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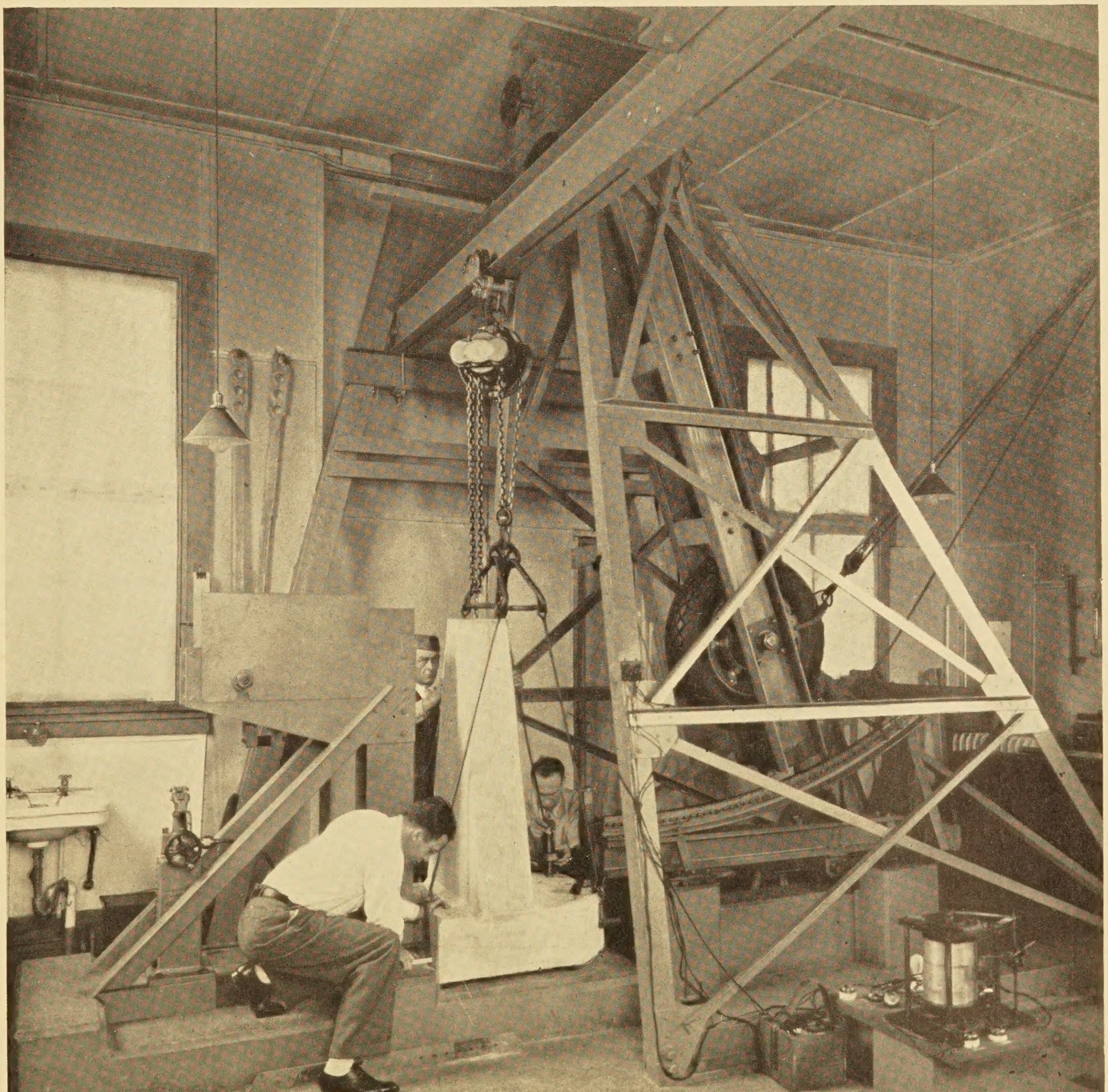

MACHINE FOR MAKING IMPACT AND SUSTAINED LOAD TESTS OF CONCRETE

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A MACHINE FOR IMPACT AND SUSTAINED LOAD TESTS OF CONCRETE

BY THE DIVISION OF TESTS, BUREAU OF PUBLIC ROADS

Reported by **L. W. TELLER**, Senior Engineer of Tests, and **JAMES A. BUCHANAN**, Associate Engineer of Tests

ONE of the fields of research in which the Bureau of Public Roads has been engaged for a number of years is that concerned with the structural action road surfaces. The development of a knowledge of the forces to which road surfaces are subjected and of the manner in which pavements react to those forces are the two major objectives in this research.

An essential part of this general investigation has been a study of the forces to which road surfaces are subjected by vehicle wheels. These forces may be of a sustained nature if the vehicle is stationary; they may be of a transient nature if the vehicle is rolling smoothly over the pavement; or they may occur quite suddenly where irregularities in the surface contour of the road induce impact reactions.

Load limitations imposed by law, and data on actual vehicle weights obtained in numerous traffic surveys, indicate the order of magnitude of the forces to which pavements are likely to be subjected by the wheels of static or slowly moving vehicles. The extensive studies of motor-vehicle impact phenomena made in this country have developed quite definite knowledge of the reactions to be expected under various conditions and of the influence of each of the major factors that determine the magnitude.[1] More recently work along similar lines has been conducted in England and the published data from these researches [2] are in general accord with those obtained in the studies mentioned above.

It can be said, therefore, that so far as the structural action of pavements is concerned, there exists a fairly adequate knowledge of the forces which may be expected from the wheels of present-day vehicles.

Of equal importance is the development of a knowledge of the effects of these forces on pavement structures of various types. Both theoretical analyses [3] and experimental researches [4] have furnished a considerable amount of fundamental information concerning the effects of static or slowly applied forces on pavements of the rigid type, out of which there is evolving a better understanding of the principles of design for such pavements for loads of this nature. When, however, the effects of suddenly applied forces (such as impact reactions) are considered, the information available is much less satisfactory. In spite of the attention that has been given in recent years to the impact testing of materials of all sorts, the fact remains that the underlying principles of stress development under impact are not well understood.

TEST METHOD SHOULD MEET SEVERAL REQUIREMENTS

It is generally agreed that static and impact forces may differ in their effects and it is believed that the degree of difference depends upon the time rate of deformation in the material being stressed, and probably upon other factors. However, the extent to which the various phenomena apply in the case of motor-vehicle impact has not been known.

The approach to a study of the effects of impact forces on rigid pavements is somewhat simplified by two facts: First, portland cement concrete is the principal material to be considered; and second, the total elapsed time of the impact reaction varies from about 0.05 to 0.15 second, which is neither very short nor is the range great as compared with those encountered in other fields. The absolute maximum force developed during any impact reaction is, of course, instantaneous.

As a first step in its program of research on this particular subject the Bureau of Public Roads undertook a thorough study, under controlled conditions in the laboratory, of the behavior of large-size flexure specimens of concrete when subjected to sustained and to impact forces of comparable magnitudes. This work has now been in progress for about 2 years and many tests have shown the method of test and the testing equipment to be quite satisfactory. It is the purpose of this paper to outline the method of test that was adopted and to describe the testing machine that was designed and built for this purpose, a machine that is unique in several respects. The presentation of the results of the tests that are being made will be left to a subsequent discussion.

Since the object of the study was to determine the relative effects of sustained and impact forces on the flexural behavior of concrete, the selection of a method of test and the design of the testing machine were guided by the following basic requirements:

1. It should be possible to apply both sustained and impact forces of known magnitudes to a specimen in such a manner that the only variable present would be the duration of the applied force.
2. The specimens should be flexure specimens of portland cement concrete.
3. The magnitude of the forces, the duration of the forces, the masses involved in the impact, the flexing of the concrete and other conditions surrounding the test, should not depart too widely from conditions that obtain when a pavement slab sustains a wheel load.
4. The testing machine must be of a practicable size.
5. The force system should be determinate.

A consideration of these requirements led to the adoption of the method of test shown schematically in

[1] The Motor Truck Impact Tests of the Bureau of Public Roads, by Earl B. Smith, Public Roads, vol. 3, No. 35, March 1921.
Motor Truck Impact as Affected by Tires, Other Truck Factors and Road Roughness, by James A. Buchanan and J. W. Reid, Public Roads, vol. 7, No. 4, June 1926.
Motor Truck Impact as Affected by Rubber Tread Thickness of Tires, by James A. Buchanan, Public Roads, vol. 11, No. 7, September 1930.
Road Impact Produced by a Heavy Motor Bus, by James A. Buchanan, Public Roads, vol. 13, No. 9, November 1932.

[2] Impact of Wheels on Roads, by Aughtie, Batson, and Brown. Proceedings, Institution of Civil Engineers (Br.), vol. 237, Session 1933-34, part I.
The British researches are also reported in the Reports of The Ministry of Transport, Roads Department, beginning with that for the year 1930.

[3] Stresses in Concrete Pavements Computed by Theoretical Analysis, by H. M. Westergaard, Public Roads, vol. 7, No. 2, April 1926; and Analytical Tools for Judging Results of Structural Tests of Concrete Pavements, by H. M. Westergaard, Public Roads, vol. 14, No. 10, December 1923.

[4] Stress Measurements in Concrete Pavements, by L. W. Teller, Proceedings, Fifth Annual Meeting, Highway Research Board, December 3-4, 1925.
The Six-Wheel Truck and the Pavement, by L. W. Teller, Public Roads, vol. 6, No. 8, October 1925.
The Structural Design of Concrete Pavements, by L. W. Teller, and Earl C. Sutherland, Public Roads, vol. 16, Nos. 8, 9, and 10, and vol. 17, Nos. 7 and 8.

figure 1. A compound pendulum, carrying a vehicle wheel at its center of percussion, is used to apply horizontal forces to a test specimen that is arranged as a vertical cantilever. Impact forces are developed by swinging the pendulum, and sustained forces are developed by exerting a pull on the axle of the wheel on the pendulum.

It is readily apparent that this arrangement can, by proper design, be made to satisfy the basic requirements stated above. Either sustained or impact forces can be applied to the specimen at the same point and through the same contact medium, the tire. The horizontal application of forces is advantageous, as the gravitational effect is eliminated; and, in the case of the impact force, a single blow without secondary rebound can be applied. The cantilever specimen has definite advantages and the use of a beam of varying width gives a section that has useful stress variation characteristics and simulates to a fair degree the unsupported corner of a pavement slab. The characteristics of the pendulum as a dynamic machine can be theoretically established and empirically checked. By designing a machine and specimen of suitable proportions, the forces and masses involved in the test reach satisfactory values.

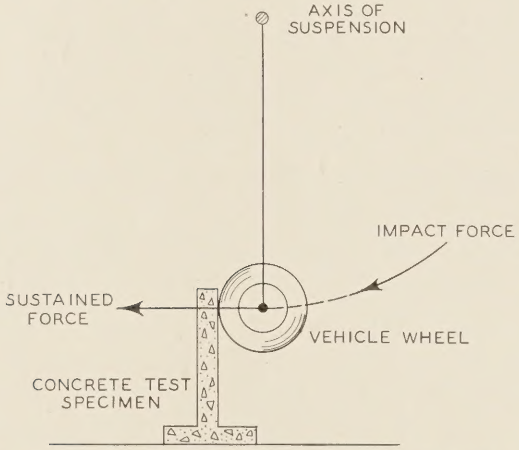


FIGURE 1.—SCHEMATIC DIAGRAM SHOWING THE PRINCIPLES OF THE TEST.

The design of the test specimens determined the arrangement of the testing machine and certain of its principal dimensions. It is logical, therefore, to describe these specimens before discussing the machine itself.

FOUR TYPES OF TIRE SELECTED FOR TESTING IN MACHINE

The general form of the test specimens is shown in figure 2. Except for the thickness of the vertical cantilever the dimensions of all specimens are the same. The cantilever beams are 48 inches long, measured from the top surface of the heavy integral base to the free end. The cantilever portion is 30 inches wide at the base, 5 inches wide at the top or free end, and of a constant thickness. Specimens 4, 6, 8, 10, and 12 inches thick have been made. The integral base portion used to anchor the test specimen to the floor is 10 inches thick. No reinforcing steel has been used in any of the specimens.

The center of the area of force application is 40 inches above the base, at which point the facial area of the beam is sufficient to accommodate the entire area of contact for any of the tire and load conditions that it is planned to use.

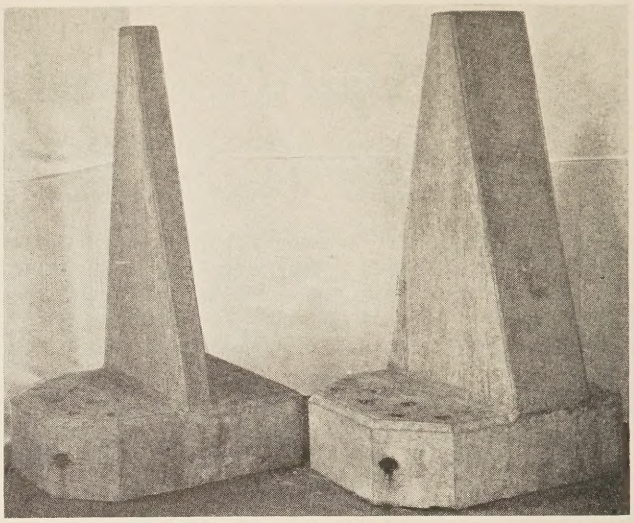


FIGURE 2.—CANTILEVER TEST SPECIMENS. THIS VIEW SHOWS THE CONTRAST BETWEEN SPECIMENS 4 AND 12 INCHES THICK AND ALSO SHOWS A NUMBER OF DETAILS TO WHICH REFERENCE IS MADE LATER.

Because of their direct effect on the design of the machine, the probable maximum bending stress and the probable deflection of the cantilever at the center of the area of force application were calculated for each of the five thicknesses of specimen. These calculated values for a range of forces are shown in figure 3. The values indicate that a maximum force of about 15,000 pounds would probably be required, assuming concrete of normal strength. A large percentage of the testing, however, would involve forces of 6,000 pounds or less.

The range in forces required for testing the beams indicated that appropriate tire equipment would have a normal or rated load-carrying capacity of about 2,000 pounds. In order to make it possible to vary the durations of the impact reactions and also the areas of contact over which the forces were applied to the test specimen, interchangeable tires of the four types commercially available were chosen. In order that the effect of tread design variations might be minimized, tires with a uniform tread design were obtained. The final selection of actual sizes was made in conformity with the recommendations of the Tire and Rim Association, for truck and bus equipment, in effect at the time of purchase and are described as follows:

Solid—34 by 4 inch, low profile, solid center, capacity 1,900 pounds.

Cushion—36 by 5 inch, high profile, hollow center, capacity 1,900 pounds.

Pneumatic—32 by 6 inch, 10 ply, high pressure, capacity 1,900 pounds at 78 pounds per square inch inflation pressure.

Balloon—7.50/20 inch, 8 ply, low pressure, capacity 1,900 pounds at 49 pounds per square inch inflation pressure.

The general load deformation characteristics of the tires were determined by sustained load tests in which the radial deflections were measured and areas of contact on a plane surface were obtained. A smooth envelope drawn around the tire imprint determined the length, width, and gross area of contact. The results of these tests are shown in figure 4. The tires were put in a "cyclic state" by repeated loadings before obtaining the test values for this figure. The inflation pressures for the two pneumatic tires were measured with the tires free from applied load.

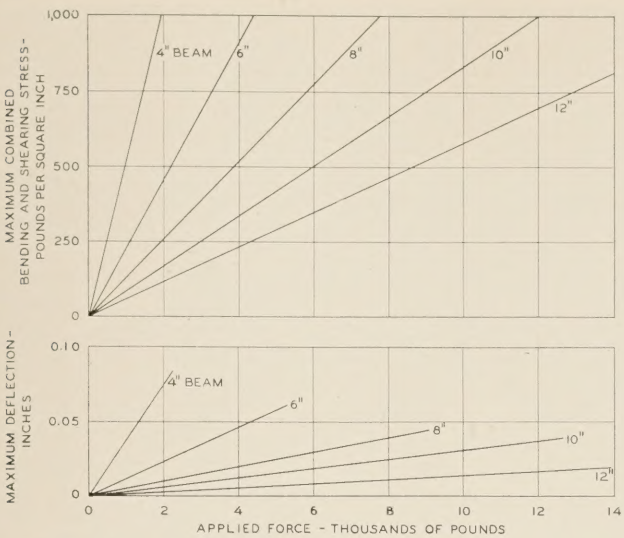


FIGURE 3.—MAXIMUM STRESSES AND DEFLECTIONS OF CONCRETE CANTILEVER TEST SPECIMENS.

In the case of the impact test by this method the effective weight of the pendulum[5] corresponds to the unsprung component of the static wheel load of a vehicle. Provided the effective pendulum weight is not unduly out of proportion to the unsprung weight ordinarily encountered in vehicles equipped with tires of 1,900 pounds capacity, it is not necessary to restrict the pendulum mass to a particular value, so long as this mass is sufficient to develop the desired maximum force with a reasonable pendulum length and arc of swing. These requirements led to an early tentative selection of an effective weight which was of the order of one third of the tire capacity.

Before proceeding further with the practical aspects of the design it is desirable to review briefly the elementary theory of the pendulum, as it has a bearing upon much of the discussion that follows.

MATHEMATICAL RELATIONSHIPS FOR COMPOUND PENDULUM OUTLINED

If a heavy mass could be concentrated at a point and suspended by a perfectly flexible and weightless cord it would constitute what is known as a simple pendulum. When such a pendulum oscillates or swings in a vertical plane it behaves according to definite physical laws. The velocity, acceleration, kinetic energy, and other properties, may be exactly determined. For example, the period or time of oscillation of the simple pendulum is known to depend upon (1) the length of the pendulum, (2) the force of gravity, and (3) the angle of swing according to the following relation:

$$T=2\pi\sqrt{\frac{l}{g}\,\frac{\theta}{\sin\theta}}$$

in which

T=the time required for one complete oscillation, in seconds.
l=the length of the pendulum.
g=the acceleration of gravity.
θ=the angular displacement from the vertical, in radians.

It is physically impossible to construct a simple pendulum. Frequently an actual pendulum is built up of a number of parts, rigidly connected. Each particle of this composite whole, being at a fixed distance from the axis of support, tends to oscillate as a simple pendulum in its own natural period. The rigid connection with other particles prevents this, and all of the particles that constitute the composite structure are forced to oscillate in a common period. Some of the particles will oscillate more rapidly, others more slowly than their natural periods would demand. However, there will be some whose natural period will be the same as that of the composite mass and the distance from the axis of support to these particles will be the length of the equivalent simple pendulum, that is, the hypothetical pendulum having the same period of oscillation.

The point at which the particles in the rigid or compound pendulum, as it is called, are oscillating in their own natural period is known as the center of oscillation because it is interchangeable with the center of suspension without affecting the period of the pendulum. An interesting and important property of the pendulum is that if it is struck a blow at the center of oscillation, normal to the pendulum and in the plane of oscillation, rotation will be produced about the axis of suspension but the blow will create no force reaction at this axis. For this reason the center of oscillation is often referred to as the center of percussion.

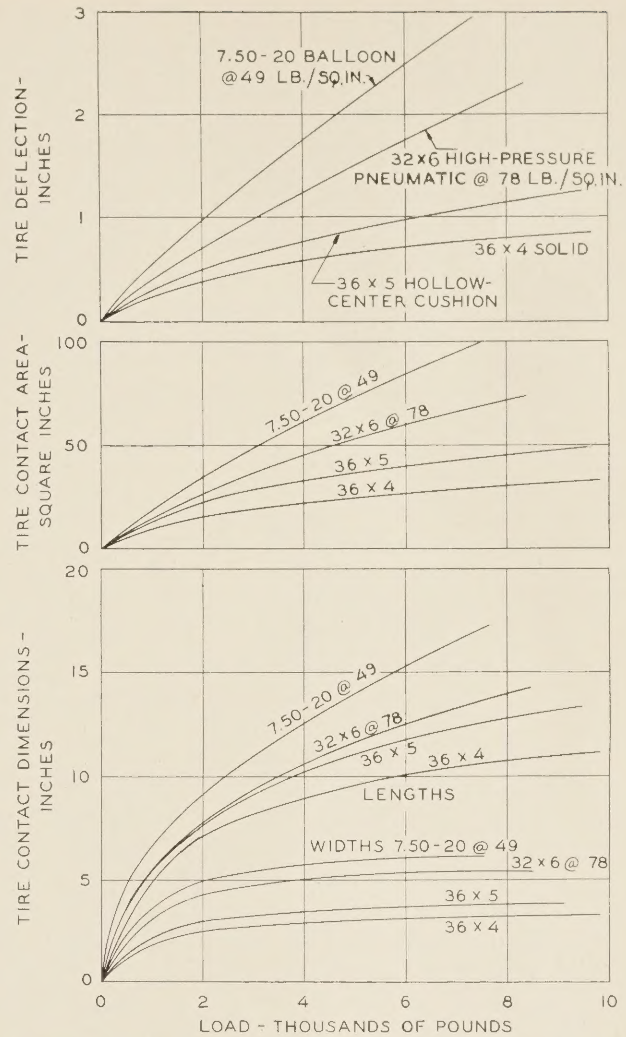
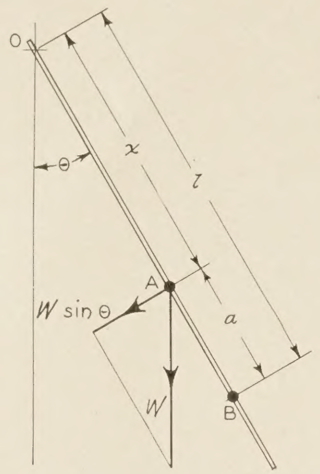


FIGURE 4.—CHARACTERISTICS OF TIRES UNDER SUSTAINED LOADS.

[5] By effective weight is meant that part of the total pendulum weight which is used in calculating the mass effective in developing force against the specimens during impact.

Because a compound pendulum does not conform to the defined requirements of a simple pendulum, its motion cannot be predicted directly from the laws of the simple pendulum. These laws must be modified to satisfy both the laws of an oscillating body and those of a rotating body. As such, they apply to the equivalent simple pendulum. It is with a compound pendulum that the present discussion is concerned and the principles of its motion are those to be considered.



O - CENTER OF SUSPENSION
A - CENTER OF GRAVITY
B - CENTER OF OSCILLATION OR PERCUSSION
x AND a - DISTANCES FROM THE CENTER OF GRAVITY TO "O" AND "B", RESPECTIVELY
l - LENGTH OF EQUIVALENT SIMPLE PENDULUM
W - WEIGHT OF PENDULUM
θ - ANGLE OF DISPLACEMENT FROM THE VERTICAL

FIGURE 5.—DIAGRAM OF THE COMPOUND PENDULUM.

Referring to figure 5, which shows in diagrammatic form the essentials of a compound pendulum, it is apparent that the position of the center of gravity, A, can be determined by taking the moments of the various individual masses which go to make up the total weight, W, about the center of suspension, O, and dividing their summation by this weight

$$x=\frac{\Sigma M_0}{\Sigma W}$$

The moment of inertia of the pendulum about O, ΣI_0, is determined in a similar manner by taking the moments of inertia of the individual masses about O, and adding them.

The radius of gyration, k_0, may be determined from the moment of inertia and weight of the pendulum by dividing, as follows:

$$k_0=\sqrt{\frac{\Sigma I_0}{\Sigma W}}$$

The distance, l, from the center of suspension, O, to the center of oscillation (or percussion), B, may then be determined by the following relation

$$l=\frac{k_0^2}{x}$$

This, it will be remembered, is the length of the equivalent simple pendulum, a pendulum having the same natural period as the compound pendulum. This period, as previously stated, is determined by the formula

$$T=2\pi\sqrt{\frac{l}{g}\frac{\theta}{\sin\theta}}$$

In connection with the use of this formula the value of the term $\frac{\theta}{\sin\theta}$ is of interest. It is shown in table 1 for several angles within the range under consideration.

TABLE 1.—*Values of* $\frac{\theta}{\sin\theta}$ *for several angles*

Value of angle θ: Degrees	Value of angle θ: Radians	Sin θ	$\frac{\theta}{\sin\theta}$	$\sqrt{\frac{\theta}{\sin\theta}}$
0° 30′	0.008727	0.008727	1.0000	1.0000
1°	.017453	.017452	1.0001	1.0000
2°	.034907	.034900	1.0002	1.0001
3°	.052360	.052336	1.0005	1.0002
10°	.174533	.173648	1.0051	1.0025
20°	.349066	.342020	1.0206	1.0102
30°	.523599	.500000	1.0472	1.0233

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These values indicate that in computing the period of oscillation this term may be omitted, if the angle of swing is small, for all but the most precise determinations.

If the sum of the moments about O is divided by the length, l, the resulting value is the weight to be used in computing the mass effective at the center of percussion during impact. This effective weight, converted to a mass value, and multiplied by the deceleration [6] developed when the motion of the pendulum is reversed, gives the dynamic force of the reaction.

The angular velocity, ω, of the oscillating pendulum may be obtained from its period of oscillation by the expression

$$\omega=\frac{2\pi}{T}=\sqrt{\frac{g}{l}\frac{\sin\theta}{\theta}}$$

The maximum tangential velocity, v_t, of the center of percussion (at the bottom of the swing) is

$$v_t=\omega\theta l$$

the value of θ being expressed in radians.

The angular acceleration, α, similarly stated, is

$$\alpha=-\frac{gx\sin\theta}{k_0^2}\text{ or }-\frac{g\sin\theta}{l}$$

the tangential acceleration at the center of percussion a_t, being

$$a_t=-g\sin\theta.$$

Having decided upon the type and sizes of the test specimens, upon the types and sizes of the tires, and upon the general order of mass required in the pendulum, there still remained other and more detailed requirements to be considered in the design, for example:

The center of percussion of the pendulum and the center of the area of force application on the specimen

[6] Acceleration is defined as the second derivative of the displacement-time relation, or as the time rate of change of velocity. As such it must be either positive or negative. Deceleration is a term frequently employed when referring to a negative or retarding acceleration. The instruments used for measuring acceleration, either positive or negative, are called accelerometers.

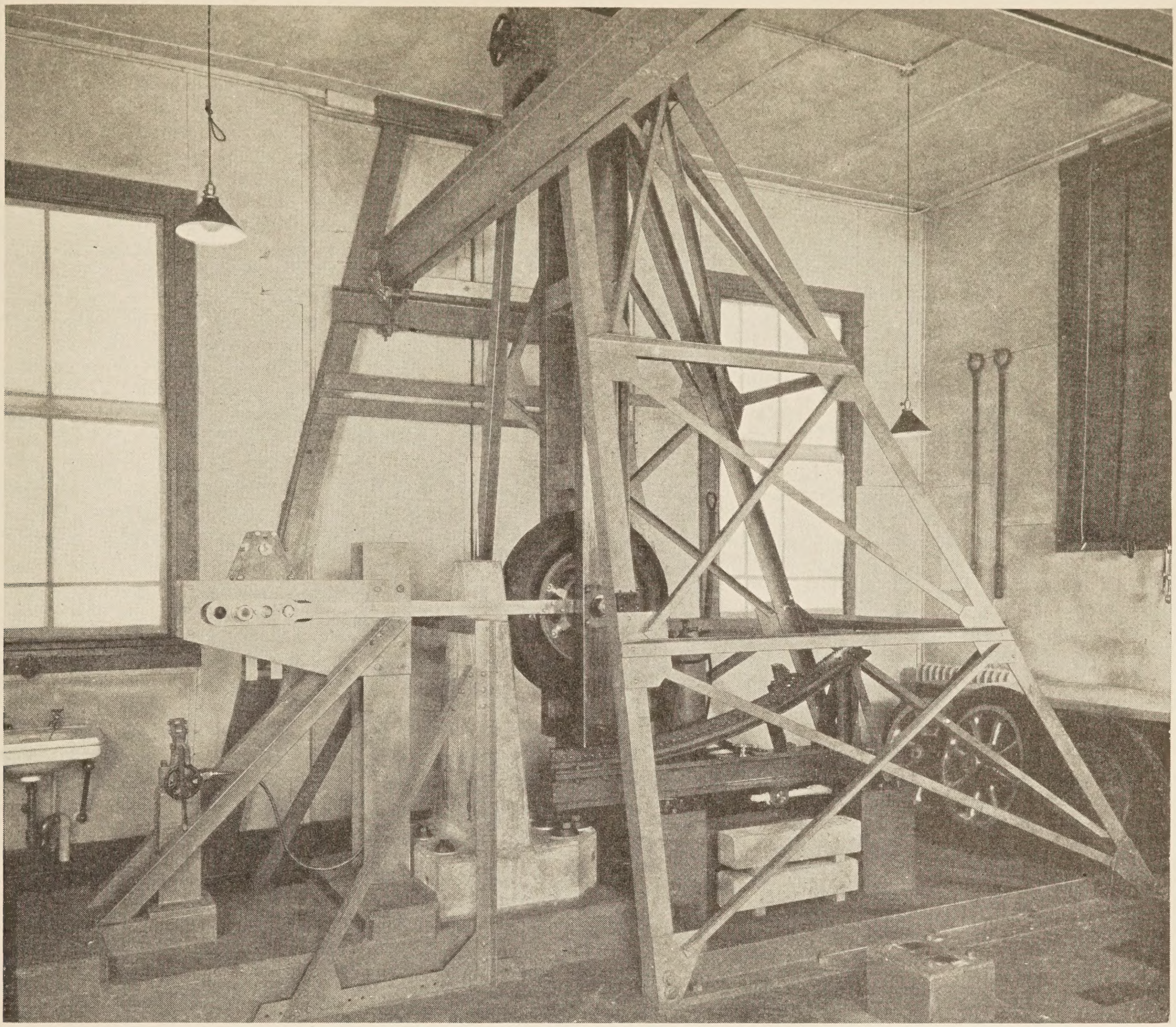


Figure 6.—General View of the Complete Testing Machine.

should coincide. The pendulum length should be practicable yet sufficient to develop the desired forces with the mass available within a reasonable arc of swing. The maximum arc was arbitrarily set at 30°. Sufficient stiffness should be provided to minimize bending and torsional deformation in the pendulum proper.

The center of suspension should be so designed that the pendulum would swing true yet with the minimum of frictional damping. It should be rigidly supported yet adjustably mounted.

Acceleration of the pendulum mass at the center of percussion should be readily measurable.

Means should be provided for easily applying and accurately measuring sustained forces on the test specimens. Conversion from one method of force application to the other should be possible in a minimum of elapsed time.

Wheel and tire assemblies should be readily interchangeable, yet should be securely mounted in the pendulum. The range in tire diameters likely to be used would be 30 to 42 inches and the maximum sectional width of tire about 10 inches.

A positive pendulum release mechanism is necessary to permit exact duplication in the dynamic tests. An automatic rebound catch should be provided to permit single applications of dynamic force. Safeguards are necessary to prevent the accidental release of the pendulum.

The concrete cantilever beams should be conveniently placed and securely mounted in the testing apparatus. Access to all parts of the test specimen must be provided for strain and deflection measurements.

THE TESTING MACHINE DESCRIBED

The testing machine that was designed to meet the various requirements that have been described is shown in figure 6.

The pendulum element proper consists of two 10-inch aluminum alloy channel members arranged back to back and separated 11½ inches by four box-type stiffening diaphragms. The striking wheel is mounted at the center of percussion. Each of the four wheels is fitted with a special hub designed to receive steel centering cones through which passes a 2-inch diameter

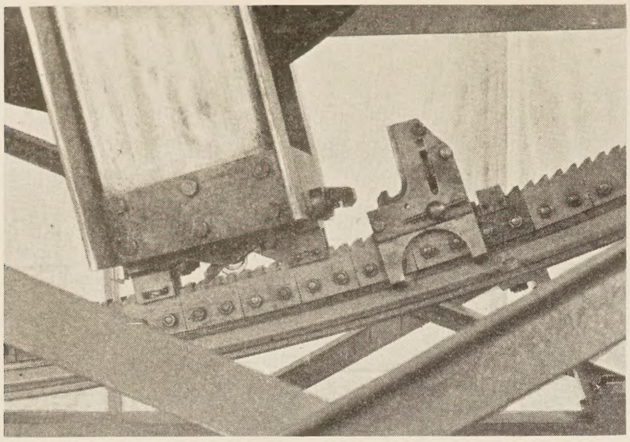


Figure 7.—The Pendulum Release Mechanism. Note the Rebound Catch Engaged With the Rack, Under the Center of the Pendulum.

steel shaft. Nuts on the ends of this shaft clamp the wheel assembly securely to the channel members.

At the upper end of the pendulum element is the axis of suspension or pivot. This consists of a steel shaft 1½ inches in diameter, which oscillates in two annular ball bearings.

The aluminum channel members are reinforced with steel plates at the points where the wheel and pivot shafts pass through them. One of these plates at each side of the wheel mounting is extended to serve as a support for one of the two accelerometers used in determining the impact forces. The length of the pendulum between the center of the pivot shaft and that of the striking wheel shaft is 100.00 inches.

The pivot bearings are carried in a heavy casting which is supported at the top of a pyramidal, structural steel frame or tower. The mounting permits adjustment of the pendulum to swing in a vertical plane properly oriented with respect to the test specimen and also permits a horizontal adjustment of the pivot in this plane of more than 6 inches. This traversing of the pivot is accomplished by the hand-wheel near the top of the tower (fig. 6) and the adjustment is made to compensate for differences in the radii of the several tires. After adjustment, the pivot casting is rigidly clamped to the tower by means of heavy bolts.

A rebound catch is mounted at the center of the lower end of the pendulum element. It consists of a pawl or detent mechanism which is automatically released during the impact. During rebound this pawl slides over the teeth of the large curved rack arranged along the arc of swing (see fig. 6) and engages at the end of swing, thus preventing a second or rebound blow from being struck. The large rack is independently supported on a heavy concrete pedestal and, since its position with respect to the axis of suspension must always be the same, it also is fitted with adjustments that allow this relation to be maintained whenever the pendulum pivot is moved.

In order to release the pendulum and allow it to swing through any desired angle (up to the 30° limit previously mentioned), a trip mechanism was designed that would free it suddenly, positively, and without imparting any tangential force. This mechanism, shown in figure 7, is adjustably mounted on the rack and can be locked at any desired point. A hook or catch on the back of the pendulum engages a similar element in the release device until freed by operation of the trigger lever. A pin lock prevents accidental operation of the release.

In order to make the wheel and tire assemblies dynamically interchangeable, the weights of these assemblies were adjusted to a common value by filling portions of the hollow steel wheels with lead. This additional weight was so distributed as to produce the least possible effect on the moment of inertia. The striking mass of the pendulum is thus the same regardless of the wheel and tire assembly used. The unavoidable variation in the moment of inertia of the wheel and tire assemblies produces a variation in the location of the center of percussion within the limits of 0.2 inch above to 0.3 inch below the center of the area of impact force application. The horizontal forces developed at the axis of suspension by this lack of complete coincidence are so small as to be without significance.

Each component part of the pendulum assembly when completed was carefully measured and weighed by two observers. From these data, the distance from the axis of suspension to the center of gravity of the part, the mass moment, and moment of inertia of the part about this axis, were computed. The observed weights, measured distances, and computed moments for all of the individual elements of the pendulum assembly are shown in table 2.

Table 2.—*Properties of the pendulum elements*

Element	Weight	Distance from center of gravity of element to axis of suspension	Moment of element about axis of suspension	Moment of inertia of element about axis of suspension
	Pounds	*Inches*	*Pound-inches*	*Pound-inches*[2]
Aluminum side channels	186.43	61.29	11,426	956,596
Pivot shaft, nuts, washers	15.40	.00	0	8
Pivot plates, bolts, etc	37.30	.13	5	397
Upper aluminum diaphragm, bolts, etc	15.12	8.00	121	1,179
Intermediate aluminum diaphragm, bolts, etc	14.94	42.00	627	26,563
Lower aluminum diaphragm, bolts, etc	15.03	76.00	1,142	87,023
Bottom cast-iron diaphragm, plates, screws	62.13	123.89	7,697	954,157
Wheel shaft plates, bolts, etc	31.60	100.00	3,160	316,429
Wheel shaft, cones, washers, nuts	28.22	100.00	2,822	282,235
Wheel and tire assembly	310.59	100.00	31,059	[1] 3,149,034
Accelerometers, bolts, etc	7.29	99.39	726	72,347
Accelerometer wires, binding posts	1.20	57.83	69	5,504
Rebound catch and trip assemblies	10.94	125.49	1,373	172,416
Summation—by first observer	736.19		60,227	6,023,888
Check—by second observer	736.24		60,226	6,023,307
Average values	736.22		60,226	6,023,598

[1] This is an average value for the 4 tire and wheel assemblies, which had values as follows: Balloon—3,136,683; high pressure—3,136,011; cushion—3,165,487; solid—3,157,954. This causes a variation in the location of the center of percussion of about −0.2 to +0.3 inch.

FORMULAS GIVEN FOR COMPUTING IMPACT FORCES

From the data in this table, the physical properties of the completed rigid pendulum were determined by means of the theoretical relations that were described earlier in the paper, as follows:

$\Sigma W = 736.22$ pounds, the total pendulum weight, acting at the center of gravity (point A, fig. 5).

$\Sigma M_0 = 60{,}226$ pound-inches, the summation of moments of all of the pendulum parts about the axis of suspension (point O, fig. 5).

$\Sigma I_0 = 6{,}023{,}598$ pound-inches,[2] the summation of moments of inertia of all of the pendulum parts about the axis of suspension.

k_0=90.453 inches, the radius of gyration of the composite pendulum about the axis of suspension.

$x=\frac{\Sigma M_0}{\Sigma W}$=81.804 inches, the distance of the center of gravity from the axis of suspension.

$l=\frac{k_0^2}{x}=\frac{\Sigma I_0}{\Sigma M_0}$=100.015 inches, the distance of the center of oscillation (or percussion) from the axis of suspension, also the length of the equivalent simple pendulum. (The designed length of l was 100.00 inches.)

$w=\frac{\Sigma M_0}{l}$=602.26 pounds, the pendulum weight effective at the center of percussion during impact.

g=32.155 feet per second per second, gravitational acceleration at the site of the pendulum.[7]

As previously stated the design of the pendulum is directly related to that of the test specimen because it must be capable of developing the forces required for the test. Mention has also been made of the fact that this force is dependent not only on the mass of the pendulum but on the acceleration or the time rate at which the velocity is reduced during impact, as well. In order to design successfully it was necessary to consider the complete interrelationship of the specimen, pendulum, and tire characteristics. The manner in which this was done will be described.

The concrete cantilever specimens are so placed that when the pendulum hangs freely at rest, the tire just touches the face of the specimen. The falling pendulum will therefore attain its maximum tangential velocity at the instant that the tire first makes contact with the specimen.

For the purpose of estimating probable acceleration maxima during impact, for use in the design, it was assumed that: (1) The combined cushioning of the tire and the specimen deflections will decelerate the pendulum in accordance with the laws of simple harmonic motion, (2) the influence of gravity on the deceleration of the pendulum is negligible, and (3) the center of percussion of the pendulum coincides with the center of the area of contact between the tire and the specimen. It is realized that none of these assumptions is rigorously true. However, each is sufficiently near the true condition to permit equating the maximum tangential velocity of the pendulum with the maximum linear velocity for simple harmonic motion to determine the probable maximum deceleration values for various tire types, angles of swing, and other conditions of test, with an accuracy that is sufficient for designing.

The physical relations involved during impact are:

d=the combined deflection of the tire and specimen or distance in which the pendulum is brought to zero velocity.

$v=\omega\theta l$, the maximum tangential velocity of the pendulum mass.

$A=\frac{v^2}{d}$, the maximum deceleration of the pendulum mass. (The velocity of the falling pendulum and that for the simple harmonic motion are assumed to be equal at the instant of contact.)

$F=\frac{wA}{g}$, the maximum force developed against the specimen (w being the effective weight of the pendulum).

$\frac{t}{2}=\pi\sqrt{\frac{d}{A}}$, the time duration of the contact between the tire and the face of the test specimen.

As already noted in connection with table 1, the influence of the angle of swing, as expressed by the term $\frac{\theta}{\sin\theta}$, or its reciprocal, is negligible for small angles and not important for angles of moderate magnitude [8] except in calculations where great precision is desired. In computing the probable values of the maximum accelerations the value of this term was assumed to be unity, permitting desirable simplification in a number of instances.

VIBRATION AND DEFLECTION OF SUPPORTING FRAME INSIGNIFICANT

Because of the fact that during the period of tire and specimen deflection the pendulum mass is not moving exactly horizontally but in a rising arc, the effective force applied to the specimen is slightly less than the apparent force. This difference amounts to about 6 pounds per inch of tire deflection and applies to both the sustained and the impact forces.

By the method outlined, the relations between the angle of swing, tire and beam deflection, and maximum impact force, were calculated and these are shown graphically in figure 8. To this chart have been added the curves from figure 4 that show the static load deflection characteristics of the tires, and the curves from figure 3 that show the theoretical bending stress and deflection characteristics of the cantilever test specimens. Thus figure 8 presents a composite graph showing the general relationships that exist between the various factors that are involved in a test. This graph has proved to be quite useful in connection with actual testing operations as a guide for selecting the conditions necessary to produce certain desired results.

As a check on the correctness of the calculated values for mass and center of percussion, the location of the center of gravity and the period of free swing were determined empirically. This was done after the pendulum was substantially completed. At the time of the determination the computed distance between the axis of suspension and the center of gravity was 81.36 inches. The mean value of several experimental balancings gave a value of 81.33 inches, a difference of less than 0.04 percent. Subsequent slight changes in the pendulum changed the value to that given earlier in the paper (81.804 inches).

The period of swing was determined experimentally by means of stop-watch measurements of the time required for 50 cycles. The damping caused by air and pivot friction as reflected in the reduction in the length of arc for successive swings (measured horizontally) was observed also. The data obtained from each of these observations for series of swings of small amplitude are shown in figure 9. It is noted that the measured period varies with the angle of swing and that it is within about 0.005 second of the theoretical value. No attempt was made at a precise determination. The damping also varies with the angle of swing although within the range covered by the experiments the loss per cycle is quite small.

By measuring the maximum deceleration that occurs at the center of mass concentration when the mass of

[7] From the Smithsonian Physical Tables, 8th edition, p. 564.

[8] For an angle of 10° the error caused by assuming the value of this term to be unity amounts to less than ¼ percent.

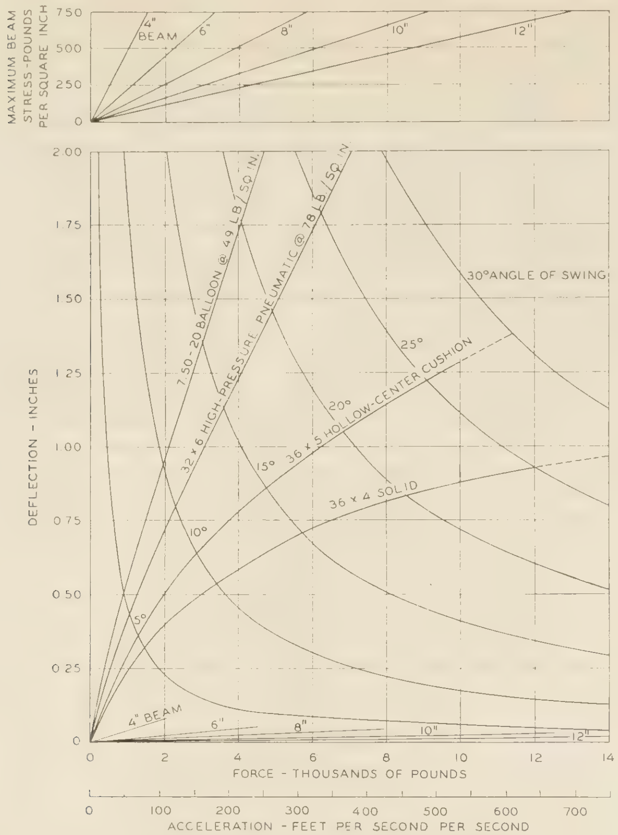


FIGURE 8.—COMPOSITE GRAPH SHOWING INTERRELATED CHARACTERISTICS OF TEST SPECIMENS, TIRES, AND PENDULUM.

the pendulum is brought to rest during the deflection of the tire and the specimen, the maximum dynamic force can be calculated. For measuring the maximum acceleration values, accelerometers were mounted on the pendulum in the manner shown in figure 10. The accelerometers are mounted one on each side opposite the center of impact. The instruments are of the single-element, electrically indicating, contact type described fully in earlier publications.[9]

The design and construction of the pendulum are such that there is essential coincidence of the center of percussion with the center of the area of dynamic-force application. It was mentioned earlier that when this condition obtains no force reaction is created at the center of suspension during impact. Since there is a negligible change in the forces acting at the center of suspension, vibration and deflection in the supporting frame are reduced to a minimum.

APPARATUS FOR APPLYING SUSTAINED LOADS DESCRIBED

Earlier in the paper mention was made of the development of sustained forces against the specimen by a pull exerted on the wheel shaft of the pendulum. The general appearance of the apparatus for this purpose can be seen at the left of the specimen in figure 6. Briefly, it consists of a suitably arranged hydraulic jack and pump for developing the force and an elastic system for measuring its magnitude.

A heavily reinforced concrete platform serves as a base for the sustained-load apparatus and as an anchorage for the test specimens. At a position in the rear

[9] See Calibration of Accelerometers for Use in Motor Truck Impact Tests, by J. A. Buchanan and G. P. St. Clair, PUBLIC ROADS, vol. 11, no. 5, July 1930; also, Impact Reactions Developed by a Modern Motor Bus, by J. A. Buchanan, PUBLIC ROADS, vol. 12, No. 2, April 1931.

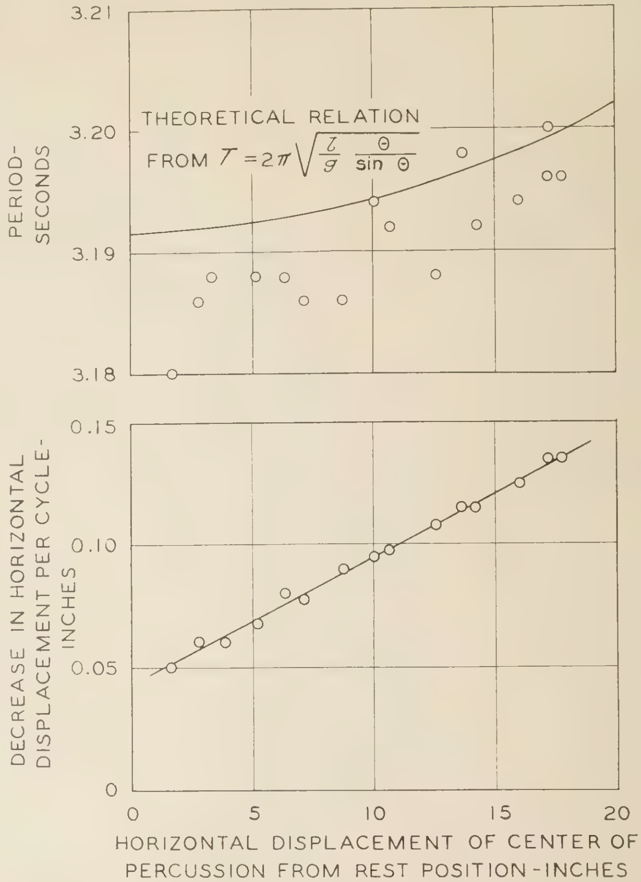


FIGURE 9.—FREE SWINGING NATURAL PERIOD AND DAMPING OF PENDULUM FOR VARIOUS DISPLACEMENT DISTANCES.

of the specimen, a post consisting of an 8-inch H-section is erected and rigidly braced to structural steel members encased in the concrete base. This post supports the sustained load apparatus and also carries a buffer to take the test specimen should it fail unexpectedly. A hydraulic jack of 5-inch diameter is secured to the post in such a manner that its axis lies in the plane of the pendulum and at the height of the wheel shaft. The line of travel of the jack ram is horizontal and its direction of motion is away from the test specimen.

The H-beam post supports also a pair of heavy wing plates, one on either side, which carry the load-measuring equipment. Horizontal slots or openings cut in these plates provide level runways for a carriage that rolls on annular-ball-bearing wheels. A sturdy transverse reaction pin or shaft receives the thrust of the jack through the load-measuring device, also supported by the carriage. Connection between the reaction pin and the wheel shaft of the pendulum is effected by a pair of steel tie bars, one of which can be seen in figure 6. A series of eyes at one end of each of these bars provide engagement at any one of four positions for the reaction pin while at the other end of the bars quickly detachable latches furnish a simple means for attachment to the wheel shaft through the conical blocks that center the wheel hub. The connection between the tie bars and the reaction pin is made through adjustable eccentric bushings which permit exact parallelism to be obtained between the reaction pin and the wheel shaft of the pendulum.

The load-measuring device consists of a pair of opposed, simply supported, vertically mounted steel

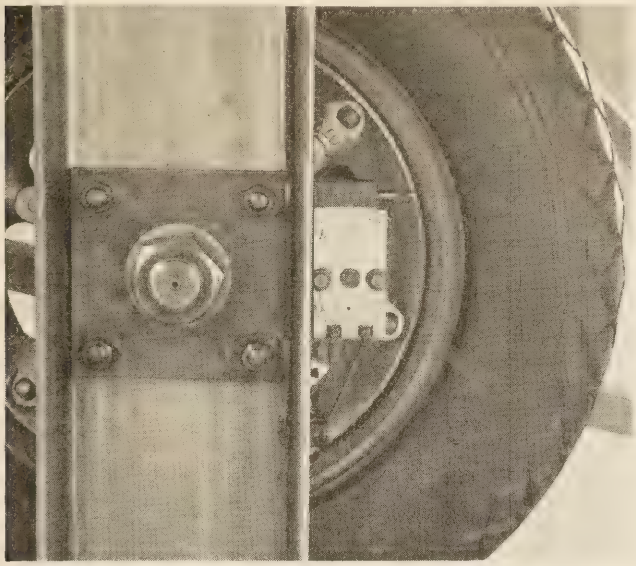


FIGURE 10.—CONTACT ACCELEROMETER MOUNTED ON CHANNEL MEMBER DIRECTLY OPPOSITE CENTER OF PERCUSSION.

beams of heat-treated alloy steel, whose combined deflections, under a centrally-applied load is measured with a dial micrometer as an index of the load or force. At each end of the inner faces of the beams there is a common reaction bearing in the form of a ground pin or roller in suitable seating blocks. At the upper end of the beams as they are mounted, the bearing pin is extended in both directions and these extended ends provide centers for a pair of annular ball bearings. These bearings roll on short horizontal tracks provided in the carriage which bears the reaction pin. This suspension permits the pair of beams to hang freely and vertically between the jack and the reaction pin. The details of the suspension are shown in figure 11.

The head of the hydraulic jack is fitted with a bronze face 3.8 inches square for bearing on one of the beams. The reaction pin which bears on the opposed beam is of heat-treated alloy steel ground to a diameter of 2 inches. Each beam is of alloy steel, heat treated and ground to a true rectangular prism 3.8 inches wide and 1.8 inches deep. The distance between reactions (or span) is 20 inches.

The combined deflection of the two beams is measured at the midpoint with a dial micrometer which reads directly in ten-thousandths of an inch. One division on the dial of the micrometer corresponds to an applied load increment of about 15 pounds, this being determined by calibration in a testing machine. The safe load-carrying capacity of this particular pair of beams is ample for all requirements and their elastic behavior has been constant.

The cantilever test specimens are securely mounted on the reinforced concrete platform in the position shown in figure 6. Horizontal sliding is prevented by friction on the base and by the reaction developed at the low transverse abutment that is an integral part of the platform. Moments tending to overturn the specimen are resisted by 10 large anchor bolts that pass through the base of the specimen. Seven of these bolts, each 1½ inches in diameter, are on the side nearest the pendulum, while three bolts, each 1 inch in diameter, are on the side nearest the vertical post. The stress in these bolts is relatively small, their design being such that, for the conditions assumed,

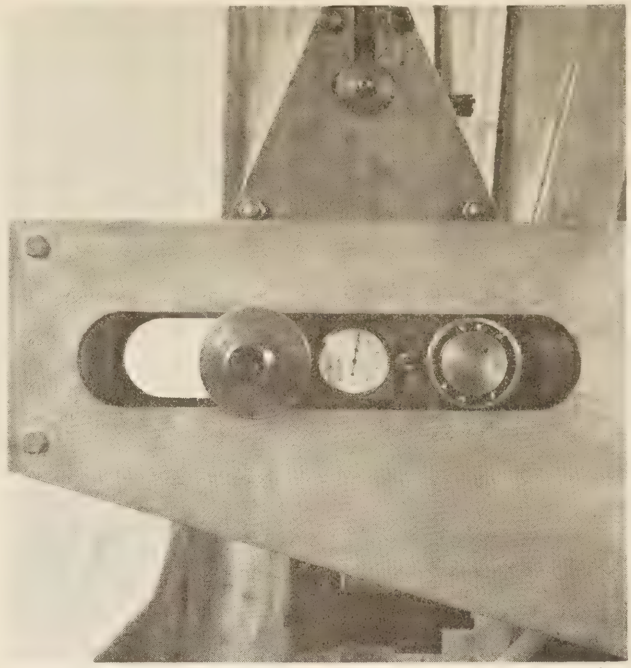


FIGURE 11.—DETAILS OF THE SUSTAINED LOAD DEVICE. SEEN THROUGH THE OPENING OF THE WING PLATE FROM RIGHT TO LEFT ARE RAM OF THE JACK, BALL-BEARING CARRIAGE WHEEL, DIAL MICROMETER BETWEEN THE PAIR OF BEAMS, AND THE END OF THE REACTION PIN. ABOVE IS THE CENTER OF SUSPENSION FOR THE LOAD-MEASURING BEAMS.

their maximum elongation would not cause an increase in the measured deflection of the specimen of more than about 5 percent. In the great majority of the tests the effect would be considerably less than this.

STRAINS AND DEFLECTIONS OF CONCRETE BEAMS MEASURED

An independently supported trolley and a geared hoist afford means for swinging the cantilever test beams into position on the platform anchorage. The beams are lifted by a sling attached to the ends of two steel bars which are passed through longitudinal holes in the specimen base (see cover illustration). The specimens are first placed on temporary wedges and carefully adjusted for elevation, alinement, perpendicularity, and orientation. The anchor bolts are then inserted but not tightened. The entire space between the platform and the base of the specimen (which rests on the wedge supports) and the space between the specimen base and the rear abutment is next filled with portland cement mortar placed with special tools and thoroughly compacted. After this bedding material has set sufficiently to bear the weight of the specimen without flowing, the wedges are removed. The anchor bolts are tightened after the mortar under the base has set thoroughly. The bolt heads bear on the top of the base through large cast iron washers set in mortar.

The elastic behavior of the test specimen is studied by comparing strains in the compression and tension faces of the cantilever and deflections under both sustained and suddenly applied forces of corresponding magnitudes.

The strains are measured with the recording strain gage developed by the Bureau and described in detail elsewhere.[10] In addition to the static calibration described in this reference, the behavior of the gages

[10] An Improved Recording Strain Gage, by L. W. Teller, PUBLIC ROADS, vol. 14, No. 10, December 1933.

FIGURE 12.—STRAIN GAGES ON THE COMPRESSION FACE OF THE TEST SPECIMEN. A CORRESPONDING PAIR ON THE OPPOSITE FACE MEASURES TENSILE STRAINS.

under suddenly applied strains has been investigated in several ways and for the rates of force development that are encountered in motor-vehicle impact there appears to be no significant difference between the behavior under displacements developed slowly and those developed suddenly.

Strain gages are placed at corresponding positions near the bottom of the compression and tension faces of the cantilever beam, but sufficiently above the base to avoid significant modification of stress due to the influence of the base. Strain gages located on the compression face are shown in figure 12.

The deflection of the specimen is usually measured as a horizontal displacement of a point on the compression face opposite the center of the area of force application. The apparatus consists of a light stylus arm extending from a plug set in the concrete of the specimen and arranged tc record the displacement directly on a smoked glass plate that is held in guides on an independent structural steel frame. This frame can be seen astride the concrete platform, just in back

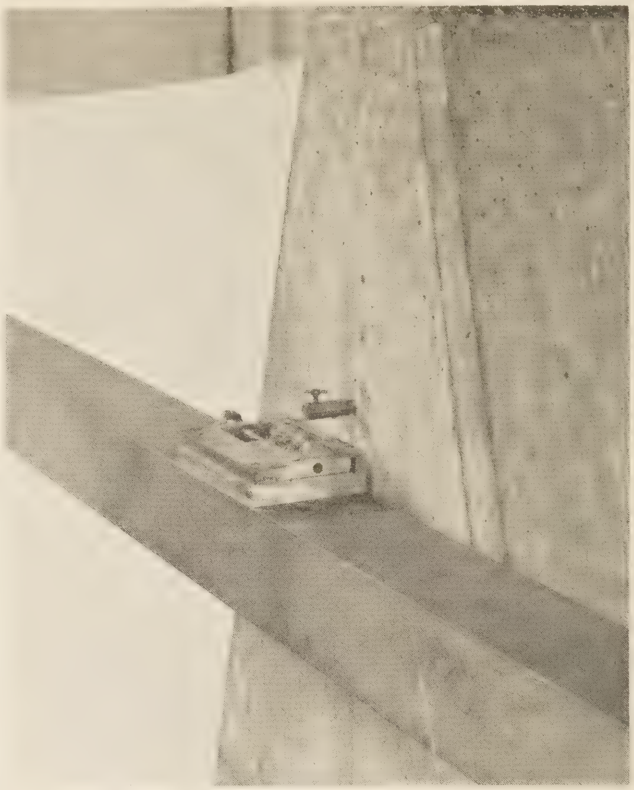


FIGURE 13.—BEAM DEFLECTION IS RECORDED ON A SMOKE-COATED GLASS PLATE BY MEANS OF A STYLUS.

of the specimen in figure 6, and the device itself is shown in figure 13.

The machine is erected in a building having a very heavy floor. This floor consists of a 12-inch reinforced concrete floor slab cast with heavy integral girders and supported on concrete walls at the edges and short columns at the third points in both directions. While no provision is made for maintaining a constant temperature in this laboratory, an effort is made to avoid widely fluctuating temperatures, particularly sudden changes.

The first group of specimens fabricated contained 26 units, the majority of which were of intermediate thickness. Of this group a considerable number have been tested and many thousands of load applications have been made. This experience has demonstrated that the testing equipment is satisfactorily producing the results for which it was designed.

DETERMINATION OF VARIATION IN UNIT PRESSURE OVER THE CONTACT AREA OF TIRES[a]

BY THE DIVISION OF TESTS, BUREAU OF PUBLIC ROADS

Reported by L. W. TELLER, Senior Engineer of Tests, and JAMES A. BUCHANAN, Associate Engineer of Tests

ONE of the lesser but persistently recurring questions that arise in considering the effects of wheel loads on pavement surfaces is that of the variation in pressure intensity over the area of contact between the tire and the pavement.

In the early attempts at stress analysis in pavement slabs it was assumed that the load was applied either at a point or along a line whose length equaled the width of the tire. Later, as pneumatic tires came into more widespread use, it was recognized that these assumptions were far from true. When Westergaard presented his theory of stresses in pavement slabs before the Highway Research Board in 1925,[1] he used the assumption that wheel loads are applied to pavement slabs through a circular area of contact over which the unit pressure is uniform.

This assumption had the advantage of simplicity; and, for purpose of analysis, it seemed to be not unreasonable, particularly as applied to pneumatic tires. It was known, of course, that the contact area was not actually circular, nor could the unit pressure over it be strictly uniform because of inherent characteristics in the form and construction of the tire. However, at that time there was little or no information that could be used as a basis for a better assumption. The shapes of the areas of contact have since become more generally known, but the manner in which unit pressure varies over such areas has remained relatively unknown.[2]

The reason for this scantiness of available data probably lies in the lack of a simple and satisfactory means for making reliable determinations.

In an effort to obtain additional information on this subject, the Bureau of Public Roads has recently developed the following method for determining these unit pressures—a method which requires but little addition to readily available equipment and which appears to give reasonably good results.

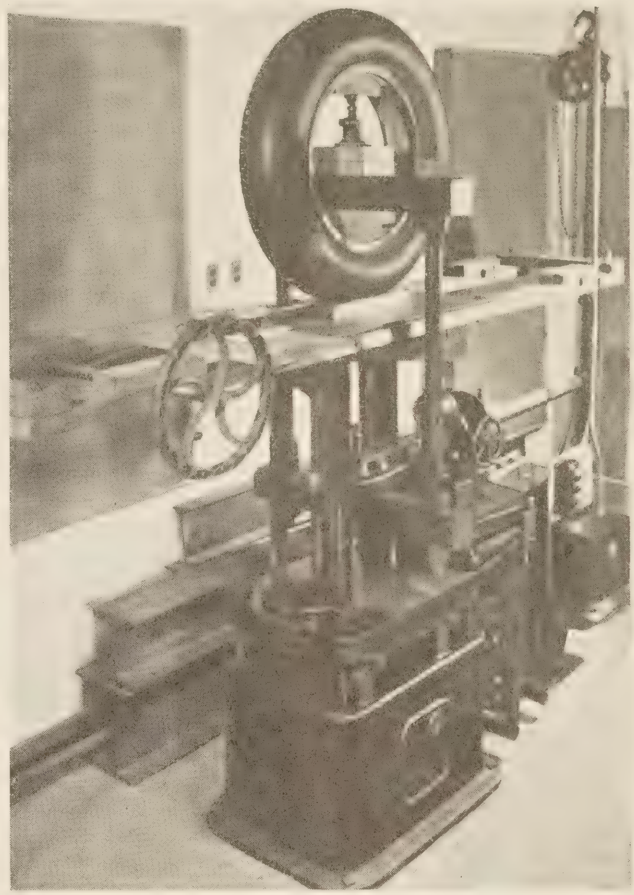


FIGURE 1.—TIRE MOUNTED IN A UNIVERSAL TESTING MACHINE EQUIPPED TO MAKE UNIT PRESSURE DETERMINATIONS.

TEST EQUIPMENT AND PROCEDURE DESCRIBED

The tire is mounted in a universal testing machine in the manner usually employed in load-deflection tests. A smoothly ground steel plate of adequate size is placed to receive the reaction. Between the tire and the reaction plate is placed a ⁵⁄₁₆-inch square bar of polished brass, arranged longitudinally along the area of contact. On either side of this bar are set steel filler plates of the same thickness as the bar and of sufficient size so that the bar and plates cover the entire area of contact. A windlass and spring dynamometer provide a means for applying and measuring the force required to overcome initial friction in sliding the bar longitudinally while load is applied to the tire. No lubricant is used, but a sheet of steel shim stock 0.0018 inch thick is placed between the tire and the friction bar to provide a uniform contact surface for sliding. The manner in which the equipment is set up in the testing machine is shown in figure 1 and the details of the friction bar, filler plates, and shim stock cover, are shown in figure 2.

In making the test the procedure is to place the friction bar successively in parallel positions ⁵⁄₁₆ inch apart until the entire area of contact has been covered. In each position the force required to move the bar longitudinally in each direction is measured and the two determinations are averaged. The test load is

[a] Paper presented at the Seventeenth Annual Meeting of the Highway Research Board, Washington, D. C., 1937.
[1] Computation of Stresses in Concrete Roads, by H. M. Westergaard. Proceedings, Fifth Annual Meeting, Highway Research Board, 1925.
[2] Glimpses of Balloon Tire Progress, by B. J. Lemon. Journal of the Society of Automotive Engineers, February 1925.
Distribution of Wheel Loads Through Various Rubber Tires, by Samuel Eckels. Proceedings, Eighth Annual Meeting, Highway Research Board, 1928.
Tests to Determine the Behavior of Tires on a Smooth Surface (Onderzoekingen Omtrent het Gedrag van Autobanden op een Effen Weg) Engineering Thesis, Technische Hoogeschool, Delft, by Hendrik Misset, January 20, 1932. (Particularly chapter 3—The distribution of pressure in the contact area between the tire and the road.)
Ground Contact Area of Tires Varies Directly with Deflection, by P. M. Heldt, Automotive Industries, July 23, 1932.
Semi-Pneumatic Tires and Specific Pressures (Semipneumatici e Pressioni Specifiche) by Raffaele Ariano. Le Strada (Italian), November 1933.

Figure 2.—Details of the Apparatus. The Friction Bar Lies Between Filler Plates of Equal Thickness and is Attached to the Spring Dynamometer. The Friction Surfaces, When the Tire is Under Load, Are Between the Bottom of the Friction Bar and the Lower Plate of Ground Steel, and Between the Top of the Friction Bar and the Steel Shim Placed Under the Tire.

assumed to be distributed to the respective elemental strips or sections thus obtained, in proportion to the ratio of the average frictional force for each section to the summation of the average frictional forces for all sections. In effect, this amounts to a first differentiation of the test load and is diagrammatically represented in figure 3, A.

To obtain the distribution along each of the longitudinal sections ⁵⁄₁₆ inch wide, the differences between the frictional resistances for successive increments of contact length on the sliding bar are determined, the remaining portion of the longitudinal strip being filled with a separate and stationary square bar. A convenient length increment is 1 inch and, as in the first differentiation, the average frictional resistance for both directions of movement is obtained.

The total load on each elemental strip is apportioned along its length according to the force differences for its respective subsections. By the double differentiation thus accomplished, the total load over the entire contact area is broken down into a great many small loads on elemental areas each ⁵⁄₁₆ inch wide and 1 inch long, except in the boundary areas where irregular shapes are unavoidable. These elemental areas are shown in figure 3, B.

Assuming the unit pressure at the center of gravity of each of the elemental areas to be the average intensity of pressure over that area, the unit values are computed and plotted in their respective positions on an outline of the actual gross contact area for the load being used in the test. The construction of the

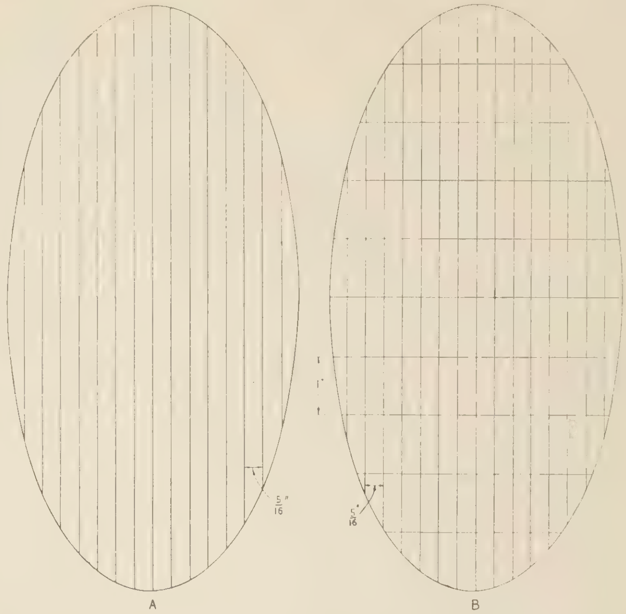


Figure 3.—Diagram Showing how Contact Areas Were Divided (A) Into Sections ⁵⁄₁₆ Inch Wide, and (B) Into Areas ⁵⁄₁₆ Inch by 1 Inch, for the First and Second Differentiations of the Test Load, Respectively.

Figure 4.—The Solid and Balloon Tires Used in Making Unit Pressure Determinations.

lines or contours of equal unit pressure within this area is then a simple process.

Two tires have been tested by this method—one a 36- by 6-inch solid truck tire, the other a 36- by 8-inch airplane pneumatic tire. Each was tested under a 4,000-pound load, corresponding approximately to the normal rated capacity. These two tires were selected for the initial tests for two reasons. First, they both had smooth treads, thus eliminating any question as to the effect of tread design and giving more basic data on the distribution of unit pressure. Second, they represent about as wide a range in cushioning quality as will be commonly found in tires of their capacity on the highway today.

The solid tire had become hardened somewhat with age, although it had been but slightly used. The pneumatic tire was mounted on a 36- by 8-inch truck tire rim and inflated to 60 pounds per square inch. They are shown in figure 4.

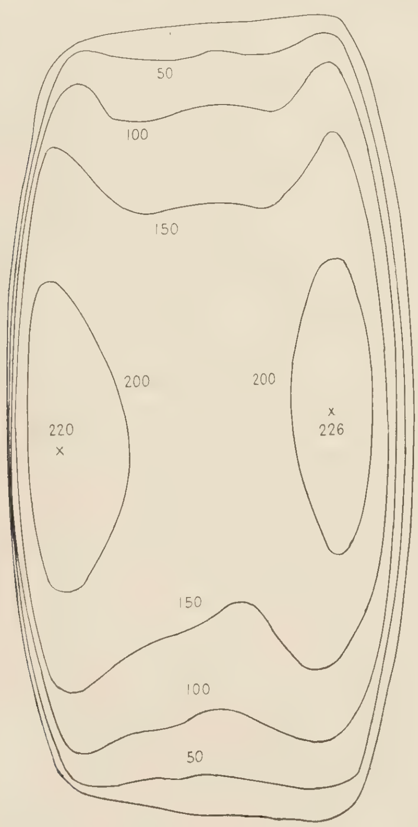
MAXIMUM INTENSITY OF PRESSURE UNDER SOLID TIRE THREE TIMES THAT UNDER PNEUMATIC TIRE

The unit pressure distributions over the contact areas of the two tires, obtained in the manner described, are shown in figures 5 and 6.

The intensity distribution data for the pneumatic tire show that over a considerable portion of the contact area the intensity of pressure is essentially uniform and not greatly different from the inflation pressure of the tire. For the solid tire, the zone of approximately constant unit pressure covers a much smaller percentage of the total area and the intensity within the zone is much higher than for the pneumatic tire. These relations are as might be expected from a consideration of the nature of the cushioning media involved.

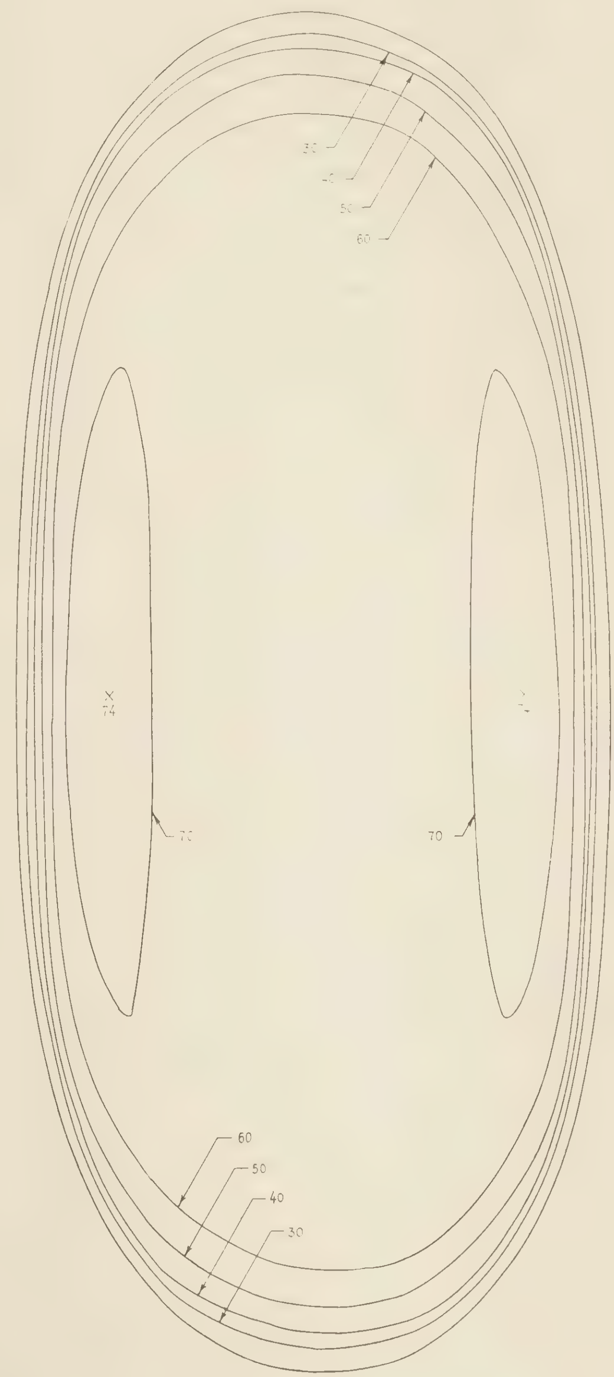
However, when consideration is given to the extreme difference in type represented by these two tires, the similarity between their unit pressure patterns is rather remarkable. The variations in unit pressure have the same general characteristics and the maximum values occur in two transversely symmetrical areas rather than at the geometric centers of the contact areas. The solid volumes defined by the contours of figures 5 and 6 represent the total loads supported by the tires. As a check on the work, these volumes were computed and converted into equivalent load. The results of these integrations are indicated on the figures.

The dimensions of the two contact areas and corresponding unit pressure values are given in table 1. In addition to the maximum measured intensities, values are also given for the corresponding uniformly loaded circular areas previously mentioned and for assumed ellipsoidal distributions in which the intensities of



TIRE SIZE 36 BY 6 INCHES
APPLIED LOAD____4,000 POUNDS
INTEGRATED LOAD___3,992 POUNDS

Figure 5.—Unit Pressure Distribution Over the Area of Contact for a Regular Solid Tire. Figures Show Pressure in Pounds per Square Inch.



TIRE SIZE 36 BY 8 INCHES
INFLATION PRESSURE__60 POUNDS PER SQUARE INCH
APPLIED LOAD_____4,000 POUNDS
INTEGRATED LOAD___3,914 POUNDS

Figure 6.—Unit Pressure Distribution Over the Area of Contact for a Low-Pressure Pneumatic Tire. Figures Show Pressure in Pounds per Square Inch.

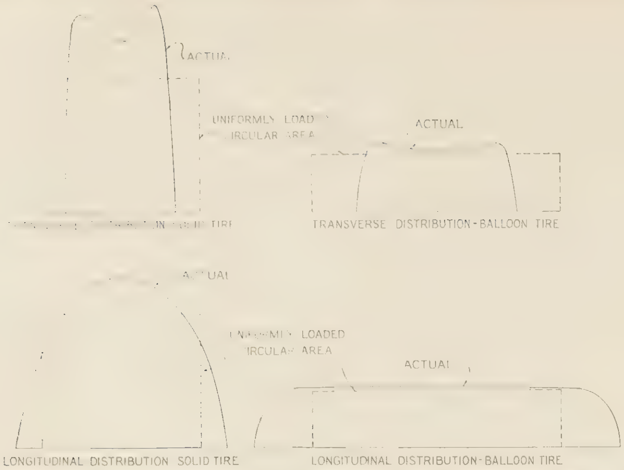


Figure 7.—Comparison of Actual Unit Pressure Distribution With That for a Uniformly Loaded Circular Area.

pressure are assumed to vary as the ordinates to semi-ellipsoidal surfaces placed over the areas of contact.

Table 1.—*Contact dimensions and pressure intensities for a load of 4,000 pounds on two types of tire equipment*

	Smooth-tread solid tire [1]	Smooth-tread pneumatic tire [2]
Contact area ... sq. in.	27.3	64.7
Contact length ... in.	7.8	13.3
Contact width ... do.	4.0	5.8
Diameter of equivalent circular area ... do.	5.82	9.05
Diameters of equivalent elliptical area ... do.	8.32—4.16	12.76—6.38
Average unit pressure over contact area ... lb./sq. in.	146	62
Maximum measured unit pressure ... do.	226	74
Maximum unit pressure for ellipsoidal distribution ... do.	220	93

[1] 36 by 6 inches.
[2] 36 by 8 inches, inflated to 60 pounds per square inch.

In figures 7 and 8 the actual transverse and longitudinal distributions are compared with those obtained by the uniformly loaded circular area method and by the suggested ellipsoidal distribution method. For the circular distribution, the area of the circle is equal to the actual area of contact; and the intensity, being the average pressure, does not vary. For the ellipsoidal distribution the area of the ellipse is equal to the actual area of contact; the diameters of the ellipse are in the ratio of 2 : 1; and the intensity varies over the area of contact in the manner described above. While it is obvious that neither of these conventionalized representations is an exact definition of actual conditions, it is evident from a comparison of figures 7 and 8 that the ellipsoidal distribution conforms much more closely to the actual than does the uniformly loaded circular area. This is also evident in the actual contour patterns of figures 5 and 6 and in the numerical data of table 1.

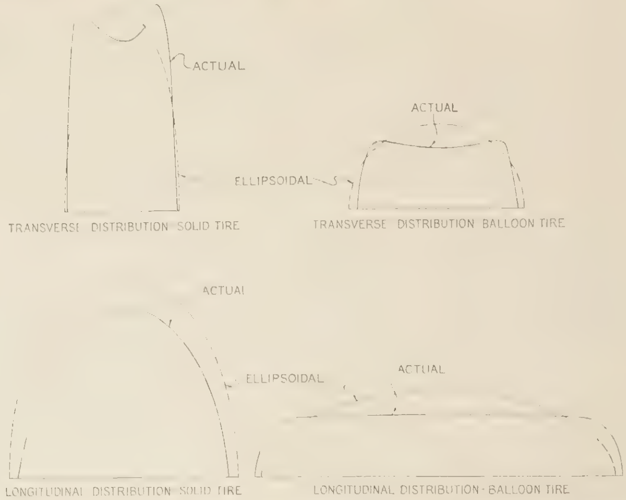


Figure 8.—Comparison of Actual Unit Pressure Distribution with Ellipsoidal Pressure Distribution.

The 2:1 ratio between the diameters of the ellipse is a simplification which was selected after a review of tire impressions obtained in nearly 400 tests of tires, tests that involved a wide variety of types, sizes, loads, and degrees of cushioning. This study showed an average ratio of contact length to contact width of approximately 2:1.[3]

The assumption that a vehicle tire transmits its load to the pavement through a uniformly loaded circular area is a generalization that may be of sufficient accuracy for most analytical purposes. The ellipsoidal distribution, however, is also capable of mathematical expression and, from the data obtained in these tests, appears to express more accurately the true condition. Whether or not differences in the manner of pressure distribution over the area of contact would lead to significant differences in pavement stress is a matter which may be studied either analytically or by physical tests. If such a study should reveal that variation in intensity of pressure is a matter of sufficient importance to be considered, then the ellipsoidal representation suggested by these tests offers a conveniently expressed generalization that reasonably approaches actual conditions of unit pressure distribution.

[3] For solid tires the average ratio was 1.59; for cushion tires, 1.95; for high-pressure pneumatic tires, 2.22; and for balloon tires, 2.12.

STATUS OF FEDERAL-AID HIGHWAY PROJECTS

AS OF NOVEMBER 30, 1937

STATE	COMPLETED DURING CURRENT FISCAL YEAR			UNDER CONSTRUCTION			APPROVED FOR CONSTRUCTION			BALANCE OF FUNDS AVAILABLE FOR NEW PROJECTS
	Estimated Total Cost	Federal Aid	Miles	Estimated Total Cost	Federal Aid	Miles	Estimated Total Cost	Federal Aid	Miles	
Alabama	$ 583,100	$ 281,550	19.6	$ 2,470,361	$ 1,235,180	127.5	$ 2,838,620	$ 1,419,305	129.5	$ 4,911,145
Arizona	1,112,079	813,048	47.7	1,594,385	1,134,923	72.5	223,721	160,767	10.3	1,208,220
Arkansas	1,983,112	1,981,278	120.8	2,188,072	2,183,586	148.9	23,505	23,177		2,210,878
California	5,498,435	2,946,065	113.8	5,991,087	3,221,670	112.7	2,992,449	1,581,385	55.4	1,166,154
Colorado	2,579,505	1,440,469	101.6	1,106,257	608,843	34.2	1,200,474	651,578	43.4	1,680,199
Connecticut	783,198	389,196	9.3	231,520	108,123	.8				1,513,589
Delaware	231,150	115,530	5.8	305,782	152,847	15.2	219,285	107,624	13.3	1,117,011
Florida	301,250	149,953	8.2	2,716,172	1,358,086	65.2	508,170	254,085	7.8	2,655,059
Georgia	1,827,545	906,734	108.9	4,133,172	2,066,581	179.8	3,719,426	1,859,713	159.9	3,838,348
Idaho	1,939,457	1,157,184	162.3	1,152,613	688,471	79.6	324,130	192,020	15.4	934,782
Illinois	8,049,929	3,966,266	216.3	6,435,061	3,213,423	200.8	4,328,700	2,051,850	86.3	1,742,718
Indiana	4,201,035	2,094,684	97.5	4,143,915	2,071,841	120.7	1,561,323	761,349	35.9	1,721,973
Iowa	6,363,111	2,885,898	206.5	4,319,091	1,918,113	123.4	2,313,087	1,074,650	68.2	208,627
Kansas	2,871,112	1,407,608	164.4	3,397,445	1,698,621	108.0	2,028,609	1,013,032	103.1	3,213,482
Kentucky	1,358,902	629,451	40.5	3,598,362	1,799,181	105.6	1,697,590	848,795	82.8	2,405,857
Louisiana	382,218	186,969	9.3	6,555,335	1,430,435	45.2	5,293,838	836,455	12.5	1,729,722
Maine	1,898,980	949,490	52.6	1,831,416	915,708	41.7	444,429	222,210	10.0	191,803
Maryland	184,006	92,003	2.1	2,371,418	1,183,926	37.0	212,000	106,000	3.2	1,712,879
Massachusetts	1,405,641	702,820	9.6	4,834,281	2,417,139	13.9	107,346	53,673		1,914,700
Michigan	5,373,810	2,666,746	142.4	5,765,110	2,883,055	140.2	2,122,309	986,329	30.4	185,756
Minnesota	5,277,952	2,627,429	242.7	3,116,681	1,411,211	133.2	2,074,604	1,037,302	83.8	1,330,548
Mississippi	1,242,210	621,100	59.3	4,464,490	2,231,430	206.0	2,339,440	1,209,020	94.9	2,571,626
Missouri	6,061,226	3,007,848	353.2	5,848,769	2,806,918	182.9	3,802,495	1,576,761	88.0	1,421,548
Montana	3,721,299	2,093,228	277.7	1,648,307	889,815	60.1	1,158,967	651,907	67.9	1,850,663
Nebraska	2,098,156	1,049,078	205.5	4,602,599	2,289,278	440.4	2,576,516	657,149	79.6	2,136,261
Nevada	1,600,322	1,376,084	83.8	1,240,211	1,072,530	62.1	194,286	168,479	27.0	752,301
New Hampshire	217,825	107,616	3.3	495,198	244,730	9.8	196,021	96,847	2.9	985,626
New Jersey	1,674,609	760,009	20.2	841,670	420,515	2.0	1,033,280	515,805	10.7	1,784,824
New Mexico	2,400,300	1,473,364	191.6	2,827,723	1,824,005	137.3	114,353	69,739	1.8	280,256
New York	9,785,560	4,596,809	185.1	16,454,138	7,867,325	258.0	2,360,540	1,178,770	31.0	398,519
North Carolina	2,329,608	1,161,503	180.5	5,870,974	2,764,937	284.5	1,491,140	709,000	61.5	2,469,315
North Dakota	1,012,110	1,012,110	186.5	1,192,830	1,172,320	77.4	294,572	294,572	36.4	3,243,231
Ohio	2,426,602	1,139,650	30.1	9,995,419	4,966,354	112.7	2,077,260	1,038,630	25.6	4,945,536
Oklahoma	2,507,319	1,315,152	103.1	2,655,498	1,373,237	121.7	1,510,965	794,868	66.3	3,194,210
Oregon	3,380,430	1,994,181	122.8	1,536,520	937,055	54.1	842,035	475,028	44.4	877,471
Pennsylvania	7,979,161	3,970,260	121.5	8,236,069	4,098,185	105.7	5,180,236	2,580,379	83.3	2,226,381
Rhode Island	551,258	266,814	5.7	1,350,904	675,452	15.4	8,930	4,465		757,990
South Carolina	2,304,729	965,203	188.8	4,792,449	1,991,505	211.5	1,405,994	535,535	74.9	1,364,461
South Dakota	2,414,306	1,367,392	242.3	1,638,946	906,260	173.5	360,716	199,460	48.1	3,074,150
Tennessee	1,039,736	518,939	47.7	1,648,644	824,322	45.0	1,508,780	754,390	47.0	4,445,866
Texas	8,827,775	4,409,623	639.9	10,160,260	5,062,609	458.6	3,325,551	1,571,242	198.8	5,301,352
Utah	686,534	493,723	55.5	1,332,090	949,451	121.9	97,770	69,230	7.9	1,077,280
Vermont	799,081	390,183	23.3	1,747,230	772,795	43.7	92,330	45,969	3.1	45,277
Virginia	2,054,457	1,027,228	107.3	2,824,737	1,356,968	69.9	1,186,836	584,968	36.4	1,960,360
Washington	1,660,432	855,474	72.3	2,718,731	1,424,512	39.3	1,135,160	597,200	5.5	612,992
West Virginia	804,265	406,733	23.1	1,673,601	835,894	43.9	400,792	369,031	14.3	2,081,497
Wisconsin	7,443,878	3,523,739	233.8	4,081,871	2,001,677	118.3	1,344,401	525,500	20.2	1,097,198
Wyoming	2,329,852	1,424,305	235.9	1,356,222	819,929	154.6	167,241	103,140	18.5	179,548
District of Columbia										
Hawaii	459,087	226,420	8.0	504,649	248,452	9.9	443,946	202,633	6.9	1,150,702
Puerto Rico				534,707	265,374	10.7	451,130	225,565	9.4	134,061
TOTALS	134,017,654	69,944,139	5,889.7	172,532,992	86,794,797	5,537.0	71,335,298	33,046,581	2,163.5	90,013,952

CURRENT STATUS OF UNITED STATES WORKS PROGRAM HIGHWAY PROJECTS

(AS PROVIDED BY THE EMERGENCY RELIEF APPROPRIATION ACT OF 1935)

AS OF NOVEMBER 30, 1937

STATE	APPORTIONMENT	COMPLETED			UNDER CONSTRUCTION			APPROVED FOR CONSTRUCTION			BALANCE OF FUNDS AVAILABLE FOR NEW PROJECTS
		Estimated Total Cost	Works Program Funds	Miles	Estimated Total Cost	Works Program Funds	Miles	Estimated Total Cost	Works Program Funds	Miles	
Alabama	$4,151,115	$3,944,757	$3,903,775	136.9	$250,100	$243,478	7.8				$3,862
Arizona	2,569,841	3,156,647	2,507,267	193.7	38,548	38,548					24,026
Arkansas	3,352,061	3,150,524	3,123,198	350.5	181,216	180,523	9.7				48,339
California	7,747,928	7,527,537	7,171,405	257.5	572,929	515,656	6.4				60,868
Colorado	3,395,263	2,385,907	2,294,163	99.4	89,597	89,596	6.0	$8,200	$8,200		1,003,304
Connecticut	1,418,709	1,180,464	1,102,094	17.9	232,031	206,330	4.5	124,130	64,435	0.2	45,850
Delaware	900,310	665,348	639,615	56.6	219,462	219,462	10.3	26,712	26,712	4.4	14,521
Florida	2,597,144	2,598,621	2,529,126	99.1	38,957	38,957					29,061
Georgia	4,988,967	1,819,038	1,777,667	98.4	2,450,184	2,181,804	116.5	442,460	442,460	29.6	587,036
Idaho	2,222,747	2,267,396	2,161,480	185.9	39,559	39,559					21,707
Illinois	8,694,009	8,028,820	7,786,908	456.2	781,010	781,010	32.0				126,091
Indiana	4,941,255	4,872,511	4,578,655	219.7	331,419	331,419	18.7				31,181
Iowa	4,991,664	5,223,196	4,879,904	528.3	110,198	107,765	.3	3,474	3,470		525
Kansas	4,994,975	4,552,153	4,482,802	356.3	419,276	417,103	34.2	15,620	15,620	1.6	79,449
Kentucky	3,726,271	3,465,086	3,307,265	349.8	394,471	394,471	8.7				24,535
Louisiana	2,890,429	2,666,658	2,373,902	165.1	428,080	367,449	2.6	97,087	97,070	10.4	52,008
Maine	1,676,799	1,579,111	1,566,938	72.6	112,460	109,861	3.5				
Maryland	1,750,738	637,876	631,089	23.4	581,095	581,095	11.6	193,134	159,302	6.4	379,253
Massachusetts	3,262,885	2,183,676	2,183,676	18.0	829,930	439,160	.4	1,136,726	568,363	.8	71,686
Michigan	6,301,414	6,520,746	5,948,097	288.6	290,686	289,571	3.3	44,500	38,000	.3	25,746
Minnesota	5,277,145	6,403,838	5,170,514	896.0	96,799	88,962	5.9				17,669
Mississippi	3,457,552	2,984,094	2,979,471	209.8	395,446	394,406	25.9	10,800	10,800	.6	72,874
Missouri	6,012,652	5,294,154	5,144,127	776.9	790,784	736,082	.7	34,390	32,294		100,149
Montana	3,676,416	3,597,216	3,536,508	200.7	95,385	95,385	.1	8,462	8,462		36,062
Nebraska	3,870,739	3,311,029	3,203,989	346.4	587,690	587,686	24.3	70,270	70,270	1.8	8,795
Nevada	2,243,074	2,291,627	2,196,238	110.1	84,970	38,146	1.7				8,690
New Hampshire	945,225	876,181	843,257	37.7	121,092	101,301	5.7				667
New Jersey	3,129,805	1,240,345	1,220,240	23.7	1,874,079	1,874,079	11.7	34,468	34,468		1,018
New Mexico	2,871,397	2,809,901	2,804,735	213.7	43,071	43,071		14,681	12,196		11,395
New York	11,046,377	10,762,211	10,305,153	170.0	349,700	309,700	2.0	69,000	69,000	.2	362,524
North Carolina	4,720,173	4,451,805	4,379,902	267.8	285,491	285,491	22.9	37,900	37,900	.5	16,881
North Dakota	2,867,245	2,493,139	2,452,479	378.9	107,799	107,799	1.2	282,834	282,834	36.0	24,134
Ohio	7,670,815	6,281,178	6,195,424	280.2	1,431,922	1,396,497	16.4	30,200	30,200	1.9	48,694
Oklahoma	4,580,670	4,477,024	4,268,653	396.4	293,970	293,970	12.0	8,800	5,000	.1	13,047
Oregon	3,038,642	3,216,548	2,950,598	164.6	45,580	45,580		11,846	11,846	1.0	30,618
Pennsylvania	9,347,797	5,969,960	5,727,602	235.1	3,521,971	3,075,081	40.2	497,593	388,504	10.6	156,609
Rhode Island	989,208	1,111,600	989,136	18.8							72
South Carolina	2,702,012	2,261,423	2,111,864	224.6	617,029	545,752	24.9	9,664	9,664	1.5	34,731
South Dakota	2,976,454	2,596,836	2,593,052	476.1	369,237	369,237	28.6				14,165
Tennessee	4,192,460	3,190,715	3,139,537	126.1	1,020,913	1,020,913	24.6	35,710	22,060	2.6	9,951
Texas	11,989,350	12,592,804	11,526,736	1,105.8	458,571	346,058	11.9	83,337	83,100	7.2	33,455
Utah	2,067,154	2,119,876	1,904,505	206.7	131,655	131,655	1.6				30,994
Vermont	924,306	1,061,113	915,284	23.2	8,000	8,000					1,022
Virginia	3,652,667	3,377,912	3,260,504	990.8	178,541	173,644	14.3				218,519
Washington	3,026,161	3,313,249	2,936,854	164.3	45,392	45,392		25,465	20,845	.7	23,070
West Virginia	2,231,412	1,970,965	1,876,155	81.0	380,572	322,457	14.5	32,800	32,800	5.0	
Wisconsin	4,823,884	5,240,385	4,727,518	343.4	94,268	93,900	.3				2,466
Wyoming	2,219,155	2,172,672	2,167,016	152.4	33,287	33,287					18,852
District of Columbia	949,496	950,000	949,496	8.8							
Hawaii	926,033	678,560	655,700	10.4	277,293	215,689	7.0	10,000	10,000		44,644
TOTALS	195,000,000	177,524,432	168,081,273	12,614.3	22,131,715	20,352,037	574.9	3,400,263	2,595,875	123.4	3,970,815

PUBLICATIONS of the BUREAU OF PUBLIC ROADS

Any of the following publications may be purchased from the Superintendent of Documents, Government Printing Office, Washington, D. C. As his office is not connected with the Department and as the Department does not sell publications, please send no remittance to the United States Department of Agriculture.

ANNUAL REPORTS

Report of the Chief of the Bureau of Public Roads, 1931. 10 cents.

Report of the Chief of the Bureau of Public Roads, 1933. 5 cents.

Report of the Chief of the Bureau of Public Roads, 1934. 10 cents.

Report of the Chief of the Bureau of Public Roads, 1935. 5 cents.

Report of the Chief of the Bureau of Public Roads, 1936. 10 cents.

Report of the Chief of the Bureau of Public Roads, 1937. 10 cents.

DEPARTMENT BULLETINS

No. 1279D. . Rural Highway Mileage, Income, and Expenditures, 1921 and 1922. 15 cents.

No. 1486D. . Highway Bridge Location. 15 cents.

TECHNICAL BULLETINS

No. 55T. . . Highway Bridge Surveys. 20 cents.

No. 265T. . . Electrical Equipment on Movable Bridges. 35 cents.

MISCELLANEOUS PUBLICATIONS

No. 76MP. . The Results of Physical Tests of Road-Building Rock. 25 cents.

No. 191MP. Roadside Improvement. 10 cents.

No. 272MP. Construction of Private Driveways. 10 cents.

No. 279MP. Bibliography on Highway Lighting. 5 cents.

The Taxation of Motor Vehicles in 1932. 35 cents.

Guides to Traffic Safety. 10 cents.

Federal Legislation and Rules and Regulations Relating to Highway Construction. 15 cents.

An Economic and Statistical Analysis of Highway-Construction Expenditures. 15 cents.

Highway Bond Calculations. 10 cents.

Single copies of the following publications may be obtained from the Bureau of Public Roads upon request. They cannot be purchased from the Superintendent of Documents.

SEPARATE REPRINT FROM THE YEARBOOK

No. 1036Y. . Road Work on Farm Outlets Needs Skill and Right Equipment.

TRANSPORTATION SURVEY REPORTS

Report of a Survey of Transportation on the State Highway System of Ohio (1927).

Report of a Survey of Transportation on the State Highways of Vermont (1927).

Report of a Survey of Transportation on the State Highways of New Hampshire (1927).

Report of a Plan of Highway Improvement in the Regional Area of Cleveland, Ohio (1928).

Report of a Survey of Transportation on the State Highways of Pennsylvania (1928).

Report of a Survey of Traffic on the Federal-Aid Highway Systems of Eleven Western States (1930).

UNIFORM VEHICLE CODE

Act I.—Uniform Motor Vehicle Administration, Registration, Certificate of Title, and Antitheft Act.

Act II.—Uniform Motor Vehicle Operators' and Chauffeurs' License Act.

Act III.—Uniform Motor Vehicle Civil Liability Act.

Act IV.—Uniform Motor Vehicle Safety Responsibility Act.

Act V.—Uniform Act Regulating Traffic on Highways.

Model Traffic Ordinances.

A complete list of the publications of the Bureau of Public Roads, classified according to subject and including the more important articles in Public Roads, may be obtained upon request addressed to the U. S. Bureau of Public Roads, Willard Building, Washington, D. C.

CURRENT STATUS OF UNITED STATES WORKS PROGRAM GRADE CROSSING PROJECTS

(AS PROVIDED BY THE EMERGENCY RELIEF APPROPRIATION ACT OF 1935)

AS OF NOVEMBER 30, 1937

STATE	APPORTIONMENT	COMPLETED		NUMBER			UNDER CONSTRUCTION		NUMBER			APPROVED FOR CONSTRUCTION		NUMBER			BALANCE OF FUNDS AVAILABLE FOR NEW PROJECTS
		Estimated Total Cost	Works Program Funds	Grade Crossings Eliminated by Separation or Relocation	Grade Crossing Structures Reconstructed	Grade Crossings Protected by Signals or Otherwise	Estimated Total Cost	Works Program Funds	Grade Crossings Eliminated by Separation or Relocation	Grade Crossing Structures Reconstructed	Grade Crossings Protected by Signals or Otherwise	Estimated Total Cost	Works Program Funds	Grade Crossings Eliminated by Separation or Relocation	Grade Crossing Structures Reconstructed	Grade Crossings Protected by Signals or Otherwise	
Alabama	$ 4,034,617	$ 3,557,534	$ 3,539,035	47	1	12	$ 378,519	$ 378,519	2			$ 175,300	$ 109,000	4			$ 8,063
Arizona	1,256,099	1,211,282	1,162,221	14		1	91,216	68,807	1		3	2,280	2,280			1	22,791
Arkansas	3,574,060	2,823,031	2,815,830	51	6	4	673,549	672,844	5		13	46,324	46,324			16	39,062
California	7,486,362	7,154,760	6,966,713	45	8		470,205	470,204	2			10,000	10,000	5			39,445
Colorado	2,631,567	1,907,899	1,828,075	23	1		821,664	791,096	7	2	21						12,397
Connecticut	1,712,684	297,379	297,379	2	1	1	1,341,801	1,320,440	8								94,865
Delaware	418,239	130,000	130,000	1			277,993	277,993	2								10,246
Florida	2,827,883	2,411,470	2,381,150	30	5	2	119,018	119,018	1			133,770	133,770			73	193,945
Georgia	4,895,949	599,219	597,715	14	5	5	1,386,449	1,386,449	24	7	6	776,040	776,040	11	2	41	2,135,745
Idaho	1,674,479	1,396,104	1,385,766	21	2	9	256,458	256,458	2		9	4,261	4,261			3	27,994
Illinois	10,307,184	8,808,994	8,687,750	65	8		1,516,234	1,509,830	8	2		127,000	78,000	2			31,604
Indiana	5,111,096	4,112,984	3,998,799	37	12		1,108,824	1,108,824	5		162						3,474
Iowa	5,600,679	4,379,206	4,285,042	95	9	8	1,300,569	1,294,781	11								20,856
Kansas	5,246,258	3,955,532	3,932,379	54		5	1,273,119	1,253,704	4	1		115,024	22,020	2		3	38,155
Kentucky	3,672,387	1,358,648	1,350,795	19	5		1,984,219	1,694,487	5	6		612,579	612,579	3	1		14,526
Louisiana	3,213,467	1,328,350	1,328,343	14	1		1,120,878	1,120,859	11	1		649,599	600,110	4	1		164,155
Maine	1,426,861	1,265,372	1,263,094	19	1	4	160,788	142,540	1	1							21,227
Maryland	2,061,751	501,739	501,739	3	3	16	850,670	850,670	6	2	4	476,862	404,393	2	1	10	304,949
Massachusetts	4,210,833	2,488,969	2,486,539	20	4		1,305,790	1,305,790	6			249,991	249,991	1			168,513
Michigan	6,765,197	6,877,565	6,579,490	44	8		101,477	101,477				43,500	43,500		1		40,730
Minnesota	5,395,441	4,776,166	4,677,484	81	14	50	709,458	703,408	5								14,550
Mississippi	3,241,475	2,040,404	2,040,235	45	4	6	690,038	690,038	10	3	3	78,300	78,300	2		14	432,902
Missouri	6,142,153	3,012,558	2,944,928	31	1		3,293,571	3,190,761	18		1	1,650	1,650			1	4,814
Montana	2,722,327	2,588,865	2,532,327	37	7		245,576	185,931	1	1							4,069
Nebraska	3,556,441	2,810,180	2,774,426	74	3	19	554,167	554,167	7		1	203,027	203,027	3		4	24,821
Nevada	887,260	886,182	857,417	8	3		13,308	13,308				3,630	3,630			5	12,905
New Hampshire	822,484	737,902	737,857	8	4	6	106,670	82,521	2			2,106	2,106			1	
New Jersey	3,983,826	3,058,834	3,046,907	20	5	1	831,570	831,570	4	4		44,730	44,730		1		60,619
New Mexico	1,725,286	1,644,472	1,637,584	18	1		82,384	82,384	1								5,318
New York	13,577,189	10,476,925	10,182,856	33	39		3,107,900	3,107,900	12	10		94,000	94,000				192,433
North Carolina	4,823,958	3,439,944	3,419,818	46	18		1,298,009	1,298,009	16		115						106,130
North Dakota	3,207,473	2,819,708	2,815,995	56	4		389,012	389,012		1							2,466
Ohio	8,439,897	2,110,287	1,990,344	15	5	2	5,875,528	5,407,381	32	3	2	866,660	851,660	12		10	190,512
Oklahoma	5,004,711	3,668,274	3,656,660	57	9	7	1,356,324	1,223,224	8	3	10	75,190	75,190			24	49,637
Oregon	2,334,204	2,329,684	2,242,111	16	6	2	74,215	74,215	1								17,878
Pennsylvania	11,483,613	7,590,688	7,077,760	64	18	9	4,335,166	3,968,178	22	5		150,473	150,473	5			287,202
Rhode Island	699,691	700,500	695,685	4	3												4,006
South Carolina	3,059,956	1,567,463	1,548,229	31	11	9	1,013,267	1,005,837	15	4	23	192,517	192,517	1	1	16	313,373
South Dakota	3,249,086	2,559,860	2,559,162	53	6	53	473,336	473,336	12			269,350	210,393	17			6,195
Tennessee	3,903,979	1,305,258	1,296,577	25	3	26	2,212,100	2,212,100	20	1	2	151,380	151,380	1		11	243,922
Texas	10,855,982	9,803,489	9,792,408	125	14	77	552,216	552,216	3	1	12	182,969	182,969			60	328,389
Utah	1,230,763	1,118,408	1,111,512	15	1	1	103,523	103,523	2								15,728
Vermont	729,857	749,535	705,319	10	7	18	14,100	14,100			2						10,438
Virginia	3,774,287	2,496,532	2,382,064	42	15	15	1,341,858	1,335,233	10	5	3	40,980	38,966		1	6	18,024
Washington	3,095,041	2,719,882	2,696,786	22	11	11	368,494	368,069	1	1							30,186
West Virginia	2,677,937	862,164	860,198	8	1	4	1,735,439	1,734,936	18	2	2	78,855	78,855	2		1	3,948
Wisconsin	5,022,683	4,017,481	3,988,449	37	5		923,521	922,766		2	6	123,559	61,711			27	49,756
Wyoming	1,360,841	1,161,714	1,152,337	12			189,365	189,365	2								19,139
Dist. of Columbia	410,804	417,779	410,804	3													
Hawaii	453,703	284,891	284,005	3			226,162	158,370	2								11,327
TOTALS	196,000,000	140,322,066	137,636,098	1617	298	383	48,655,717	46,996,648	335	68	400	5,981,906	5,513,825	77	9	327	5,853,429