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ROLLING RESISTANCE FORCES IN PNEUMATIC TIRES

JANUARY 1976

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16. Abstract

An anlysis is presented for the influence of test drum curvature on stress levels and resulting rolling resistance forces in pneumatic tires. The influence of test method on the measurement of rolling loss is also considered, and expressions are derived allowing rolling resistance measurements made on cylindrical drums to be transformed to flat surface values. Experimental verification is presented for the major conclusions.

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I. INTRODUCTION

The analyses given below attempt to clarify the rolling loss phenomena from the point of view of the principles of mechanics. The approach is an elementary one but is necessary as a prelude to more complete understanding of the various observed phenomena.

All rolling loss considered here is assumed to be measured in the steady state, i.e., temperature equilibrated condition. This allows us to draw on the concept that steady state rolling loss is well approximated by a linear dependence upon tire load, a crucial fact which is well documented in the literature [2].

A limited amount of experimental data is presented to verify some of the formulations presented here. For this reason these formulations should be considered provisional and subject to later revision if warranted.

II. SUMMARY

An analysis is presented for the interaction of tire rolling loss with drum curvature effects for the technically important problem of measuring this rolling loss on a cylindrical dynamometer wheel or test drum. This is the test method most commonly used in such measurements, and it is important to understand this interaction so that tire loss values may be reduced to a common base-line state, namely, those values obtained when running on the highway.

The analyses are divided into two parts. The first concerns the relation between maximum tire stress and tire load when the tire is in contact with a surface of given curvature. The second part concerns the resolution of tire and roadwheel forces and torques, so that one may see the relationship between actual tire loss and measured forces and torques.

The results of the two analyses may be summarized as follows:

(A) For the same tire load and inflation pressure the rolling resistance of a tire on a curved drum of radius R is greater than on a flat surface due to increased stress levels in the tire. The relationship between these is

$$F_{x_R} = F_x\left(1 + \frac{r}{R}\right)^{1/2} \qquad (1)$$

where F_{x_R} = True rolling resistance on a drum of radius R

F_x = Rolling resistance on the highway

r = Outside radius of the tire

R = Drum radius

(B) The rolling resistance on a curved drum, as measured by an axle force transducer, is less than the rolling resistance F_{x_R} due to the interaction of normal forces with measured rolling loss. This is caused by elastic deformation of the tire. The relationship between the two is

$$F_{x_M} = F_{x_R}\left(1 + \frac{r_L}{R}\right)^{-1} \qquad (2)$$

where F_{x_M} = Rolling resistance obtained from axle force measurements, taken on a drum of radius R

r_L = Loaded radius (axle height) of the tire above drum surface

(C) Combining Eqs. (1) and (2) gives

$$F_{x_M} = F_x\left(1 + \frac{r}{R}\right)^{1/2} \left(1 + \frac{r_L}{R}\right)^{-1} \tag{3}$$

This means that force measurements made on a curved drum give rolling resistance values less than those observed on a flat surface by the ratio

$$\left(1 + \frac{r}{R}\right)^{1/2} \left(1 + \frac{r_L}{R}\right)^{-1}$$

Same as using [illegible] [illegible] torque measurements on drum

(D) Torque measurements on a curved drum, such as coast down tests, measure the force F_{x_R} of Eq. (1) directly, without interaction of forces due to elastic deformation. For this case the rolling resistance as inferred from torque or coast down measurements must be reduced by the factor

$$\left(1 + \frac{r}{R}\right)^{1/2}$$

[illegible] using coast-downs on drum

in order to obtain the rolling resistance of the tire on a flat surface.

(E) If maximum tire deflection is used as a loading parameter instead of total tire load, then the rolling resistance is independent of drum radius and takes a form linearly proportional to deflection as shown in Eq. (28a).

III. TIRE LOSSES

LOAD-DEFLECTION ANALYSIS-CARCASS

Assume that the tire diameter conforms to the rigid roadwheel as shown by the dark lines in Figure 1. The contact patch length L is given by

$$L = 2R\sin\gamma = 2r\sin\theta \tag{4}$$

Assume both γ and θ to be small angles. Then to a first approximation

$$L \simeq 2R\gamma \simeq 2r\theta \qquad \theta = \frac{L}{2r} \quad \theta^2 = \frac{L^2}{4r^2} \tag{5}$$

The maximum tire deflection δ is given by

$$\delta = r(1 - \cos\theta) + R(1 - \cos\gamma) \tag{6}$$

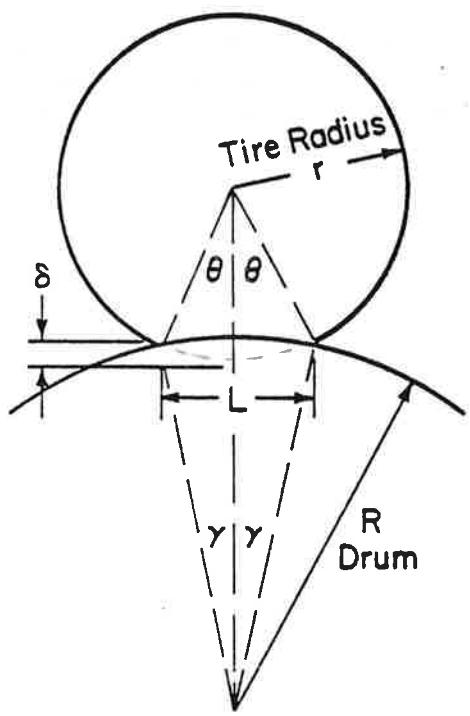

FIGURE 1. GEOMETRY OF TIRE AND TEST DRUM

Note that for an inside roadwheel one uses Eq. (6) but now with a negative value for R.

Again assuming small angles, Eq. (6) may be written

$$\delta = r\frac{\theta^2}{2} + R\frac{\gamma^2}{2} = \frac{L^2}{2}\left(\frac{1}{r} + \frac{1}{R}\right) \tag{7}$$

or

$$\frac{L}{2} = \delta^{1/2}\left(\frac{1}{2r} + \frac{1}{2R}\right)^{-1/2} \tag{8}$$

Consider next the cross-section of the tire at its center-plane, as shown in Figure 2. The width b of the contact patch is given by

$$b = 2r_1 \sin\beta \simeq 2r_1\beta \simeq w\beta \tag{9}$$

where w = tire section width

r_1 = radius of tire cross section

Also

$$\delta = r_1(1 - \cos\beta) \simeq \frac{w}{2}\frac{\beta^2}{2} \tag{10}$$

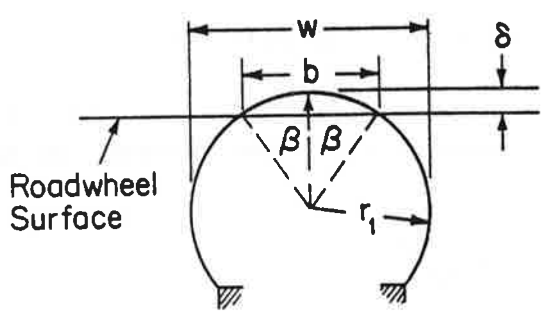

FIGURE 2. TIRE GEOMETRY

Combining (9) and (10) gives

$$\frac{b}{2} = (\delta w)^{1/2} \tag{11}$$

We now assume, as has been done in the past [3], that the load carried by the tire is the product of its contact area and inflation pressure p_o, and further that the contact area is an ellipse of semi-major axis $\frac{L}{2}$ (Eq. (8)) and semi-minor axis $\frac{b}{2}$ (Eq. (11)). Using F_z for the tire load

$$F_z = \pi p_o \frac{L}{2}\frac{b}{2} = \frac{\pi}{\sqrt{2}} p_o \, \delta(w)^{1/2} \left(\frac{1}{r} + \frac{1}{R}\right)^{-1/2} \tag{12}$$

or

$$\delta = \frac{\sqrt{2}}{\pi}\left(\frac{F_z}{p_o}\right)\frac{1}{\sqrt{w}}\left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \tag{13}$$

By general consideration of linear elasticity the strain in a body is proportional to the deflection at a point divided by a characteristic length. Equation (13) describes the maximum deflection. We choose tire radius r as the characteristic length and write the maximum strain e_{max} as

$$\begin{aligned} e_{max} &= \beta_1 \left(\frac{\delta}{r}\right) \\ &= \beta_2 \left(\frac{F_z}{p_o}\right)\frac{1}{r\sqrt{w}}\left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \end{aligned} \tag{14}$$

where β_j are dimensionless constants. The corresponding stress σ_{max} is

$$\sigma_{max} = \beta_3 E \left(\frac{F_z}{p_o}\right)\frac{1}{r\sqrt{w}}\left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \tag{15}$$

where E is a material modulus inserted to maintain dimensional homogeneity.

It is known that energy losses associated with cyclic flexing of materials exhibiting hysteretic loss characteristics are governed by an equation of the form

$$\text{Loss per cycle} = (\sigma_{max})^k$$

where k depends on material properties [1].

For this analysis we consider only temperature equilibrated values of rolling resistance. These show a linear dependence of loss with tire load [2]. For this reason we choose k = 1 here.

The energy loss per unit volume may now be written

$$\text{Energy Loss/Unit Volume} = \beta_4 \sigma_{max} \tag{16}$$

The power loss in the tire is a product of the carcass volume S_c and the energy loss per unit volume, all multiplied by the frequency with which a unit volume of the carcass passes through the contact patch. This frequency is given by the relation

$$f = \frac{\omega_t}{2\pi} = \frac{V}{2\pi r} \text{ cycles/unit time} \tag{17}$$

where V = forward velocity and ω_t = angular velocity of the tire. The total power loss in the carcass P_{L_c} is

$$P_{L_c} = \beta_5 E \left(\frac{F_z}{p_o}\right) \frac{V S_c}{r^2 \sqrt{w}} \left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \tag{18}$$

where S_c is the volume of the carcass. It is also possible to express this power loss in terms of tire deflection δ

$$P_{L_c} = \beta_5 E \frac{\delta}{r^2} S_c V \tag{18a}$$

LOAD-DEFLECTION ANALYSIS OF THE TREAD

Stresses in the tread are almost entirely compressive in nature and due to contact with the drum or roadway surface. Consider the tread supported by the carcass as shown in Figure 3. This is a series combination of springs.

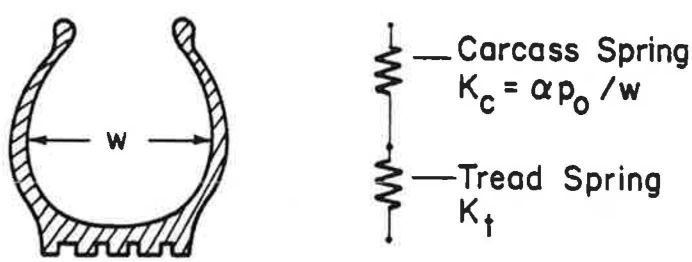

FIGURE 3. CROSS SECTION OF TIRE

The carcass spring has a spring rate proportional to inflation pressure p_o, and approximately inversely proportional to radius of curvature, or section width w. Its spring rate K_c may be approximated by the relation $K_c = \alpha p_o/w$, where α is a constant of proportionality. The equivalent series spring is given by

$$K_{eq} = \frac{\alpha p_o K_t/w}{\dfrac{\alpha p_o}{w} + K_t} \tag{19}$$

Consideration of numerical values of these spring rates leads to the conclusion that $K_t \gg \frac{\alpha p_o}{w}$ so that

$$K_{eq} \simeq \frac{\alpha p_o K_t}{w K_t} \simeq \alpha p_o/w$$

For a given tire deflection δ the compressive stress level in the tread is given by

$$\sigma_t = \delta K_{eq} = \frac{\alpha p_o \delta}{w}$$

Assuming that hysteretic loss is once again proportional to stress to the first power for temperature equilibrated conditions, and using the value of δ from Eq. (13), one obtains

$$\sigma_{max} \sim F_z \frac{1}{w^{3/2}} \left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \tag{20}$$

Using Eq. (17) the total power loss in the tread is now given by

$$P_{L_t} = \beta_6 F_z \frac{V \cdot S_t}{r w^{3/2}} \left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \tag{21}$$

where S_t is the volume of the tread. This may also be expressed in terms of tire deflection δ as

$$P_{L_t} = \beta_6 \frac{p_o \delta}{wr} V S_t \tag{21a}$$

TOTAL POWER LOSS

The total power loss in the tire is now given by the sum of Eqs. (18) and (21), or (18a) and (21a).

The power input to the tire is given by the product of rolling resistance F_{x_R} and forward velocity V

$$P_{IN} = F_{x_R} \cdot V \tag{22}$$

In the steady state the power loss given by Eqs. (18) and (21) must be equaled by the power supplied, given by Eq. (22). Equating and solving for F_{x_R} gives

$$\begin{aligned} F_{x_R} &= \beta_5 E \left(\frac{F_z}{p_o}\right) \frac{S_c}{r^2\sqrt{w}} \left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \\ &\quad + \beta_6 F_z \frac{S_t}{rw^{3/2}} \left(\frac{1}{r} + \frac{1}{R}\right)^{1/2} \\ &= \left[\beta_5 E \left(\frac{F_z}{p_o}\right) \frac{S_c}{\sqrt{r^5 w}} + \beta_6 F_z \frac{S_t}{\sqrt{r^3 w^3}}\right] \left(1 + \frac{r}{R}\right)^{1/2} \\ &= F_x \left(1 + \frac{r}{R}\right)^{1/2} \end{aligned} \tag{23}$$

where F_x is the rolling resistance on a flat surface.

$$F_x = \beta_5 E \left(\frac{F_z}{p_o}\right) \frac{S_c}{\sqrt{r^5 w}} + \beta_6 F_z \frac{S_t}{\sqrt{r^3 w^3}} \tag{24}$$

We assume here that the volume S_c of the tire carcass is the sum of all carcass elements undergoing cyclic stressing during rolling, and to a first approximation this is the total tire carcass excluding beads. The tread volume S_t is the total tread.

If the power input to the tire is in the form of a torque then it has the value

$$P_{IN} = T_t \, \omega_t \tag{25}$$

where ω_t is the true angular velocity from Eq. (14). The tire power loss in terms of angular velocity is, from Eqs. (14), (15), and (18)

$$P_L = \frac{\beta_5 E}{2}\left(\frac{F_z}{p_o}\right)\frac{\omega_t S_c}{r\sqrt{w}}\left(\frac{1}{r}+\frac{1}{R}\right)^{1/2}$$

$$+ \frac{\beta_6 F_z}{2}\ \frac{\omega_t S_t}{w^{3/2}}\left(\frac{1}{r}+\frac{1}{R}\right)^{1/2} \qquad (26)$$

Equating Eqs. (25) and (26) gives

$$T_{t_R} = \frac{\beta_5 E}{2}\left(\frac{F_z}{p_o}\right)\frac{S_c}{r\sqrt{w}}\left(\frac{1}{r}+\frac{1}{R}\right)^{1/2}$$

$$+ \frac{\beta_6}{2} F_z \frac{S_t}{w^{3/2}}\left(\frac{1}{r}+\frac{1}{R}\right)^{1/2}$$

$$= \left[\frac{\beta_5 E}{2}\left(\frac{F_z}{p_o}\right)\frac{S_c}{\sqrt{r^3 w}} + \frac{\beta_6}{2} F_z \frac{S_t}{\sqrt{rw^3}}\right]\left(1+\frac{r}{R}\right)^{1/2} \qquad (27)$$

This represents the torque associated with operating the tire on a convex surface of radius R. For a concave surface, a negative value of R is required.

For operating the tire on a flat surface

$$T_t = \frac{\beta_5 E}{2}\left(\frac{F_z}{p_o}\right)\frac{S_c}{\sqrt{r^3 w}} + \frac{\beta_6}{2} F_z \frac{S_t}{\sqrt{rw^3}} \qquad (28)$$

which is the torque needed to just overcome the internal losses in a tire.

If the power loss is expressed in terms of tire deflections as in Eqs. (18a) and (21a) then the rolling resistance may be obtained by again using Eq. (22) to give

$$F_x = \beta_5 E \frac{\delta}{r^2} S_c + \beta_6 \frac{p_o \delta}{wr} S_t$$

$$= \delta\left[\beta_5 E \frac{S_c}{r^2} + \beta_6 \frac{p_o}{wr} S_t\right] \qquad (28a)$$

This value of rolling resistance is independent of surface curvature and depends only on the maximum deflection of the tire. Equation (28a) implies that if deflection is maintained constant then the same rolling loss should be generated on a flat surface as on a curved drum.

IV. FORCE AND TORQUE ANALYSIS

The rolling resistance of a tire as calculated from stress considerations, as in Eqs. (23) or (24), may not be the rolling resistance as seen on conventional test measuring equipment. The following analyses attempt to examine the technically important test configurations in order of increasing complexity. For clarity all computations use force resultants only.

FREELY ROLLING TEST TIRE—POWER SUPPLIED FROM TEST SURFACE

Case 1—Flat Surface

The test surface moves to the right with velocity V. Figure 4 shows a free body diagram of the forces acting on the tire and wheel.

F_x = true rolling resistance

F_{x_m} = rolling resistance as measured by an axle force transducer

$$\sum F_x = 0 : \quad F_{x_m} = F_x$$

$$\sum F_z = 0 : \quad F_z = N$$

$$\sum M_o = 0 : \quad F_z \varepsilon_1 = F_x \cdot r_L$$

$$\varepsilon_1 = \frac{F_x}{F_z} \cdot r_L \tag{29}$$

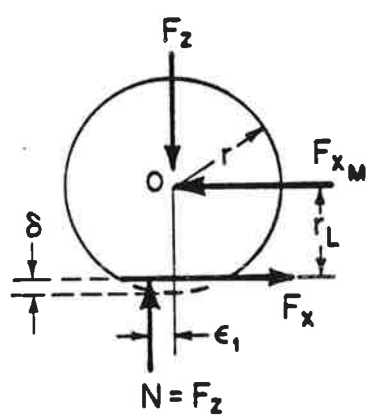

FIGURE 4. FORCES ON TIRE ON FLAT SURFACE

The measured rolling resistance is the true rolling resistance. The vertical force resultant moves forward by an offset ε_1 given by Eq. (29).

CASE 2—Cylindrical Surface

The tire is assumed to conform to the cylindrical drum as shown in Figure 5. The drum rotates clockwise due to a clockwise torque T_R. The wheel is also subject to a driving torque T_W.

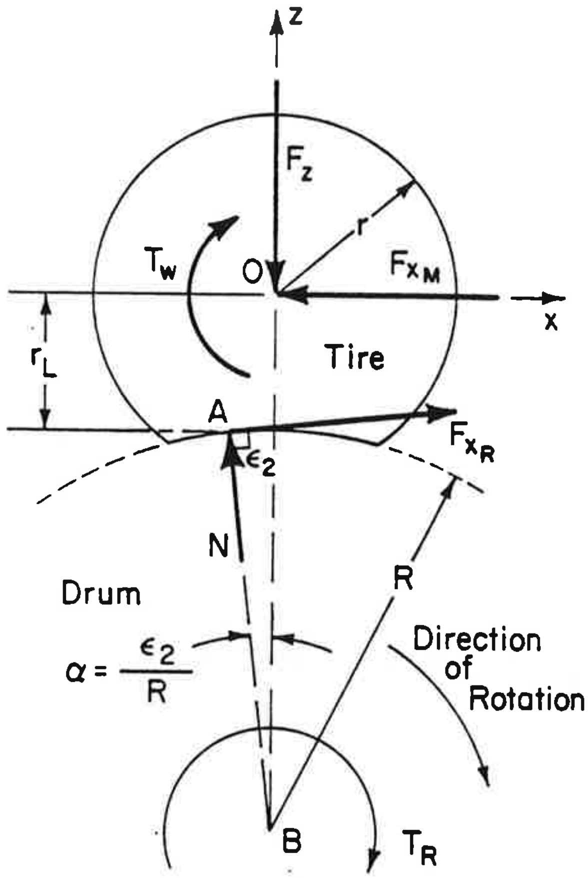

FIGURE 5. FORCES ON TIRE ON TEST DRUM

The free body diagram of the tire in Figure 5 shows forces acting on the tire. These are normal and tangential to the drum surface, being the normal force N and the rolling resistance F_{x_R}, now offset by an arc length ε_2 along the drum surface from the vertical line of centers.

The argument behind this assignment of force directions is simply that each element of drum surface has acting on it a pressure component normal to the surface and one tangential to it. The component normal to the surface passes through the drum center, causing no moment about the center. Hence the resultant of the normal forces, made up of the sum of the small incremental normal components, cannot cause any moment about the drum center, and must be perpendicular to the drum surface. The sum of tangential components forms a resultant tangential force at the drum surface, essentially perpendicular to the normal force resultant, i.e., tangent to the drum.

The measured horizontal force F_{x_M} is as shown, at right angles to the centerline between drum and tire. Considering the tire itself:

$$\sum F_z = 0 : \quad -F_z + F_{x_R} \sin\alpha + N \cos\alpha = 0$$

Assume

$$F_{x_R} << F_z$$

and

$$\alpha \text{ small}$$

Then

$$F_z \simeq N.$$

$$\sum F_x = 0 : \quad -F_{x_M} + F_{x_R} \cos\alpha - N \sin\alpha = 0$$

or

$$F_{x_M} = F_{x_R} - \alpha F_z. \tag{30}$$

$$\sum M_A = 0; \qquad -F_z \cdot \varepsilon_2 + F_{x_M}(r_L) - T_W = 0 \tag{31}$$

where r_L is the axle height above the drum surface.

Using Eq. (30) and the relationship $\varepsilon_2 = R\alpha$ gives

$$F_z R\alpha = (F_{x_R} - \alpha F_z)(r_L) - T_W$$

or

$$\alpha = \frac{F_{x_R}}{F_z} \frac{r_L}{R+r_L} - \frac{T_W}{F_z(R+r_L)} \tag{32}$$

This allows Eq. (30) to be solved for the measured rolling resistance

$$F_{x_M} = F_{x_R}\left(1 + \frac{r_L}{R}\right)^{-1} + T_W(R+r_L)^{-1} \tag{33}$$

For an internal drum one again uses a negative value for R.

Finally the torque input T_R to the dynamometer drum can be obtained from a free body diagram of that drum, and by taking moments about point B of Figure 5 of the drum, which yields

$$T_R = F_{x_R} \cdot R \tag{34}$$

One special case of technical interest can now be considered separately, namely, that of the freely rolling tire where $T_W = 0$.

From Eq. (33)

$$F_{x_M} = F_{x_R}\left(1 + \frac{r_L}{R}\right)^{-1} \tag{35}$$

From this we conclude the following:

(1) Axle force transducer measurements made on a powered drum misrepresent the rolling resistance of the tire on the curved drum, giving a rolling resistance smaller than the true value on a convex drum and larger than the true value on a concave drum. The reason for this is interaction of the contact pressure force resultant with the rolling resistance measurement.

(2) It is seen from Eq. (34) that torque measurements on a powered drum, either convex or concave, reflect the true value of the rolling resistance of the tire on the curved surface, although the rolling resistance may be different from that on a flat surface. Coast down and shaft torque measurements fall within this class.

(3) In order to assess the effects of increased rolling resistance due to increased tire stresses on a curved drum, along with the force interaction

effect discussed in (1) above, we must combine the hypothesis of Eq. (23) and substitute the term F_x from there into Eq. (35) accordingly. This is shown in Eq. (13).

(4) Finally if torque T_R on the drum is used as a measure of tire loss Eq. (34) shows no error associated with force interaction on the drum, but again Eq. (23) must be used to correct for the increased stress level in the tire. This is discussed in Section D of the Summary.

V. COMPARISON WITH EXPERIMENT

While extensive experimental data is not available to assess the validity of the conclusions given here, some rather accurate measurements have recently been made on a small group of passenger car tires at Calspan Corporation, Buffalo, New York. From their data, a comparison is made of the ratio of rolling loss measured on a 67-inch diameter drum to rolling loss on a flat belt, to that same ratio as predicted by Eq. (3) of this report. This comparison is presented in Table I. In general agreement is quite good.

A further verification of the correction factor due to increased stress levels, described in Eq. (1), is given in Figure 6. Here the rolling resistance of a small pneumatic tire is plotted against the ratio of the tire radius to test drum radius. This tire is an accurate scale model of a bias ply tire. It was developed and fabricated at University of Michigan facilities. The rolling resistance values were determined from coast down tests on a single shaft containing a number of test drums of different diameter. This experimental arrangement is shown in Figure 7, while the test tire itself is shown in Figure 8.

Each of the experimental values of rolling resistance was measured for the same vertical load, same inflation pressure, and same surface speed. Under these conditions the rolling resistance as corrected to a flat surface by Eq. (1) should be a constant value. As can be seen from Figure 6, the corrected drum values are indeed nearly constant. This gives additional verification for the accuracy of the correction factors developed in Eq. (1) for increased stress levels in the tire due to drum curvature.

Additional verification of the general analysis presented here can be obtained by comparing experimental data for rolling loss of tires to Eq. (28a), taking care to express the rolling loss in terms of tire deflection and inflation pressure. If Eq. (28a) is plotted on a carpet plot as shown in Figure 9 a series of straight lines is obtained connecting rolling loss values at varying pressures and at varying tire deflections. To check this, two small scale pneumatic tires illustrated in Figure 8 were also used to obtain measured rolling loss data on a single drum under a variety of inflation pressures and tire deflections. This data has been plotted in Figures 10 and 11 again in the form of a carpet plot. This shows clearly that the form is very close to that given in Eq. (28a), and serves as further verification of the loss function derived in Section III of this report. It also tends to verify the form of the tread loss term which appears as the second term in Eq. (28a).

A different and somewhat more sensitive test of the general theory presented here is obtained directly from Eq. (28a). Examination of this formulation shows that it predicts that the rolling resistance force depends only on maximum tire deflection and is independent of the radius of the test drum upon which the measurement is made. This hypothesis can be tested by measuring the rolling resistance of the same tire on test drums of two different radii. This has been done using the experimental apparatus illustrated in Figure 7. The results are shown in Figure 12, where it is seen that rolling resistance is essentially the same at equal tire deflections for two test drums of quite different size, the larger drum being 2.67 times the diameter of the smaller. This result serves to confirm the specific form of Eq. (28a), as well as the general formulation leading to it.

Note that Eq. (28a) also predicts a linear relationship between rolling resistance and tire deflection. The data given in Figure 12 shows two regions, each of approximate linearity, with a somewhat different slope in each. However the second region seems to be quite linear above a tire deflection of 10% of section height, and this is the region of technical importance. It is not uncommon to find such anomalies at very small tire deflections.

TABLE I.-SUMMARY OF PREDICTED AND EXPERIMENTAL RATIOS OF ROLLING RESISTANCE MEASURED ON A 67-INCH DRUM BY DIRECT FORCE MEASUREMENT TO THE ROLLING RESISTANCE MEASURED ON A FLAT SURFACE BY THE SAME METHOD

$$\text{Predicted Ratio of } F_{x_M}/F_x = \frac{(1 + r/R)^{1/2}}{(1 + r_L/R)}$$

Where:

F_{x_M} = Rolling resistance measured on roadwheel

F_x = Rolling resistance measured on flat surface

r = Tire radius

R = Roadwheel radius = 33.5 inches

r_L = Loaded radius of tire

Tire	Predicted F_{x_M}/F_x	Experimental* F_{x_M}/F_x
A78-13 Bias Ply	.871	.891
G78-14 Bias Ply	.858	.848
H78-15 Bias Ply	.856	.850
A78-13 Bias Belt	.873	.884
G78-14 Bias Belt	.860	.825
L78-15 Bias Belt	.851	.834
BR78-13 Radial	.881	.891
GR78-14 Radial	.875	.847
LR78-15 Radial	.866	.834

*All measurements at rated load, 24 psi inflation pressure and at temperature equilibrium.

Source: Calspan Corporation.

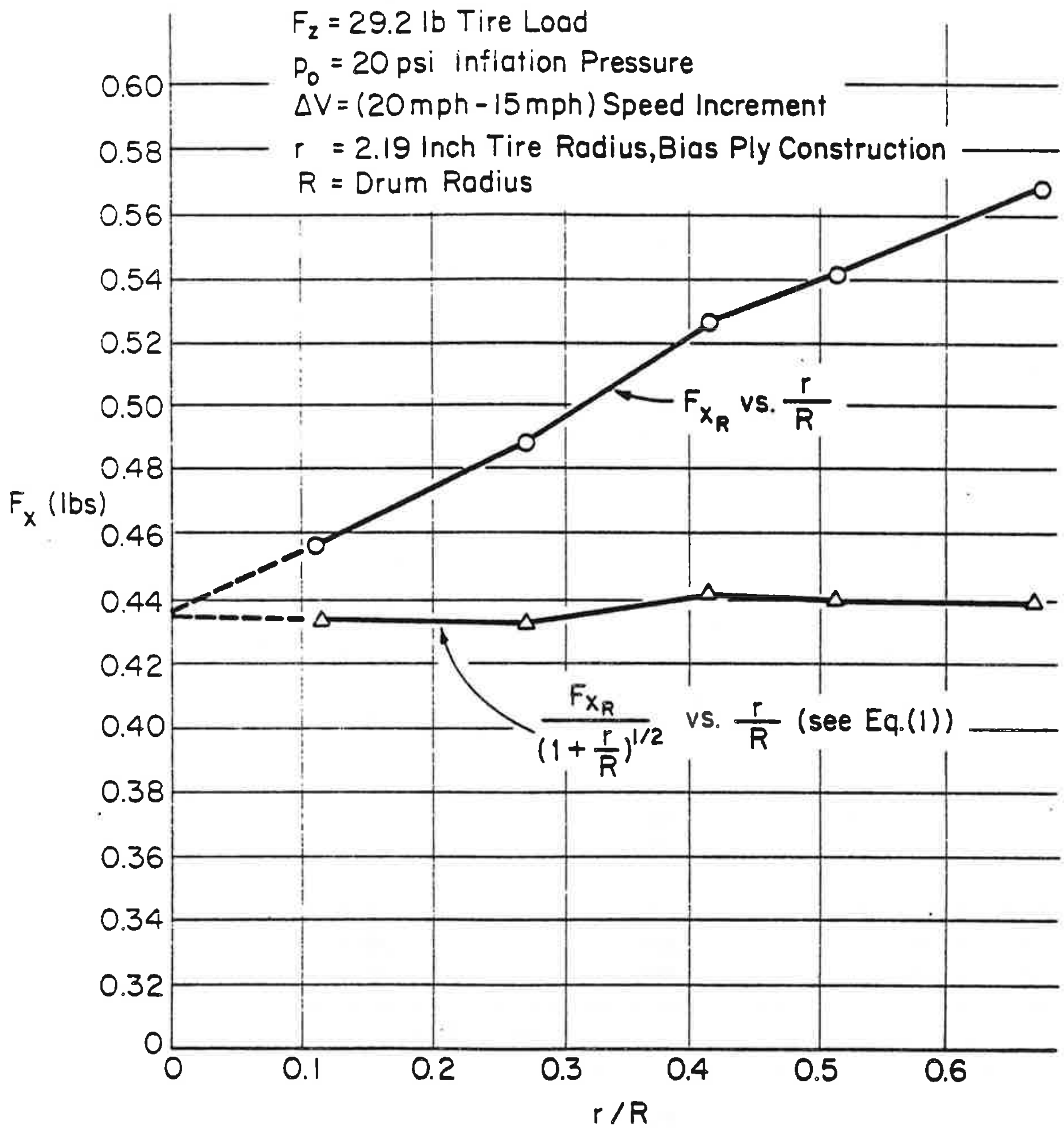

FIGURE 6. MEASURED ROLLING RESISTANCE ON VARIOUS DIAMETER TEST DRUMS

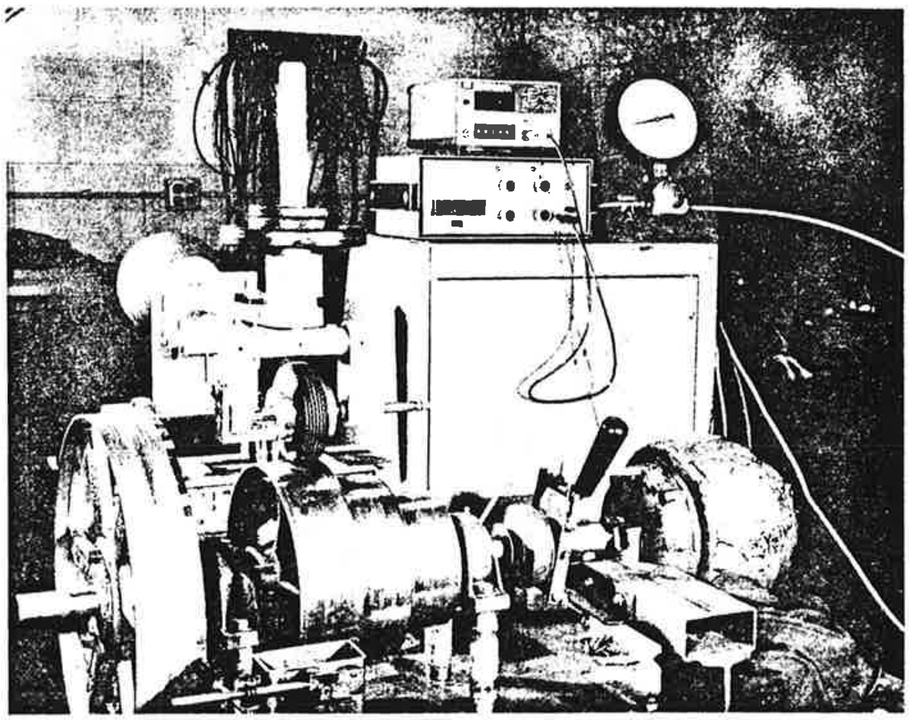

FIGURE 7. TEST ARRANGEMENT USED TO MEASURE ROLLING RESISTANCE ON DIFFERENT DIAMETER DRUMS

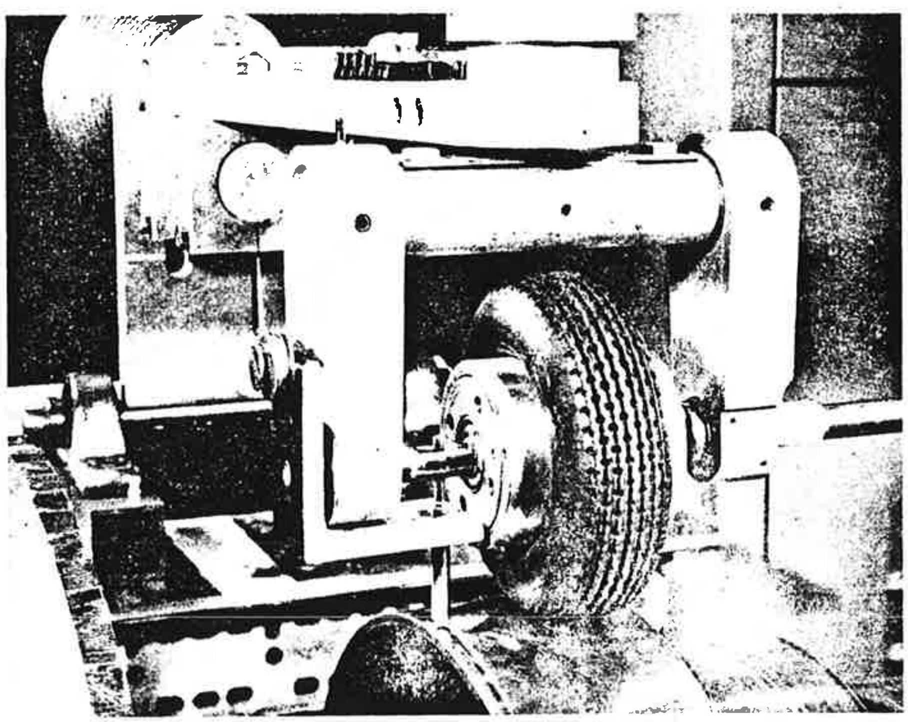

FIGURE 8. SMALL SCALE TEST TIRE

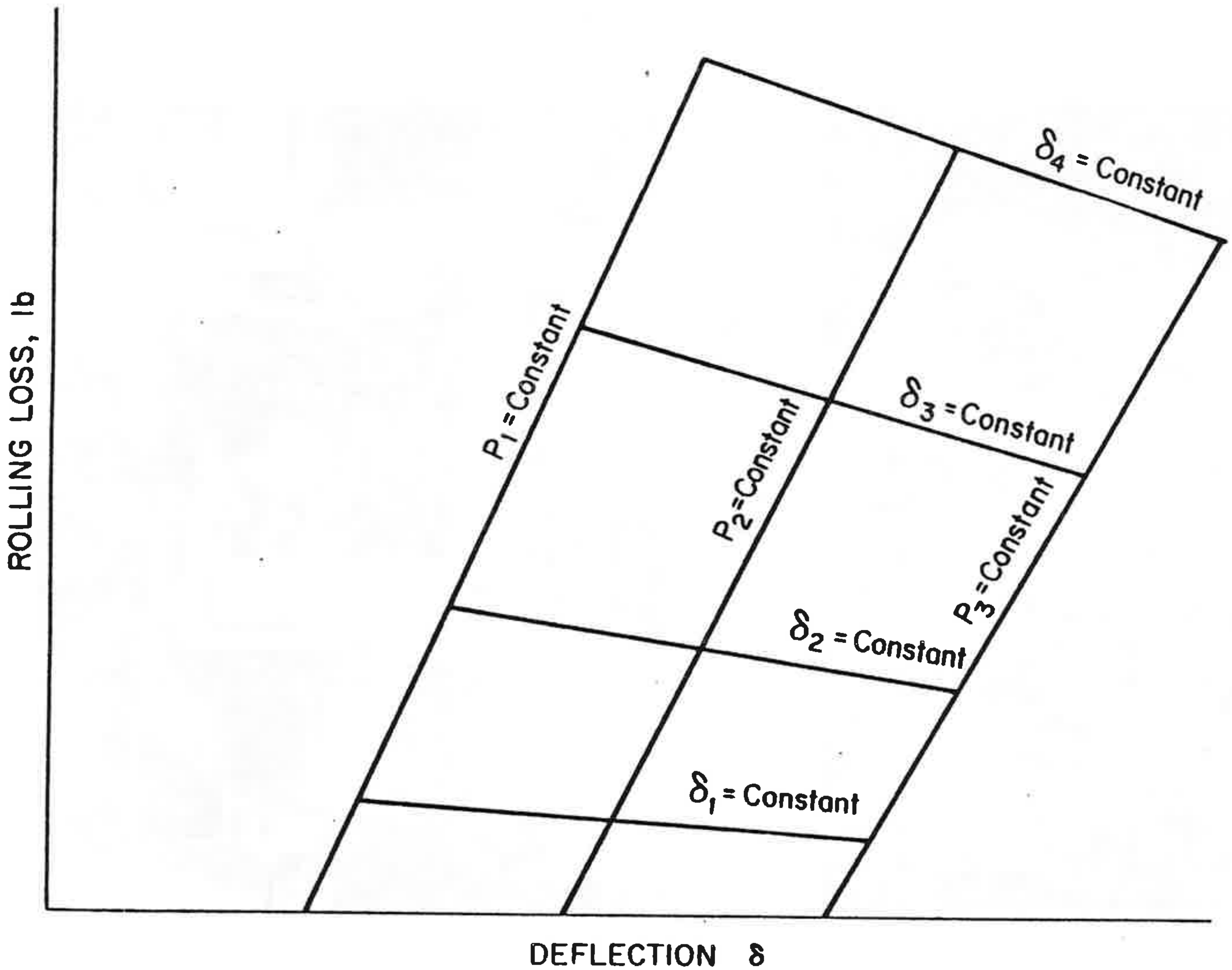

FIGURE 9. CARPET PLOT OF EQ. (28a) FOR ROLLING RESISTANCE

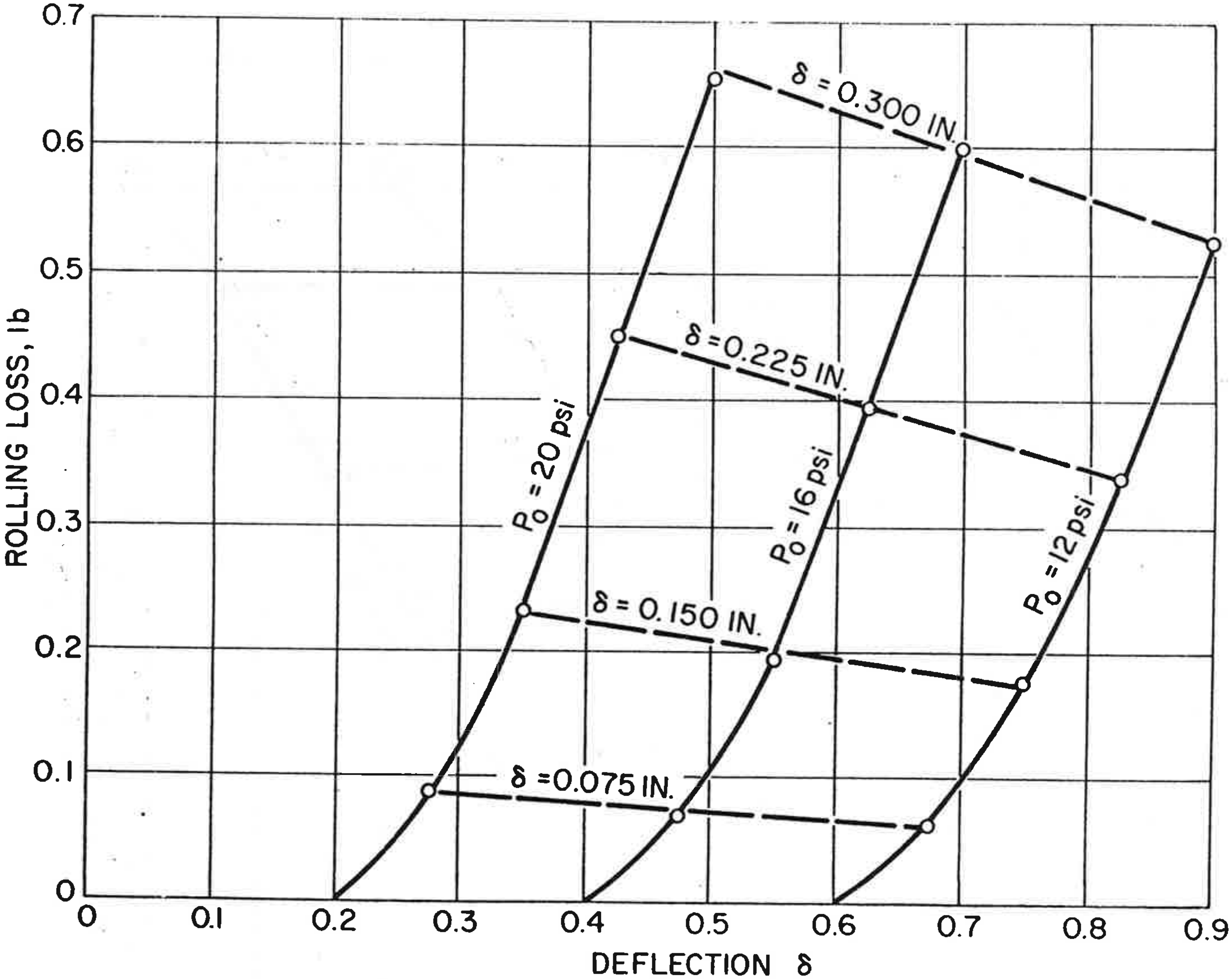

FIGURE 10. CARPET PLOT OF MEASURED ROLLING RESISTANCE AS A FUNCTION OF INFLATION PRESSURE AND TIRE DEFLECTION (TIRE A)

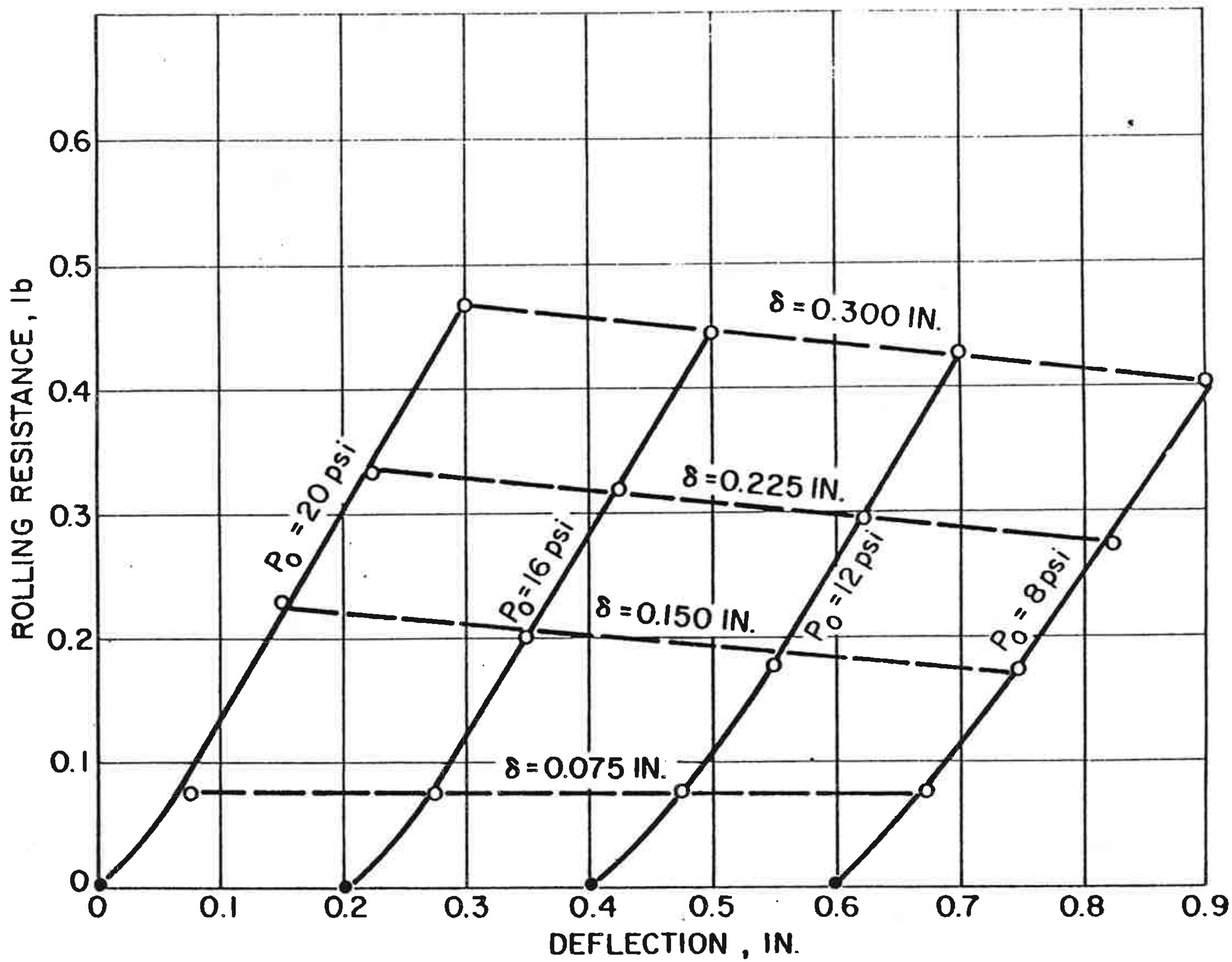

FIGURE 11. CARPET PLOT OF MEASURED ROLLING RESISTANCE AS A FUNCTION OF INFLATION PRESSURE AND TIRE DEFLECTION (TIRE B)

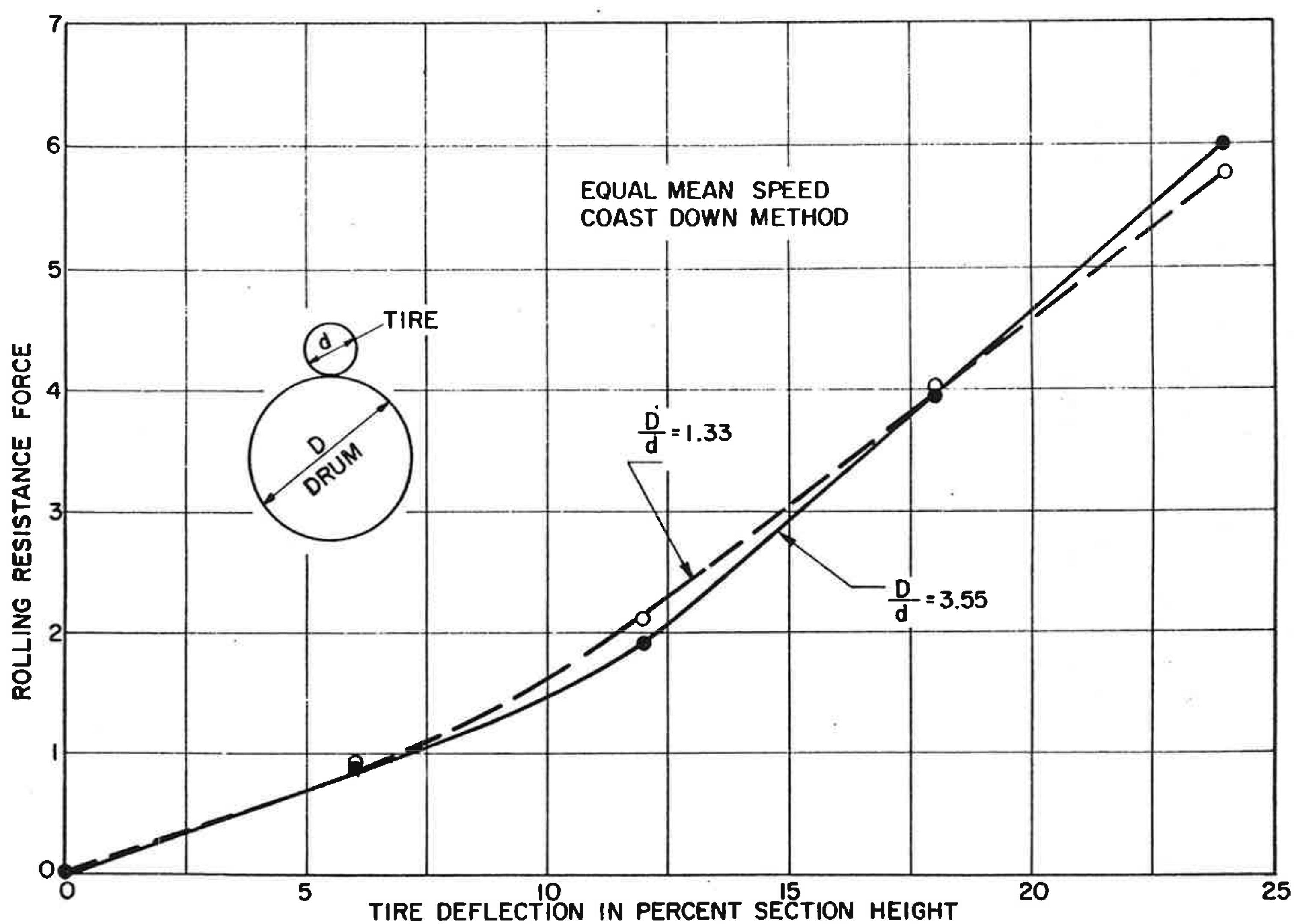

FIGURE 12. ROLLING RESISTANCE FORCE AT CONSTANT DEFLECTION ON TWO TEST DRUMS

REFERENCES

[1] "Shock and Vibration Handbook," ed. by C. M. Harris and C. E. Crede, McGraw-Hill Book Co., New York, 1961.

[2] Clark, S. K., R. N. Dodge, R. J. Ganter, and J. R. Luchini, "Rolling Resistance of Pneumatic Tires," University of Michigan Report DOT-TSC-74-2, prepared for Department of Transporation, Transportation Systems Center, Cambridge, Mass., July 1974.

[3] Clark, S. K., "Simple Approximations for Force-Deflection Characteristics of Aircraft Tires," University of Michigan Report 056080-8-T, prepared for National Aeronautics and Space Administration, Washington, D.C., December 1965.

EXECUTIVE SUMMARY

An analysis is presented for the influence of test drum curvature on stress levels and resulting rolling resistance forces in pneumatic tires. The influence of test method on the measurement of rolling loss is also considered, and expressions are derived allowing rolling resistance measurements made on cylindrical drums to be transformed to flat surface values. Experimental verification is presented for the major conclusions.

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