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SIMULATING DYNAMIC RIDE CHARACTERISTICS OF PNEUMATIC TIRES

ARMY ENGINEER WATERWAYS EXPERIMENT STATION,
VICKSBURG, MISSISSIPPI

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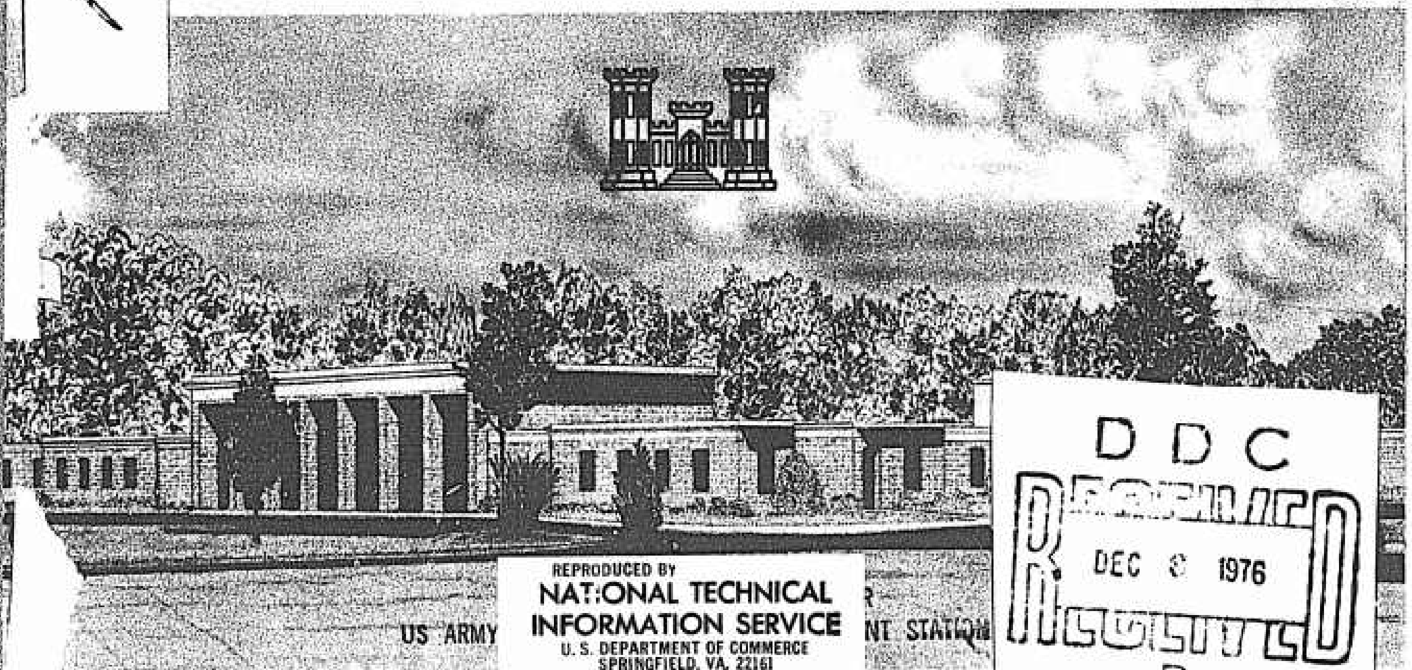
by

A. S. Lessem



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13. ABSTRACT

The data shown indicate that a mathematical representation of pneumatic tire, in terms of the deflections of many radial segments, successfully displays the essential feature of horizontal and vertical forces transmitted through the tire. It is indicated further that the segmented tire model enables realistic predictions to be made of the displacement and force time histories for a pneumatic tire towed over a rigid obstacle.

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Foreword

This study was conducted at the U. S. Army Engineer Waterways Experiment Station (WES) as a part of the vehicle mobility research program under DA Project 1T014501B52A, "Research in Earth Sciences," Task 01, "Terrain Analysis," under the sponsorship and guidance of the Development Directorate, U. S. Army Materiel Command.

Acknowledgment is made to personnel of the Simulation Laboratory of the George C. Marshall Space Flight Center, National Aeronautics and Space Administration, Huntsville, Alabama, for assistance and use of their computing facilities.

The tests were performed by personnel of the Mobility Research Branch, Mobility and Environmental Division, WES, during the period September 1966 to April 1967 under the general supervision of Messrs. W. J. Turnbull, W. G. Shockley, and S. J. Knight, and under the direct supervision of Dr. D. R. Freitag. Mr. Lessem prepared this paper for presentation at the 1968 Forest Engineering Conference of the American Society of Agricultural Engineers at Michigan State University, East Lansing, Michigan, in September 1968.

COL John R. Oswalt, Jr., CE, and COL Levi A. Brown, CE, were Directors of the WES during this study and preparation of this paper. Mr. J. B. Tiffany was Technical Director.

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Allan S. Lessem
Engineer

Mobility Research Branch
Mobility and Environmental Division
U.S. Army Engineer Waterways Experiment Station
Vicksburg, Mississippi

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SIMULATING DYNAMIC RIDE CHARACTERISTICS

OF PNEUMATIC TIRES

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A. S. Lessem

In a program of research in vehicle dynamics, the U. S. Army Engineer Waterways Experiment Station has developed and verified a mathematical model for the case of a pneumatic tire traversing nondeforming obstacles, with zero slip. Data used to calculate model parameters and to produce time histories of dynamic responses were obtained in laboratory tests with 9.00-14 tires under several conditions of ply rating and inflation pressure.

The model that has been used often in vehicle dynamics studies to represent a pneumatic tire (for example, see references 1 and 2) is shown in fig. 1. The spring element represents the ability of the tire to recover its original shape in the absence of deforming forces; the dashpot element (when it is included) represents the dissipation of energy in the tire by viscous damping. Contact between tire and terrain is at a single point, and forces are transmitted through the tire to the axle in the vertical direction only.

The basic idea for the tire model described herein is to represent the tire by many spring elements, as shown in fig. 2. This accounts for physically significant contributions of horizontal forces transmitted to the axle. For simplicity, dashpot elements were not included.

When the pneumatic tire is represented by many spring elements, the displacement forcing function for one spring element need not be the same as that for the neighboring spring. This feature permits small-size

terrain features with abrupt changes in slope to be "enveloped" by the model. As each spring element is deflected, the spring force produced and transmitted to the vehicle axle is determined. Knowledge of the magnitude and location of this force permits a force vector to be computed and resolved into horizontal and vertical components. The contributions of the individual elements are summed vectorially and used as inputs to the mathematical model that represents the vehicle.

To convert the basic idea for the tire model into practical form, procedures were developed for division of the pneumatic tire into segments, calculation of spring coefficients for the segments, and conversion of terrain profiles into separate displacement forcing functions for the segments.

That portion of the tire carcass that undergoes flexing and deformation at any instant is identified as the "active region" (see fig. 3). This region was assumed constant in size and large enough to include most anticipated fluctuations in the size of the ground contact area. In reality, as the tire traverses irregular terrain, the sizes of both the active region and the tire contact area vary. As the tire rotates, different portions of the carcass are swept into, and out of, the active region. Thus, instead of attention being focused on a particular portion of the tire as it rotates about the axle, a nonrotating area between the terrain and the axle is monitored. This nonrotating active region is divided into segments (see fig. 3). These segments are fixed in position with respect to one another, and may be regarded as boundaries through which the physical tire passes during rotation.

The load-deflection curves for a pneumatic tire, shown in fig. 4, represent conditions for which large-scale carcass flexing occurs. These

curves can be regarded as the result of joint contributions of many segments acting together, since a larger area of carcass is involved in restraining the load as more load is applied. From these curves the small-scale flexural properties of any one segment in the active region can be inferred.

A concept of "effective radial deflection" was developed as an aid in calculating segment spring coefficients. In this concept, illustrated in fig. 5, the actual segment deflection encountered during a load-deflection test is replaced by a fictitious uniform radial deflection, Δ . The effective radial deflection of each segment varies as the deflection varies at the vertical reference position.

The flexure property of the tire carcass contained within each segment boundary is represented by a linear spring with coefficient K , which is the same for all segments. The spring is positioned along the radial center line of each segment. With the location of each segment with respect to the vertical reference position known, the magnitude and line of action of the force due to deflection of each spring can be determined. In an experimental setup for recording quasi-static load-deflection curves, a load cell registers the total vertical force exerted by the deflected tire carcass. The link between the analytical representation of the tire and the physical reality of its load-deflection curve can be made by using an equation of static equilibrium as follows:

$$F = 2 \sum_{i=1}^{n} K\Delta_i \cos \phi_i \tag{1}$$

where

F = Load cell reading, lb

K = Segment spring coefficient, lb/in.

Δ_i = Effective radial deflection, i^{th} segment, in.

ϕ_i = Force direction angle, i^{th} segment, radians

n = Number of segments each side of vertical reference position

To apply this analytical representation to a specific tire, one point on the load-deflection curve for that tire must be used to calculate K in equation 1. If a value of carcass deflection, δ, at the vertical reference position is selected (for instance $\delta = 1$ in.), and if the load cell output, F_1, corresponding to this selection is read from a load-deflection graph, then K can be computed as:

$$K = \frac{F_1}{2 \sum_{i=1}^{n} \Delta_i \cos \phi_i} \tag{2}$$

In this equation, the values of Δ_i corresponding to $\delta = 1$ in. are read from a graph (fig. 6) of effective radial deflection versus carcass deflection, and the values of $\cos \phi_i$ are fixed by section geometry.

Once K is calculated, this value can be used in equation 1 to define the analytical load-deflection curve. Thus, any other δ may be selected and corresponding Δ_i obtained from the effective radial deflection graph. These values of Δ_i, together with the known values of $\cos \phi_i$, are put into equation 1 to compute the load corresponding to the selected deflection. An experimental load-deflection curve is compared in fig. 7 to several points produced by equation 1 using a K value determined at

$\delta = 1$ in. This value was chosen on the ascending portion of the experimental load-deflection curve. The use of a linear, constant-coefficient spring as representative of the flexing of a segment of tire carcass appears to be reasonable.

With the rotating tire mathematically represented by a stationary segmented active region, the nondeformable terrain profile encountered by the tire is represented as a displacement function that traverses the active region. The segments are deflected in sequence, by the displacement function, as illustrated in fig. 8. The time of application and the deflection amplitude are respectively different for each segment.

The computer implementation of the model must provide for generation of a terrain profile function. This function must be shifted in time to account for its sequential encounters with the segments, and it must be changed in amplitude to account for dynamic motions of the tire-vehicle system.

To evaluate the segmented-tire model, controlled tests were conducted in the laboratory with a single-wheel test carriage. The model of the tire and the model of the carriage were combined, as shown in fig. 9.

The model of the tire was organized with the active region divided into 10 segments, and the segment spring coefficients were calculated for several inflation pressures. Numerical values for the carriage spring and damping coefficients were obtained by repeating the obstacle tests using a rigid aluminum wheel, thus revealing the dynamic properties of the carriage alone. The values obtained were reasonably independent of carriage speed.

In writing the equations of motion, it was assumed that (a) the obstacle does not deform, (b) there is no slip between tire and obstacle under towed conditions, (c) no forces are generated parallel to the wheel axle, (d) the

carriage towing speed remains constant, and (e) the pneumatic wheel load remains constant. The equations are:

$$m\ddot{z} = \sum_{i=1}^{10} K\Delta_i \cos \phi_i - B_V\dot{z} - g\left(m \pm \frac{F_o}{g}\right) \tag{3}$$

$$m\ddot{x} = \sum_{i=1}^{10} K\Delta_i \sin \phi_i - B_H\dot{x} - K_H x \tag{4}$$

where

$$\Delta_i = \begin{cases} Y_i - H_{ri} - z , & Y_i - H_{ri} - z \geq 0 \\ 0 , & Y_i - H_{ri} - z < 0 \end{cases} \tag{5}$$

and

B_H = Horizontal carriage damping coefficient, lb/in./sec

B_V = Vertical carriage damping coefficient, lb/in./sec

F_o = Pneumatic load applied to tire in excess of, or in opposition to, deadweight load

g = Acceleration of gravity

H_{ri} = Threshold height of i^{th} segment in equivalent rigid wheel, in.

K = Segment spring coefficient, lb/in.

K_H = Horizontal carriage spring coefficient, lb/in.

m = Inertial mass of test carriage, lb sec^2/in.

x = Horizontal axle displacement, in.

Y_i = Vertical obstacle height beneath i^{th} segment, in.

z = Vertical axle displacement, in.

δ = Center-line static deflection, in.

Δ_i = Deflection of i^{th} segment, in.

ϕ_i = Location angle for i^{th} segment

Equation 3 is used to calculate the resultant vertical force, equation 4 the resultant horizontal force, and equation 5 the deflection of each segment. This segment deflection, Δ_i , is permitted to have positive values only; negative values are replaced by zero. This corresponds to an assumption that the tire may only be compressed by the obstacle and not stretched by it.

The rigid-wheel threshold height, H_{ri} , is illustrated in fig. 10. Each H_{ri} gives the height of the i^{th} segment contact point above the ground when the tire has no static deflection. The H_{ri} concept serves two purposes: first, to permit the tire model to display realistic static deflections by invoking as many segments as needed to restrain an applied load, and second, to modify the height of obstacle displacement functions as required by the height of the contact point of each segment.

Computed and observed horizontal restraining force exerted on the axle by the carriage and vertical displacement of the axle, during traversal of a 2- by 8-in. rigid, rectangular obstacle, are compared in fig. 11. The comparison was essentially qualitative, this being consistent with the use of the overly simplified representation of the test carriage. The most desired feature of the composite tire-carriage model, the ability to reproduce the basic features of the response wave shapes as seen in the laboratory, was realized; the composite model, through the segmented representation of the pneumatic tire, was capable of producing responses with realistic wave shapes. The positively and negatively directed horizontal restraining forces and the vertical axle displacements were displayed with basically correct form. If a point-contact model had been used, the duration of the response would have been identical with the time for the point to traverse

the obstacle. The multiple-contact model under discussion produces responses whose durations are realistically extended beyond the corresponding obstacle traversal time. Horizontal input to the axle from a point-contact model can be obtained only by special computing techniques for estimating the angular position of the resultant force acting on the wheel. The prediction of horizontal responses is a natural product of the segmented system.

A quantitative comparison of the computed and observed responses revealed several discrepancies that were related to the simple linear model of the test carriage and to the finite number of segments in the tire model. These can be overcome by more refined modeling (reference 3).

SUMMARY

The data shown indicate that a mathematical representation of a pneumatic tire, in terms of the deflections of many radial segments, successfully displays the essential feature of horizontal and vertical forces transmitted through the tire. It is indicated further that the segmented tire model enables realistic predictions to be made of the displacement and force time histories for a pneumatic tire towed over a rigid obstacle.

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3. Lessem, A. S., "Dynamics of Wheeled Vehicles; A Mathematical Model for the Traversal of Rigid Obstacles by a Pneumatic Tire," Technical Report No. M-68-1, Report 1, May 1968, U. S. Army Engineer Waterways Experiment Station, CE, Vicksburg, Miss.

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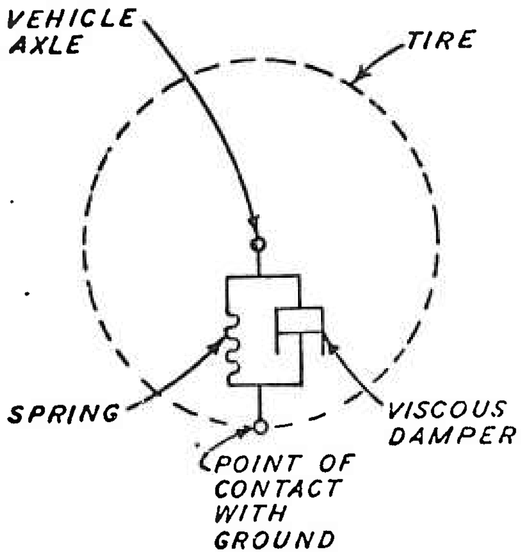


Fig. 1. Point-contact model

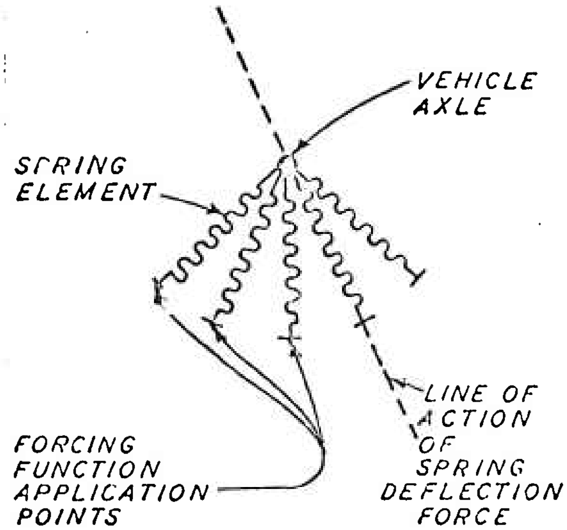


Fig. 2. Extended-contact model

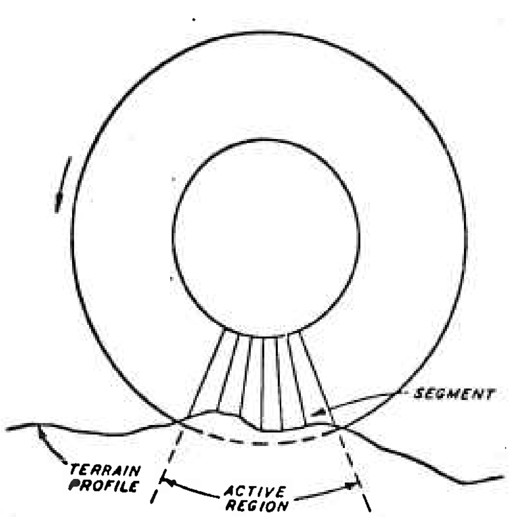


Fig. 3. Segmented, nonrotating, active region

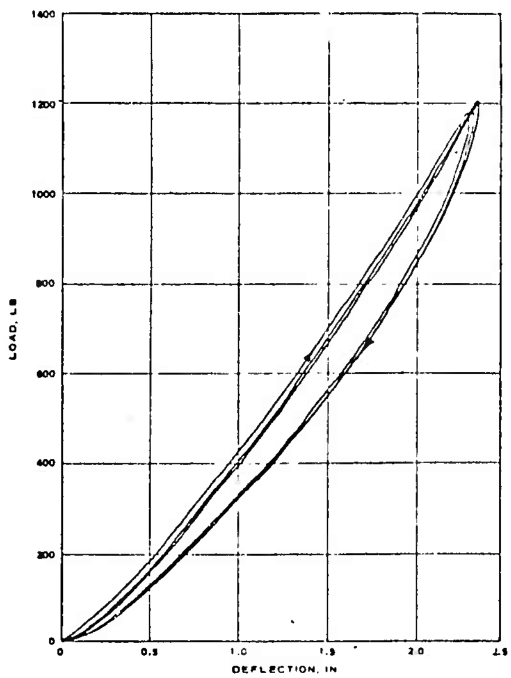


Fig. 4. Representative load-deflection curves; 9.00-14, 8-PR tire, 10-psi inflation pressure

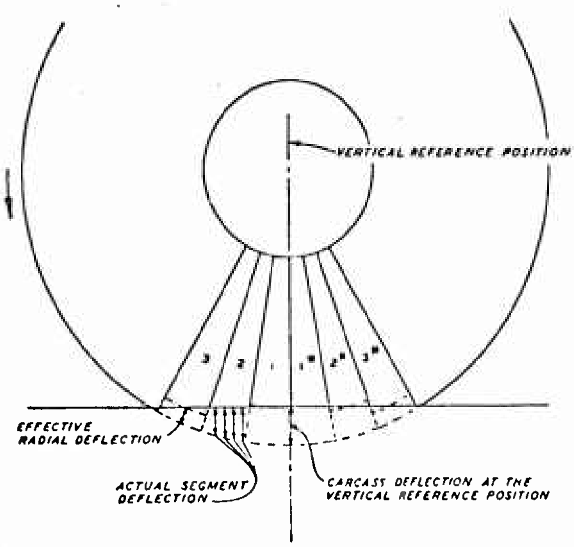


Fig. 5. Illustration of concept of effective radial deflection

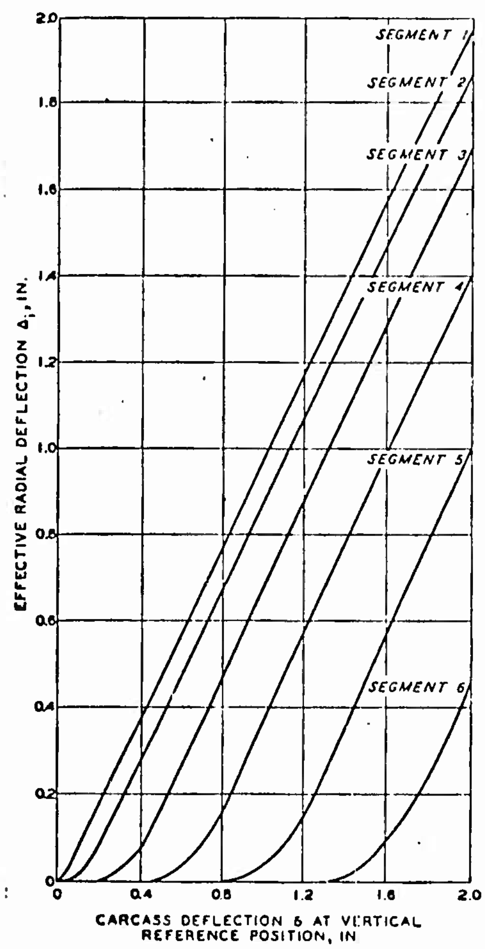


Fig. 6. Representative radial deflection curves

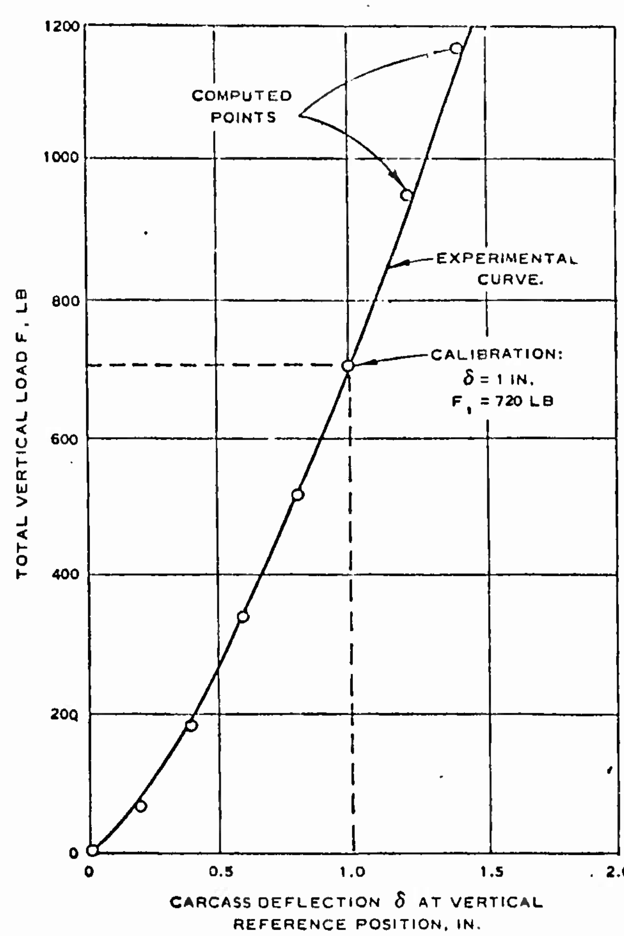


Fig. 7. Experimental load-deflection curve

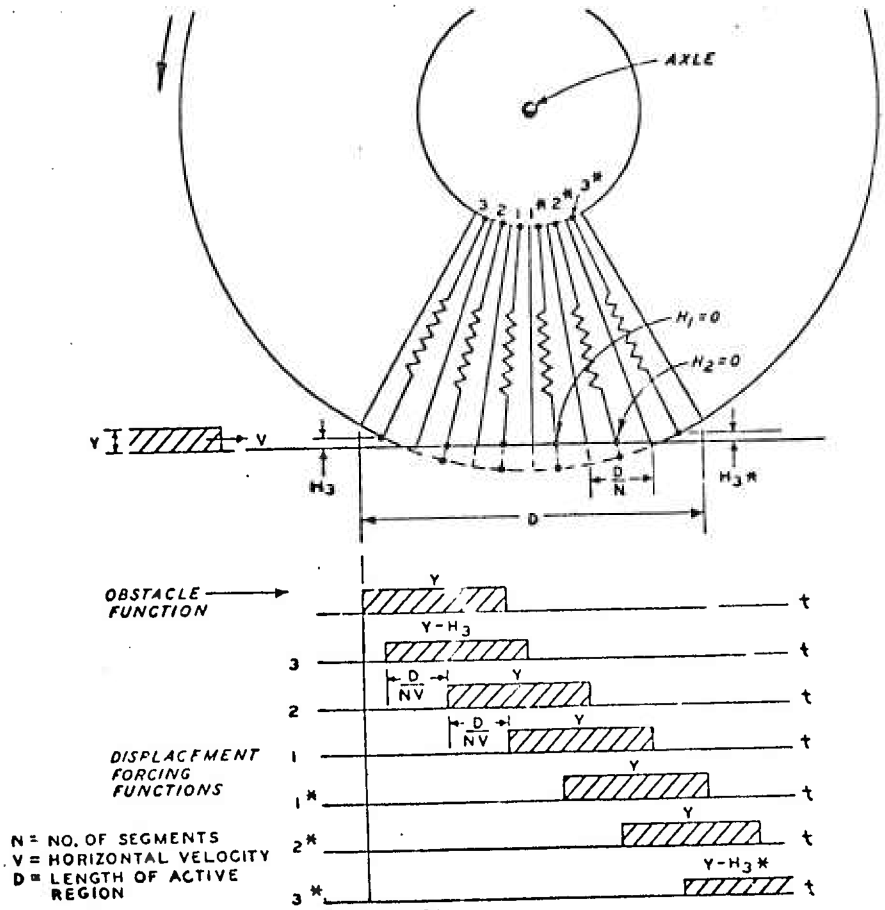


Fig. 8. Sequential deflections

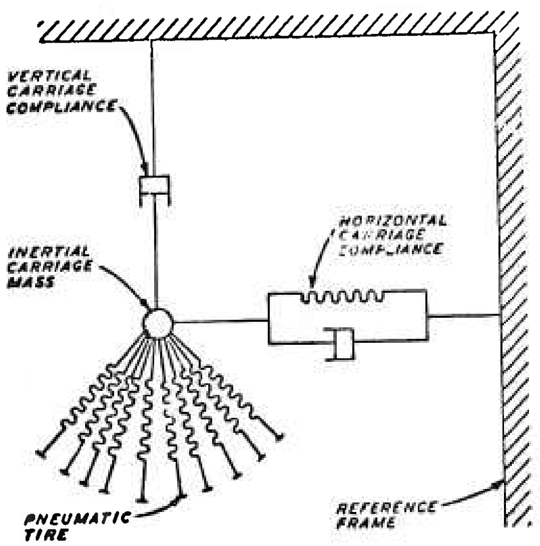


Fig. 9. Composite tire-carriage model

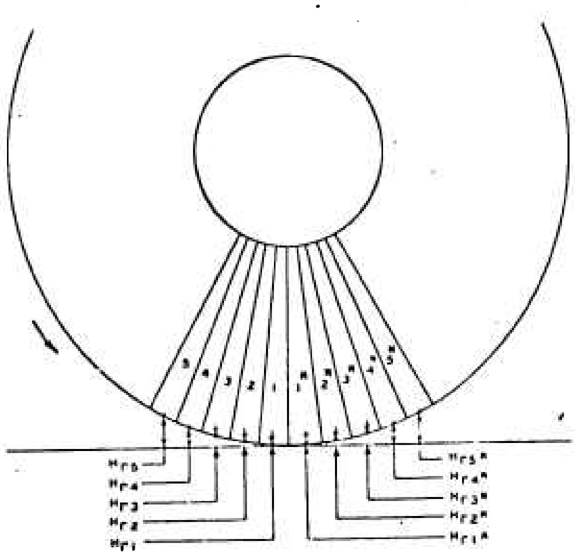
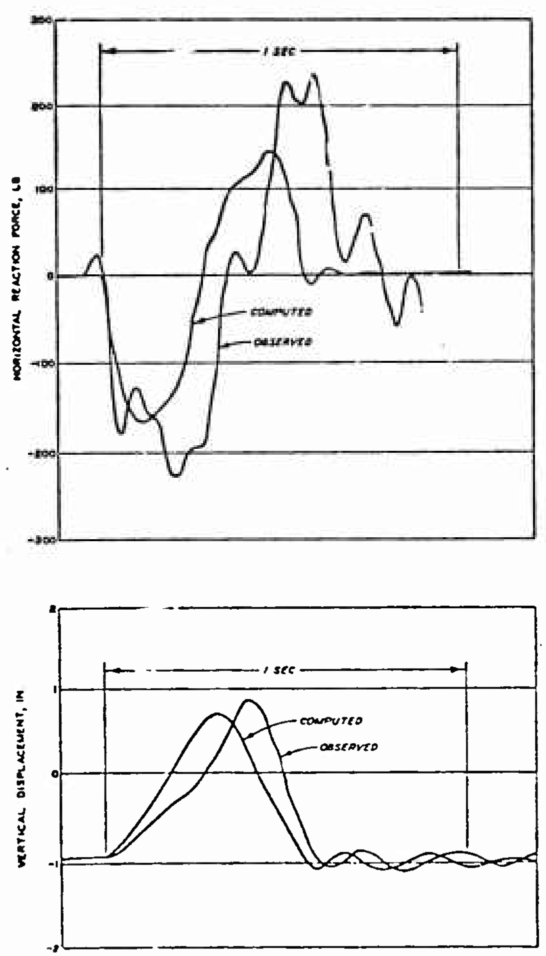
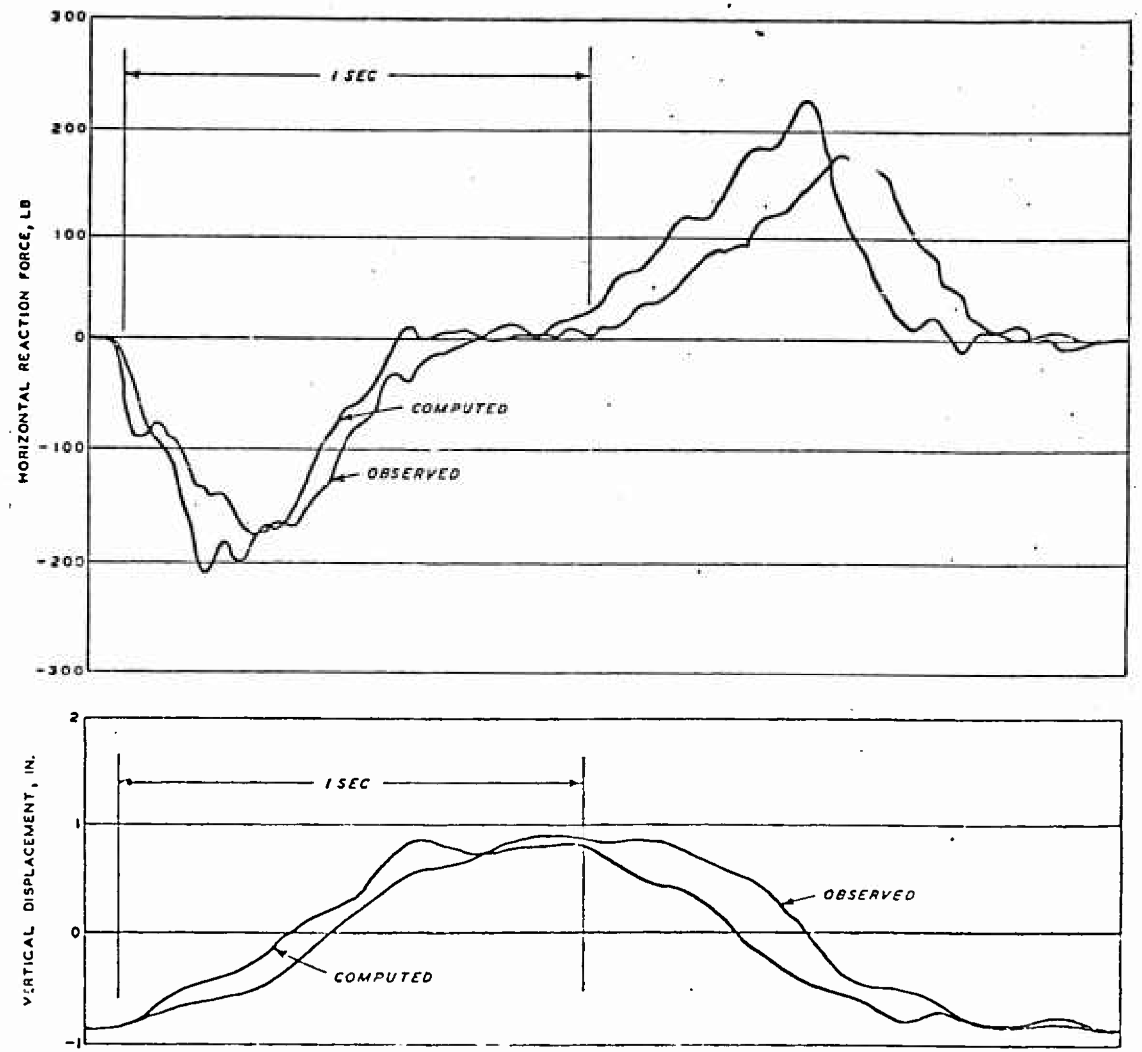


Fig. 10. Rigid-wheel threshold height



a. 1000-lb load, 20-psi inflation pressure, 3-fps carriage speed



b. 500-lb load, 30-psi inflation pressure, 1-fps carriage speed

Fig. 11. Comparison of empirical and analytical results (traversal of 2- by 8-in. rigid obstacle)