.021	0.025	1.98	0.18

The wheels were of a S-1261 section, 36-inch diameter, multiple wear, with a 7-inch wheel bore. The axles were of the standard A.A.R. type* for 5½ in. x 10 in. journal with a 7-inch nominal diameter wheel seat. The wheels and axles were finished with mounting allowances that varied from 0.0078 to 0.0085 in.

5. *Physical Properties of Steels.*—A portion of the investigation was devoted to determining the physical strength properties of the material in the heat-treated rolled steel wheels as finished for service. The purpose of these tests was to determine the strengths of repre-

*See: "Wheel and Axle Manual," Association of American Railroads, (Revised 1935 edition).

sentative samples of the steel in the hub, rim, and plate, and to obtain a value for the modulus of elasticity of the material in order that the strain measurements made during the tests might be translated to equivalent stress values. In the manufacture of a member of such complicated shape, all the steel does not receive the same amount of working in forming the various parts of the wheel. This condition, together with the unequal rates of cooling, might be expected to cause a variation in strength of the steel between hub, rim, and plate of the wheel.

For the purpose of determining the strength properties of the various parts of a wheel, test specimens were cut from finished wheels that were rolled from the same heats of steel and subjected to the same heat treatments as the test wheels but which were not subjected to the test loads. Tension and compression specimens were cut from the rim and hub in a circumferential direction in both the front* and back* faces of the wheels, and from the plate of the wheels in a radial direction. Thus the tests included a total of eight tension and eight compression specimens of material cut from a carbon-steel wheel and eight specimens of each kind cut from a silico-manganese-steel wheel. The details of the test specimens, and the portions of the stress-strain curves up to the yield point of the steel are shown in Fig. 1a for the tension tests and in Fig. 1b for the compression tests. The physical properties of the steels obtained from these data are listed in Table 1. Because of its lack of significance the ultimate compressive strengths of the steels were not found, but nearly all compression specimens were loaded to at least 110 000 lb. per sq. in. without fracture and the specimens did not show a large amount of lateral expansion at this stress.

An examination of Table 1 shows that the yield points and ultimate strengths of the specimens taken from various parts of a wheel were fairly uniform, except for those taken from the hub, which showed lower strengths and lower ductility in both steels. This is probably to be expected, since this material cools slower, receives less working during manufacture, and is more likely to contain segregated impurities from the center of an ingot, than the other parts of a wheel. The average values of modulus of elasticity were found to be 29 300 000 for the compression tests and 29 700 000 for the tension tests.

*The "front" refers to the outside face of the wheel, and the "back" is the inside face as the wheels are assembled on an axle.

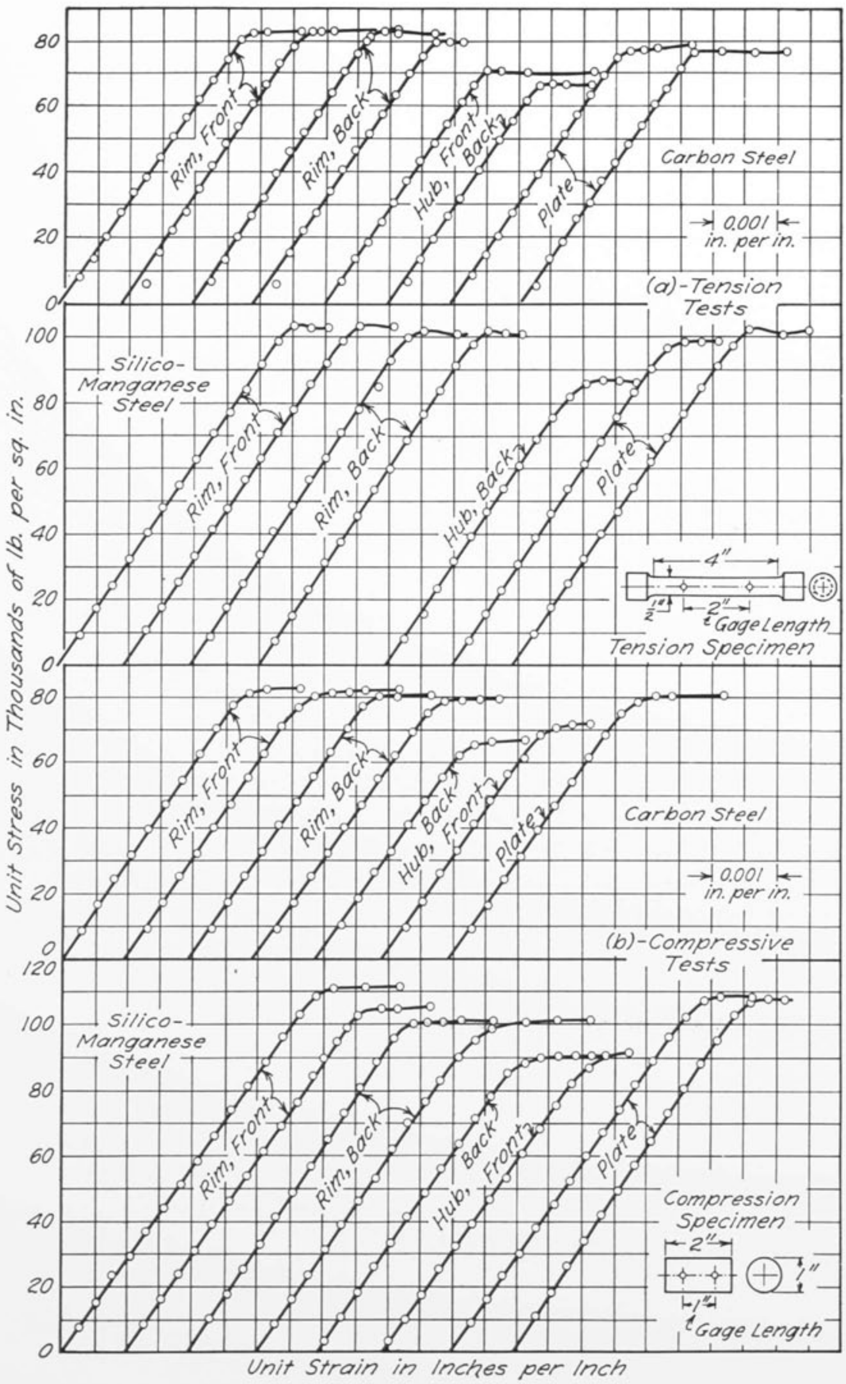
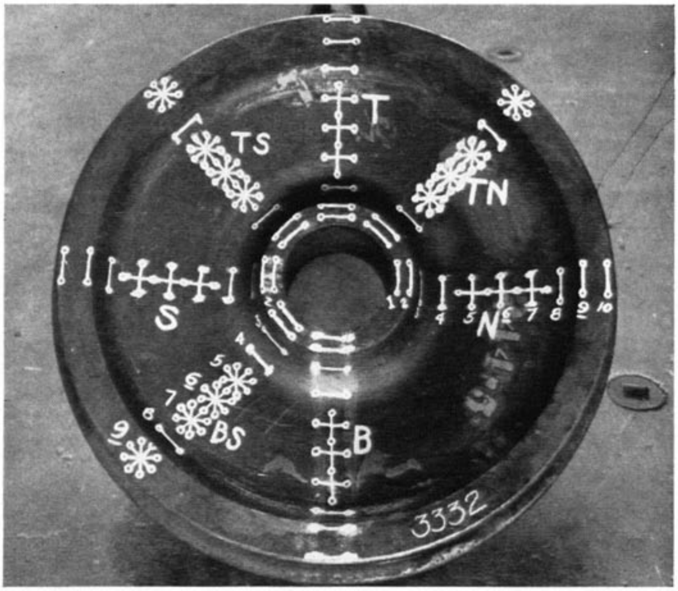


FIG. 1. TYPICAL TENSION AND COMPRESSION STRESS–STRAIN CURVES FOR STEEL FROM WHEELS

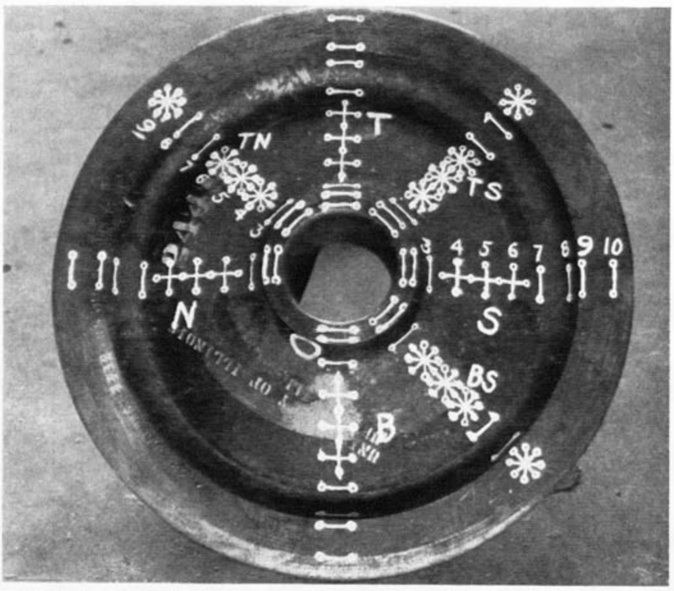
TABLE 1
PHYSICAL PROPERTIES OF WHEEL STEEL SPECIMENS

Specimen No.	Yield Point lb. per sq. in.	Tensile Strength lb. per sq. in.	Per Cent Elongation in 2 in.	Per Cent Reduction in Area	Average Modulus of Elasticity Millions of lb./sq. in.
Carbon Steel in Tension					
C2-RF*	82 500	134 900	17.5	37.8	29.3
C2-RF	83 000	134 800	18.0	36.9	30.2
C2-RB	82 000	133 000	17.0	38.0	30.2
C2-RB	80 000	131 000	18.5	38.2	29.7
C2-W	77 500	130 100	18.5	41.8	30.1
C2-W	77 000	135 000	19.0	41.7	28.9
C2-HF*	71 000	123 200	12.5	16.0	29.7
C2-HB	66 800	120 100	16.0	26.1	29.3
Average.......	77 480	130 260	17.2	34.6	29.7
Carbon Steel in Compression					
C2-RF	83 000				29.5
C2-RF	82 000				29.7
C2-RB	81 000				30.3
C2-RB	80 000				29.3
C2-W	80 000				28.7
C2-W	80 000				....
C2-HF	72 000				28.4
C2-HB	67 000				29.1
Average.......	78 120				29.3
Silico-Manganese Steel in Tension					
S2-RF	103 000	163 000	18.5	40.2	29.9
S2-RF	103 000	162 000	19.0	42.7	29.5
S2-RB	101 500	161 000	18.5	41.4	29.8
S2-RB	101 300	161 400	17.5	43.2	29.8
S2-W	98 600	159 000	18.5	43.8	29.6
S2-W	102 000	160 000	17.0	40.3	29.0
S2-HB	86 800	148 500	16.0	34.3	29.6
Average.......	99 430	159 300	17.9	40.8	29.6
Silico-Manganese Steel in Compression					
S2-RF	111 800				29.7
S2-RF	105 000				28.7
S2-RB	101 000				31.0
S2-RB	102 000				29.4
S2-W	109 000				28.4
S2-W	108 000				30.3
S2-HF	91 000				28.1
S2-HB	91 000				29.2
Average.......	102 350				29.3

*R, W, and H indicate rim, plate and hub, respectively; F and B refer to front and back faces of the wheel.



(a)



(b)

FIG. 2. LOCATION OF GAGE LINES FOR STRAIN MEASUREMENT ON WHEELS
(a) Outside face of wheel
(b) Inside face of wheel

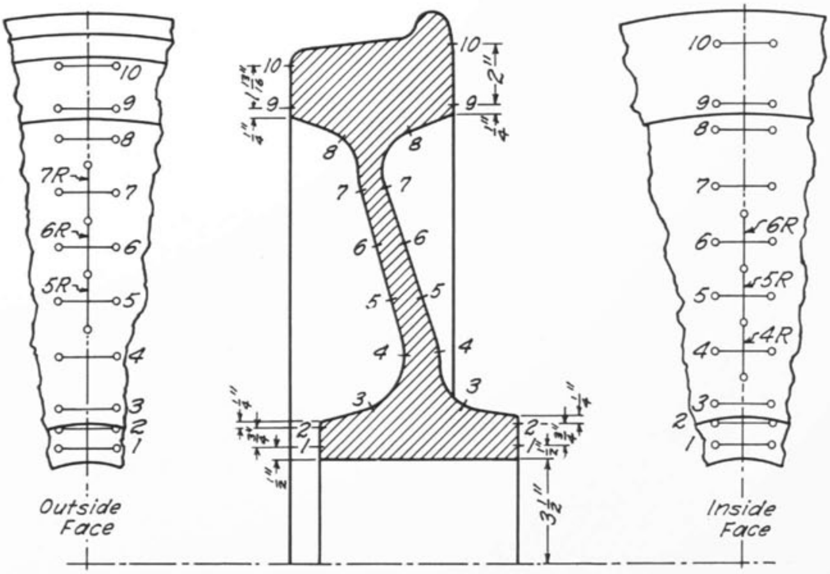


Fig. 3. Diagram Showing Location of Tangential and Radial Gage Lines

III. General Test Procedure

6. *Preparation of Wheels for Testing.*—The only preparation of the wheels and axles for testing consisted of locating and drilling strain gage holes, and of calipering of the wheel bore and axle diameters to determine the mounting tolerances. On each wheel tested, 230 strain gage lines of 2 in. gage length were located. A photograph of a wheel giving the location of these gage lines is shown in Fig. 2, and a diagram of the locations on a section of the wheel is shown in Fig. 3. In each wheel 7 radial sections or rows of strain gage lines were laid out on both faces of the wheel. With the wheel set in a vertical plane (as used in the loading tests) these rows consisted of a vertical diameter (T-B), a horizontal diameter (N-S), each containing 20 circumferential and 6 radial gage lines, and three diagonal rows (BS, TS, and TN). Each diagonal row contained 9 circumferential, 4 radial, 4 horizontal, and 4 vertical gage lines. Thus on the diagonal rows (on which the directions of the principal strains were not known) the strain rosettes made it possible to obtain the

strains on 4 intersecting gage lines at 45 deg. to each other at each of four points along the row; from these readings the principal strains and their directions could be obtained.

In all the data which follows, these gage lines will be designated by letters indicating the row or section of the wheel (see Fig. 2) on which the readings were taken, and by a number (1 to 10) giving the position along the row. The gage line numbers 1, 2, and 3 indicate locations on the hub, (see Fig. 3) 4, 5, 6, and 7 are on the plate, and 8, 9, and 10 on the rim of the wheel. Letters following the location number indicate the direction in which the strain was measured at the point. The subscript R is used for radial readings, while subscripts H and V indicate readings on a strain rosette in the horizontal and vertical directions with the wheel in position for test. All gage line locations designated by number only, (no subscript giving direction) indicate readings made in a tangential (circumferential) direction at that point.

A gage length of 10 in. was laid out on both the top and bottom of the axle at the center of its length, and gage holes were also located in the rim and hub of the wheel to accommodate special gages used to measure the lateral movements of the rim and the bending of the plate of the the wheel.

In laying out the gage lines templates were used to produce exactly the same locations on all four wheels tested. Since it was expected that most of the strains in the wheels would be small it was felt that the strains could be obtained with more accuracy by using a large number of gage lines and averaging the values obtained at a given point on each pair of wheels, and also (where possible) by averaging the values obtained at two positions on each wheel that were symmetrically located with respect to the applied loads.

In preparing the gage lines, gage holes were drilled to a depth of about $\frac{1}{16}$ in. with a No. 54 drill. The surface of the wheel at each gage hole was then ground smooth with a fine carborundum wheel to provide a smooth plane surface and remove the burrs from the drill holes. The gage holes must be carefully prepared with axes parallel and with the edges of holes properly dressed if accurate strain measurements are desired. For several locations on the wheel it was found difficult to produce good gage holes particularly on the more sharply curved surfaces of the wheel.

Before proceeding with the mounting test the wheel bore and axle diameters were obtained by calipering at three positions along the length of the seat on each of four planes containing the geometric axis of the axle. The averages of these twelve readings in each case were used in computing the mounting tolerances.

7. *Testing Equipment.*—A 2-in. Berry strain gage was used on the wheels and a 10-in. Whittemore gage was used on the axles. The observations of strain made with these gages were obtained with a precision of about 0.00005 in. per in. on the 2-in. gage and 0.00001 in. per in. on the 10-in. gage. Both gages were calibrated while the tests were in progress. Every observed reading was estimated to the nearest tenth of a division and check readings were then made to reproduce the original reading within about half a division.

For measuring deflections of the axle and lateral movements of the wheel rim, Ames dials were attached to special direct reading instruments or holders. On these dials one division represented a movement of 0.001 in.

All of the tests were made in a 3 000 000 lb. capacity Southwark-Emery hydraulic testing machine. While most of the loads employed were a small proportion of the capacity of the machine, it has been calibrated carefully, and the measured loads are believed to be correct within a tolerance of one-half of one per cent.

8. *Determination of Principal Strains.*—The principal strains at a point on the surface of a body may be defined as the maximum and minimum algebraic values of strain at that point. The negative (compressive) strains are interpreted as being smaller algebraically than the positive (tensile) strains even though they may be larger in numerical magnitude. The strains measured on the vertical diameter of the wheel (sections B and T) were principal strains because of their symmetrical locations with respect to the applied loads in all tests. The strains measured on the horizontal diameter (sections N and S) were not in the directions of principal strain because of the slight dissymmetry of loading in some of the tests. However, the observed strains on sections N and S were small in all cases, and it is assumed that they will serve as satisfactory values for comparison with the strains on other sections, but this was not true of the strains measured on the diagonal sections (BS, TS and TN) of the wheel. For this reason two additional strain gage lines (forming a strain rosette) were added at each of four locations along the diagonal sections; with the aid of these additional readings the principal strains could be obtained at these locations.

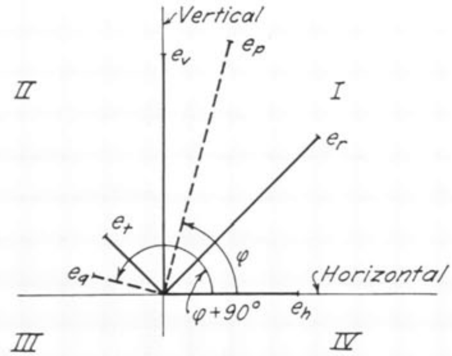
There are several different methods of obtaining the principal strains from the strains measured on the rosettes. A graphical method* employing the "Dyadic Circle" and an algebraic method† were used in this investigation; a number of the values reported in Tables 4 and 6 were obtained graphically and checked algebraically.

For measurements made on the strain rosettes (see Fig. 2) with four intersecting gage lines at 45 deg. to each other, algebraic equations for the maximum and minimum principal strains are as follows:

$$e_p = \frac{e_h + e_v + e_r + e_t}{4} + \tfrac{1}{2}\sqrt{(e_h - e_v)^2 + (e_r - e_t)^2}$$

$$e_q = \frac{e_h + e_v + e_r + e_t}{4} - \tfrac{1}{2}\sqrt{(e_h - e_v)^2 + (e_r - e_t)^2}$$

$$\text{and} \qquad \tan 2\varphi = \frac{e_r - e_t}{e_h - e_v}, \text{ in which:}$$



e_p = algebraic maximum principal strain
e_q = algebraic minimum principal strain
e_h = measured strain on horizontal gage line
e_v = measured strain on vertical gage line
e_r = measured strain on radial gage line
e_t = measured strain on tangential gage line
φ = angle between the horizontal and the direction of the principal strain e_p; φ is measured in a clockwise sense if the strain e_r lies in the second quadrant, or in a counterclockwise sense if e_r lies in the first quadrant, as indicated in the diagram. (Also $\varphi \pm 90°$ is the angle between the horizontal and the direction of the principal strain e_q.)

*W. R. Osgood and R. G. Sturm, "The Determination of Stresses from Strains on Three Intersecting Gage Lines and Its Application to Actual Tests," Journal of Research, Bureau of Standards, May 1933, Vol. 10 No. 5, p. 685.

†W. R. Osgood, "The Determination of Principal Stresses from Strains on Four Gage Lines 45 deg. Apart," Journal of Research, Bureau of Standards, December 1935, Vol. 15 No. 6, p. 579-581.

G. E. Beggs and E. K. Timby, "Interpreting Data from Strain Rosettes," Engineering News-Record, Vol. 120 No. 10, March 10, 1938.

In using this equation tensile strains are assigned a plus sign and compressive strains a negative sign; all values of e are unit strains (in. per in.).

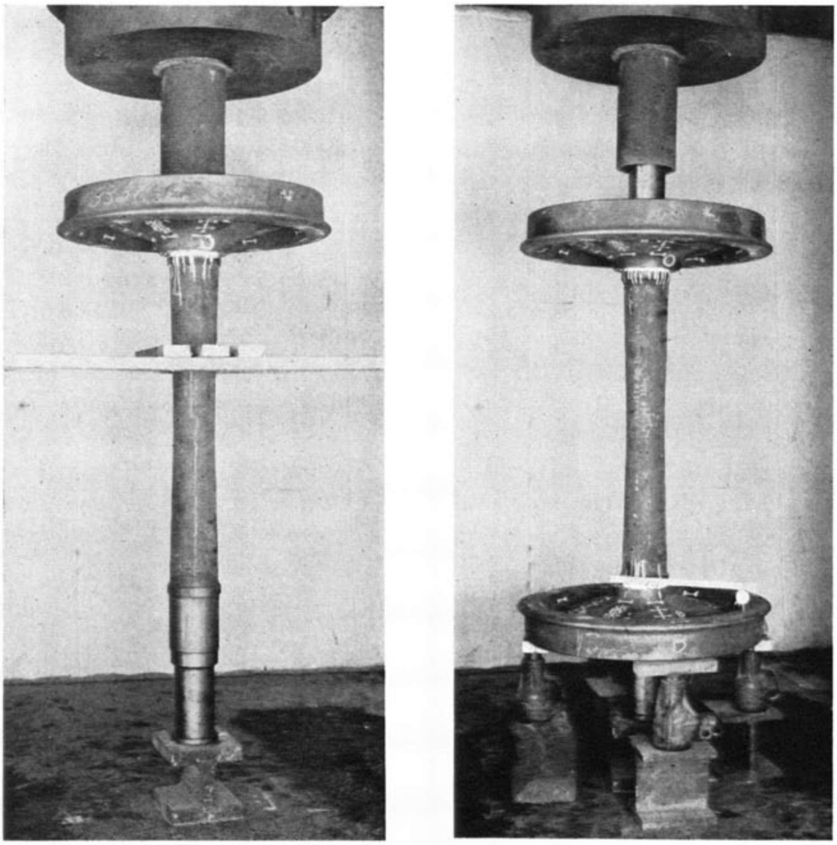
The process of determining the stresses in wheels from measured strains is far more complex than that of measuring a strain in a given direction and determining a stress–strain relationship, since the strain in a given direction is influenced by the stresses in the metal in other directions at that point. At a given point in the material the principal stresses are a function of Poisson's ratio, the modulus of elasticity, and of the three principal strains at that point (measured in three mutually perpendicular directions). In this investigation the strains were measured on the surface of the wheel and the principal strain normal to the surface of the wheel was negligible; the other two principal strains tangent to the surface were measured (or computed from measurements). To simplify the presentation, the data will be discussed mainly from the viewpoint of measured strains and the principal strains, but in several cases the important strains are converted to equivalent principal stresses.

IV. Mounting Tests

9. *Mounting Allowance and Procedure.*—In service, a railway car wheel is pressed on the axle and held there by friction. To develop the high radial pressure between wheel hub and axle that prevents the wheel from working loose, it is common practice to make the axle wheel seat diameter larger than the bore of the wheel by an amount known as the mounting allowance. In these tests no variation was made in the mounting allowance; it was approximately 0.008 in. for all four wheels tested. During mounting this difference in diameters is taken up by expanding the wheel bore (causing strains to be set up in the wheel), by compressing the axle slightly, and by a small amount of localized deformation at the sliding surfaces of contact. During the mounting the measured allowance is probably decreased somewhat by a smoothing of the sliding surfaces of contact involving some permanent set that removes the high portions of the tool marks left by machining.

The mounting tests were made on four wheels consisting of two carbon-steel wheels numbered 3331 and 3332 and two silico-manganese-steel wheels numbered 4891 and 4892 that were mounted in pairs on two normalized and tempered steel axles.

The photographs in Fig. 4 illustrate the method of mounting the wheels on the axles. The wheel bore and seat were carefully cleaned



(a) (b)

FIG. 4. MOUNTING PROCEDURE; UPPER WHEEL BEING PRESSED ON AXLE
(The jacks on the lower wheel were used only to prevent the assembly from toppling over.)

and coated with the standard A.A.R. lubricant, consisting of a mixture of 12 lb. of white lead to one gallon of boiled linseed oil. With the axle standing on end on the table of the testing machine, the wheel was lowered into place at the upper end. The wheel was pressed onto the axle at the rate of ½-inch travel per minute, and the load was read at each ¼-inch of movement. After each inch of movement the load was removed and a set of strain measurements taken, along two diameters at right angles to each other, and on both sides of the wheel. There were 24 radial and 56 circumferential gage lines per wheel read during these tests. Readings of strain were

TABLE 2

TEST RESULTS OF INITIAL MOUNTING

Values are average of readings on four gage lines, 90 deg. apart; unit strains are in hundred-thousandths of in. per in.; + indicates tension, − compression.

Wheel Number	Composition	Weight of Wheel lb.	Mounting Allowance in.	Maximum Mounting Load lb.	Maximum Circumferential Strains in Hub During Mounting		Maximum Radial Strains in Plate During Mounting		Maximum Strains After Mounting			
									Circumferential		Radial	
					Outside Face	Inside Face	Outside Face	Inside Face	Outside Face	Inside Face	Outside Face	Inside Face
3331	Carbon Steel	840	0.0078	141 000	+55.3	+49.5	−13.0	− 7.6	+55.3	+37.6	−13.0	− 7.0
3332	Carbon Steel	840	0.0079	130 600	+53.2	+55.1	−13.3	− 4.8	+53.2	+33.9	−13.3	− 3.3
4891	Si-Mn Steel	807	0.0084	81 100	+65.6	+66.1	−12.1	− 5.4	+65.6	+51.2	−12.1	− 4.2
4892	Si-Mn Steel	807	0.0085	162 600	+37.6	+69.5	−13.0	−18.0	+37.6	+44.6	−13.0	−17.3

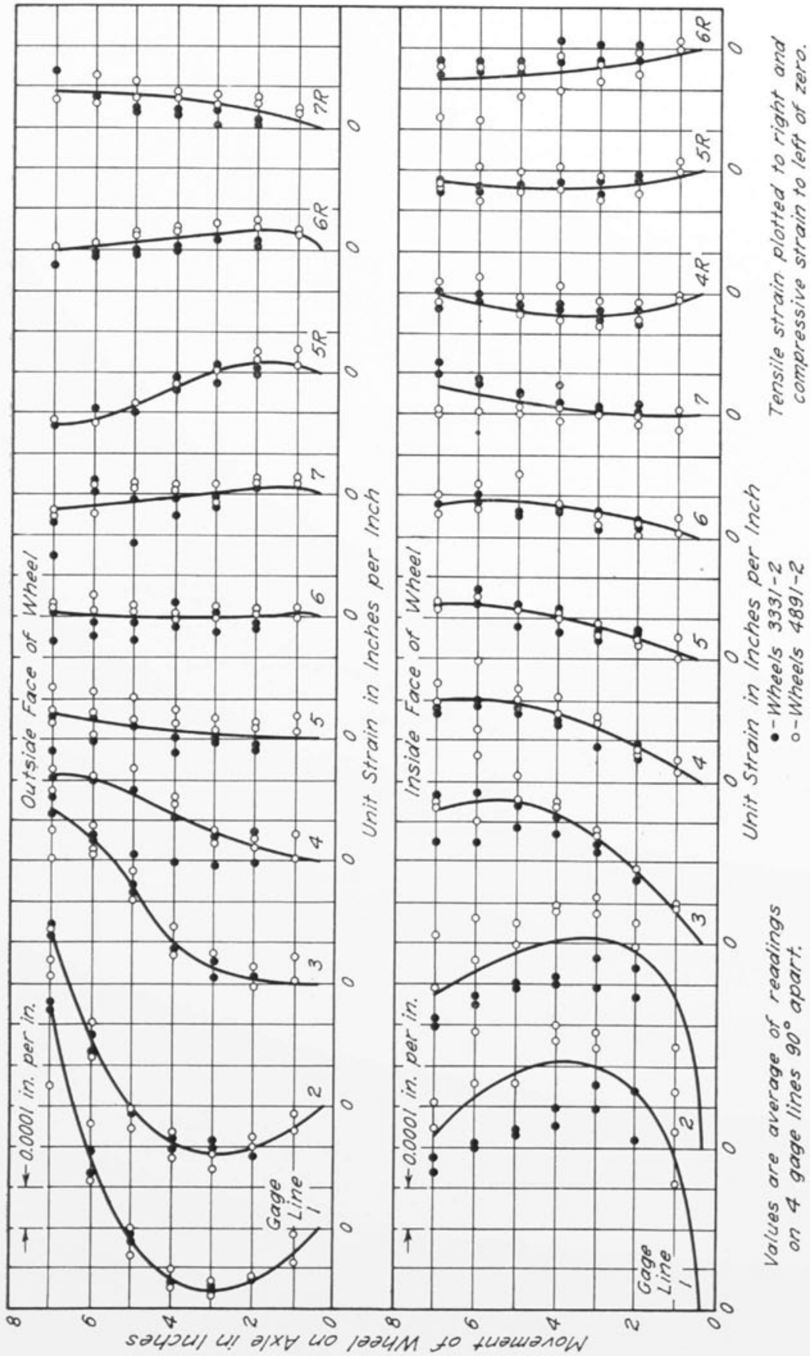


FIG. 5. RELATION OF WHEEL STRAINS TO MOVEMENT OF WHEEL ON AXLE DURING MOUNTING

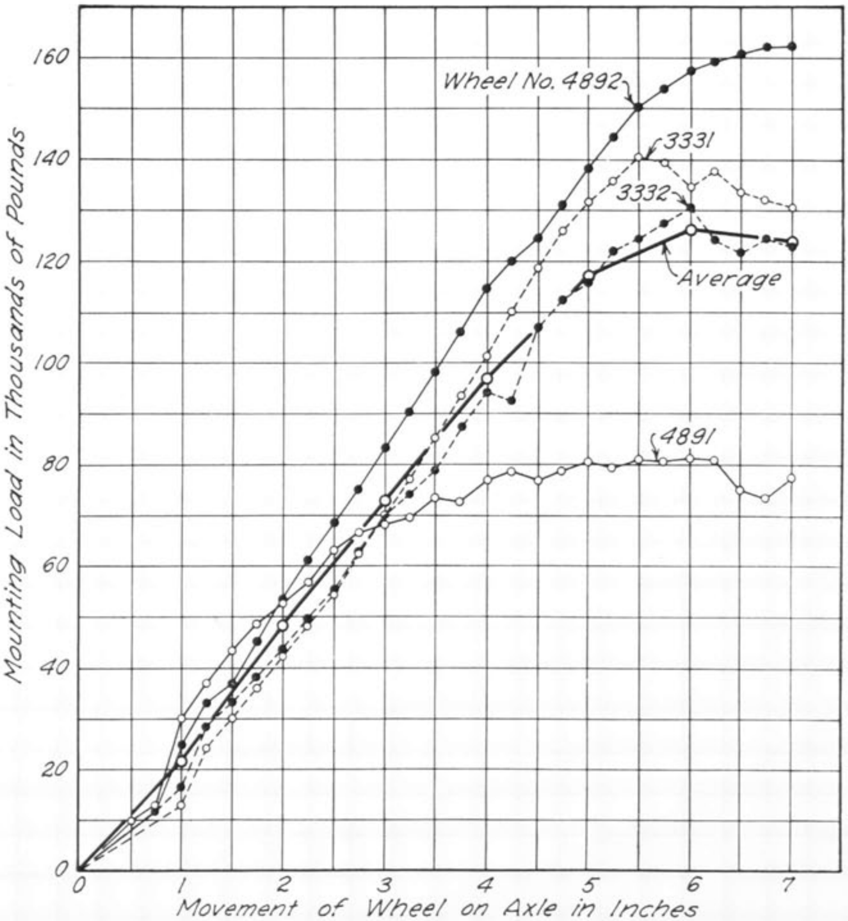


FIG. 6. RELATION OF MOUNTING LOAD TO MOVEMENT OF WHEEL ON AXLE

not observed in the rim of the wheel as it was felt that these strains would be negligible.

10. *Results of Mounting Tests.*—The principal results of the mounting tests are given in Table 2. This table gives the initial mounting allowance, the maximum load required to mount the wheel, and the average maximum strains developed in each wheel during mounting. More complete data giving the individual variations of strains and mounting loads as each wheel was pressed on are shown in Figs. 5 and 6, and the same data, but showing only the average strains for all four wheels, are plotted in Fig. 7. The only

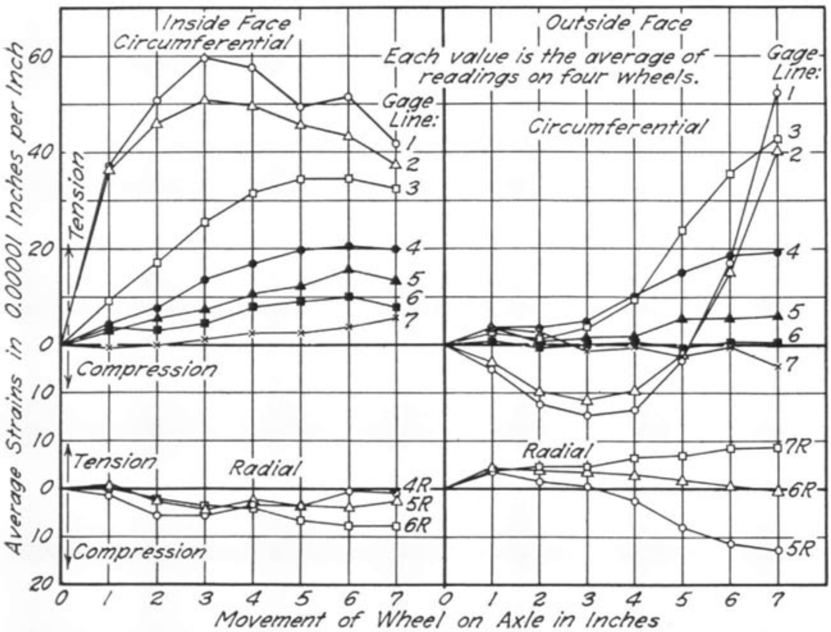


Fig. 7. Average Wheel Strains Due to Mounting

strains of importance in the mounting test were those in or near the hub of the wheel (gage lines 1, 2 and 3). In general, the final measured strains were usually slightly larger on the outside (front) face of the hub than those on the inside (back) face of the hub after the mounting was completed.

The data plotted for gage lines 1 and 2 (see Fig. 5) on the inside face of the wheel show considerable scatter for the four wheels tested, and the curves drawn in indicate only the general trend of the data. The slight differences in mounting allowance at the inside face of the wheel evidently affected these mounting strains. The maximum tensile strains on the inside face of the four wheels, developed while mounting, ranged from 0.00049 to 0.00069 (in. per in.) on gage line 1, whereas the final strains on this gage line after the mounting had been completed were somewhat smaller.

As the wheel was pressed on, the hub strains on the inside (back) face of the wheel increased rapidly to their maximum values, which were obtained after the axle had entered the bore about 3 inches. During the last four inches of movement these strains gradually decreased about 30 per cent. On the outside (front) of the wheel the hub strains gradually increased to a maximum compression during the first 3 inches of movement, and then reversed in direction, becoming tensile strains after about 5 inches movement on the axle; the maximum tensile strains in the outside hub face were reached with the wheel seated the full 7 inches on the axle. The strains measured along radial gage lines were relatively small in all cases, the maximum strain (average 0.00013) occurring as a compression on the outside face of the wheel at gage line 5.

The allowable mounting pressure for steel wheels in service on 5½ in. x 10 in. axles, as prescribed in the wheel and axle manual of the Association of American Railroads, ranges between 75 and 110 tons. It will be noted that for all wheels except No. 4892, the shapes of the load displacement curves (Fig. 6) are of a type listed as unsatisfactory in the A.A.R. manual, and the observed loads were less than the minimum prescribed mounting pressures. Of the wheels tested, No. 4892 required the greatest mounting load (162 600 lb.) and the mounting curve indicates a good fit, while the other silico-manganese-steel wheel (No. 4891), having practically the same mounting allowance, required the smallest mounting load (81 100 lb.). The exact cause of this wide variation in mounting loads is not known, but it has been pointed out by Horger and Nelson* that any one or all of the following factors may cause a wide variation in the mounting load:

(1) Shape of axle end and wheel bore, including taper and mounting allowances

(2) The lubricant, and variations in the amount of pigment present on the axle at the time of mounting

(3) The surface finish (smoothness, straightness, roundness)

(4) The materials of which the wheel and axle are made and their heat treatment

(5) The velocity of assembly of the wheel and axle

Probably the wide variation in mounting loads shown in Fig. 6 is due to a combination of several of these factors. Measurements at several points along the length indicated variations of several thousandths of an inch in the diameters of wheel bores or of wheel seats. The machined surfaces had some variations in finish, and some slight tapers or out of roundness evidently existed. The velocity with which the wheel was pushed on the axle during the mounting operation was, of course, much smaller in these tests than that commonly used in commercial practice.

*O. J. Horger and C. W. Nelson, "Design of Press- and Shrink-Fitted Assemblies," Journal of Applied Mechanics, Trans. A.S.M.E., December 1937, p. A183, and March 1938, p. A32.

In spite of the large difference in mounting loads of the four wheels the curves of mounting strains are quite similar, and the maximum strains developed differ but little except for the gage lines nearest the axle. However, the final strains (with the wheel fully mounted) in the hub of wheel 4892 were smaller than those in wheel 4891. The greatest difference was apparent on the outside face of the hub at gage line 1, where the tangential strains were 0.00035 and 0.00066, respectively, in these two wheels. The strains measured on the outside face of the hub were smaller for wheel 4892 than those obtained for any of the other wheels tested. Apparently the small differences or irregularities in surface finish and slight tapers of wheel seat or wheel bore have a great influence on the mounting loads and wheel strains, and some wheels requiring high mounting loads may not be subjected to strains as great as those existing in wheels mounted with lower mounting loads.

V. Vertical Load Tests

11. *Outline of Vertical Load Tests.*—The vertical load tests were carried out on the same four wheels used in the mounting tests, and all strains measured were those produced in addition to the mounting strains already described.

The vertical loads were intended to produce loading conditions in the wheels and axle similar to those produced by a vertical static load on the assembly in service, but to produce measurable strains in the portions of the wheel that were not appreciably stressed it was found advisable to use much higher loads than the nominal wheel loading expected under service conditions. However, none of the measured strains were large enough to indicate that yielding of the steel occurred during the test. To facilitate the testing the usual direction of loading in service was reversed; the rail pressure was applied to the top of the wheel, and the journals were supported in standard brass bearings at each end. The journal brasses in turn were held by castings (standard journal bearing wedges) turned with a 78 in. radius on the bottom which rested on a flat pedestal, allowing rotation of the ends as the axle deflected. Figure 8 shows a view of the loading assembly in the testing machine. The loads were applied through heavy steel girders and a pair of rocker supports which permitted a free inward deflection of the wheel. Two steel blocks were placed on the wheel treads and the loads were applied to these blocks by the rocker arms, which were spaced at standard gage distance. The loads were applied in increments of 40 000 to 80 000 lb. until a

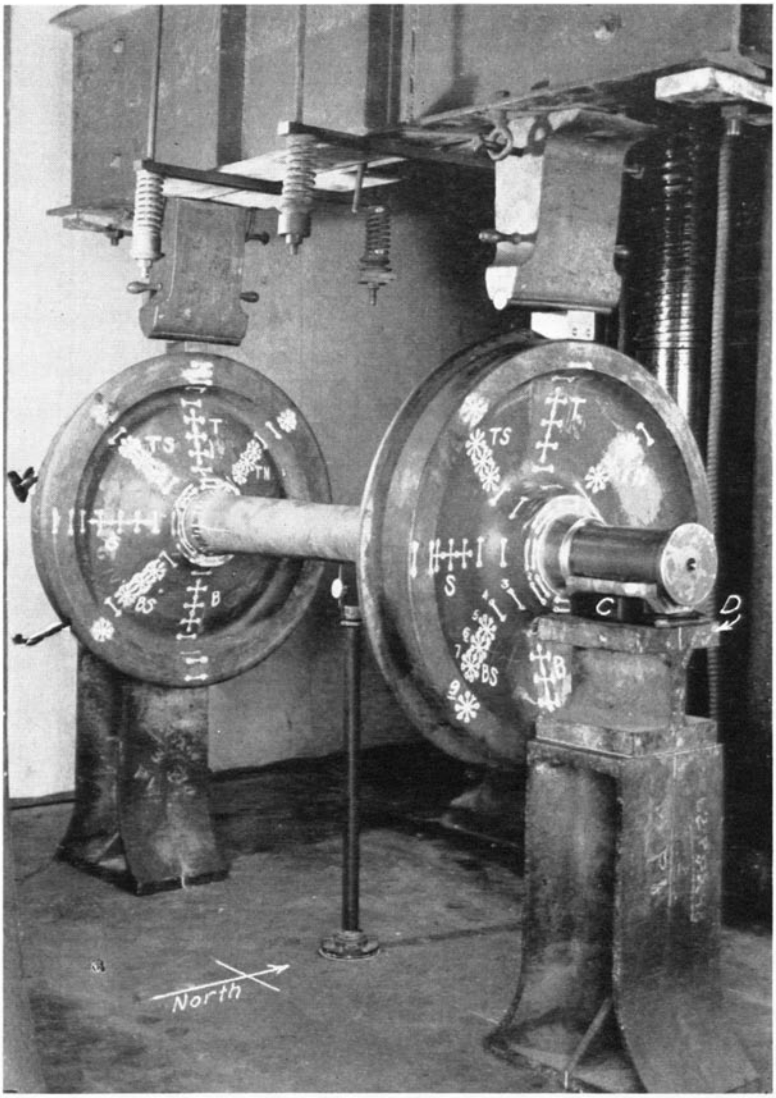


FIG. 8. WHEEL AND AXLE ASSEMBLY SUBJECTED TO VERTICAL LOAD
C is a casting having a curved bearing surface in contact with D to allow rotation of axle caused by deflection.

maximum of 300 000 lb. axle load (150 000 lb. per wheel) was reached in each case.

Strain gage measurements were taken on 7 radial rows of gage lines (located as shown in Fig. 2) on both faces of each wheel, there

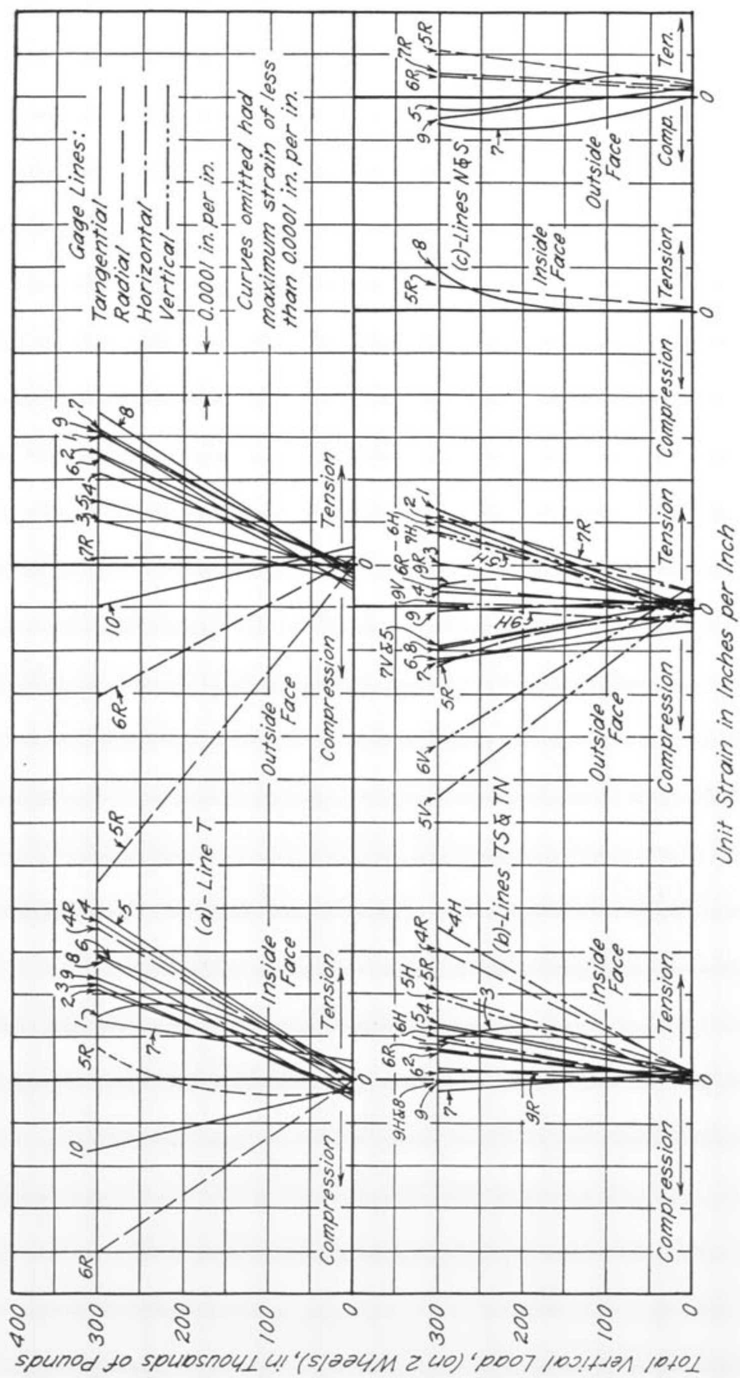


FIG. 9. AVERAGE STRAINS IN VERTICAL LOAD TEST

TABLE 3

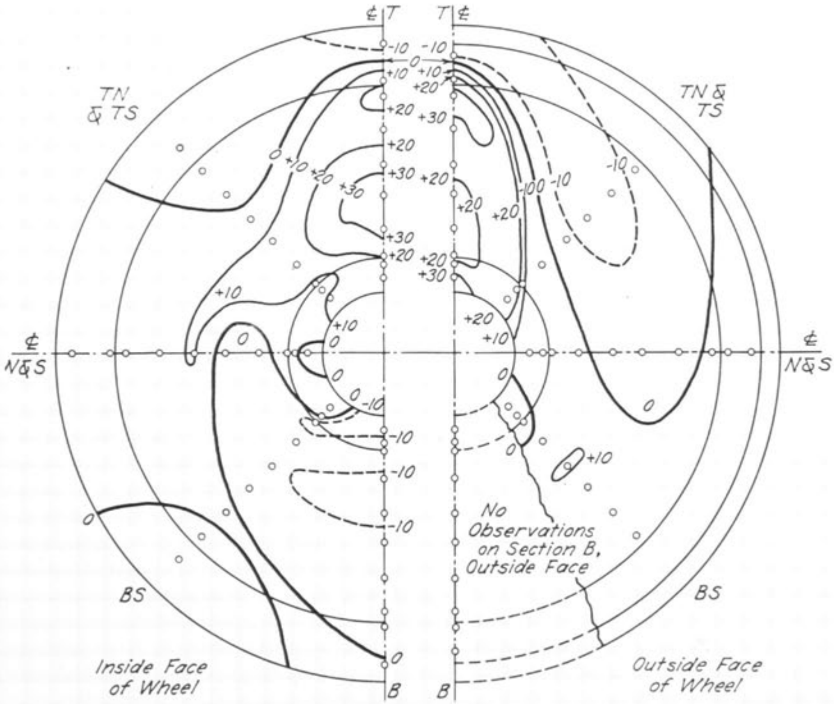
WHEEL STRAINS DUE TO VERTICAL LOAD OF 150 000 POUNDS PER WHEEL

Values are average of four wheels; strains are in hundred-thousandths of in. per in.; + indicates tension, − compression.

Gage Line	Strain on Section B	Strain on Section BS	Strain on Sections N and S	Strain on Sections TS and TN	Strain on Section T	Maximum Strain on Any Section	
						Tension	Compression
			Outside Face of Wheel				
1		......	+ 1.7	+22.4	+30.9	+30.9	......
2		......	+ 3.9	+21.1	+25.9	+25.9	......
3		− 0.5	+ 3.3	+10.9	+11.2	+11.2	− 0.5
4		+ 0.5	+ 0.5	+ 4.2	+20.2	+20.2	......
5		+12.3	− 2.5	− 9.0	+17.8	+17.8	− 9.0
6		+ 2.9	− 2.7	−11.8	+22.8	+22.8	−11.8
7		+ 6.2	− 5.6	−12.8	+23.4	+23.4	−12.8
8		+ 4.6	− 0.6	− 9.0	+36.2	+36.2	− 9.0
9		+ 1.8	− 4.1	− 0.5	+26.7	+26.7	− 4.1
10		......	− 3.0	......	−10.3	......	−10.3
5R		+22.6	+11.8	−13.6	−68.2	+22.6	−68.2
6R		+ 3.7	+ 6.0	+ 5.5	−30.0	+ 6.0	−30.0
7R		− 4.4	+ 6.9	+22.3	+ 1.8	+22.3	− 4.4
9R		+ 4.2	......	+ 4.6	......	+ 4.6	......
			Inside Face of Wheel				
1	−15.1	− 1.3	− 2.3	+12.0	+14.5	+14.5	−15.1
2	− 5.1	+ 5.5	+ 0.6	+ 7.5	+16.4	+16.4	− 5.1
3	− 4.9	− 7.0	+ 1.6	+ 9.8	+21.0	+21.0	− 7.0
4	−12.3	− 5.1	− 1.5	+13.1	+34.8	+34.8	−12.3
5	−12.3	− 9.9	− 2.6	+12.7	+37.8	+37.8	−12.3
6	− 3.8	− 1.1	+ 1.1	+ 2.2	+28.2	+28.2	− 3.8
7	− 1.6	+ 0.5	+ 2.4	− 0.7	+14.0	+14.0	− 1.6
8	− 1.1	− 0.5	+ 8.7	− 4.8	+24.8	+24.8	− 4.8
9	− 4.6	− 2.0	+ 5.1	− 0.6	+18.4	+18.4	− 4.6
10	+ 0.9	......	+ 5.6	......	−15.5	+ 5.6	−15.5
4R	−24.5	−20.4	+ 4.7	+27.9	+35.2	+35.2	−24.5
5R	− 5.7	− 4.9	+ 5.9	+21.5	+ 5.9	+21.5	− 5.7
6R	+ 5.7	+ 6.1	+ 6.6	+ 8.6	−40.3	+ 8.6	−40.3
9R	......	+ 5.3	......	+ 3.5	......	+ 5.3	......

being 217 gage lines read on each wheel for each increment of load. Strain readings were also taken on the axles at mid-span on top and bottom surfaces, and the downward deflection of the axle at mid-span was observed. Further information on wheel deformations was obtained by measuring the change in distance between wheel flanges, and the lateral movements of the wheel rim with respect to the plane of the hub, on horizontal and vertical diameters (at top, bottom, south and north). These lateral movements were read with a special gage which is shown in position on the lower wheel in Fig. 4 (b).

12. *Wheel Strains in Vertical Load Tests.*—The average strains for the four wheels observed at the maximum vertical load (150 000 lb. per wheel) are listed in Table 3. The individual strains measured



Tensile Strain ———————, (Positive Values)
Compressive Strain - - - - - -, (Negative Values)

Values given are the average strains for four wheels in hundred thousandths of an inch per inch.

Contours are symmetrical about vertical center line.

Fig. 10. Contour Lines of Tangential Strains for Vertical Load of 150 000 lb. per Wheel

for each of the four companion wheels were averaged, and these values are plotted in Fig. 9, showing the variation of strains with the total applied load for gage lines on several sections of the wheel. Similar curves were also plotted for the gage lines on rows BS and B, but these are not reproduced here as the strains were small on the lower half of the wheel.

The average distribution of the tangential strains throughout the wheel for a load of 150 000 lb. per wheel is illustrated by means of contour lines in Fig. 10. Each contour line represents a locus along which all points in the wheel have a tangential strain of the amount indicated on the line. These curves indicate that the tangential strains throughout the wheel were relatively small, especially for the lower half of the wheel. It should be noted here that readings were not made along the bottom (B) row of gage lines on the outside face of the wheel in the loading tests, since the pedestals supporting the axle journals obscured this portion of the wheel and made it impossible to use a strain gage on this section.

In general, the largest strains observed in the wheel occurred along section T which is the vertical section of the wheel lying directly between the load and the supporting journal. Along this section there was a fairly general circumferential tension developed in the wheel, reaching a maximum strain of 0.00038 in the plate for the highest loading (on gage line 5, inside face). There were some high radial compressive strains on the outside face of the wheel reaching their maximum value at the junction of hub and plate. The maximum compressive radial strain observed was 0.00068 in. per in. (on gage line 5R, outside) at 150 000 lb. wheel load.

From Table 3 it may be seen that the tangential and radial strains observed on the other sections of the wheel were generally much smaller than those on section T. It must be remembered that on the diagonal sections (BS, TS and TN) the maximum (or principal) strains do not correspond with the strains measured in tangential and radial directions. The principal strains and their directions were obtained at four locations on each of the diagonal sections from the strains measured on rosettes, and these data are listed in Table 4. Here again, however, it will be observed that the maximum values of principal strain on the diagonal sections of the wheel were not as large as the radial strains that occurred on the top section of the wheel. There were undoubtedly some localized radial compressive stresses of fairly large magnitude developed in the rim at the point of application of the load, but no means was available to determine the magnitude of these localized stresses. In service, these contact stresses might lead to localized crushing of the material in the tread of the wheel, but would not usually be of significance in causing a general failure of the wheel.

Due to the fact that the nominal load for these wheels in service is about one-fifth of that used in the tests from which the data in Tables 3 and 4 were obtained, it is felt that only very small strains (reaching maximum values of 0.00008 in tension, and 0.00014 in compression) might be expected under nominal vertical loads. These values, however, would apply only to the average strains over a 2-in. gage length at the locations for which measurements were made in the test. It is quite possible that localized concentrations of stress might occur on the sharply-curved portions of the wheel which could not be satisfactorily obtained from strain gage measurements.

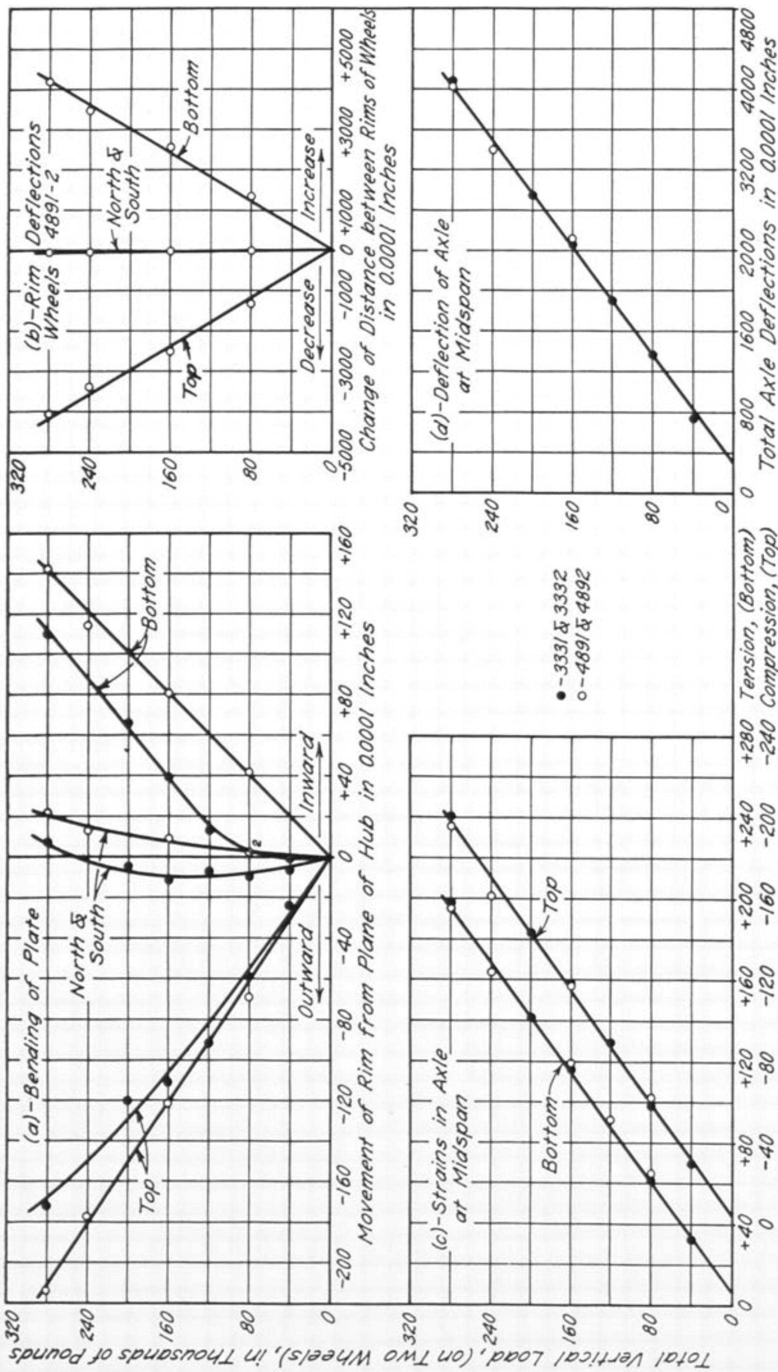


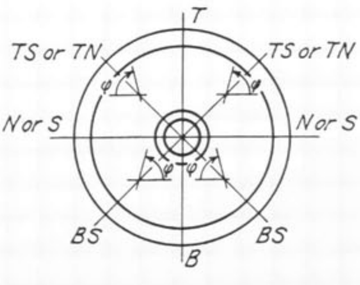
FIG. 11. STRAINS AND DEFLECTIONS OF WHEELS AND AXLES UNDER VERTICAL LOADS

TABLE 4

PRINCIPAL STRAINS DEVELOPED ON DIAGONAL SECTIONS FOR VERTICAL LOAD OF 150 000 POUNDS PER WHEEL

Values are average of four wheels; + indicates tension, − compression; unit strains are in hundred-thousandths of in. per in.; φ is angle (in degrees) between maximum principal strain and a horizontal axis (see page 18, and diagram below).

Gage Line	OUTSIDE FACE			INSIDE FACE		
	Principal Strains		Angle φ	Principal Strains		Angle φ
	Maximum	Minimum		Maximum	Minimum	
	Sections TS and TN					
4	......	......	....	+35.3	+ 7.8	16 deg. 30 min.
5	+18.5	−41.4	2 deg. 30 min.	+22.8	+ 8.8	20 deg.
6	+20.2	−27.3	10 deg. 30 min.	+11.8	− 0.7	15 deg. 30 min.
7	+26.3	−17.1	26 deg. 30 min.	......	......	....
9	+ 4.7	− 0.65	50 deg.	+ 4.1	− 0.3	57 deg. 30 min.
	Section BS					
4	......	......	....	− 5.7	−22.3	56 deg. 30 min.
5	+25.8	+ 7.2	107 deg.	− 5.6	−10.6	134 deg. 30 min.
6	+ 9.7	− 3.4	91 deg. 40 min.	+ 6.0	+ 2.1	149 deg.
7	+ 8.1	− 6.7	67 deg.	......	......	....
9	+ 6.1	+ 1.2	165 deg.	+ 6.2	− 1.3	141 deg. 20 min.



13. *Deflections of Wheels and Axles.*—The movements of the wheels and the strains and deflections of the axle during the vertical load test are shown in graphical form in Fig. 11. The bending deflection of the inside rim of the wheel measured relative to the plane of the hub (Fig. 11a) was relatively small; at the maximum vertical load (150 000 lb. per wheel) the top of each wheel (4891 and 4892) moved outward approximately 0.021 in., the bottom inward about 0.014 in., and the two sides (north and south) deviated less than 0.002 in. from their original positions relative to the plane of the hub. The *total* movement (Fig. 11b) between the rims of the two wheels, however, was quite large. The tops of the two rims moved inward

(together) a total of 0.410 in., while the bottoms moved outward (apart) 0.417 in., at the highest load. It is interesting to note that the bending of the wheel plate tends to *separate* the tops of the two wheels about 0.04 in., but the deflection of the axle produces a rotation of the wheels large enough to actually result in moving the rims *together* at the top about 0.41 in. The two sides of the wheels tended to remain approximately the same distance apart.

The strains measured on the top and bottom of the axle (Fig. 11c) give an indication of the stresses developed at mid-span. Since the strains evidently increased in proportion to the load in both axles tested the yield point of the steel probably was not exceeded by the high loadings used. Assuming the modulus of elasticity of the axle steel to be 30 000 000 lb. per sq. in., the maximum stress calculated from the measured strains would be about 59 200 lb. per sq. in.

The axle deflections (Fig. 11d) also varied in proportion to the load, the maximum value being approximately 0.40 in. A portion of this deflection is due to the deflection of the pedestals and supporting journals, and probably the fact that this curve does not go through the origin may be due to the initial seating of the contact surfaces in the supports as the load was applied. After removal of the load it was found that there was a residual deflection of 0.006 in., which also may be attributed mainly to this cause rather than to permanent set in the axles.

If the maximum vertical load imposed on each wheel in service were assumed to be 40 000 lb. it will be seen from the data in Fig. 11 that the axle may be expected to deflect about 0.14 in., and the stress in the axle at mid-span would be approximately 18 000 lb. per sq. in. Under this load the distance between wheel rims at the top and bottom of the wheel would change about 0.13 in.

VI. Combined Vertical and Lateral Load Tests

14. *Testing Procedure.*—To obtain the approximate effect of a service condition in which lateral pressure is applied to the flange of a wheel, tests were made in which combined vertical and lateral loads were used. The same loading arrangement was employed as that used in the vertical load tests except for the addition of special steel yolks and tension rods through which inward horizontal forces could be applied to the loading block on the tread of each wheel. Figure 12 shows the general arrangement used in the tests, and a diagram of the special device used to apply the horizontal forces is shown in Fig. 13.

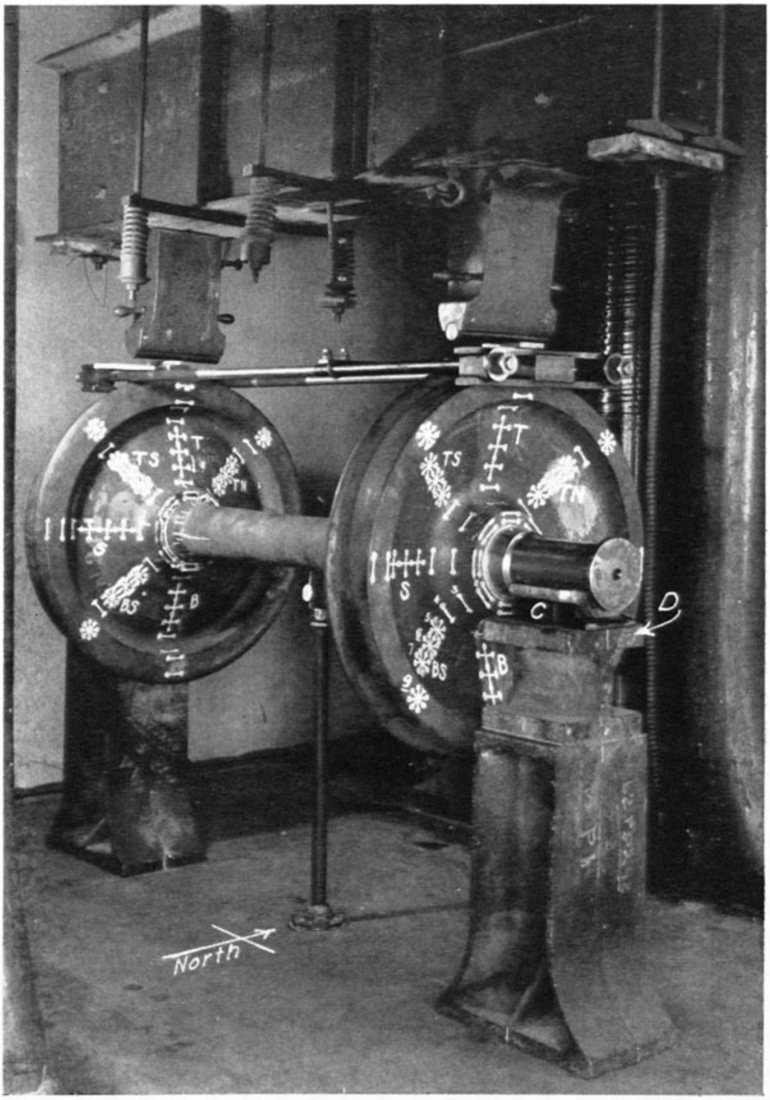


FIG. 12. WHEEL AND AXLE ASSEMBLY SUBJECTED TO VERTICAL LOADS AND FLANGE THRUSTS

C is a casting having a curved bearing surface in contact with D to allow rotation of axle caused by deflection.

The horizontal thrusts were obtained by tightening the nuts on the ends of the two $1\frac{1}{4}$-inch diameter steel tension rods. These rods were each reduced to one square inch in cross-section near the

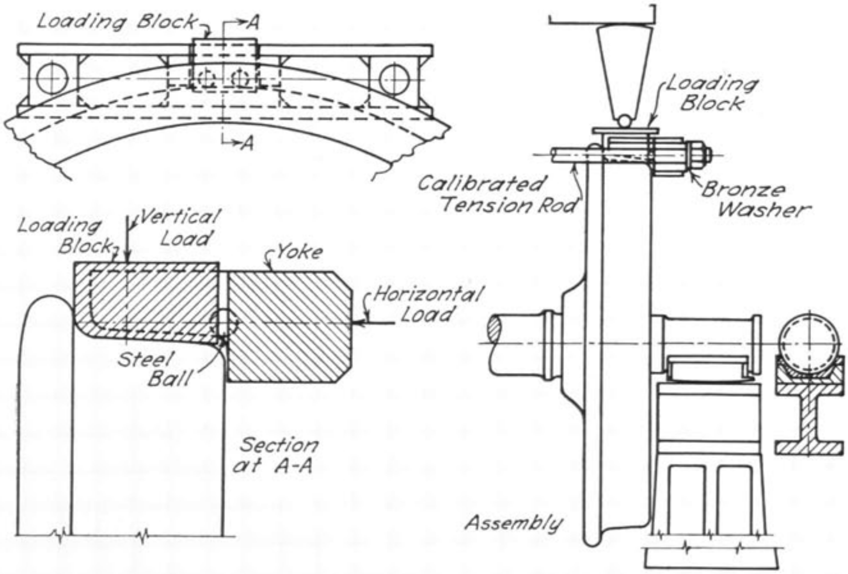


FIG. 13. LOADING BLOCK FOR TRANSMITTING VERTICAL AND HORIZONTAL LOADS TO WHEEL

middle, and two gage lines on each rod were calibrated with a Whittemore 10-in. strain gage so that the desired thrust could be obtained by tightening the nuts until a predetermined strain was reached. Calibration of these rods showed that one division of the strain gage corresponded to 311 lb. total tension in each rod and the gage readings were estimated to 1/10 of a division. It should be noted that this loading arrangement produced equal flange loads on both wheels of the assembly; this does not agree with the type of loading obtained in service, where the lateral forces usually act on only one wheel at a time. This variation in loading would produce different strains and deflections in the axle, but should not affect the wheel strains appreciably.

All readings taken in the flange thrust test were made on the same gage lines and with the wheels in the same position as in the vertical load test. For each test, zero strain readings were taken with the wheels and axle in position for test; then a total vertical load of 60 000 lb. was applied (30 000 lb. per wheel) and the strains were observed. Successive horizontal flange loads of 20 000, 40 000 and 60 000 lb. were then applied (the vertical load being held constant at 60 000 lb.) and a complete set of strain measurements were recorded for each increment of load. As in the vertical load test,

TABLE 5

WHEEL STRAINS DUE TO COMBINED VERTICAL AND FLANGE THRUST LOADS

Vertical load = 30 000 lb. per wheel, flange thrust = 60 000 lb.; values are average of four wheels; strains are in hundred-thousandths of in. per in.; + indicates tension, − compression.

Gage Line	Strain on Section B	Strain on Section BS	Strain on Sections N and S	Strain on Sections TS and TN	Strain on Section T	Maximum Strain on Any Section	
						Tension	Com-pression
Outside Face of Wheel							
1		......	+ 9.0	−16.2	−30.0	+ 9.0	−30.0
2		......	+ 7.9	−12.7	−25.2	+ 7.9	−25.2
3		+ 8.6	+ 3.6	−10.9	−11.0	+ 8.6	−11.0
4		+ 5.5	+ 0.9	− 5.2	− 0.9	+ 5.5	− 5.2
5		+ 3.1	+ 4.3	+ 6.3	+ 9.0	+ 9.0	......
6		− 7.0	+11.8	+17.2	+19.7	+19.7	− 7.0
7		− 4.8	+13.4	+22.9	+27.6	+27.6	− 4.8
8		− 2.4	+18.3	+28.7	+ 9.0	+28.7	− 2.4
9		− 3.3	+38.9	+48.2	− 7.9	+48.2	− 7.9
10		......	+26.3	......	......	+26.3	......
5R		−28.9	− 7.0	+35.6	+43.6	+43.6	−28.9
6R		− 1.4	− 9.6	− 8.9	− 7.7	......	− 9.6
7R		+20.6	−13.9	−34.5	−36.5	+20.6	−36.5
9R		+ 1.8	......	−11.4	......	+ 1.8	−11.4
Inside Face of Wheel							
1	−11.0	− 8.3	− 8.2	+13.9	+23.2	+23.2	−11.0
2	− 8.7	− 5.1	− 7.7	+12.6	+24.5	+24.5	− 8.7
3	+ 8.8	+ 7.7	− 4.3	− 5.7	−10.1	+ 8.8	−10.1
4	+29.8	+19.5	− 4.4	−32.7	−38.7	+29.8	−38.7
5	+30.2	+20.2	− 2.5	−26.2	−42.4	+30.2	−42.4
6	+15.8	+10.1	+ 1.3	−17.3	−14.5	+15.8	−17.3
7	+ 9.0	+ 2.9	+ 4.9	+ 0.9	+ 2.9	+ 9.0	......
8	+ 9.9	+ 2.2	− 6.1	−10.9	+37.2	+37.2	−10.9
9	+24.5	+10.7	−21.6	−20.5	+45.1	+45.1	−21.6
10	+33.5	......	−22.4	......	+59.5	+59.5	−22.4
4R	+67.1	+51.8	+ 2.4	−92.7	−150.5	+67.1	−150.5
5R	+32.6	+27.8	+ 3.6	−47.1	−95.8	+32.6	−95.8
6R	+ 0.7	+ 8.8	+ 4.1	−16.8	−50.8	+ 8.8	−50.8
9R	......	+ 0.7	......	+10.0	......	+10.0	......

the deflections of the axle, and the relative movements of the wheel rims, were also measured for all loadings.

15. *Wheel Strains in Flange Thrust Tests.*—The average strains observed in the four wheels at the maximum loading (30 000 lb. per wheel vertical, 60 000 lb. lateral) are listed in Table 5. The individual strains measured for each pair of companion wheels were averaged and these values are plotted in Fig. 14 showing the variation of strains on each gage line, for each increment of load. These curves do not start with zero strain, but begin at the strain due to vertical load alone. The average distribution of the tangential strains throughout the wheel is shown by means of contour lines in Fig. 15.

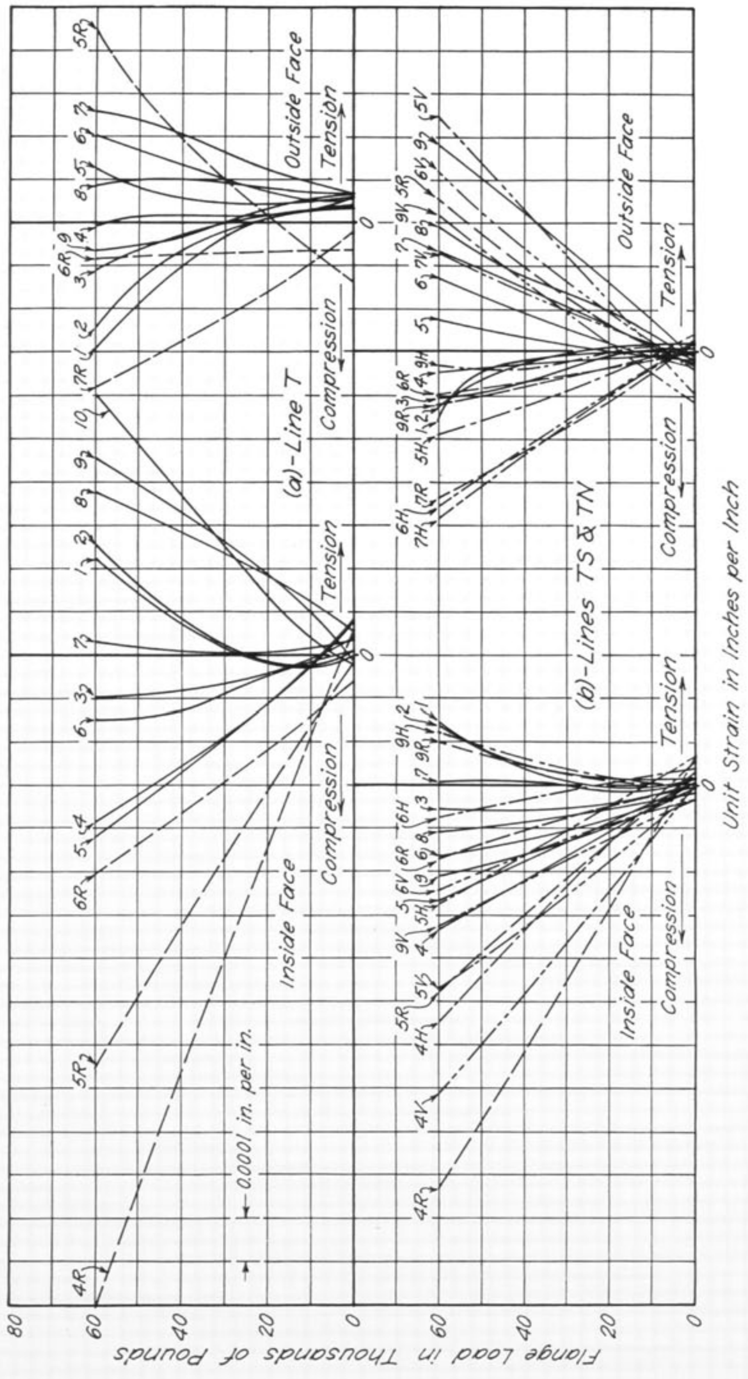


FIG. 14. AVERAGE STRAINS IN FLANGE THRUST TEST

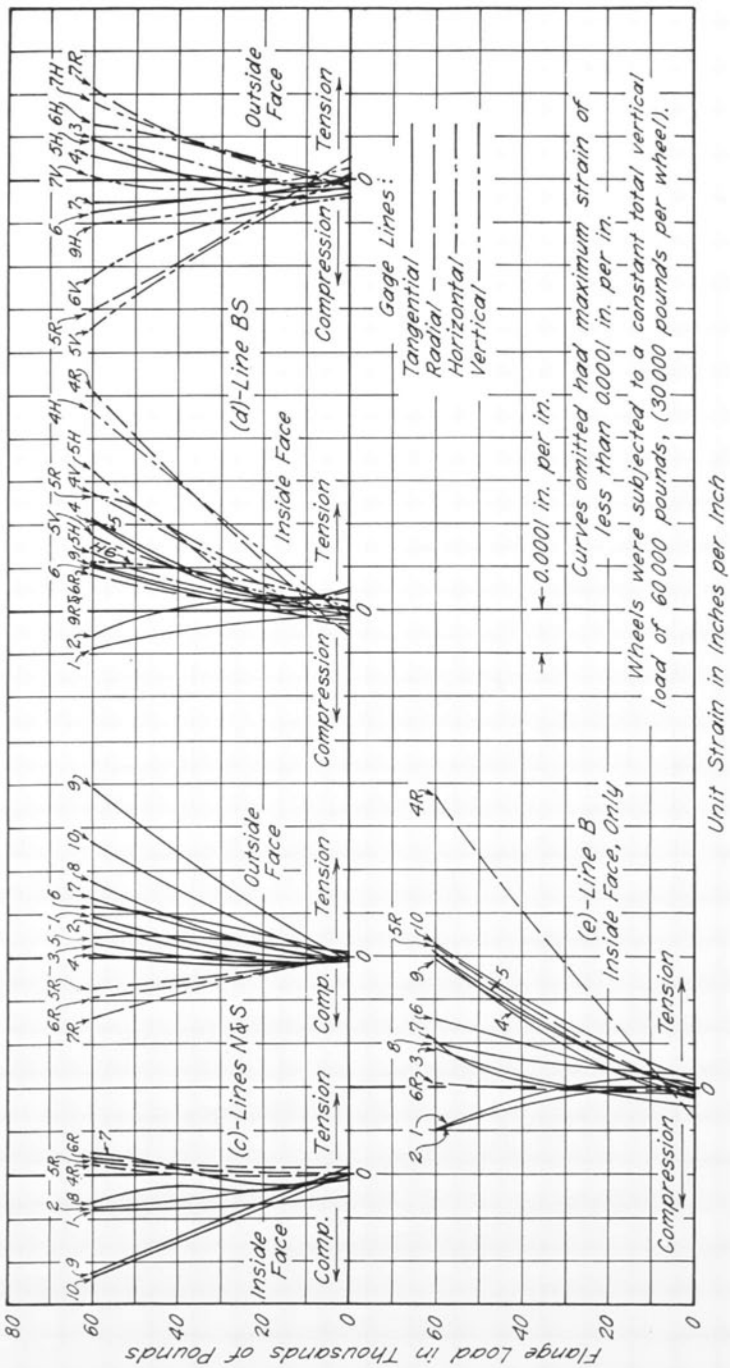


Fig. 14. (Concluded). Average Strains in Flange Thrust Test

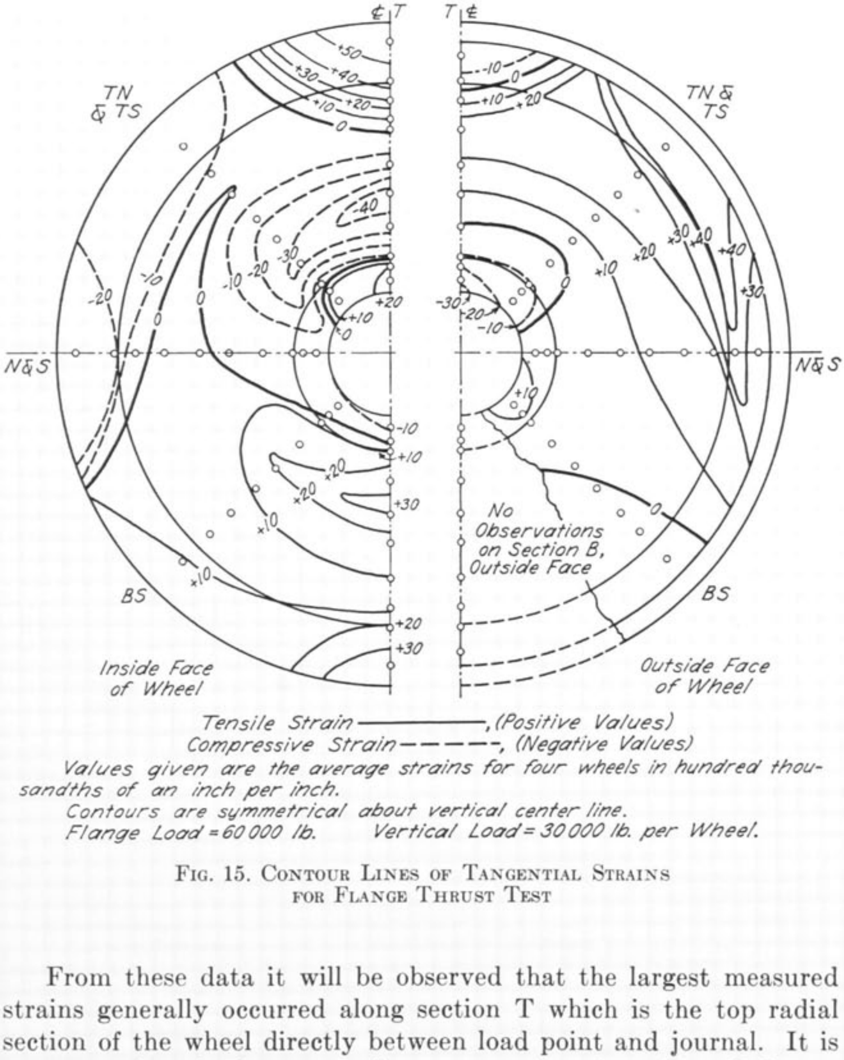


FIG. 15. CONTOUR LINES OF TANGENTIAL STRAINS FOR FLANGE THRUST TEST

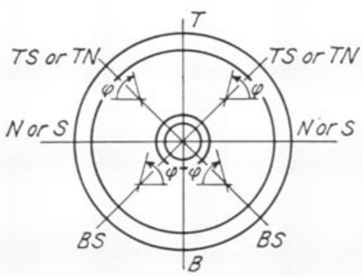
From these data it will be observed that the largest measured strains generally occurred along section T which is the top radial section of the wheel directly between load point and journal. It is apparent that the largest strains were measured on the radial gage lines on the inside face of the wheel and were compressive strains. The wheel evidently acts somewhat as a cantilever in resisting the horizontal thrust, producing radial compression along the top section on the inside face and a tension on the outside face; the combined vertical load seems to add somewhat to the radial compression. The average strain on gage line 4R reached a maximum value of 0.00150 at the highest load; this was the largest strain observed on any gage line for any of the tests made. The greatest radial tensile

TABLE 6

PRINCIPAL STRAINS DEVELOPED ON DIAGONAL SECTIONS FOR FLANGE LOAD OF 60 000 POUNDS AND VERTICAL LOAD OF 30 000 POUNDS PER WHEEL

Values are average of four wheels; + indicates tension, − compression; unit strains are in hundred-thousandths of in. per in.; φ is angle (in degrees) between maximum principal strain and a horizontal axis (see page 18, and diagram below).

Gage Line	OUTSIDE FACE			INSIDE FACE		
	Principal Strains		Angle φ	Principal Strains		Angle φ
	Maximum	Minimum		Maximum	Minimum	
	Sections TS and TN					
4	......	......	....	−31.8	−93.9	143 deg.
5	+58.4	−21.4	78 deg. 40 min.	−22.7	−51.0	156 deg. 30 min.
6	+44.0	−37.9	100 deg.	− 8.8	−24.7	1 deg. 30 min.
7	+34.9	−48.5	111 deg. 30 min.	......	......	....
9	+56.2	−18.4	116 deg. 30 min.	+20.5	−34.0	16 deg. 30 min.
	Section BS					
4	......	......	....	+55.6	+17.3	29 deg. 20 min.
5	+14.1	−39.4	161 deg.	+33.2	+15.9	13 deg. 30 min.
6	+13.6	−22.3	4 deg. 20 min.	+13.1	− 1.3	158 deg.
7	+24.1	− 6.9	27 deg. 30 min.	......	......	....
9	+ 4.9	− 8.2	77 deg. 30 min.	+13.8	+ 3.9	176 deg. 40 min.



strains observed were 0.00067 (on section B, line 4R, inside face) and 0.00044 (on section T, line 5R, outside face).

The largest tangential strains occurred along section T at gage lines 9 and 10 directly under the point of load application. Aside from these values the maximum tangential strains were all less than 0.00040 throughout the wheel and do not seem to be of particular importance, the radial strains being much more significant.

The average principal strains that occurred at strain rosettes along the diagonal sections of the wheel have been listed in Table 6. These values indicate that the largest principal strains on the diagonal

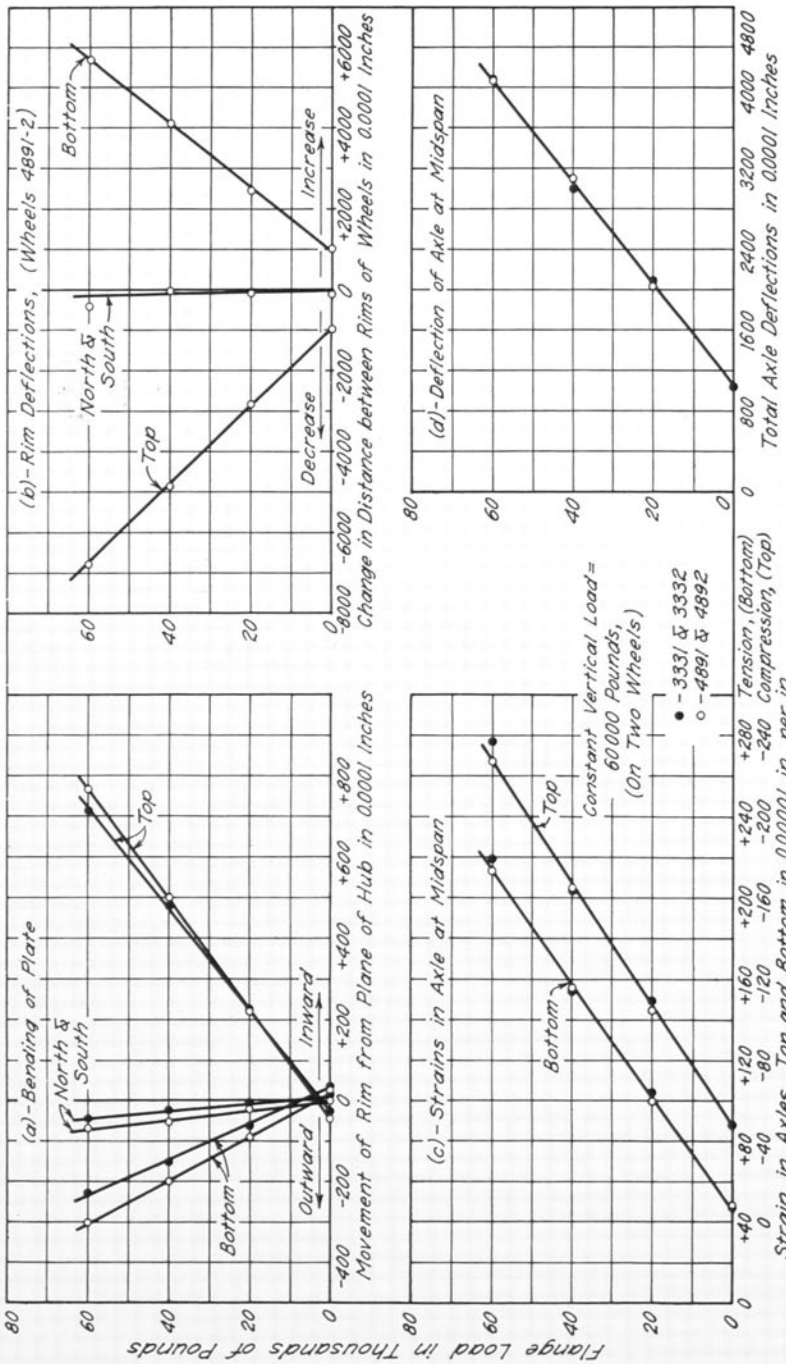


FIG. 16. STRAINS AND DEFLECTIONS OF WHEELS AND AXLES UNDER COMBINED VERTICAL AND FLANGE LOADS

sections occurred at the rosette location on the plate nearest the center of the wheel. The values of principal strain shown in this table are much larger than the observed strains for sections N and S, but were smaller in general than the maximum strains on the top section of the wheel.

An inspection of Fig. 14 shows that the strains on gage lines 1 and 2 in the hub of the wheel did not vary in direct proportion to the flange loads applied. The most abrupt change in the rate of increase of strain with load occurred at a flange load of approximately 30 000 lb. This non-linear relation between strain and load was probably caused by a localized slipping of the hub on the axle in the circumferential direction produced by excessive loads. The hub evidently tends to expand on one side and contract on the other side of the axle over small areas, and a slight circumferential slip or rubbing action between the wheel and axle takes place at these points. This type of action, if repeated a large number of times in service, would be likely to lead to a fatigue failure of the axle due to the combination of rubbing corrosion and localized stresses set up at the points of contact.

It will be observed (Fig. 14) that the wheel strains, produced by the initial application of the vertical load of 30 000 lb. per wheel, were generally very small compared with the strains subsequently developed by the horizontal flange load. Evidently lateral thrusts on the flange set up a more severe state of stresses in the wheel than do extremely large vertical loads alone. If it were assumed that for service conditions the nominal loads that might be expected to occur were a lateral flange load of 30 000 lb. and a vertical load of 30 000 lb. per wheel, then the maximum strains developed in the wheel would be approximately one-half of those listed in Tables 5 and 6.

16. *Deflections of Wheels and Axles.*—The movements of the wheel rims, and the strains and deflections of the axles, when subjected to combined vertical load and flange thrust, are plotted in Fig. 16. It will be observed that the curves shown in this figure do not go through the origin of coördinates since the values of abscissa for zero flange load are the values produced by the initial application of the 60 000 lb. vertical load.

The bending deflection of the wheel plate (Fig. 16 a) was much greater in this test than that observed in the vertical load test, and the directions of motion of the various parts of the rim relative to the plane of the hub are just the reverse of those observed when only vertical load was applied to the wheel. At the maximum flange load

(60 000 lb.) the top of each wheel had moved inward approximately 0.074 in. and the bottom and sides moved outward about 0.027 in. and 0.006 in., respectively.

The total movement between the rims of the two wheels (Fig. 16 b) was also larger than that observed under the heavy vertical loads alone. The tops of the two rims moved inward a total of 0.680 in., while the bottoms moved outward 0.569 in. at the highest flange thrust load.

The strains measured on the top and bottom of the axle (Fig. 16 c) increased in direct proportion to the flange load; judging from these results it is not believed that the yield point of the material in the axle was exceeded in the tests. The maximum strains observed would indicate a stress of approximately 67 200 lb. per sq. in. at mid-span; using the ordinary flexure formula and computing this stress from the known loads that were applied, a value of 66 700 lb. per sq. in. was obtained.*

The maximum axle deflection under the combined vertical and lateral loads of 60 000 lb. each was 0.408 in. Here again, as in the vertical load tests, a small portion of this deflection is probably due to deflection and seating of the supporting pedestals, but the residual deflection observed upon removal of the loads was only about 0.005 in.

For a closer comparison with service conditions let us assume a 30 000 lb. vertical load and a 20 000 lb. flange thrust applied to each wheel. Under these conditions the axle may be expected to deflect 0.21 in., the stress calculated from the strains in the axle at mid-span would be 31 400 lb. per sq. in., and the distance between the wheel flanges would change about ¼ in. The combination of a lateral or flange thrust load and a nominal vertical load evidently is a much more severe test of the wheel and axle than extremely heavy vertical loads alone.

17. *Combined Strains Due to Mounting, Vertical Loads, and Horizontal Flange Thrust.*—The strains noted in the data presented so far have been these developed in the wheel for each individual method of loading alone. These strains may be added algebraically to determine the approximate effect of any given combination of loadings. In particular it is of interest to determine the total strains produced by combining the strains due to mounting the wheel on an axle with those developed by external static loads.

The algebraic sum of the strains due to mounting and those developed by a vertical load of 150 000 lb. on the wheel are sum-

*See Table 9.

TABLE 7

RESULTANT STRAINS IN VERTICAL LOAD TEST

Values listed are the summations of strains due to mounting and those produced by a total vertical load of 300 000 lb. (150 000 lb. per wheel); strains are in hundred thousandths of in. per in.; + indicates tension, − compression.

Gage Line	Strain on Section B	Strain on Section BS	Strain on Sections N and S	Strain on Sections TS and TN	Strain on Section T	Maximum Strain on Any Section	
						Tension	Compression
Outside Face of Wheel							
1		......	+53.9	+74.6	+83.1	+83.1	......
2		......	+44.1	+61.3	+66.1	+66.1	......
3		+42.3	+46.1	+53.7	+54.0	+54.0	......
4		+19.6	+19.6	+23.3	+39.3	+39.3	......
5		+17.5	+ 2.7	− 3.8	+23.0	+23.0	− 3.8
6		+ 3.2	− 2.4	−11.5	+23.1	+23.1	−11.5
7		+ 2.0	− 9.8	−17.0	+19.2	+19.2	−17.0
8		+ 4.6	− 0.6	− 9.0	+36.2	+36.2	− 9.0
9		+ 1.8	− 4.1	− 0.5	+26.7	+26.7	− 4.1
10		......	− 3.0	......	−10.3	......	−10.3
5R		+ 9.7	− 1.1	−26.5	−81.1	+ 9.7	−81.1
6R		+ 3.5	+ 5.8	+ 5.3	−30.2	+ 5.8	−30.2
7R		+ 4.3	+15.6	+31.0	+10.5	+31.0	......
9R		+ 4.2	......	+ 4.6	......	+ 4.6	......
Inside Face of Wheel							
1	+26.7	+40.5	+39.5	+53.8	+56.3	+56.3	......
2	+32.4	+43.0	+38.1	+45.0	+53.9	+53.9	......
3	+27.5	+25.4	+34.0	+42.2	+53.4	+53.4	......
4	+ 7.6	+14.8	+18.4	+33.0	+54.7	+54.7	......
5	+ 1.4	+ 3.8	+11.1	+26.4	+51.5	+51.5	......
6	+ 4.4	+ 7.1	+ 9.3	+10.4	+36.4	+36.4	......
7	+ 4.1	+ 6.2	+ 8.1	+ 5.0	+19.7	+19.7	......
8	− 1.1	− 0.5	+ 8.7	− 4.8	+24.8	+24.8	− 4.8
9	− 4.6	− 2.0	+ 5.1	− 0.6	+18.4	+18.4	− 4.6
10	+ 0.9	......	+ 5.6	......	−15.5	+ 5.6	−15.5
4R	−25.5	−21.4	+ 3.7	+26.9	+34.2	+34.2	−25.5
5R	− 8.1	− 7.3	+ 3.5	+19.1	+ 3.5	+19.1	− 8.1
6R	− 2.2	− 1.8	− 1.3	+ 0.7	−48.2	+ 0.7	−48.2
9R	......	+ 5.3	......	+ 3.5	......	+ 5.3	......

marized in Table 7. In the upper half of Fig. 17 are plotted (for sections T and TS) these resultant strains and also those caused by the vertical load alone. It will be seen that superimposing the tangential mounting strains on those due to the vertical load causes an increase in the tensile strains. This increase in tangential strain is negligible in the rim, but becomes more significant as we approach the center of the wheel, reaching a maximum value in the hub. The maximum resultant tangential strain for the vertical load test was a tensile strain of 0.00083 (on gage line 1, outside face). The radial strains due to vertical loading were changed only a slight amount by

Table 8

Resultant Strains in Flange Thrust Test

Values listed are the summations of strains due to mounting, and the strains produced by simultaneous total vertical and lateral loads of 60 000 lb. each; strains are in hundred thousandths of in. per in.; + indicates tension, − compression.

Gage Line	Strain on Section B	Strain on Section BS	Strain on Sections N and S	Strain on Sections TS and TN	Strain on Section T	Maximum Strain on Any Section	
						Tension	Compression
Outside Face of Wheel							
1		......	+61.2	+36.0	+22.2	+61.2	......
2		......	+48.1	+27.5	+15.0	+48.1	......
3		+51.4	+46.4	+31.9	+31.8	+51.4	......
4		+24.6	+20.0	+13.9	+18.2	+24.6	......
5		+ 8.3	+ 9.5	+11.5	+14.2	+14.2	......
6		− 6.7	+12.1	+17.5	+20.0	+20.0	− 6.7
7		− 9.0	+ 9.2	+18.7	+23.4	+23.4	− 9.0
8		− 2.4	+18.3	+28.7	+ 9.0	+28.7	− 2.4
9		− 3.3	+38.9	+48.2	− 7.9	+48.2	− 7.9
10		......	+26.3	......	......	+26.3	......
5R		+41.8	−19.9	+22.7	+30.7	+30.7	−41.8
6R		− 1.6	− 9.8	− 9.1	− 7.9	......	− 9.8
7R		+29.3	− 5.2	−25.8	−27.8	+29.3	−27.8
9R		+ 1.8	......	−11.4	......	+ 1.8	−11.4
Inside Face of Wheel							
1	+30.8	+33.5	+33.6	+55.7	+65.0	+65.0	......
2	+28.8	+32.4	+29.8	+50.1	+62.0	+62.0	......
3	+41.2	+40.1	+28.1	+26.7	+22.3	+41.2	......
4	+49.7	+39.4	+15.5	−12.8	−18.8	+49.7	−18.8
5	+43.9	+33.9	+11.2	−12.5	−28.7	+43.9	−28.7
6	+24.0	+18.3	+ 9.5	− 9.1	− 6.3	+24.0	− 9.1
7	+14.7	+ 8.6	+10.6	+ 6.6	+ 8.6	+14.7	......
8	+ 9.9	+ 2.2	− 6.1	−10.9	+37.2	+37.2	−10.9
9	+24.5	+10.7	−21.6	−20.5	+45.1	+45.1	−21.6
10	+33.5	......	−22.4	......	+59.5	+59.5	−22.4
4R	+66.1	+50.8	+ 1.4	−93.7	−151.5	+66.1	−151.5
5R	+30.2	+25.4	+ 1.2	−49.5	−98.2	+30.2	−98.2
6R	− 7.2	+ 0.9	− 3.8	−24.7	−58.7	+ 0.9	−58.7
9R	......	+ 0.7	......	+10.0	......	+10.0	......

the addition of the mounting strains, since the radial strains produced in the rim and plate during mounting were very small.

The algebraic sum of the strains due to mounting and those developed in the flange thrust test by a vertical load of 30 000 lb. (per wheel) and a horizontal flange load of 60 000 lb. are summarized in Table 8. These resultant values are plotted on the lower half of Fig. 17 for sections T and TS of the wheel; also shown in these figures are the strains produced by the external applied loads alone. On the outside face of the hub the horizontal flange load produced tangential compressive strains, whereas the mounting strains were tensile strains, and when added algebraically they tend to balance each other. The resultant strains, therefore, were

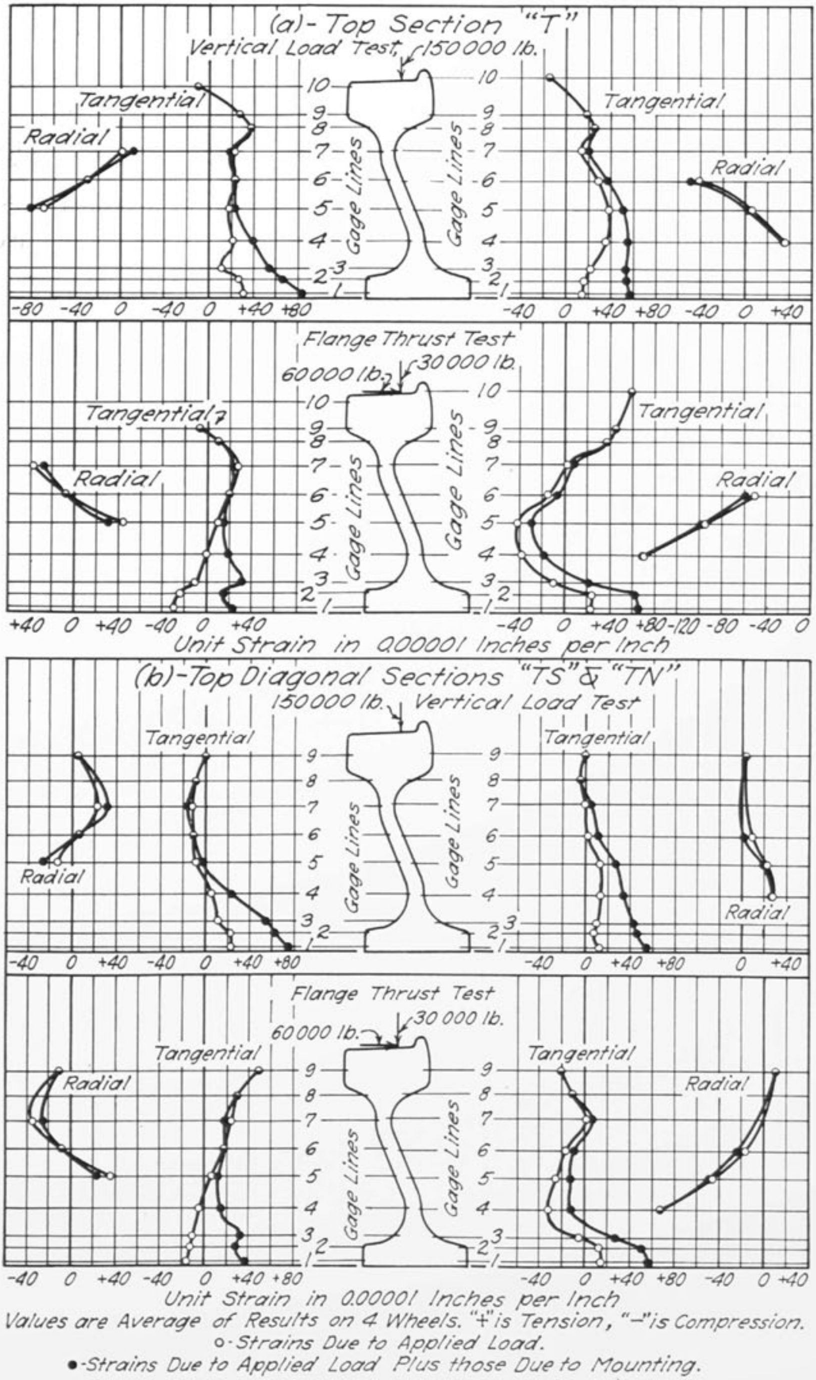


FIG. 17. DISTRIBUTION OF STRAINS ON TOP SECTION (T) AND ON TOP DIAGONAL SECTIONS (TS AND TN) OF WHEEL

smaller than those measured in either the mounting test or in the flange thrust test. On the inside face of the wheel the tangential compressive strains in the plate (caused by applied loads) were also decreased by the addition of the mounting strains. However, the tangential strains developed on the inside face of the hub were tensile strains in both tests, and resulted in a maximum combined strain of 0.00065 (on gage line 1).

The maximum combined radial strains occurred in the plate near the hub (line 4R) on the inside face of the wheel, and reached values of 0.00152 in compression (on section T) and 0.00066 in tension (on section B). These values are practically the same as those due to the external applied loads alone.

An arbitrary extrapolation of the curves of radial strain on the inside face for section T of the wheel would indicate that the radial strains developed in the vertical load test at the fillet joining the rim and plate might be much greater than any of those measured in the plate. It is quite possible that there were localized strains of fairly large magnitude at this point that could not be measured due to the impossibility of taking readings over small sharply-curved surfaces with the 2-in. strain gage employed in these tests.

The curves in Fig. 17a give a summary of the principal results of the investigation, since they show the distribution of the strains along the most highly stressed section of the wheel (section T) that were produced by all three types of tests.

18. *Principal Stresses Developed in Wheels.*—As has already been pointed out the determination of the principal stresses at any point in a wheel from the measured strains is impossible unless the three principal strains at the point are known. If three normal stresses S_x, S_y, and S_z exist in the material in the directions of three rectangular coördinates, the strain e_x in the direction of S_x is

$$e_x = 1/E\left[S_x - \mu(S_y + S_z)\right],$$

where E is the modulus of elasticity and μ is Poisson's ratio for the material. At the surface of the wheel one of these stresses (say S_z) is zero in the direction normal to the surface of the wheel. In this case, if the two principal strains at right angles to each other and tangent to the surface of the wheel are known, they may be interpreted in terms of the principal stresses as follows:

$$e_x = 1/E\,(S_x - \mu S_y)$$
$$e_y = 1/E\,(S_y - \mu S_x)$$

Solving for the value of stress we obtain

$$S_x = (e_x + \mu e_y)\frac{E}{1 - \mu^2}$$

$$S_y = (e_y + \mu e_x)\frac{E}{1 - \mu^2},$$

where S_x is the unit stress in the direction of the strain e_x, and S_y is the stress in the direction of e_y.

Unfortunately, for many of the gage lines it was not possible to obtain readings from which both principal strains could be obtained. This was particularly true on the hub, and on the more sharply-curved surfaces of the wheel, where the radial strains could not be measured. However, for most of the gage locations in the plate of the wheel both principal strains have been obtained from the observed readings. On sections B and T the principal strains occurred in tangential and radial directions corresponding with the directions of measured strains; on the diagonal sections the principal strains were obtained from strain rosettes and are listed in Tables 4 and 6.

The average value for modulus of elasticity of the steel obtained from the physical tests was 29 500 000 lb. per sq. in., and the value of Poisson's ratio for steel is approximately 0.30. Using these values and the strains measured in the flange thrust test (see Table 5) it is evident from the foregoing equations that the maximum stresses developed in the wheel by the maximum applied loads were a radial compressive stress of 52 700 lb. per sq. in. (at gage line 4R on section T of the wheel), and a radial tensile stress of 24 700 lb. per sq. in. (at line 4R, section B), both occurring on the inside face of the wheel.

For the vertical load test the principal stresses computed from the strains developed in the wheel by the 150 000 lb. load were somewhat smaller; the maximum tensile stress was only 14 800 lb. per sq. in. (on gage line 4R, inside face) and the maximum compressive stress was 20 400 lb. per sq. in. (on gage line 5R, outside face). For both the vertical load and the flange thrust test, the great bulk of strain measurements at other locations in the wheel would indicate stresses of much smaller magnitude.

It is impossible to determine accurately the circumferential tensile stresses set up in the hub of the wheel during the mounting and in the subsequent loading tests. The measured circumferential tensile strains were accompanied by radial compressive strains that could not be measured. If the stresses in these two directions in the hub were assumed equal in magnitude and the stress in the third direction (parallel to the axle) was zero, it is evident from the foregoing equations that

$$S_x = (e_x - 0.3e_x)\frac{E}{1 - 0.09} = 0.77Ee_x,$$

or the circumferential stress would be only about four-fifths as great as that which would be obtained by multiplying the measured strain by the modulus of elasticity.

On the basis of these assumptions it is evident that the strain measurements in the hub can be used to furnish only a rough estimate of the stresses developed. Referring to the resultant or combined strains developed in the vertical load test (Table 7) it appears that the maximum tensile hub strain of 0.00083 may correspond to a stress of about 19 000 lb. per sq. in. The maximum final mounting strain of 0.00066 developed in the hub of wheel 4891 would correspond to a tensile stress of approximately 15 000 lb. per sq. in.

In connection with these values, however, it should be remembered that the maximum hub stresses would occur at the point of contact of the wheel bore on the axle where it was not possible to measure the strains. Gage line 1, on which these strains were measured, was approximately ½ in. away from the wheel bore. Since the strains in this region (and the corresponding stresses) decrease rapidly as we move radially away from the wheel bore, it is felt that the maximum circumferential stresses developed at the wheel bore actually may have been somewhat larger than those reported in the foregoing.

VII. Ultimate Load Test

19. *Testing Procedure.*—A final test was made on one of the medium-carbon-steel wheels (No. 3331) to determine whether the wheel would be seriously deformed by extremely heavy vertical loads. It is realized that such static loads might not be normally expected in service, but occasional impact loads at switches and crossings, or jolts applied to the wheel due to worn spots in the rail, might set up momentary strains in the wheel similar to those produced by heavy static loads. Also it was felt that this test might help to locate any inherent weak sections in the wheel that could be corrected by redesign of the shape.

The wheel was placed in a vertical plane under the movable head of the testing machine, and compressive loads were applied along the vertical diameter of the wheel. On the tread of the wheel (at the top and bottom) were placed sections of the head of a standard 130-lb.

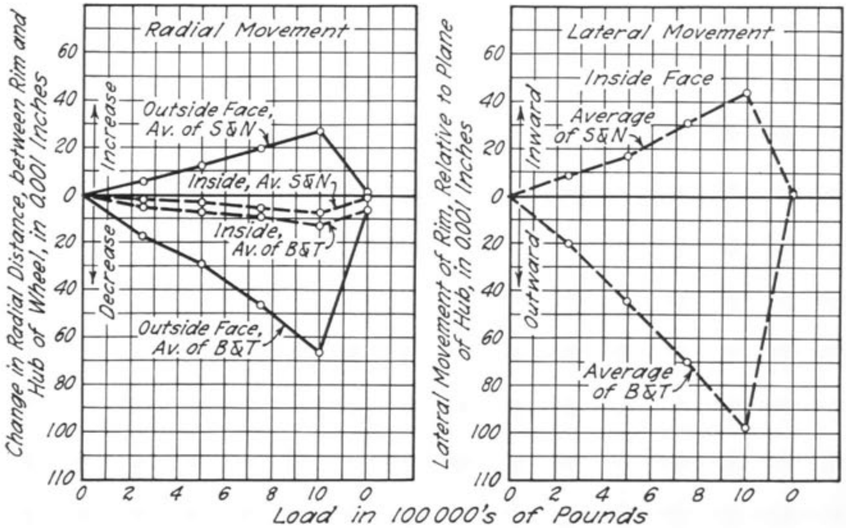


FIG. 18. MOVEMENTS OF WHEEL RIM IN ULTIMATE LOAD TEST

rail through which the load was transmitted to the wheel. This load was gradually increased to a maximum value of 1 100 000 lb. At increments of 250 000 lb. load observations were made of the movements of the rim of the wheel relative to the hub, in both lateral and radial directions.

Readings were made at both ends of the horizontal and vertical diameters of the wheel (at B, T, S and N). The movements of the rim relative to the plane of the hub (in the lateral direction) were observed in the same manner and with the same gage as that used in the loading tests already described. The radial changes in distance between the rim and the hub were measured with a special detachable gage that fitted between the hub and rim, and the movements were read directly on an Ames dial reading to 0.001 in.

20. *Results of Ultimate Load Test.*—It was originally planned to load the wheel to even greater loads than those recorded, but at 1 100 000 lb. the breakage of two large cast iron plates that were used to apply the load to the rail heads made it necessary to stop the test.

The results of this test showing the measured values of rim movements have been plotted in Fig. 18. Both the radial and lateral movements show only small deviations from a straight line relationship with the applied loads, indicating that a general yielding of the wheel did not occur even at these high loads. Upon removal of the load only small permanent sets could be detected from the measurements made; the permanent set in the radial direction was slightly less than 0.006 in., and that in the lateral direction was less than 0.003 in.

The largest deformation during the test was the lateral movement of the wheel rim at the loading points (B and T) relative to the plane of the hub; for a load of 1 000 000 lb. the top and bottom of the wheel moved outward (away from the flange side) almost 0.10 in. At the two ends of the horizontal diameter (N and S) the rim moved laterally inward approximately 0.044 in. at the highest load.

The radial measurements showed that the rim twisted or warped as the load was applied, since the observed radial movements along the outside face of the wheel were relatively much greater than those along the inside face. These radial movements and torsional twists of the rim were not large but would undoubtedly result in developing some high stresses in the wheel. The combined lateral movement and twisting of the rim might be especially effective in developing localized concentrations of stress at the fillets joining the rim and plate of the wheel.

Final visual examination of the wheel after test indicated that the damage sustained by the wheel in this static test was negligible; however, the wheel would doubtless develop a fatigue failure if subjected to a large number of cycles of such excessive overloading.

VIII. Photo-Elastic Analysis

21. *The Photo-Elastic Method.*—The photo-elastic method of stress analysis has been described in detail in many short articles and treatises,* and will be only briefly outlined in this discussion.

In this method a thin transparent isotropic model is stressed in a field of polarized light and the interference fringes formed on the image of the model by the action of the stresses may be interpreted in terms of the difference of the principal stresses in the model. At any free (unloaded) boundary of the model the stress normal to the boundary is zero and the other principal stress (tangent to the boundary) may be obtained directly. The model is made in geo-

*See for instance:
E. G. Coker and L. N. G. Filon, "A Treatise on Photo-Elasticity," Cambridge University Press, 1931.
N. Alexander, "Photo-Elasticity," Rhode Island State College, 1936.
M. M. Frocht, "Recent Advances in Photo-Elasticity," Transactions A.S.M.E. Vol. 53, No. 15, APM 53-11, p. 135-153, 1931.
E. G. Coker, "Photo-Elasticity for Engineers," General Electric Review, Vol. 23, p. 870-877, p. 966-973, 1920.
F. E. Richart, T. J. Dolan, T. A. Olson, "Tests of Reinforced Concrete Knee Frames and Bakelite Models," Bulletin 307, Engineering Experiment Station, University of Illinois, 1938.

metric proportion to the member to be studied and is similarly loaded. The method may be said to be an experimental analogy combining the principles of optics and theory of elasticity, and the stresses determined are those which would be developed in a perfectly elastic member having the same proportions as the model.

Recent experimental work has indicated that a three-dimensional stress analysis can be made by photo-elastic methods, but the process involved is still in an experimental stage. For this reason photo-elastic analyses have been limited mainly to the study of stresses in two dimensions, that is, where the loads are acting in one plane and the member is in the form of a flat plate lying in the same plane. A complete analysis of the stresses in a car wheel cannot be made by using a two-dimensional technique and the results of the analysis herein reported should be regarded only as data giving a qualitative idea of the relative distribution of radial stresses in the wheel.

22. *Outline of Tests.*—Tests were made on a flat, annealed, bakelite model ¼ in. thick whose contour was cut to the same proportions as the radial cross-section of the standard A.A.R. 36-in. diameter wrought-steel wheel but was reduced to a scale ⅓ full size. In order to study the effect of varying the radii of the fillets joining the rim to the plate of the wheel, the model was first finished and tested with larger fillets, and then cut down on one side at a time to the standard A.A.R. dimensions and re-tested. The model with large fillets on both sides of the wheel has been designated as design A, and that with a large fillet on the inside face of the wheel only, has been designated design B.

The model was supported on the edge representing the wheel bore, and loaded in the plane of the model either with vertical loads applied at different positions along the tread of the wheel or with combined vertical and lateral flange loads of equal intensity (see Figs. 19 to 22). Some difficulty was encountered in loading the model because the rotation and movement of the rim produced by the stresses tended to move the point of application of the load to different positions on the tread. For this reason, the stresses caused by the vertical loads are not directly comparable between the various models tested, but all results would represent possible types of loading in service.

The loads were applied by means of a small movable testing machine mounted in the photo-elastic polariscope, and a beam of polarized light was passed through the model while it was being stressed. The optical apparatus consisted of a mercury vapor lamp

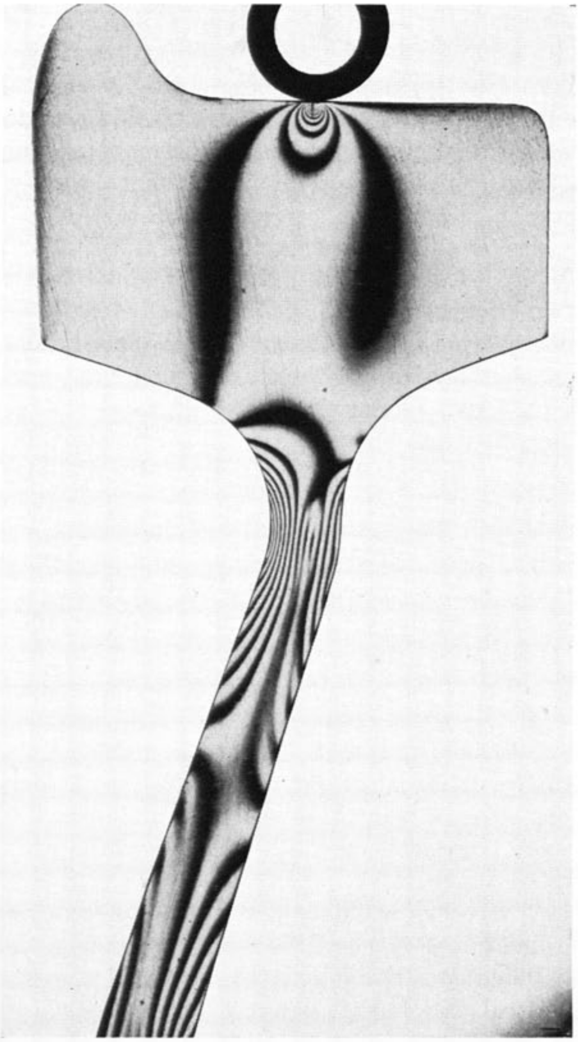


FIG. 19. FRINGE PHOTOGRAPH OF STANDARD A.A.R. WHEEL SUBJECTED TO VERTICAL LOAD ONLY

with monochromatic green filter, Polaroid disks for polarization of the light, quarter wave plates of mica to produce circularly polarized light, and a series of lenses to produce a parallel beam of light through the model and focus an image of it in a camera. Photographs were made of the interference fringe patterns formed on the image of the model when subjected to each of the loadings, and the

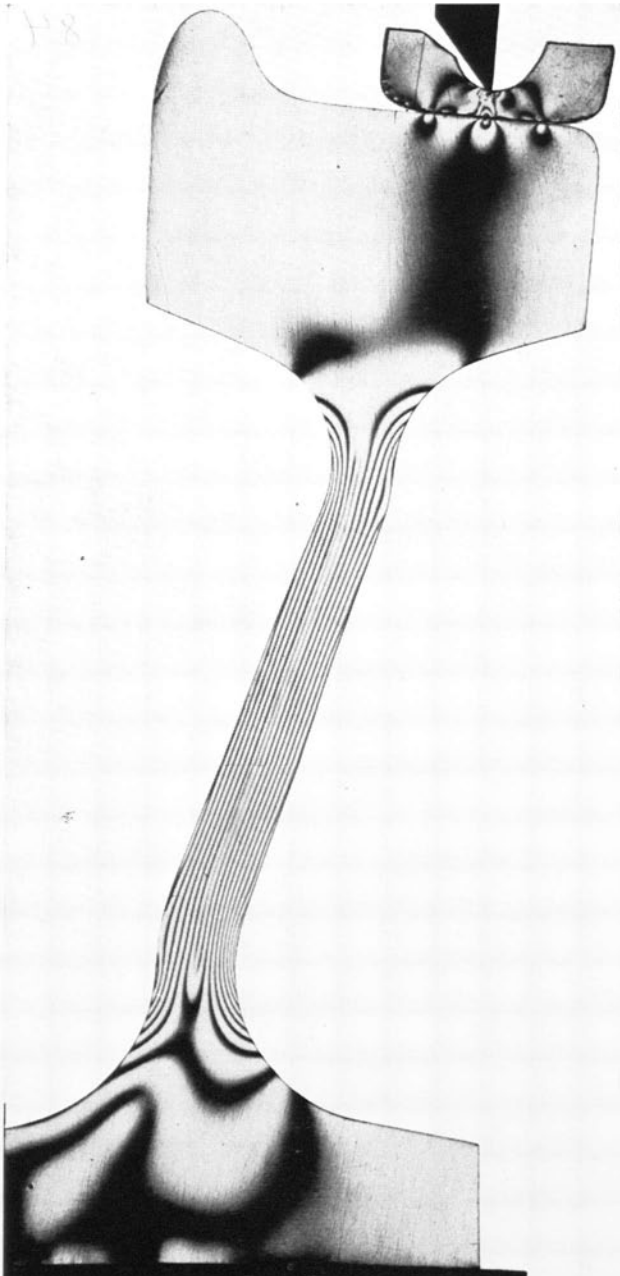


FIG. 20. FRINGE PHOTOGRAPH OF STANDARD A.A.R. WHEEL SUBJECTED TO VERTICAL LOAD NEAR OUTSIDE FACE

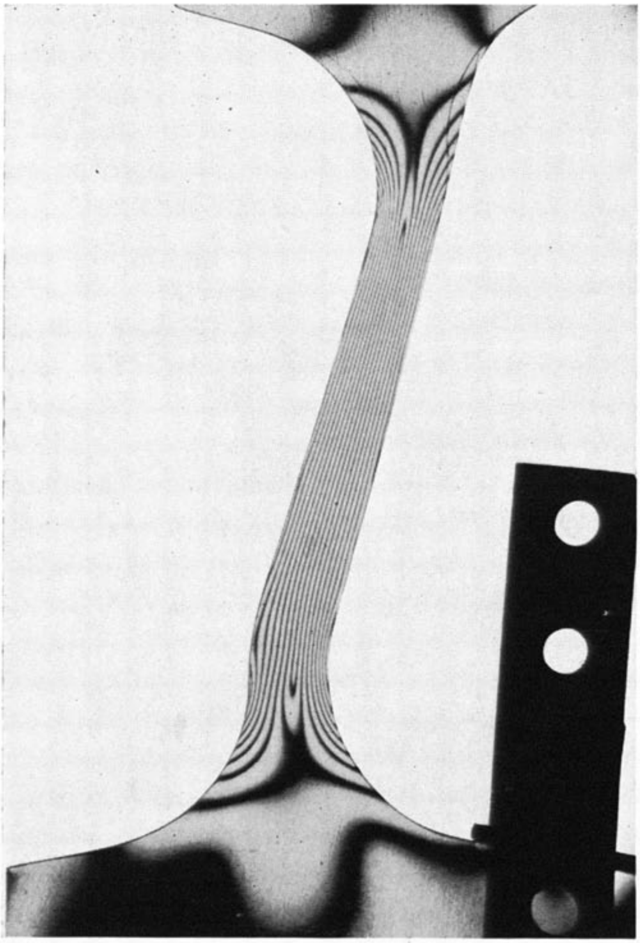


FIG. 21. FRINGE PHOTOGRAPH OF WHEEL DESIGN A SUBJECTED TO COMBINED VERTICAL AND LATERAL LOADS

order of appearance of each fringe was determined by gradually loading and unloading several times. The results of the tests were interpreted from these fringe photographs, since each new fringe to appear represented an equal increment of increase in boundary stress.

23. *Results of Photo-Elastic Analysis.*—Typical fringe photographs of the model are shown in Figs. 19 to 22, and the distribution of stresses along the boundaries of the model were obtained from photographs of this type. For convenience in interpreting the relative values of stress for the various loadings used, the values of stress as

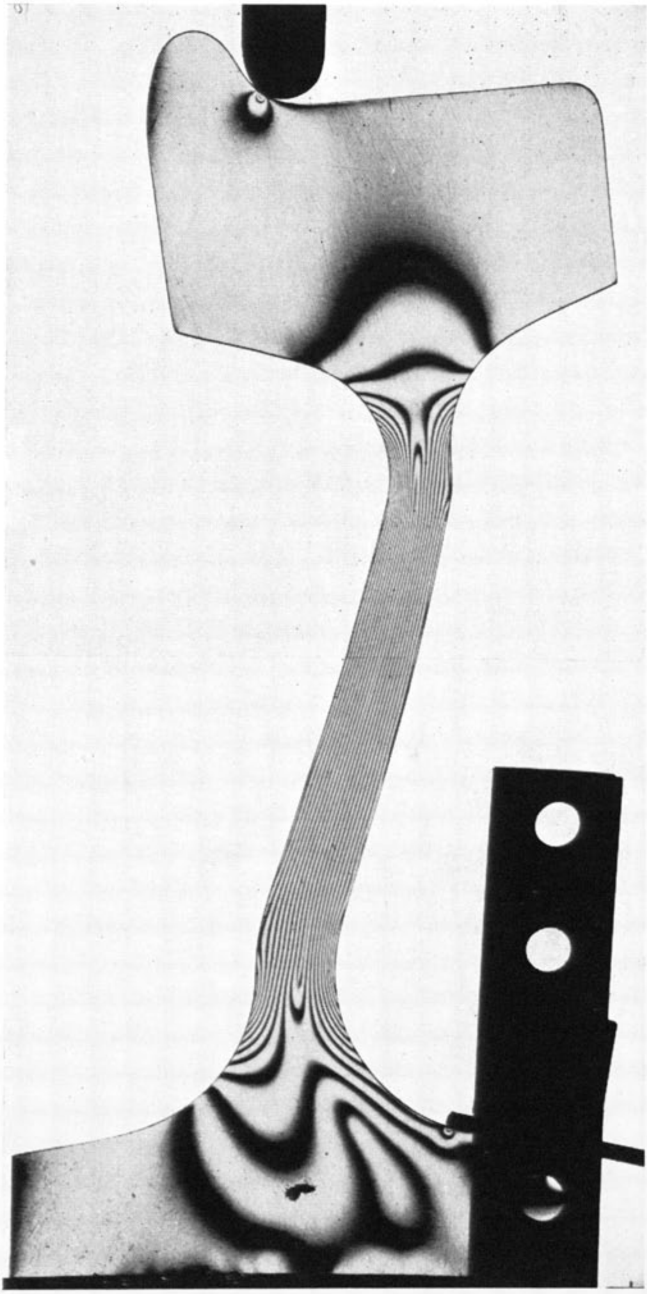


FIG. 22. FRINGE PHOTOGRAPH OF STANDARD A.A.R. WHEEL SUBJECTED TO COMBINED VERTICAL AND LATERAL LOADS

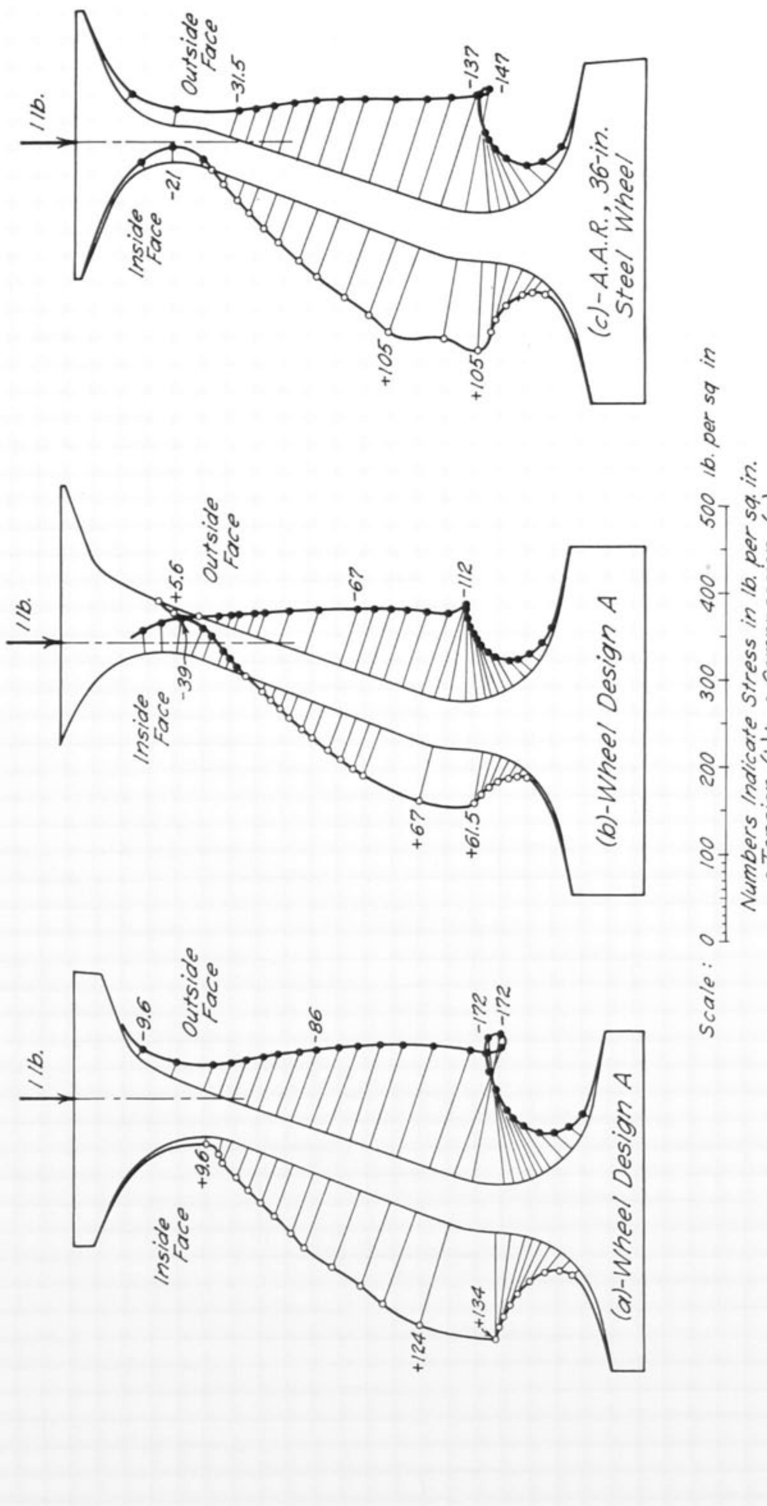


FIG. 23. PHOTO-ELASTIC STRESS DISTRIBUTION FOR VERTICAL LOAD OF ONE POUND

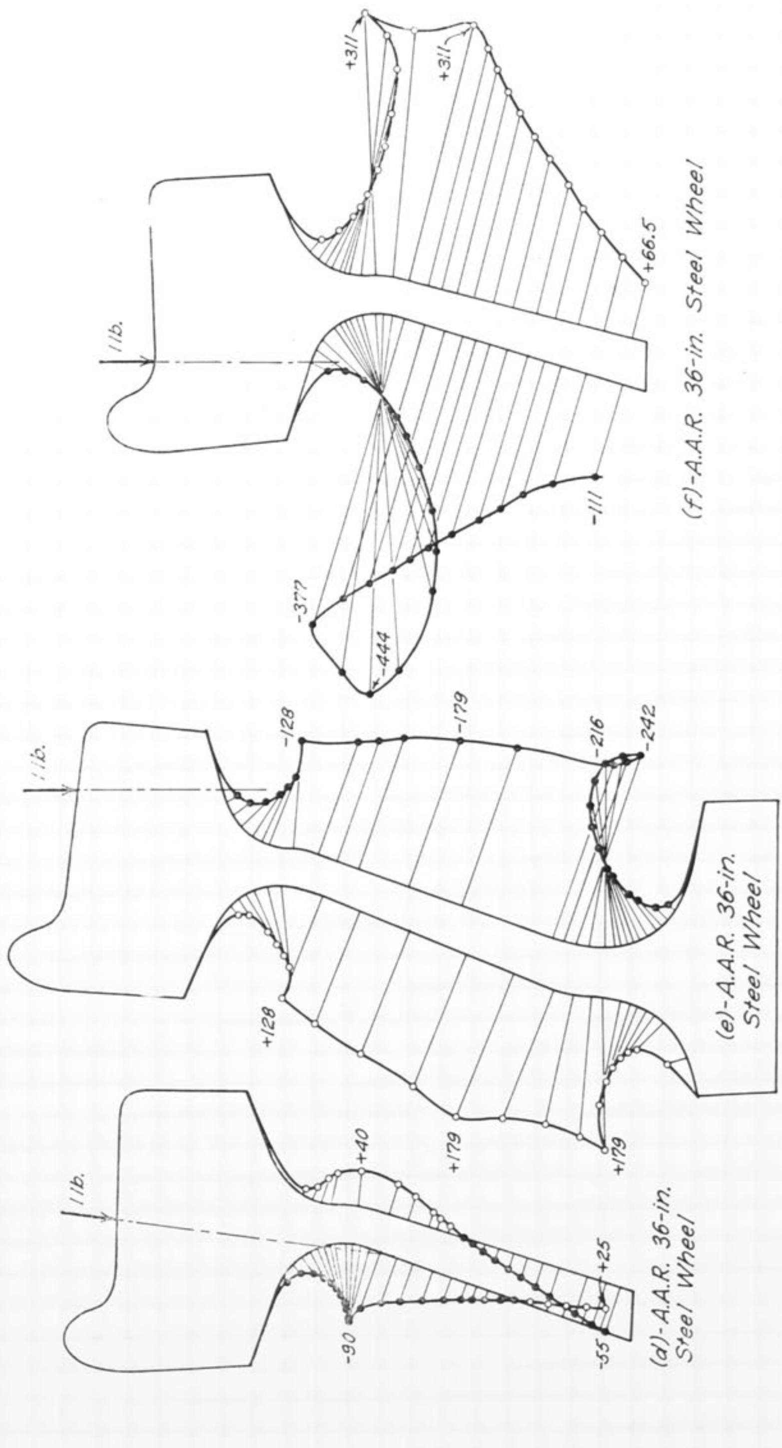
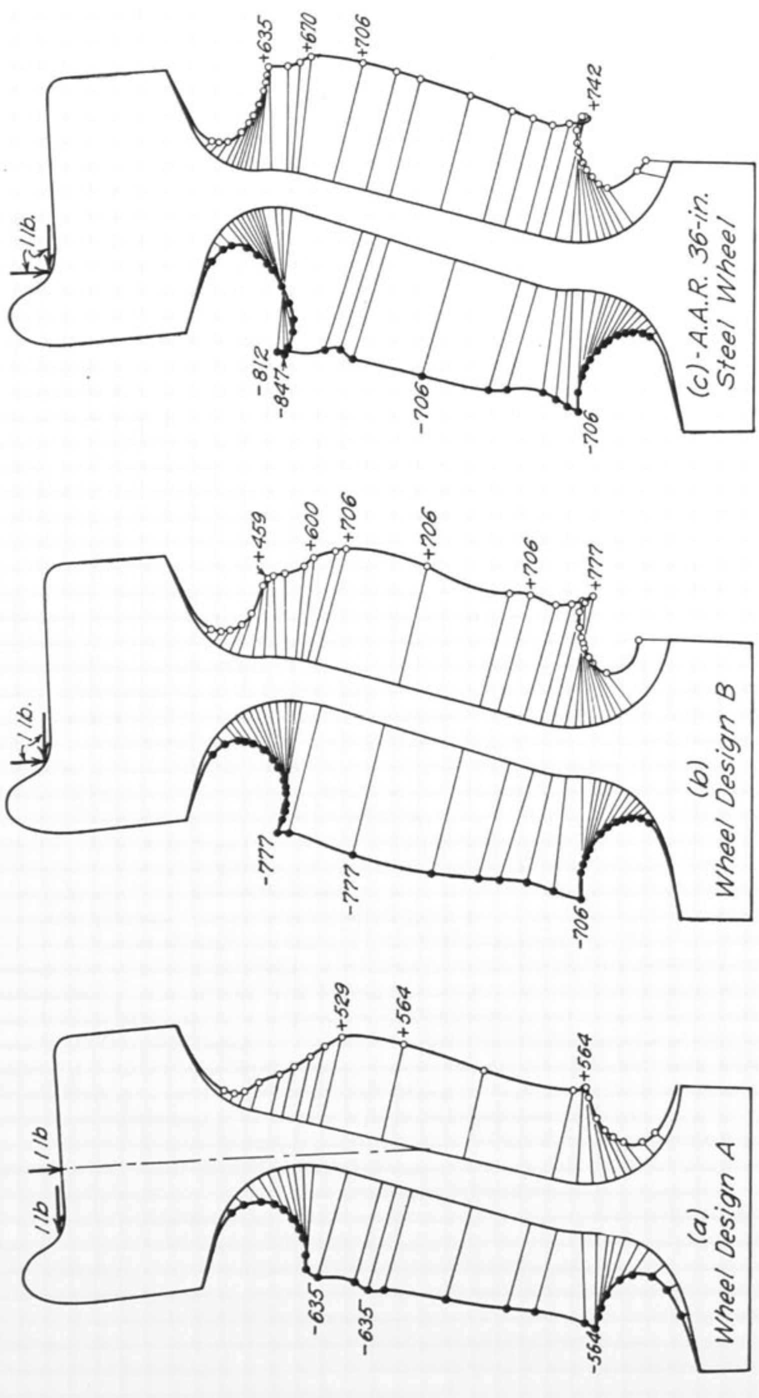


FIG. 23 (CONCLUDED). PHOTO-ELASTIC STRESS DISTRIBUTION FOR VERTICAL LOAD OF ONE POUND



Numbers Indicate Stress in lb. per sq. in. ∘ Tension, (+); • Compression, (−) Scale: 0 500 1000 lb. per sq. in.

FIG. 24. PHOTO-ELASTIC STRESS DISTRIBUTION FOR VERTICAL AND FLANGE THRUST LOADS OF ONE POUND EACH

determined from each fringe photograph were divided by the load applied to the model, to yield values of stress per pound of load. These values of stress due to a unit load (one pound) are plotted as ordinates perpendicular to the boundaries of the models in Figs. 23 and 24.

The positions of the loads applied to the model have been indicated on each stress distribution curve; it should be noted that the vertical load was not in the same position for each test. Figures 23a, 23b, and 23c indicate that for loads applied approximately at standard gage distance, a slight displacement of the vertical load toward the inside face of the wheel tends to decrease considerably the stresses in the plate near the hub. It will also be noted that the stresses in the plate near the rim are relatively small for these loadings, and a slight shift of the load causes the stresses to vary between small tensile and compressive values; it is doubtful whether the larger fillets (design A) offer any advantage over the standard design for this type of loading since these stresses are small in either case. A slight inclination of the load as in Fig. 23d causes the stresses at the junction of rim and plate to increase somewhat, and again decreases the stresses near the hub.

A comparison of Figs. 23c, 23e, and 23f shows the large variation in stresses that was caused by shifting the vertical load from a position near the flange to a position near the outside edge of the tread. A load near the flange (Fig. 23f) developed large compressive stresses on the inside face of the plate at the junction with the rim, and tensile stresses on the outside face, while the stresses near the hub were negligible. A load near the outside edge of the tread (Fig. 23e) developed tensile stresses on the inside face and compressive stresses on the outside face which reached maximum values near the hub of the wheel; these maximum stresses, however, were not as large as those that were developed at points near the rim by a vertical load applied close to the flange. In general, the maximum stresses at the upper fillets caused by the vertical loads occurred on the inside face, and those on the lower fillets occurred on the outside face of the wheel. As the vertical load was moved across the tread, the greatest variation of stress (from $-$ 444 to $+$ 120 lb. per sq. in.) occurred on the inside face of the plate near the rim; this point might therefore be the one most likely to develop fatigue cracks due to excessive repeated vertical loading.

The stresses developed by combined vertical and lateral flange loads (Fig. 24) were much larger than those due to vertical loads alone, and have been plotted to a scale only one-fifth as large. The stresses developed for design A (Fig. 24a) were smaller than those shown for the other two shapes tested, probably because the vertical load was applied approximately at gage distance, whereas for the other two shapes tested the vertical load was applied at the flange. It will be noted that vertical and lateral flange loads of equal intensity developed a compressive stress along the inside face of the plate and a tensile stress on the outside face that were both fairly uniform in their distribution. Only a slight tendency to develop localized stresses is shown at the fillers. However, in comparing the distribution of stresses in designs A and B with those for the standard wheel, it will be noted that increasing the radius of the fillet under the rim and straightening the outside face just under the rim both help to smooth out the stress distribution and to reduce the tendency to develop localized concentrations of stress at these points.

Comparing the results of Figs. 23a to 23d with those of Figs. 24a to 24c it may be estimated that equal combined vertical and lateral flange loads may be expected to develop stresses from 5 to 10 times as great as those produced by a single vertical load applied approximately at standard gage distance. Likewise the addition of a lateral load on the flange tends to develop large stresses throughout the entire depth of the wheel plate.

It has been pointed out in Section 21 that this photo-elastic analysis does not take into account the distribution of stresses through the wheel in a circumferential direction and the torsional rigidity of the rim; therefore only an approximate indication of the relative radial stresses could be obtained. This transfer of stress to adjacent portions of the wheel would materially alter the distributions shown in Figs. 23 and 24, and probably would result in decreasing the relative stresses in the plate in the region of the hub. However, it is felt that the results do indicate the large variations of stress that may be developed in the wheel by several types of loading that may be encountered in service.

IX. Conclusion

24. *Discussion of Results.*—In general the results of these tests show that the stresses developed in the standard A.A.R. steel wheel by mounting on an axle, and those due to large vertical static loads applied to the tread of the wheel, were not excessive. A vertical load of about six times the nominal design load did not produce strains in the wheel that would be considered damaging, even if repeated a large number of times. However, combined vertical and lateral flange loads of about twice the magnitude that might normally be

expected in service produced a much more serious state of stresses in the wheel than did the large vertical load alone. For lateral flange loads in excess of about 30 000 lb. the increased strains in the wheel hub have indicated that a localized circumferential slipping or rubbing action may exist between the wheel and axle that might lead to an axle failure in service. It seems reasonable to expect that increasing the speeds of rolling stock will greatly increase the lateral forces developed on the wheel flange at any abrupt change in track alignment. For this reason the most damaging stresses developed by the loads applied to a wheel in service are probably those caused by the lateral pressure on the flange at turnouts, switches, and transition curves.

In addition to the stresses discussed in this paper there may be some initial stresses present in the wheel that decrease the margin of safety somewhat. Even though the initial stresses due to the forming and heat-treating operations might in many cases be small, these stresses may be materially changed while in service. Excessive loading while in operation, or localized heating of the wheel caused by the application of brakes, might develop a new state of residual stress in the wheel.

Practically the same stress distributions were observed in the silico-manganese-steel wheels as those obtained for the ordinary carbon-steel wheels. This is to be expected since the wheels were all of the same dimensions; the stresses should be a function of the loads and geometric shape of the member provided the elastic strength is not exceeded. However, the higher elastic strength of the silico-manganese-steel indicates that these alloy steel wheels should offer an increased resistance to yielding under excessive loads that may be applied in service.

The values listed in Table 9 give a comparison of the stresses and deflections in the axle (as computed from the ordinary theory of flexure) with the values obtained by strain and deflection measurements in the actual tests. The rotations of the wheel and journals produced by the loading caused slight lateral movement of the points at which the loads were applied and this may have affected the computed bending moments somewhat. In spite of this fact, the agreement between observed and computed values is quite good. This would indicate that the elastic strength of the axle was not exceeded by the large stresses imposed during the test; furthermore, if the loading conditions are known, the stress and deflection in an axle at mid-span can be accurately computed by using the ordinary theories of flexure.

Table 9

Stress in and Deflection of Car Axle

Values determined for A.A.R. standard 5½ in. x 10 in. axle on journals 77 in. c. to c., loaded through wheels at two points 57½ in. c. to c. Horizontal flange thrust loads were applied to wheels at a point 18½ in. from axis of axle.

Load at Each Wheel 1000's of lb.		Moment at Midspan 1000's of in.-lb.	Extreme Fiber Stress at Midspan 1000's of lb./per sq. in.		Deflection at Midspan in.	
Vertical	Horizontal		Computed	From Strain Reading on 10-in. Gage	Computed	Observed
100	0	975	46.4	42.6	0.300	0.295 0.295
150	0	1463	69.5	59.2	0.450	0.409 0.402
30	0	293	13.9	14.3	0.090	0.104 0.104
30	20	663	31.6	31.4	0.200	0.209 0.204
30	40	1033	49.0	48.0	0.310	0.300 0.309
30	60	1403	66.7	67.2	0.420	0.408 0.407

The flange thrust tests indicated that the position of maximum compressive stress in the wheel (on the inside face of the plate near the hub, line T) would also become the position of maximum tensile stress (on line B) as the wheel revolved in service. This large range of stress would start a fatigue fracture at this location if the lateral load was applied during a large number of revolutions of the wheel. The results of the photo-elastic test indicated that a similar, though perhaps a smaller, variation of stress may occur on the inside face of the wheel at the fillet joining the rim and plate; this variation would be caused by the shifting position of the vertical load due to the lateral nosing of the wheel and axle assembly in service. Since the results of the photo-elastic test also show that the stresses in the wheel are materially altered by slight variations in the positions of the applied loads, the results of laboratory tests on steel wheels probably do not give a complete picture of the stresses which are developed in service under continually varying conditions of loading.

25. *Summary of Results.*—The principal results of this investigation may be stated as follows:

(1) The steel of which the wheels were made, as indicated by tests of samples cut from companion wheels, was of relatively high strength and had considerable ductility. Specimens cut from different portions of the wheel showed the properties to be fairly uniform except for the material in the hub, which generally had a slightly

lower strength and ductility. The average values of yield point and ultimate tensile strength for the carbon-steel were 77 480 and 130 260 lb. per sq. in., respectively, and for the silico-manganese-steel were 99 430 and 159 300 lb. per sq. in., respectively. The complete physical properties of the steels are listed in Table 1.

(2) The mounting tests, in which each wheel was pressed on an axle which had a mounting allowance of 0.0078 to 0.0085 in. produced large circumferential strains in the hub, but only small strains in the other portions of the wheel. The circumferential strains decreased as the distance from the hub increased. However, the hub strains due to mounting were generally greater than those produced in the hub by the subsequent external static loads applied in the vertical load and flange thrust tests. The results of the mounting test are shown in Table 2 and Figs. 5 to 7.

(3) For three of the four wheels tested the loads required to mount the wheel on an axle were below the minimum prescribed loads specified in the A.A.R. Wheel and Axle Manual.

(4) The application of vertical loads up to 150 000 lb. per wheel produced rather large radial compressive strains, especially on the gage lines nearest the hub of the wheel on the outside face of the plate. The circumferential strains produced by this loading were generally tensile strains of relatively small magnitude. The maximum radial compressive strain observed was 0.00068, and the maximum circumferential tensile strain was 0.00038. The observed strains are plotted in Figs. 9 and 10 and the average strains at maximum load are listed in Table 3.

(5) The combined or resultant strains due to mounting and those developed by vertical loading, as indicated by an algebraic summation of separately measured strains, showed increased circumferential tensile strains in the hub, reaching a maximum value of 0.00083 on the outside face of the wheel. The radial strains due to vertical loading were changed only a slight amount by the addition of the mounting strains. In Table 7 are tabulated the resultant strains in the vertical load test.

(6) The flange thrust test employing a constant vertical load of 30 000 lb. per wheel and horizontal flange loads up to 60 000 lb. produced a much more severe state of stress in the wheels and axle than did the heavy vertical loads alone. The largest compressive strains occurred along radial gage lines in the plate between the hub and the flange at which the load was applied, (on the top section) and reached a maximum value of 0.00150 on the inside face of the wheel. The maximum tensile strain of 0.00067 also occurred on a

radial gage line on the inside face of the wheel, but on the bottom section of the wheel near the hub. The circumferential strains reached a maximum value in the rim at the point of application of the loads. Aside from this location the circumferential strains were less than 0.0004 throughout the wheel. The strains observed in the flange thrust test are plotted in Figs. 14 and 15 and summarized in Table 5.

(7) The combined or resultant strains due to mounting and those developed by vertical and horizontal flange loads reached maximum tensile values of 0.00065 in the hub, and 0.00066 in the plate, whereas the maximum compressive combined strain reached a value of 0.00152 in the plate. The last two values were measured in the radial direction at the locations just mentioned. The resultant strains in the flange thrust test are listed in Table 8.

(8) The lateral bending of the wheel plate and the total movement between the rims of the two wheels was much greater in the flange thrust test than that observed for heavy vertical loads alone. The deflections and strains observed for the axles apparently varied in direct proportion to the applied loads for each test. Even though the largest measured strains (in the flange thrust test) would indicate a stress of 67 200 lb. per sq. in. at mid-span, the yield point of the steel was probably not exceeded. The observed data on wheel and axle movements are plotted in Figs. 11 and 16.

(9) The average modulus of elasticity obtained from tests of tensile specimens taken from companion wheels together with the measured strains observed in the wheels have been used to estimate the magnitudes of some of the maximum wheel stresses occurring in the various types of test. From this study it appears that the maximum measured strains in the hub during mounting would correspond to tensile stresses of less than 15 000 lb. per sq. in., and the maximum combined strains developed in the vertical load test would indicate tensile stresses of approximately 19 000 lb. per sq. in. However, the indicated stresses developed in the hub would be somewhat larger than these values if measurements could be obtained at the wheel bore. The strains developed in the flange thrust test (see item 6) indicate a maximum compressive stress of 52 700 lb. per sq. in. and a tensile stress of 24 700 lb. per sq. in.; both of these values were radial stresses on the inside face of the plate near the hub. At these locations of highest stress the tangential stresses were less than about ⅔ of the radial stresses. The great bulk of strain readings throughout the wheels would indicate stresses generally less

than half the magnitudes of these maximum stresses. The principal stresses developed in the wheel are discussed in Section 18.

(10) When subjected to a diametrical compressive load of 1 000 000 pounds the rim of the wheel moved laterally almost 0.10 in., and twisted about a circumferential line. The permanent set produced in the wheel by such excessive overload was, however, quite small; the permanent radial shortening across the plate was less than 0.006 in., and the final lateral movement of the rim was less than 0.003 in. after removal of the load. The data observed in this test are plotted in Fig. 18.

(11) It has been pointed out that there may exist localized concentrations of stress on the sharply-curved fillets of the wheel surface that could not be accurately determined by strain gage measurements. Only the average strains over 2-in. gage lengths were obtained, and it was not possible to locate radial gage lines on the fillets joining the plate to the rim and the hub.

(12) A photo-elastic stress analysis was made of a bakelite model of the cross-section of the standard A.A.R. wheel to determine in a qualitative manner the relative radial stresses and stress concentrations existing. The results indicated that only small concentrations of stress were developed at the fillets in the model, but that considerable variations in the stress were produced by slight changes in the position of the loads. The stresses developed by combined vertical and lateral loads of equal intensity were approximately 5 to 10 times as great as those produced by a vertical load at standard gage distances. The results of this test are shown in Figs. 23 and 24.

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