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SURFACE VEHICLE INFORMATION REPORT

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LABORATORY TESTING MACHINES AND PROCEDURES FOR MEASURING THE STEADY STATE FORCE AND MOMENT PROPERTIES OF PASSENGER CAR TIRES

***Foreword*—**This Document has not changed other than to put it into the new SAE Technical Standards Board Format.

1. ***Scope*—**This Information Report presents background and rationale for SAE Recommended Practice J1106, Laboratory Testing Machine and Procedures for Measuring the Steady Force and Moment Properties of Passenger Car Tires. The purpose of SAE J1106 is to define standards for equipment design and test procedures so that data from different laboratories can be directly compared. Whereas such standardization is not a requirement for testing associated with tire development, it is necessary in the context of vehicle design and tire selection problems.

The basic approach employed in developing SAE J1106 was to consolidate and document existing technology as embodied in equipment and procedures currently employed for routine tire evaluations. Equipment and procedures whose current use is restricted to research applications were not considered. Research experience is discussed in this Information Report, however, to the extent deemed necessary to provide background and rationale for SAE J1106. Material is therefore included on speed effects, contaminants dynamic testing, traction, surface geometry, and other subjects not considered in SAE J1106. The scope was expanded in an effort to anticipate questions raised by SAE J1106.

1.1 **Introduction—**The motor vehicle industry is working toward a more complete understanding of the factors contributing to the motions of vehicles on the roadway. This understanding is expressed in the form of a variety of techniques for prediction of vehicle motion responses to road and operator inputs. The ability to predict vehicle motion responses is desirable if the vehicle traffic system is to be controlled and designed for optimum safety and utility.

The pneumatic tire is the primary control element in present ground vehicle systems. Forces and moments developed at the tire-road interface provide the functions of support, directional control, braking and accelerating capability. The manner in which these forces and moments are developed can be influenced through changes in tire design. Tire input-output properties must be completely identified if vehicle motion properties are to be predicted. These tire force and moment properties are analogous to airfoil properties used in the design of aircraft. The tire force and moment tests discussed in this report are, therefore, analogous to wind tunnel tests applied to various airfoil designs and have similar utility.

The report begins with a general discussion of the factors affecting tire forces and moments, and a brief historical review of associated research and testing activities. A discussion is presented of the principal requirements for tire force and moment data as they derive from considerations of vehicle dynamics and tire design. Factors affecting test machine selection and design are reviewed in a broad context ranging from overall system structure to specific subsystem requirements. Methods for testing, data processing, and presentation and interpretation of test results are discussed in detail. The report concludes with a discussion of equipment and procedures for “special test,” not currently performed on a routine basis, but which provide data of significant and growing interest to the vehicle designer.

1.2 Factors Affecting Tire Forces and Moments—The steady state force and moment properties of pneumatic tires may be considered to comprise two phenomenologically distinct categories: (1) the geometric and material properties affecting shear-force potential (that is, friction) at the tire-road-interface, and (2) the elastic properties determining the shape of, and pressure distribution in, the interfacial contact area.

Measurements of frictional tire properties, which are severely confounded by synergistic interactions of tire, road surface, interface contamination (primarily water), temperature, slip and speed effects, must presently be considered to constitute research experimentation. Laboratory procedures most commonly used for routine tire evaluations are restricted to the measurement of elastic tire properties. Specifically, the object of these routine procedures is to characterize the mechanical response of the rolling tire to steady state deformations from its free inflated shape. The deformations to which the tire is subjected in the laboratory and idealizations of the deformations produced on a vehicle’s tires during steady state maneuvers over a perfectly smooth road. For the measurements to be meaningful, it is necessary to establish controlled test conditions which do not differ from over-the-road operating conditions in any aspect which significantly influences the tire’s elastic response. In terms of the specific factors affecting tire performance, this requirement has the following implications:

1.2.1 SPEED—The speed dependence of the steady state elastic properties of pneumatic tires is not great over the range of normal highway driving. There is probably a threshold speed, however, below which a tire’s force and moment response is speed dependent. This threshold speed has not been definitely measured, but several laboratories test successfully at speeds of 1 mph.

1.2.2 TEMPERATURE—Some physical properties of rubber are known to vary with temperature so elastic properties of tires are expected to vary with temperature. The effects of temperature on elastic properties has not been carefully investigated.

1.2.3 ROAD SURFACE—The elastic response of a pneumatic tire is dictated by the compliance of its constituent parts (cords, tread, etc.) and by the relative configuration of these parts under the conditions of external loading. A major determinant of a loaded tire’s configuration is the shape (macrogeometry) of the road surface. It is found experimentally that the distortions produced when a tire is loaded against a curved (either concave or, especially, convex) surface are different from those produced by a flat surface as to exert an influence on the tire’s elastic response. The significance of this influence depends on the degree of curvature. Consequently, laboratory machines employed to measure tire force and moment properties for vehicle design and tire selection purposes frequently incorporate flat, rigid test surfaces.

A secondary determinant of the configuration of a loaded tire is the degree of relative motion (sliding) between tire and roadway surfaces in the interfacial contact patch. For a given loading condition, the degree of relative motion is a function of the interfacial friction coefficient, hence of the microgeometery and material of both the tire and the roadway surface. Thus, the frictional properties and elastic response of the pneumatic tire are not physically independent. While the measurement of frictional tire properties has not yet advanced beyond the realm of research, procedures for controlling the factors influencing friction in the interfacial contact patch are sufficiently well developed that the confounding influence of these factors on measurements of elastic tire response can be effectively eliminated.

1.2.4 CONTAMINANTS—The presence of contaminants (such as water, dust, or oil) at the tire-roadway interface may exert a substantial effect on the interfacial friction coefficient. Great care must accordingly be exercised to eliminate the possibility of spurious interface contamination during tire force and moment testing.

In the sections which follow, the factors mentioned above will be discussed in detail, and relevant experimental data will be presented to illustrate their implications with respect to laboratory measurement of the steady state force and moment properties of tires. First, however, a brief review will be made of some of the major historical developments relative to the characterization and measurement of tire forces and moments.

1.3 Historical Background—Although the pneumatic tire was first invented in 1845, and no less a classicist than Osborne Reynolds examined the rolling behavior of rubber wheels as early as 1876 (1)[1], it was not until 1925 that Broulhiet(2) advanced the concept of slip angle which is the cornerstone idea behind current understanding of the mechanics of tire force and moment generation. The concept arose during Broulhiet's consideration of the automobile shimmy problem. This was also the subject of subsequent investigations in Germany which gave rise to the earliest known measurements of tire side forces in yawed rolling on a drum-type tester, by Fromm(3), in about 1931. More extensive drum measurements of tire forces and moments were performed later in the 1930's by Evans(4) and Bull(5). The latter investigator examined conditions of combined longitudinal and side slip. Early measurements of tire forces and moment generated on flat surfaces were obtained in road tests by Bradley and Allen(6) and Forster(7). Differences between tire test results obtained with flat and curved surface conditions were first documented by Bull(5).

Early theoretical analyses of the forces generated by longitudinally and side slipping tires were performed by Julien(8) and Fromm(9,10). These analyses, which did not consider carcass deformation, nonetheless produced results which tend to agree qualitatively with experimental tire data over the whole spectrum of operating conditions from free rolling to full sliding. The basis for modern analytical treatment of the elastic response of the "running-band" theory, developed independently by von Schlippe(11) and Hadekel(12) which assumes that the tire's elastic characteristics can be determined by considering the deformation of the equatorial line; that is, the intersection of the tire surface, when undeformed, with the wheel center plane. Fiala(13) employed running-band theory, with a model accounting also for the flexibility of tread elements and the presence of sliding in the tire-road contact area, to derive expressions for steady state tire side forces and yawing moments, as functions of normal force, sideslip, and inclination angle, which agree well with experimental data over a broad range of operating condtiions(14,15). More recently, several analysts(16,17,18) have adopted approaches similar to Fiala's for the case where the tire moves with combined longitudinal and sideslip, to derive general tire force and moment characterizing relationships which agree qualitatively with the limited experimental data available for the combined slip condition.

All of the so-called theoretical tire models cited above are in fact semi-empirical; that is, they involve one or more parameters whose values must be obtained by measuring tire forces and moment under prescribed and controlled test conditions. Hence, the development of these theories, rather than obviating the need for tire force and moment testing, actually adds to it.

Recent developments in tire force and moment testing methodology have proceeded along several different lines. Vehicle-towed dynamometers permitting road measurements of tire forces and moments under the broadest range of dynamic and environmental conditions have been developed both in this country(19,20) and abroad(21,22). Modern laboratory equipment for measuring tire forces and moments has evolved primarily from the low speed, "flat-plank" tester designed and built by the Dunlop Tire Company in England(23). Derivative machines have been described in the literature by Nordeen and Cortese(24), Ginn and Marlowe(25), Dugoff and Brown(20) and van Eldik Thieme(22). These flat bed machines are now employed routinely to evaluate tires for vehicle design and development purposes, and it is the technology of their design and use which constitutes the subject of the SAE Recommended Practice J1106.

1.Numbers in parentheses designate References at end of report.

Current research experimentation relative to the mechanics of tire force and moment generation includes measurements with internal drum(23) and continuous belt(26) tire testers. The influence of drum curvature on tire forces and moments is much less pronounced with the internal-track drum than with external-track machines(27). The internal-track machine also has other features (namely, variable test surface material, controllable surface water layer thickness) which are advantageous from the viewpoint of the tire mechanics researcher. The moving belt tester provides the principal advances of drum type equipment (namely, continuous, high-speed operating capability), plus a flat test surface. Its principal shortcoming for research purposes has been difficulty of water layer control for wet surface testing(28) but these problems have been solved in more recent versions.

Much of this technology has been somewhat slow to move from the research laboratory and be adopted as an integral part of the product development process. Better techniques for instrumentation, data processing, and vehicle simulation had to precede wide application of tire force and moment technology. The organization of the industry has also tended to retard this growth. The tire is an integral component of the vehicle and cannot be evaluated out of the context of the vehicle. Tire data were, therefore, of limited usefulness to the tire industry without full knowledge of vehicle properties and defined performance objectives. A more widespread understanding of vehicle dynamics and the activities of several independent research organizations with knowledge of vehicles and tires have combined to overcome many of these problems. The industry appears to be entering a time of rapid expansion in the application of tire force and moment technology.

The Ad Hoc Tire Force and Moment Subcommittee was formed with people experienced in the tire industry, vehicle industry and independent research establishment. After considering alternatives, the Committee agreed that SAE J1106 should be a concise and specific document directed at low speed laboratory testing on high friction surfaces since these tests are regularly conducted by many laboratories and produce results that are useful for design and development. The Committee anticipated that SAE J1106 would generate many questions from workers in the field who are concerned with test validation, speed effects, contaminants, surface geometry, test machine configurations, and other factors considered in Committee deliberations but not included in SAE J1106. The Information Report was drafted by the Committee to provide this additional background. Individuals on the Committee wrote particular sections which were combined with minimum editing. Most of the technical writing was done during the first half of 1973 and is an indication of technology existing prior to that time.

2. References—The following publications form a part of the specification to the extent specified herein. Unless otherwise indicated the latest revision of SAE publications shall apply.

2.1 Applicable Publications—The following publications form a part of the specification to the extent specified herein. Unless otherwise indicated the lastest revision of SAE publications shall apply.

2.1.1 SAE PUBLICATIONS—Available from SAE, 400 Commonwealth Drive, Warrendale, PA 15096-0001.

SAE J670—Vehicle Dynamics Terminology
SAE J1106—Laboratory Testing Machine and Procedures for Measuring the Steady Force and Moment Properties of Passenger Car Tires

2.2 Related Publications—The following publications are provided for information purposes only and are not a required part of this document.

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3. Reference Axes and Terminology

3.1 Introduction—Understanding of terminology, a common axis system, and a consistent set of sign conventions are required for any discussion of tire force and moment evaluation methodology. Material from the SAE Vehicle Dynamics Terminology are reproduced here for reference purposes.

3.2 Tire Axis System—The selection of the Tire Axis System described in SAE Vehicle Dynamics Terminology J670 is probably not obvious to many vehicle engineers who prefer to think of tire forces and moments in terms of axes oriented in the wheel spindle or kingpin. Forces and moments in these alternative axis systems can be easily derived from complete data reported in the reference system with relatively simple transformations if the orientation of these alternative axes is known.

Since the motions of spindle and kingpin axes are complicated and unique to particular vehicles and situations, there systems are less suitable for reporting of reference data on tire performance. Experience has shown that complete data recorded with the tire axis system have been generally applicable to analyses of vehicle motions and component stresses. The problem of axis transformation is one of the reasons for measuring complete, six component, data even though some of the individual force and moment components are of limited immediate interest.

Terminology and sign conventions consistent with SAE Vehicle Dynamics Terminology J670 should be used for all tire force and moment testing. The othogonal coordinate system shown in the figure and the following definitions form the basis for a force and moment testing vocabulary.

These definitions are based on the current proposed revision to SAEJ670.

3.3 Rolling Characteristics

3.3.1 LOADED RADIUS—(R) is the distance from the *center of tire contact* to the *wheel center* measured in the *wheel plane*.

3.3.2 STATIC LOADED RADIUS—The loaded radius of a stationary tire inflated to normal recommended pressure.

NOTE—In general, static loaded radius is different from the radius of a slowly rolling tire. Static radius of a tire rolled into position may be different from that of the tire loaded without being rolled.

3.3.3 SPIN AXIS—The axis of rotation of the wheel. (See Figure 1.)

3.3.4 SPIN VELOCITY—(Ω) The angular velocity of the wheel on which the tire is mounted, about its *spin axis*. Positive *spin velocity* is shown in Figure 1.

3.3.5 FREE ROLLING TIRE—A loaded rolling tire operated without application of *driving* or *braking torque*.

3.3.6 FREE STRAIGHT ROLLING TIRE—A *free rolling tire* moving in a straight line at zero *inclination angle* and zero *slip angle*.

3.3.7 LONGITUDINAL SLIP VELOCITY—The difference between the *spin velocity* of the driven or braked tire and the *spin velocity* of the *free straight-rolling tire*. Both spin velocities are measured at the same linear velocity at the wheel center in the X' direction. A positive value results from *driving torque*.

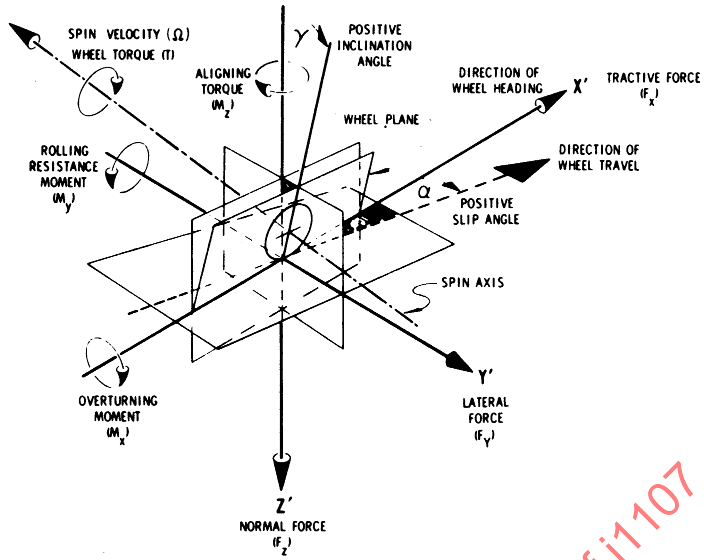


FIGURE 1—TIRE AXIS SYSTEM

3.3.8 LONGITUDINAL SLIP (PERCENT SLIP)—The ratio of the *longitudinal slip velocity* to the *spin velocity* of the *free straight rolling tire* expressed as a percentage.

NOTE—This quantity should not be confused with the slip number that frequently appears in kinematic analysis of tires in which the *spin velocity* appears in the denominator.

3.3.9 EFFECTIVE ROLLING RADIUS—(R_e) is the ratio of the linear velocity of the *wheel center* in the X' direction to the *spin velocity*.

3.3.10 WHEEL SKID—The occurrence of sliding between the tire and road interface which takes place within the entire contact area. Skid can result from braking, driving and/or cornering.

3.4 Tire Forces and Moments

3.4.1 TIRE AXIS SYSTEM (FIGURE 1)—The original of the tire axis system is the *center of tire contact*. The X' axis is the intersection of the *wheel plane* and the road plane with a positive direction forward. The Z' axis is perpendicular to the road plane with a positive direction downward. The Y' axis is in the road plane, its direction being chosen to make the axis system orthogonal and right-hand.

3.4.2 TIRE ANGLES

3.4.2.1 Slip Angle—(α) The angle between the X' axis and direction of travel of the *center of tire contact*.

3.4.2.2 Inclination of Angle—(γ) The angle between the Z' axis and the wheel plane.

3.4.3 TIRE FORCES—The external force acting on the tire by the road having the following components:

3.4.3.1 Longitudinal Force (F_x)—The component of the *tire force* vector in the X' direction.

3.4.3.2 Driving Force—The *longitudinal force* resulting from *driving torque* application.

3.4.3.3 *Driving Force Coefficient*—The ratio of the *driving force* to the *vertical load*.

3.4.3.4 *Braking Force*—The negative *longitudinal force* resulting from *braking torque* application.

3.4.3.5 *Braking Force Coefficient (Braking Coefficient)*—The ratio of the *braking force* to the *vertical load*.

3.4.3.6 *Rolling Resistance Force*—The negative *longitudinal force* resulting from energy losses due to deformations of a rolling tire.

3.4.3.7 *Rolling Resistance Force Coefficient (Coefficient of Rolling Resistance)*—The ratio of the *rolling resistance force* to the *vertical load*.

3.4.3.8 *(Lateral Force (F_y)*—The component of the *tire force* vector in the Y' direction.

3.4.3.9 *Lateral Force Coefficient*—The ratio of the *lateral force* to the *vertical load*.

3.4.3.10 *Slip Angle Force*—The *lateral force* when the *inclination angle* is zero and plysteer and conicity forces have been subtracted.

3.4.3.11 *Camber Force (Camber Thrust)*—The *lateral force* when the *slip angle* is zero and the plysteer and conicity forces have been subtracted.

3.4.3.12 *Normal Force (F_z)*—The component of the *tire force* vector in the Z' direction.

3.4.3.13 *Vertical Load*—The normal reaction of the tire on the road which is equal to the negative of *normal force*.

3.4.3.14 *Central Force*—The component of the *tire force* vector in the direction perpendicular to the direction of travel of the *center of tire contact*. *Central force* is equal to *lateral force* times cosine of *slip angle* minus *longitudinal force* times sine of *slip angle*.

3.4.3.15 *Tractive Force*—The component of the *tire force* vector in the direction of travel of the *center of tire contact*. *Tractive force* is equal to *lateral force* times sine of *slip angle* plus *longitudinal force* times cosine of *slip angle*.

3.4.3.16 *Drag Force*—The negative *tractive force*.

3.4.4 TIRE MOMENTS—The external moments acting on the tire by the road having the following components:

3.4.4.1 *Overturning Moment (M_x)*—The component of the *tire moment* vector tending to rotate the tire about the X' axis, positive clockwise when looking in the positive direction of the X' axis.

3.4.4.2 *Rolling Resistance Moment (M_y)*—The component of the *tire moment* vector tending to rotate the tire about the Y' axis, positive clockwise when looking in the positive direction of the Y' axis.

3.4.4.3 *Aligning Torque (Aligning Moment) (M_z)*—The component of the *tire moment* vector tending to rotate the tire about the Z' axis, positive clockwise when looking in the positive direction of Z' axis.

3.4.4.4 *Wheel Torque (T)*—The external torque applied to the tire from the vehicle about the spin axis; positive *wheel torque* is shown in Figure 1.

3.4.4.5 *Driving Torque*—The positive *wheel torque*.

3.4.4.6 *Braking Torque*—The negative *wheel torque*.

3.5 Tire Force and Moment Stiffness—(May be evaluated at any set of operating conditions).

3.5.1 CORNERING STIFFNESS—The negative of the rate of change of *lateral* force with respect to change in slip angle, usually evaluated at zero *slip angle*.

3.5.2 CAMBER STIFFNESS—The rate of change of *lateral force* with respect to change in *inclination angle*, usually evaluated at zero *inclination angle*.

3.5.3 BRAKING (DRIVING) STIFFNESS—The rate of change of *longitudinal force* with respect to change in *longitudinal slip*, usually evaluated at zero *longitudinal slip*.

3.5.4 ALIGNING STIFFNESS (ALIGNING TORQUE STIFFNESS)—The rate of change of *aligning* torque with respect to change in *slip angle*, usually evaluated at zero *slip angle*.

3.6 Normalized Tire Force and Moment Stiffness (Coefficients) Cornering Stiffness Coefficient (Cornering Coefficient)—The ratio of *cornering stiffness of a free straight-rolling tire* to the *vertical load*.

NOTE—Although the term cornering coefficient has been used in a number of technical papers, for consistency with definitions of other terms using the word coefficient, the term *cornering stiffness coefficient* is preferred.

3.6.1 CAMBER STIFFNESS COEFFICIENT (CAMBER COEFFICIENT)—The ratio of camber stiffness of a *free straight-rolling tire* to the *vertical load*.

3.6.2 BRAKING (DRIVING) STIFFNESS COEFFICIENT—The ratio of *braking (driving) stiffness* of a *free straight-rolling tire* to the *vertical load*.

3.6.3 ALIGNING STIFFNESS COEFFICIENT (ALIGNING TORQUE COEFFICIENT)—The *ratio of aligning stiffness* of a *free straight-rolling tire* to the *vertical load*.

3.7 Tire Traction Coefficients

3.7.1 LATERAL TRACTION COEFFICIENT—The maximum value of *lateral force coefficient* which can be reached on a *free-rolling tire* for a given road surface, environment and operating conditions.

3.7.2 DRIVING TRACTION COEFFICIENT—The maximum value of *driving force coefficient* which can be reached on a given tire and road surface for a given environment and operating conditions.

3.7.3 BRAKING TRACTION COEFFICIENT—The maximum value of the *braking force coefficient* which can be reached without locking a wheel on a given tire and road surface for a given environment and operating conditions.

3.7.3.1 Sliding Braking Traction Coefficient—The value of the *braking force coefficient* of a tire obtained on a locked wheel on a given tire and road surface for a given environment and operating condition.

4. Vehicle Dynamics and Tire Design Considerations

4.1 Introduction—Since the tire is a component of a vehicle, any tire performance evaluation must be related to its influence on vehicle performance. Vehicle factors must be considered in selection of components to be measured, accuracy requirements, and other test conditions. Some of these vehicle considerations are summarized in this section.

4.2 **Applicability of Data—**Vehicle directional control is of primary importance to automobile designers. With advances in instrumentation and computer technology, some sophisticated vehicle directional simulation procedures have developed. That tire forces and moments affect directional control is well documented(24). This is emphasized in reports comparing radial to bias ply tires. Gough(29) wrote on the difference in aligning torque and cornering stiffness between radial and bias ply tires and how this difference affects handing behavior. Nordeen, Bidwell and Rasmussen(30) reported on a test for a complete set of tire force and moment data. They interpreted the data to show relationships between force and moment variables and describe how the variations affect vehicle directional responses.

Tire force and moment data have aided in tire development and evaluation. The recent evaluation of the radial tire is an example of this area of application. The literature contains numerous reports describing how the force and moment properties of the radial tire make it different from bias belted tires with respect to vehicle handling. Nordeen(31) has published a very comprehensive report on this area of application.

Tire force and moment data have helped in the understanding of the physical principles of tire operation over a very broad range of conditions. Gough(29) describes aligning torque and cornering stiffness during transient cornering maneuvers. Peterson and Rasmussen(32) show the frequency dependence of vertical and longitudinal force variations in a steady state operating environment. Bidwell(33) reports on the coupling between lateral force and tractive force in braking or accelerating conditions.

The evaluation of road roughness is another area where tire force and moment data are applied. Butkunas(34) describes techniques for measuring the vehicle road interface transfer function for the vertical force direction. Lippmann(35,36) has demonstrated that an increase in vertical and longitudinal force will result when the tire envelopes small amplitude bumps or pot holes in the road surface.

4.3 **Accuracy Requirements—**Tire force and moment measurements should be within prescribed accuracy ranges. In those cases where the force and moment data are used solely for tire development, the measurements may serve as ends in themselves; and, while maximum accuracy is always desirable, a nominal 5% accuracy range may be tolerable, especially if only approximate numbers are sought. However, in the automotive industry, tire force and moment data are often used as major inputs to mathematical models of vehicle response simulations, and experience has shown that errors in tire force and moment measurements can be multiplied to produce significant errors in vehicle response simulation. Table 1 shows the vehicle response error multiplication resulting from a worst case combination of 5% errors in tire force and moment measurements.

TABLE 1—EFFECT OF 5% ERRORS IN TIRE DATA ON COMPUTED VEHICLE RESONSE PROPERTIES

	Typical Car	Worst Case 5% Error in Tire Data	% Change In Response
Understeer	6.1°/g	5.2°/g	17
Steering Sensitivity (Front Wheel Angle)	8.1°/g	6.9°/g	15
Lateral Acceleration Response Time	0.49 s	0.53 s	8
Yaw Velocity Response Time	0.15 s	0.17 s	13

It can be seen from the Table 1 that errors in force and moment data can be multiplied by a factor of 3 to produce large errors in vehicle response simulation. This error multiplication would have serious negative consequences for the chassis design engineer. Obviously the force and moment data must fall within prescribed accuracy ranges.

In order to obtain an acceptable overall system simulation accuracy, the accuracy of measurement of each component subsystem must be considered. If an overall system simulation accuracy of 1% full scale is required, subsystem measurement accuracies of less than 1% must be achieved.

Overall system measurement inaccuracies can arise from several sources. The following list represents the major causes of tire force and moment measurement variations.

1. Inaccuracies in the instrumentation
    a. Single transducer inaccuracy
    b. Transducer channel interactions
2. Test repeatability inaccuracies

TABLE 2—ACCURACY REQUIREMENTS

	Transducer Range	Accuracy After System Compensation	% Full Scale Accuracy
Normal Force	–4000 lb (18000 N)	10 lb (44 N)	0.25%
Lateral Force	± 4000 lb (18000 N)	10 lb (44 N)	0.25%
Longitudinal Force	± 200 lb (900 N)	1 lb (44 N)	0.5%
Aligning Torque	± 500 lb-ft (700 N·m)	2 lb-ft (2.7 N·m)	0.4%
Overturning Moment	± 1000 lb-ft (1400 N·m)	10 lb-ft (14 N·m)	1.0%
Rolling Resistance Moment	± 200 lb-ft (270 N·m)	1 lb-ft (1.4 N·m)	0.5%
Slip Angle	± 30°	± 0.05°	0.17%
Inclination Angle	± 15°	± 0.05°	0.33%
Rolling Radius	7.5-16 in (191 mm)	0.05 in (1.3 mm)	0.31%

3. Tire force and moment property variation
    a. Individual tire nonuniformities
    b. Tire to tire variations

An overall system accuracy of 2% is a goal if tire force and moment data are to be meaningfully applied to vehicle dynamics considerations.

The electronic instrumentation must be designed to have a minimal effect on overall system inaccuracy. To this end SAEJ1106 gives the acceptable limits of measurement variations for the control parameters and for the output force and moment measurements. It should be noted that these limits are expected to be achieved after transducer channel interaction, or crosstalk, has been reduced by linear compensation. Engineers have found it possible to reduce channel interaction inaccuracies from 5% to less than 0.2% using linear compensation techniques.

The acceptable limits for measurement variations are given in Table 2.

Good instrumentation design and frequent calibration checks are necessary in order that the measurements stay within acceptable limits of accuracy. It must be recognized, however, that variations in tire force and moment data will arise when conducting repeated tests of the same tire or when conducting single test on each of many tires of the same brand and construction. The data variation due to these sources is less easy to control, necessitating a judicious interpretation of the test data. It is felt that while a nominal overall system accuracy of 2% is required, it is realistic to interpret the data as being only 4% accurate.

It has been found that tire force and moment variations have occurred as the same tire is tested repeatedly. In those cases where the tire-wheel assembly is taken off the test fixture or, further, where the tire is removed from the wheel, measurement variation can arise due to different tire-wheel assembly/test fixture or tire/wheel orientations. Part of this source of variation can be removed if care is taken to index tire, wheel and test fixture orientations.

If a particular tire undergoes a large number of force and moment test or experiences significant usage between tests, force and moment data may show appreciable variation due to tread wear. Rasmussen and Cortese(37) have reported that many tire properties are significantly changes as tread depths are reduced by usage. The wet traction properties may undergo significant change; and while the effect of reduced tread depth on dry road force and moment performance is less well known, Rasmussen and Cortese have shown that the magnitudes of the lateral force and the aligning torque are significantly increased as the tire goes from a new to a half-worn condition.

If the inflation pressure is not carefully monitored during testing, variations in the force and moment data can be expected. The relationships between tire properties and inflation pressure have been well reported in the tire and automotive industries. Nordeen(31) has reported that a 2 psi variation in inflation pressure is detectable by a chassis engineer, and he shows the effect of a 4 psi inflation pressure change on the relationship between lateral force and slip angle.

Design engineering groups do not usually design a vehicle for operation on a single brand tire; rather, it must accept several brands of original equipment tires. It has been found that a range of tire properties exist for different brands of OE tires rather than unique values(31). This must be taken into account when tire force and moment data are used as input to vehicle handling simulations.

Even if precautions are taken to minimize data variation due to inflation pressure or tire-wheel assembly mountings between tests, there exist measurement variations when different tires representing the same original equipment brand and tire construction are tested. The B. F. Goodrich Company has encountered tire to tire variability in the measurement of aligning torque and cornering coefficients. Table 3 represents their findings for variation between 70 tires in 17 repeated tests.

TABLE 3—TEST VARIABILITY

	Standard Deviations (% of Mean) Cornering Coefficient	Standard Deviations (% of Mean) Aligning Torque Coefficient
Tire to Tire	3.1%	6.0%
Test to Test	2.8%	2.4%

Tests conducted at the Goodyear Tire Company on eight tires representing the same type of construction have shown that when ply steer variations are removed, a lateral force difference of 5% between tires can be expected at a 1 deg slip angle. An aligning torque variation of 25% was found at 1 deg slip angle and 10% at 4 deg slip angle.

Tire production variations, or tire uniformity properties, will also have an effect on the accuracy of tire force and moment data. Curtiss has found that at small sip angles the peak-to-peak variation in forces and moments can cause form 20-30% error in force and moment measurements. For this reason, it is necessary that tire intrinsic nonuniformities be compensated for by suitable averaging and filtering techniques.

4.4 Tire Symmetry and Non-Symmetry—An important consideration in tire force and moment testing is the fact that forces can be generated during tire rotation due to either mass, stiffness or dimensional non-symmetries of the tire. These are commonly called tire nonuniformity effects and are a result of manufacturing non-uniformity or intentional design. Manufacturing nonuniformities can generate significant radial first harmonic (38) and lateral first harmonic forces. Non-symmetries due to fabrication and design are “conicity” and “ply steer.”

There is a possible source of non-symmetrical force generation other than those intrinsic to the tire. If the weighing system, or mechanism supported on the measurement transducers, is of appreciable inertia and located where motion is experienced, inertial forces will be generated 939).

The effects of intrinsic tire non-symmetrical forces and forces due to weighing system interaction should be separated from and not included in the force and moment characteristic data. In order to correct for conicity and ply steer forces, it is necessary to test by steering and rotating in both directions for at least three conditions and solving for tire conicity force, ply steer force and apparent machine camber. Machine steer due to misalignment cannot be separated from tire ply steer force.

5. Factors Affecting Test Machine Section

5.1 Introduction—Fundamental requirements for a tire force and moment machine are a tire mounting and positioning system to generate steering modes, a running surface for the tire and a system to measure the resulting forces and moments.

The specific approach selected is based on the intended application of the acquired data. A method for rank ordering tires on a comparative basis can be different than that to provide data for direct application to vehicle directional simulations. Likewise, equipment to measure only forces and moments at small slip angles at minimum load variation might be designed differently from equipment to provide data through a range of slip angles and loads that could be expected in all modes of vehicle operation.

There have been many types of force and moment machines built and utilized for development purposes. These machines have ranged from extremely simplified devices to some newer devices of great complexity and sophistication. All can be considered useful in the study of tire forces and moments if care is taken in understanding equipment limitations and data application.

5.2 Test Machine Location—In consideration of the basic design approach to be taken, the first question is whether the facility should be laboratory or outdoor equipment. At first, an outdoor facility would seem to have a very distinct advantage of having a wide variety of road surfaces on proving grounds and public highways for testing.

Close consideration of the frictional characteristics of public roads and even specially constructed skid pads shows that existing road use is actually a disadvantage. A great deal of literature is devoted to the problems of maintaining road surface frictional characteristics (40). Contaminates such as dusty, oil, moisture, loose material, road roughness and variations in degree of pavement polish all lead to relatively poor data collection and repeatability. These problems can be reduced by repeated tests and statistical data analysis, but the resolution of small differences is limited. Data exemplifying these variations have been reported by Gengenbach(41).

Temperature variations is another disadvantage of an outdoor location. Not only does temperature affect the frictional characteristics of the road surface, but it also affects the relative stiffness of the tire and possibly the transducer and signal conditioning equipment used for data collection. This, of course, adds another dimension to the accuracy and resolution problem.

Because of road surface effects on low slip angle resolution and the potential variations that could occur at high slip angle on skid conditions due to frictional variation of roads, outdoor testing does not seem attractive. The requirements for tire evaluations can better be satisfied by testing under idealized and controlled conditions such as those provided in a laboratory environment.

5.3 General Arrangement of Machine—The next basic design consideration is the relative movement of the tire and road surface. Here we have the option of moving the tire over the road surface or moving the test surface past a fixed tire.

5.3.1 FIXED ROAD SURFACE-MOVING TIRE—This type of arrangement offers several alternatives. The first could be a tire test stand moved on wheels or rails in a straight line over a flat surface. The length of travel would have to accommodate acceleration of the tire carriage up to test speed, acquisition of data at a constant speed and then deceleration of the test stand. This could be a real disadvantage, especially in consideration of a laboratory facility. In most cases the time utilized for data acquisition would be very short due to space considerations. This would greatly reduce test efficiency and accuracy: the short test time might eliminate the ability to measure forces and moments during a complete revolution. A device of this nature has been built and used by NASA and others.

Rather than running the tire in a straight path, the tire could be run in a circular path, such as around a disc. The advantage of this system is a closed loop test cycle so that efficiency is very good with no limitation in test time duration. However, the relative curvature of the road surface path under the tire distorts the tire footprint which induces other forces and moments, even at zero slip and camber angles. To minimize this distortion, a circular path would have to be made large enough; that is, 120 ft (36.6m) radius which, though feasible, is not practical.

Both of these systems have other disadvantages. Since movement of the tire will also mean movement of the transducers and measuring system, inertial errors can be introduced, as well as errors caused by transmission of signals through slip rings or rotary transformers. These can be quite serious in light of the accuracies required in measurement.

5.3.2 FIXED TIRE-MOVING ROAD SURFACE—The most obvious choice for this type of general machine configuration would be to use a dynamometer wheel for the test surface. Flywheels or drums have been the standard test machine approach in the industry for years, mainly due to their test efficiency and test mode flexibility.

The next choice for a moving road surface would be a flat one. This, of course, is the most ideal situation. Two approaches can be made; a flat bed roadway of finite length which is moved under the tire; a continuous belt which forms a flat running surface under the tire. The first type of machine has been built and used by General Motors, B. F. Goodrich, the University of Michigan (24,25) and others for a number of years. This type of equipment is reliable in terms of flat surface measurements. The main disadvantages are productivity and limitations in test modes that can be used to the finite length of the roadway and the start and stop test cycle.

The continuous belt-type surface offers the advantage of a continuous operating surface where time for a complete test cycle can be minimized and there are no limitations on the types of test modes (stepped, swept, sinusoidal) that can be run. Existing equipment has shown the feasibility of this machine design approach(42,43,44).

5.4 Tire Positioning System—The function of this system is the application of vertical load to the tire against the running surface and of slip (steering) angle and inclination (camber) angle.

The positioning system must be designed with two basic considerations: the mode of test operation and the techniques used for measurement of the forces and moments.

Alternatives in setting slip and inclination angles include fixed position with manual settings of input parameters, fixed position with servo control of input parameters, and dynamic testing based either on manual input or servo control.

The simplest machine would consist of load application by dead weight load and manually setting the tire to the desired slip and inclination angles and locking in position. The disadvantages of this system are very low productivity due to test set-up time and lack of capability for running transient tests.

In a servo control system where the test input parameters are set automatically based on output data selection and sequence, selection of the variable input is important. The choice to design the machine with all variables being controlled (vertical load, slip angle, inclination angle) is difficult since structural rigidity and minimum frame coupling are critical to obtaining accurate data. Of the three potential variables, inclination angle is the most likely candidate for setting and locking while the other two can be variably controlled. Successful machines have been built with servo control of all modes, however(26).

5.4.1 TEST MODES—The test sequence for servo controlled systems can include fixed point testing (set all three variables and readout data), ramp-type input (that is, sweeping slip angle or vertical load) or an oscillating input at a constant sinusoidal frequency. Steady state testing implies that during a test period when data are being collected the independent variable remains fixed. The simplest way for doing this is on a fixed point basis, resting at each test value long enough to record data. This time period is based on the relaxation length of the tire and the nonuniformities of the tire. Published data(24) show that for a set slip angle condition, a steady state lateral force is reached in approximately one-half of a wheel revolution. since the desired output data for force and moment measurement are the d-c levels of the tire generated force, the lateral force variation of the tire also has to be considered. Data for one revolution of the tire should be averaged.

Fixed point testing has the main disadvantage of being time consuming. More data can be taken in a limited time if the independent variable is swept with time. Care must be taken in establishing sweep rates low enough to minimize errors due to dynamic characteristics of the test system and the tire. This type of testing is especially important when it is desired to minimize effects of temperature and wear which might occur in the fixed test mode. High slip angle sweep rates will minimize wear at a given test velocity, but it will also reduce the accuracy of force and moment measurement. Work by Segel(44) indicates lateral force as a function of load or slip angle is roughly a first order time lag in which the time constant is velocity dependent and equal approximately to the relaxation length of the tire divided by the velocity. The maximum permissible sweep rate is then approximately determined by the allowable error in tire lateral force at a given value of slip angle or load.

Criteria for independent variable sweep rates which assure near steady state forces and moments have not been firmly established. Complete enough knowledge of tire relaxation characteristics does not exist to allow a closed form solution of this problem. Current practice at most laboratories is to use empirical methods to establish acceptable sweep rates.

The third type of test mode is transient or dynamic testing. A positioning system for transient testing requires low inertia with rigid support which will probably limit its load range capability. This type of testing is usually only applicable when speed capability is available on the machine for direct vehicle simulation. There has not been much experimental work to date, but the new Calspan machine will have this capability along with existing dynamometers at B. F Goodrich and Uniroyal.

Sinusoidal input of independent variables can be used for steady work, especially if high machine speeds are used, but excessive test time would cause rapid tire wear and temperature changes. Published work939,44) has shown that to obtain lateral force and aligning torque responses equivalent to 99% of the steady state value, the tire spatial frequency must be approximately 0.016 c/ft (0.053 c/m) or less. At 60mph (98 km/h), the maximum slip angle rate would have to be 500 deg/s or 1.4 Hz. Careful consideration of hardware design should always be made when considering this type of testing in light of these frequency response constraints for accurate tire testing and potential machine resonance interactions.

5.5 Size and Independent Variable Considerations—The requirements for sizing the machine are based on the range of tire sizes to be tested. Present passenger tire sizes range from an “A” size tire on a 13 in nominal rim diameter to an “N” size on a 15 in rim. Future automotive requirements based on smaller car development work could extend the low end of the range to smaller cross section tires (approximately 5 in (127 mm) and rim diameters less than 13 in. The high end of the range could increase to 16 in diameters for increased brake capacity. The maximum tire width should be based on the widest tire available on the market which will be an N50-14 size. The loaded radius or spindle height above the test interface is based on this tire size range with the added consideration of deflection due to maximum expected test load.

From these tire size considerations, the machine should be capable of testing tires up to 32 in (813 mm) in diameter, 13 in (330 mm) in width and handle rims from below 13 in (330 mm)—possibly 10 in (254 mm)—to a maximum of 16 in (406 mm) in diameter. Rim width would be a minimum of 4 in (102 mm) wide to a maximum of 9 in (221 mm) for passenger tire testing. Since the machine is to be designed to accurately measure *tire* forces only, the wheels should be a special heavy-duty construction for minimum wheel deflection. The wheels should also be uniform to minimize wheel induced nonuniformities. This is important because the forces that will be used as data output are based on the average of the forces that will be used as data output are based on the average of the force variations through one revolution of the tire. Any distortion in the tire forces due to the wheel could give erroneous data.

The total tire envelope required for the machine should be based on the above dimensions and providing space for the tire to be positioned through ± 30 deg slip angle and ± 15 deg inclination angle.

5.5.1 SLIP ANGLE—The ranges of the independent test variables should be based on vehicle operation. In the case of slip angle the usual range falls within 5 deg; however, in the case of accident avoidance maneuvers, angles are reached where pure sliding occurs. A great deal of data has shown that once slip angles of between 20 and 30 deg are reached, the lateral force is mainly a function of the frictional characteristics of the tire/road interface. A 30 deg angle ensures that the data will be obtained at the frictional limit. Angle sweep in both directions (+ and –) should be designed into the machine to permit averaging of asymmetries in the tire and to prevent wear conditions that would bias the data. Angle adjustment capability should be in 1 deg increments (or less) up to 4 deg, because of the critical nature of this range, and no longer than 4 deg increments at the higher slip angles (above 12 deg).

5.5.2 INCLINATION ANGLE—The operating range of inclination angle on vehicles is up to 10-5 deg under some extreme maneuvers. Machine range should include both plus (+) and minus (–) sweep direction again from considerations of asymmetry averaging and wear biasing and should be in 1 deg increments (or less) for the full range.

5.5.3 LOAD—The load range for the machine is based on the tire sizing range and the variation in load as seen on the vehicle. Load transfer from the inside wheels to the outside wheels caused by lateral acceleration is important to vehicle dynamics. Since the amount of load transfer depends on a particular vehicle’s suspension, dimensions and location of the center of gravity as well as lateral acceleration levels, it is important that a standard load range be chosen for testing which will cover the loading and unloading conditions in vehicle turning. Vehicle work has shown that a range of 40-160% of the tires rated load covers these conditions. Since data will often be plotted as a function of load, a maximum of 400 lb (1780 N) increments and no less than five increments over the load range should be used to ensure accurate interpolation.

5.5.4 INFLATION PRESSURE—Inflation pressure can have a significant effect on lateral forces and aligning torque due to its stiffening of the tire structure. A variation of one psi can vary cornering stiffness by 5% and camber stiffness by 3%(24). For this reason it is important that inflation be controlled accurately during a test cycle and that standard settings be used for data comparison. The base inflation for the rated load of a passenger tire is established as a cold setting. During road operation, the temperature buildup of the tire causes an increase in pressure of about 4 psi (27.6kPa). This “hot” inflation would, therefore, be the actual running condition of the tire, and force and moment data generated at the “hot” condition would be more representative of vehicle conditions. The inflation pressure for testing should, therefore, be based on the recommended cold inflation for rated load conditions of the tire plus 4 psi (27.6 kPa) for low speed testing.

5.6 Test Surface—Width and length of the test surface must be considered. Thickness is also important if the belt’s structural stiffness is a factor in preventing distortion and lateral movement which could cause erroneous readings for the tire lateral force. The surface width is based on the maximum tire tread width and contact length which will be tested.

The surface width is based on the rotation of the rectangle formed by the widest tread width and footprint length to be tested through the maximum possible slip angle. Some additional test width is needed due to lateral deformation of the tire which occurs during cornering and inclination. For the range of tire sizes previously mentioned, a width of 18 in should be adequate to meet this criteria.

The length requirements are based on a minimum rolling distance for obtaining steady state force data and suitable averaging of tire nonuniformity.

The distance dependence of the lateral force buildup is due to the basic mechanics of the tire behavior. When a tire is rolled at a fixed slip angle, the tread and carcass are deformed laterally. The amount of lateral deformation increases for a given tread element as the tire rolls and the element passes from initial road contact back through the contact length. At some point it reaches a maximum deflection after which it slips back toward the wheel plane.

The force due to this elastic lateral deformation is the lateral force of the tire. Starting from rest, a finite rolling distance is required to develop the steady state lateral force for a given slip angle. It has been shown that this distance is approximately one-half a tire circumference.

In addition, there are variations in tire forces and moments, depending upon which portion of the circumference of the tire is in the contact patch. Because this variation changes the force level during one revolution, one complete revolution should be made to obtain a mean value of the varying lateral force. The total minimum length for accurate, steady state data should, therefore, be one and one-half revolutions of the tire.

6. Road Simulation**—**There are many factors which affect the forces and moments developed by rolling or sliding tires. This section is concerned with roadway and tire-roadway interface factors; in particular, surface (texture, curvature, condition), speed and temperature. These discussions are based on the best data available at the time of writing. Some of the questions raised are not yet resolved and await a thorough investigation.

The effects of test speed and surface curvature are discussed in some detail. These discussions are based on the best data available at the time of writing. But these questions are not entirely resolved. Testing for the effects of curvature requires that speed, surface texture, and weighing system characteristics be held constant. Only the new Calspan high speed belt machine is capable of this type of testing. Testing for speed effects is complicated by tread temperature changes and treadwear. All of these phenomena await a thorough investigation.

6.1 Environmental Factors—One of the potential advantages of testing within the laboratory is that the environment and test surface conditions are subject to measurement and control so that comparative data for different tires or different operating conditions will not be confounded by changes in these test factors. The importance of these factors cannot be overemphasized as may be illustration by consideration of some of the variability encountered in testing out-of-doors.

The notion that roads possess some unique friction coefficient has been discussed by Kummer and Meyer (reference 40). Force transmission between a tire and either a wet or dry road is a frictional process which involves the road, the tire and any interface contaminates (water, for example). Hence, the terms pavement (or tire) friction coefficient, slipperiness, etc., are misleading since they are not pavement (or tire) properties alone.

Any dry pavement free of dust and loose material will produce high skid or slip resistance with normal tires, regardless of speed. When a surface is wet, however, the ski or slip resistance is significantly reduced as the sliding speed increases. Data from steady state skid test on wet and dry Portland cement concrete pavements are shown in Figure 2.

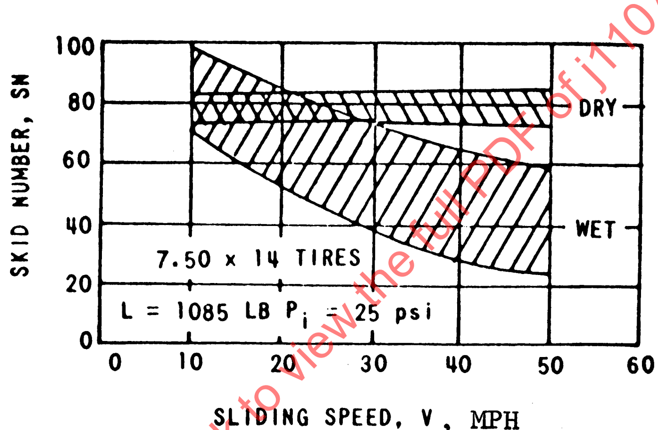


FIGURE 2—STEADY STATE SKID TESTS WITH DIFFERENT TIRES ON DRY AND WET CEMENT CONCRETE SURFACE (40)

NOTE— Most of the figures in this text have been taken from previously published literature. It was impractical to provide SI units for most of these figures due to space limitations. The figures are, therefore, reproduced as originallly published without SI units.

The macroscopic and microscopic roughness of pavement surfaces have large effects on wet skid resistance. Sharply tipped, unpolished aggregate of adequate void width produces a high sip and skid resistance, whereas rounding of the tips due to traffic polishing drastically reduces the resistance. This pattern is frequently found on four-lane highways where the traffic lane and the passing lanes will have quite different characteristics. Figure 3 shows skid number characteristics as a function of sliding speed for various wet surface types.

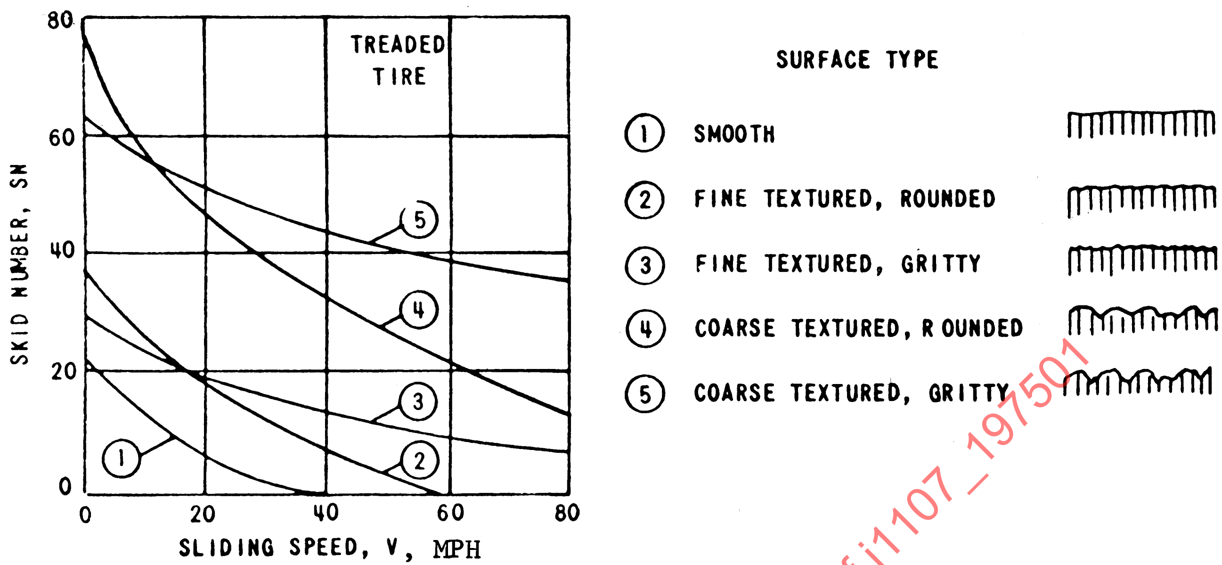


FIGURE 3—CLASSIFICATION OF PAVEMENT SURFACES ACCORDING TO THEIR FRICTION AN DRAINAGE PROPERTIES (40)

Temperature has been found to be a significant factors influence steady state skid resistance. This is shown in Figure 4, where both temperature and skid number are shown over a 24 h period. During this period a 39 °F (-1.1 °C) temperature change (from 70 ° (21.1 °C) to 100 ° (37.8 °C)) produced a decrease in skid number of nearly 20%. The skid number dependence on temperature is further illustrated in Figure 5. Here the change in British Pendulum number[1] from a reference condition at 70°F (21.1°C) is shown as a function of temperature.

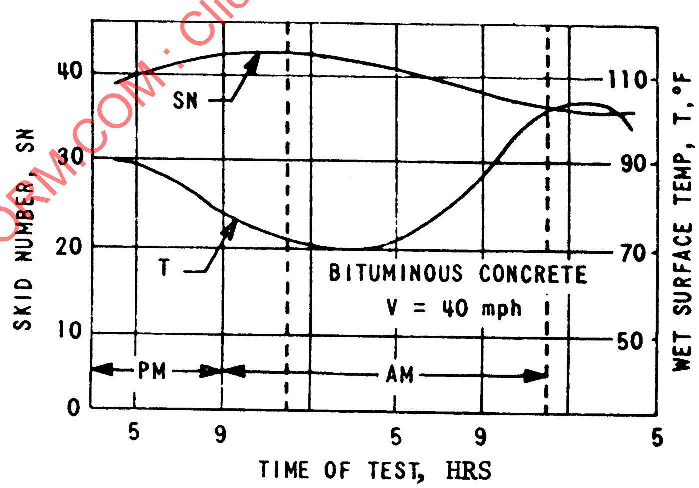


FIGURE 4—INFLUENCE OF HOURLY CHANGES IN WET SURFACE TEMPERATURE ON STEADY STATE SKID NUMBER (40)

1.The British Pendulum Number (BPN) is derived from the British Portable Tester, a device for determiing skid resistance properties of pavements. For the present purposes, the BPN may be considered roughly equivalent to skid number. (reference 40)

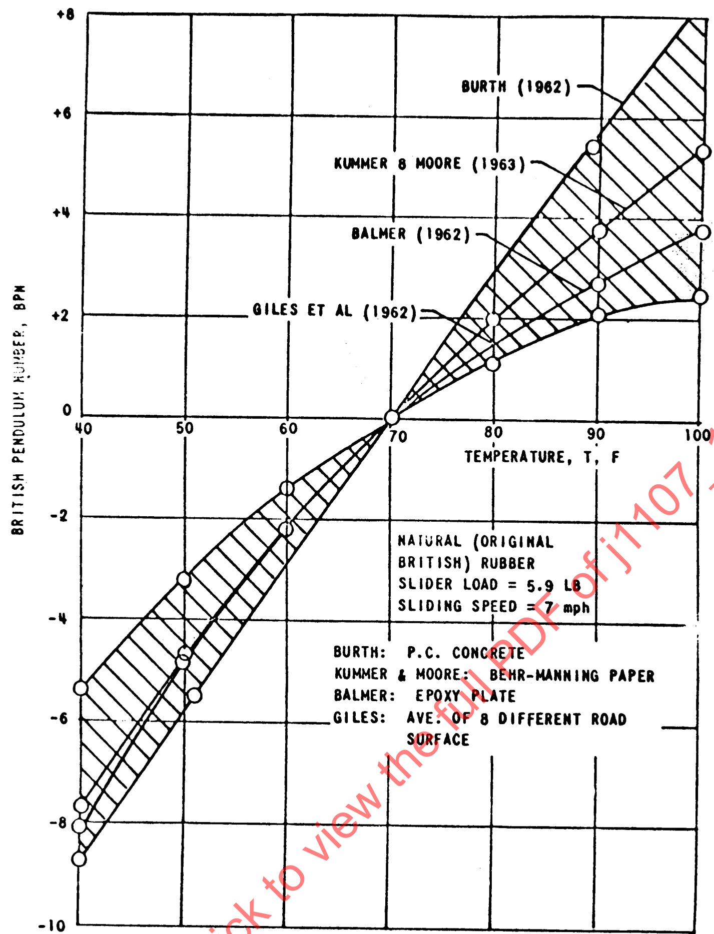


FIGURE 5—COMPARISON OF WET PAVEMENT TEMPERATURE CORRECTION VALUES, FOUND BY DIFFERENT WORKERS. (SPREAD IS LARGELY DUE TO DIFFERENCES IN SURFACE GEOMETRY AND CORRESPONDING DIFFERENCE IN ADHESION AND HYSTERESIS CONTRIBUTION.)

Difficulties in road testing on wet surfaces may be seen in the data of Figure 6 (reference 41). Here the lateral traction coefficients are shown as a function of speed. The road test results showed considerable scatter as indicated by the shaded bands while laboratory machine (internal drum) data are shown by well defined lines. Water depths from 02.-2.0 mm were tested in the machine. There is considerable difficulty in assigning a water depth for the road tests. First, an appreciable paved area is required over which there may be considerable variation. Second, these maxima are the result of several runs at different speed with the possibility that water depths were not the same during all of them. Nonetheless, from estimates made of the water depth in an area believed to be typical, a value of 1.5 mm was assigned. While this in general agreement with the machine data, the scatter precludes any quantitative understanding of the effect of water depth.

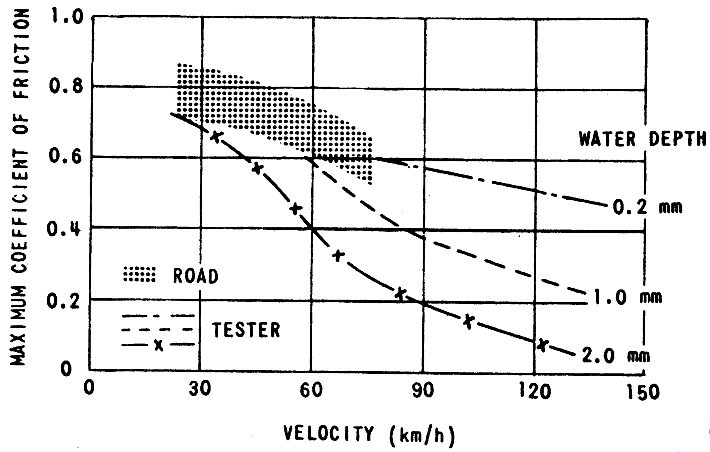


FIGURE 6—COMPARISON OF ROAD TEST AND LABORATORY (DRUM) DATA MAXIMUM CORNERING FORCE VS SPEED WET SURFACES—5.60-15 TIRES (BIAS PLY)

These few illustrations show that even for a particular tire, the skid numbers of actual pavements will not be constant but will vary from place to place on the pavement (different degrees of polish), from day to day (precipitation history), and from hour to hour (temperature changes). They will be vastly different depending upon whether the pavement is wet or dry. When wet, the skid number will depend upon how wet. Outdoor roadways are not attractive surfaces upon to do precision tire research at or near limiting slip or skid conditions because of the practical impossibility of controlling environmental conditions. The laboratory offers the opportunity of providing good control and measurement provided surfaces generally representative of roadways are provided.

6.2 Surface Texture Factors—Numerous testing groups have used carbide surfaces on laboratory machines. Gengenbach (41) has reported that an abrasive paper (carborundum) surface was stable and that after an initial wear-in it maintained its properties over an extended period. At the Technische Hochschule, University of Delft, Spaink has carried out an extensive program in making friction and drainage measurements on various carborundum abrasive papers (45). Spaink made measurements of the coefficient of friction of these papers and rubber slider, and correlated outflow measurements using a drainage meter. The same outflow meter was then used to obtain outflow measurements of road surfaces occurring in practice.

A commercial material called Scotchwalk[1] has been used on low speed testers with satisfactory results in terms of basic characteristics and stability. Rasmussen and Cortese (37) have reported on their investigations in this regard. Their conclusions are summarized as follows:

"Routine tire force and moment testing is done with an artificial road surface consisting of medium grit Scotchwalk material. This has a number of advantages:

1. The friction coefficient is similar to that of a new concrete road surface.
2. The high friction coefficient provides a maximum strain to the tire structure at high slip angles and loads. This tends to magnify the influence of various tire design factors.
3. The friction properties of the material are stable with use and can be easily cleaned with a vacuum device.
4. Minor contamination of the tire or road surface does not affect the test results."

1.Manufactured by the 3M Company.

It is also noted that there are significant differences between Scotchwalk and smooth steel. Several tires were run on a steel surface that had also been tested on Scotchwalk. Differences in force and moment properties between the two surfaces were quite variable with different tire configurations.

It is not known whether these differences were related to tread compounds or the presence of minor amounts of contamination. The tires were not given special handling or cleaning before testing on steel. No special cleaning is required in order to obtain repeatable data with Scotchwalk.

Tungsten carbide grit bonded to a steel surface has also been used. This type of material has also been applied to high speed tire uniformity machine road wheels with some success. Sheets of this material were obtained for a flat-bed machine in various grit sizes. Four tires of a set were tested on the front surfaces with results reported (37). Grit size varied between 46 and 80. The finest grit tested provided data closest to that obtained on Scotchwalk. Trends indicate that a still finer grit size would be a optimum match to Scotchwalk.

Tests with the various tungsten carbide grits involved checking of tire performance before and after testing. These spot checks determined when the abrasion to the tread surface had been sufficient to alter performance during a test. These spot checks indicated that the finer grit produces the least change in performance. The coarser grit appeared to cause severe shredding of the tread surface which usually results in some reduction in lateral force capability.

Krempel(46) has described difficulties encountered on a smooth steel surface at length. According to his observations valid results on smooth steel can only be obtained if the tire tread surface does not become sticky during the test.

A Gough plot is shown in Figure 7. Although the test time for a curve from 0-10 deg was only 25 s, Krempel observed that the tread surface became sticky at higher slip angles even under small loads. Discontinuities in the curves occurred when the slip angle exceeded 4 deg. They are caused by changes of the tread surface properties with increases in temperature. At larger slip angles (approximately 6 deg), the discontinuities in the test curves could be observed acoustically. With increasing slip angle, frequency and magnitude of the tire screeching increased continuously. Discontinuities in the curves, however, caused sudden changes in pitch and volume of the sound. Running on silicon carbide surfaces showed no tendencies for discontinuities. Figure 8 compares the data obtained for a conventional cross-ply on three different surfaces. Considerable differences can be observed between the curves obtained on the smooth steel and silicon carbide surfaces. The values of the lateral force on fine aggregate (K80) are generally higher than on coarser aggregate (K30). The differences, with the cornering force on K80 as a base, are shown in Figure 9. The differences between lateral forces on the two silicon carbide surfaces are small in comparison to the one obtained on the smooth steel surface.

As shown in Figure 10, the peak of the aligning torque of the radial tire on silicon carbide surfaces is 100% higher than on the smooth steel surface and 50% higher for the conventional tire.

Krempel makes the following general statements on the influence of the test surface:

1. Lateral force and aligning torque are larger on a silicon carbide surface than on smooth steel, as long as no tires are taken into consideration whose tread surface was already sticky or became sticky during the measurements.
2. The lateral force is generally larger on silicon carbide surfaces with finer aggregate size. However, the values of the lateral force on K80 (fine) and K30 (coarse) do not differ much.
3. No tendency can be established regarding the influence of the aggregate size on the aligning torque.
4. Up to 2 deg slip angle, the test surface effect is insignificant; however, it has a considerable effect by 4 deg.
5. The rank order of tires in their performance when operated at slip angles may be different on smooth steel and silicon carbide surfaces (especially at larger slip angles). No change in rank order could be found on silicon carbide of different aggregate size.
6. The tire temperature is of lesser influence on silicon carbide than on smooth steel surfaces.

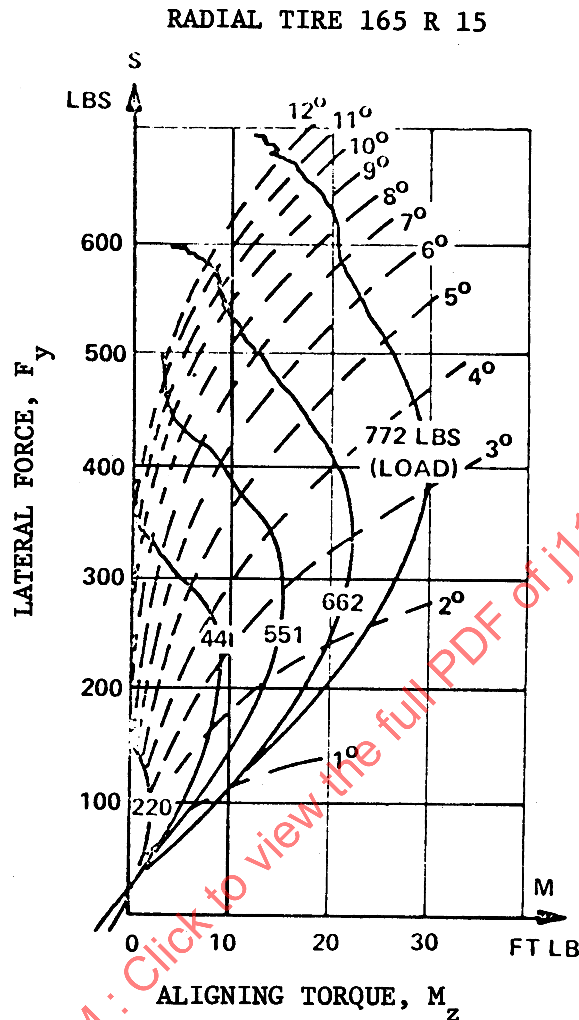
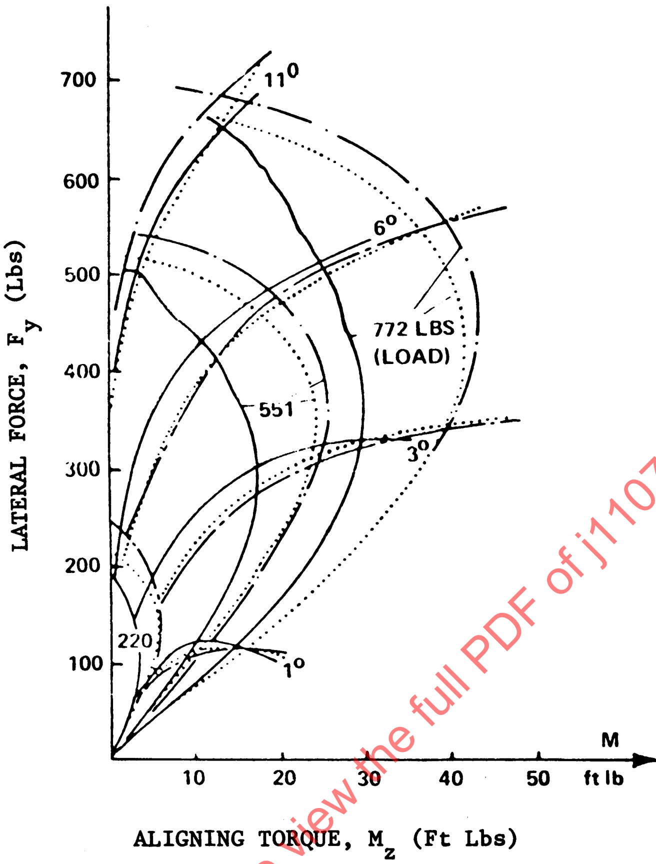


FIGURE 7—GOUGH PLOT O0BTAINED ON SMOOTH STEEL DRUM TREAD CONDITION 100%; I.P. = 25.6 PSI; SPEED = 31.2 MPH



———————— SMOOTH STEEL
.... SILICON CARBIDE K30 (COARSE GRIT)
—— —— —— SILICON CARBIDE K80 (FINE GRIT)
TREAD CONDITION 90%—100%; I.P.—25.6 PSI;
SPEED = 31.2 MPH

FIGURE 8—COMPARISON OF GOUGH PLOTS OBTAINED FOR CONVENTIONAL TIRE 6.00-15 ON THREE DIFFERENT DRUM SURFACES

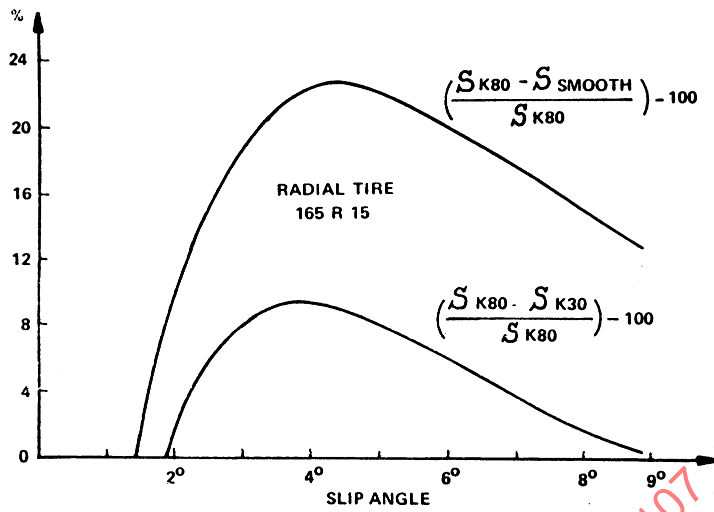


FIGURE 9—LATERAL FORCES ON TWO SILILCON CARBIDE SURFACES

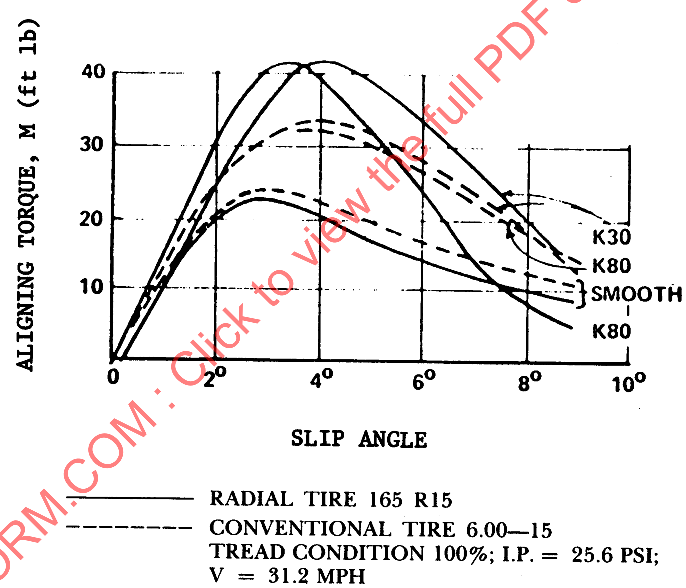


FIGURE 10—INFLUENCE OF THE DRUM SURFACE ON ALIGNING TORQUE (M)

6.3 Surface Curvature Factors—Drums have been used for structural integrity testing for many years and they have also been used for mechanical performance testing. The most common arrangement is the external drum; however, one machine has been developed at Karlsruhe in which testing is done inside the drum (27). In addition, Porsche has such a drum.

Running on a curved surface is not equivalent to running on one which is flat. Drums of various sizes are often used in tire testing. Tests comparing tire performance on flat and curved surfaces have been carried out by Krempel (46), Bull (5), Freudenstein (47), and Killmann (48), leading to different results.

To determine the geometric influence of the drum curvature on the effective rolling radius it was first assumed by Kollmann that cross section A (see Figure 11) of the tire deflection is the same in all three cases.

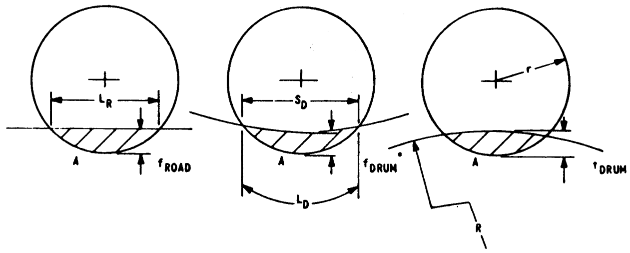


FIGURE 11—DRUM GEOMETRY

NOTE— f_{Drum} is different on exterior an interior drums; however, the values will be evaluated from the same equation.

This assumption leads to

$$f_D = f_R \sqrt[3]{1 \mp \frac{r}{R}}; \quad R > r >> f_r \quad \begin{array}{l}\text{upper sign for exterior drum}\\ \text{lower sign for interior drum}\end{array} \qquad \text{(Eq. 1)}$$

Tire deflection was measured under constant and variable tire pressure on drums of different radius. The results show a good correlation between measurement and mathematical evaluation (Figure 12). This implies that the length of the contact patch is also influence by the drum curvature and can be determined using the above assumption.

$$\text{Therefore } = L_D = L_R \sqrt[3]{\frac{1}{1 \pm \frac{r}{R}}} \; R > r > \; > f_R \qquad \text{(Eq. 2)}$$

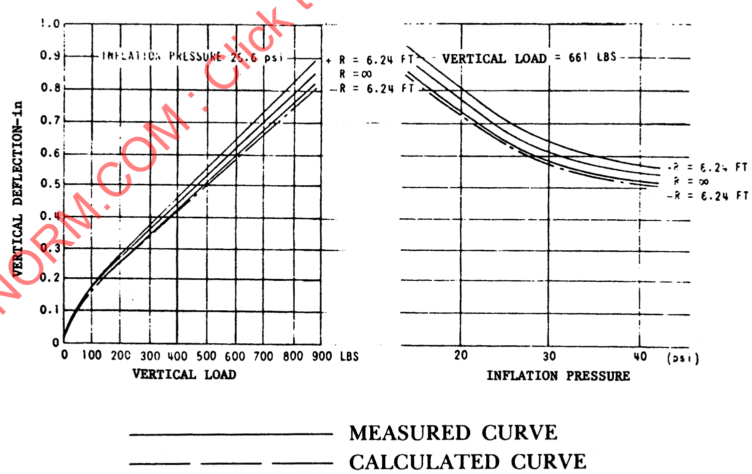


FIGURE 12—VERTICAL DEFLECTION OF TIRE 6.00-15: TREAD CONDITION 100%; AS A FUNCTION OF INFLATION PRESSURE, DRUM CURVATURE AND VERTICAL LOAD

With $L_D \approx S_D$

(that is, the length of the contact patch L_D, as measured on the curved surface was set equal to the chord S_D. The error introduced by this assumption does not exceed 0.1% even for extreme contact patch lengths and small drum diameters.) Again, calculations and test results show reasonable correlation (46).

$$f_D = f_R \sqrt[3]{\frac{\pm R + r}{\pm R}} \quad \text{(Eq. 3)}$$

The difference is contact patch lengths for a tire on a flat road surface and a tire on a drum (D≈12 ft (3.7m) is of the order of 5-10% depending on type of tire, inflation pressure and load. Krempel determined the ratios of f_D/f_R and L_D/L_R as a function of R/r. (R = radius of drum; r = radius of undeflected tire) as shown in Figure 13.

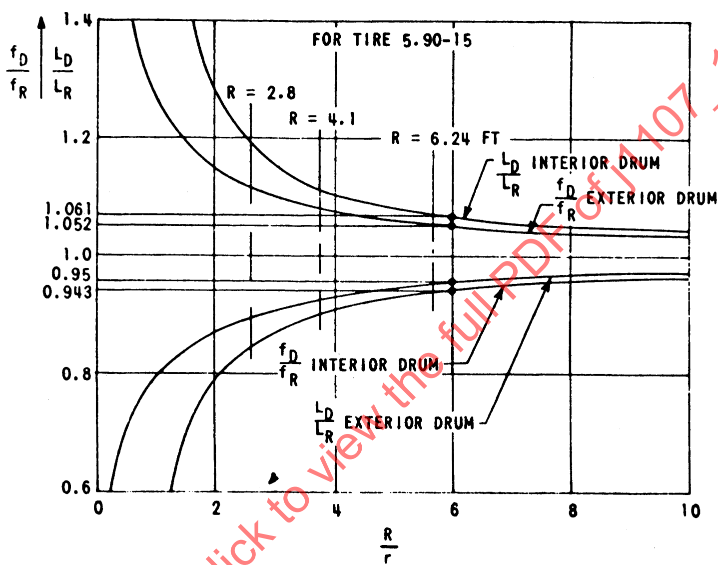


FIGURE 13—CALCULATED INFLUENCE OF DRUM CURVATURE ON VERTICAL TIRE DEFLECTION F AND LENGTH OF CONTACT PATCH L

The width of the contact patch is influence only sightly by the drum diameter. Thus the change in the area of the contact patch is mainly determined by the change in the length (L). Hence L seems to be the major factor for the differences in lateral force of a tire on a flat and a curved surface. Figure 14 shows the influence of drum curvature on the area of the contact patch.

A 6.00-15 tire under a load of 661 lb (2940 N) and 25.6 psi (176.5 kPa) inflation pressure has a contact area of 17.8in^2 (0.01148 m^2) on an exterior drum of R - 6.4 ft (1.902m).; The inflation pressure must be increased to 30.5 psi (210.3 kPa) to obtain the same area on an interior drum of the same radius, otherwise the contact area on the interior drum will be 5-10% higher (In this particular case 7%). If the inflation pressure is the same, a load of 610 lb (2710N) for the interior and 661 lb (2940 N) for the exterior drum will generate the same contact area. The differences in contact area between the tire on a flat surface and on an interior drum are approximately half of those mentioned above. Corresponding results can be determined for the theoretical mean contact pressure:

$$P = \frac{\text{vertical load}}{\text{area}} \quad \text{(Eq. 4)}$$

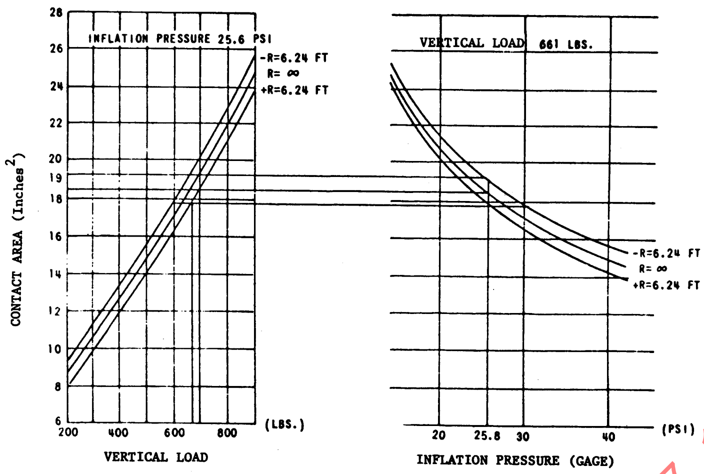


FIGURE 14—INFLUENCE OF DRUM CURVATURE ON THE AREA OF THE CONTACT PATCH AS AFFECTED BY WHEEL LOAD AND INFLATION PRESSURE FOR A 6.00-15 TIRE

As the interior drum gives a greater contact area, the calculated mean pressure is smaller than on a flat surface. Krempel found that in the case of the interior drum an increase in inflation pressure (approximately 3%—5%) would give the same calculated mean pressure as on a flat surface, whereas a decrease of the same amount would have the same effect for the interior drum.

Tests comparing lateral force on exterior drums and flat surfaces have been carried out by Bull (5) and Freudenstein (47). It was found that the lateral forces obtained on exterior drums were always smaller than those measured on flat surfaces. Bull, assuming a minimal speed effect, conducted his tests at 1 mph (1.6 km/h) on the flat surface and 29 mph (47 km/h) on a steel drum of 2.8 ft (0.85 m) radius. As can be seem from Figures 15 and 16, cornering force on a flat surface (covered with allundum) is approximately 10% higher than on the allundum covered drum surface. Camber force, however, seems to be more significantly influenced by the shape of the surface. Unfortunately Bull evaluated the influence of the test curvature on camber force on different surfaces, thus the conclusion may be confounded by the surface variable.

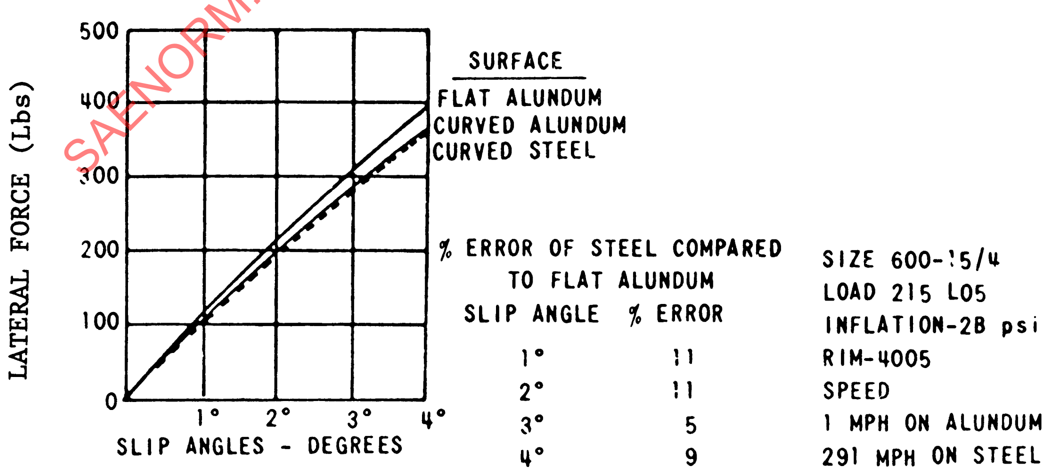


FIGURE 15—COMPARISON OF FLAT AND CURVED SURFACES

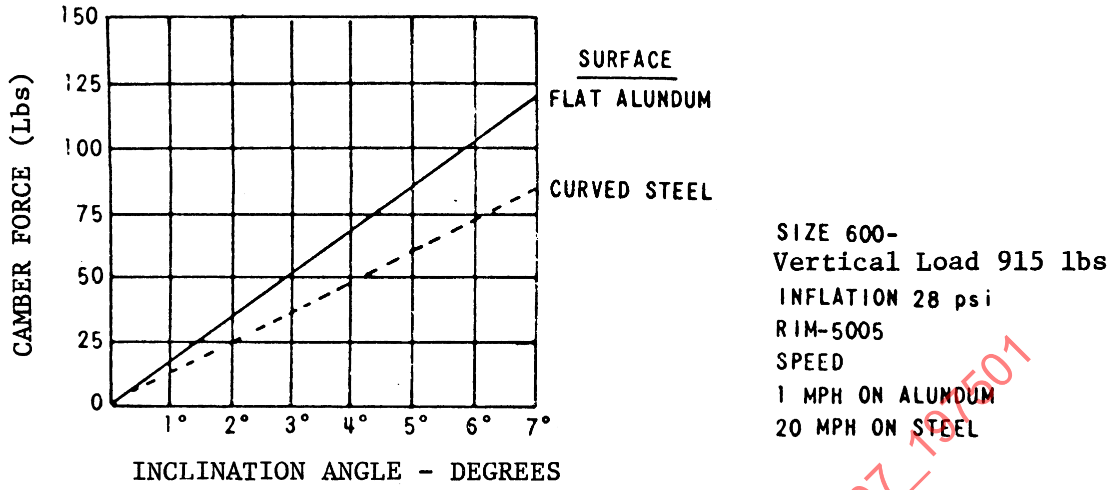


FIGURE 16—EFFECT OF SURFACE ON CAMBER FORCE

It is clear that the pressure and shear force distribution in the contact patch are also influenced by surface curvature. Figure 17 shows the pressure distribution in the contact patch of a tire on a flat surface (solid line). It is believed that the center of pressure will shift further forward on an exterior drum a d fall back on an interior drum in comparison to the flat surface due to hysteretic losses in the rubber. As the vertical deformation on an exterior drum is larger than on an interior, the losses should be greater causing a greater shift of the center of the pressure. The altered pressure distributions will, of course, have effects on the self-aligning torques of the test tires.

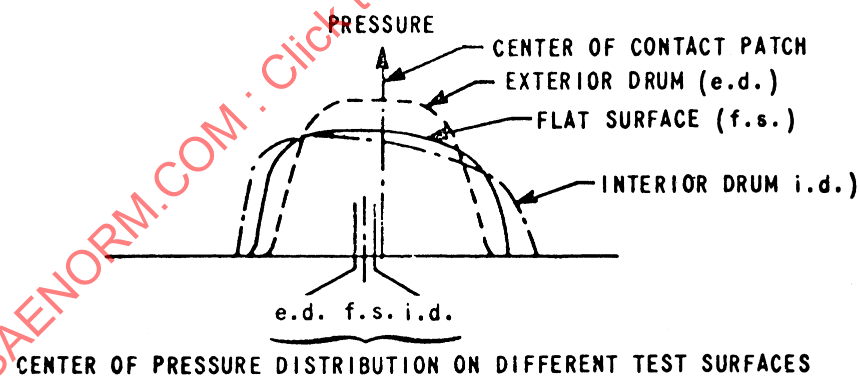


FIGURE 17—CROSS SECTIONS IN LONGITUDINAL DIRECTION THROUGH CENTER OF PRESSURE DISTRIBUTION

If the center of the pressure on an exterior drum is shifted forward, the rolling resistance of the tire must increase in order to compensate for the increased rolling resistance moment M = pdA·e (W = vertical load, e = forward shift of c.p. of normal force). Experimental verification of this reasoning appears to be missing.

6.4 Effect of Speed—It is well established that speed has a pronounced effect on the force and moment properties of a free rolling tire operating under wet roadway conditions as illustrated by Figure 18. On dry roads speed also affect forces and moments where appreciable sliding takes place in the tire-road interface. It has generally been believed that the velocity of travel over the usual passenger car ranges has a small influence on the basic force and moment characteristics of tires on flat roads, as long as the tire operates in the "frictional mode"—that is, as long as there is no gross slippage in the contact patch.

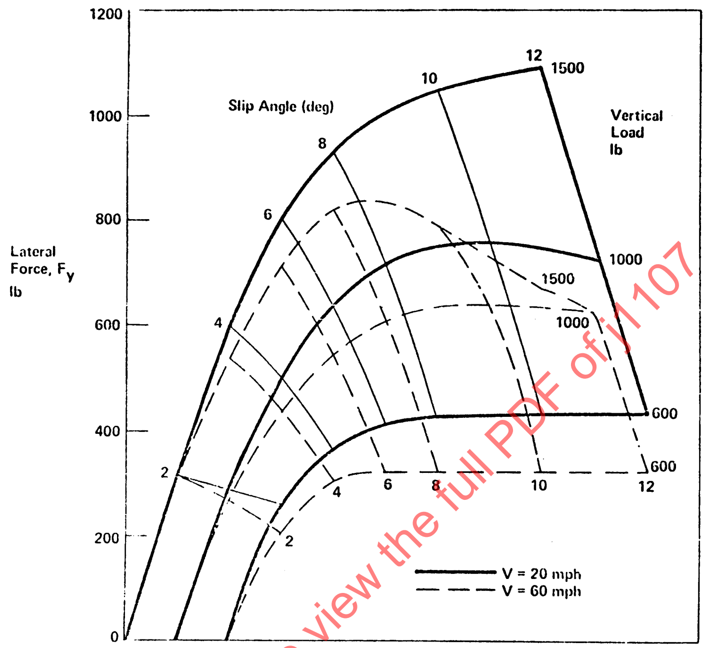


FIGURE 18—INFLUENCE OF SPEED ON LATERAL FORCE GENERATED BY A TIRE ROLLING FREELY ON WET CONCRETE

Recent experimental data (26) provides new information on this matter. Runs were made on a G78-15 bias belted tire at the design load (1380 lb (6100 N)). Inflation pressure was 24 psi (165.5 kPa) cold. Velocities of 5, 15, 30 and60 mph (8.0, 24.1, 48.3 97 kM/h) were run. The measured lateral forces and aligning torques are shown in Figures 19 and 20. A carpet plot presentation is used, with lateral force plotted against slip angle for the various speeds; thus lines of constant velocity and constant slip angle are shown. Qualitatively, the lines of constant slip angle show that at low slip angles (70 to 6 deg), the lateral force increases slightly as the velocity increases. Quantitatively, the increase of cornering stiffness between 5 mph (8.0 km/h) and 0 mph (97 km/h) is 10%; at 12 deg slip angle, the later force is reduced by about 10% over this same range.

The aligning torques do not show a slope variation at the small slip angles, but there is a reduction of torque from about 117ft·lb (159 Nm) at 5 mph (8.0 km/h) to 109 lb (485 N) at 60 mph (97 km/h). The angle of maximum aligning torque is reduced from about 4.6 deg at 5 mph (8,0 km/h) to 4.1 deg at 60 mph (97 km/h).

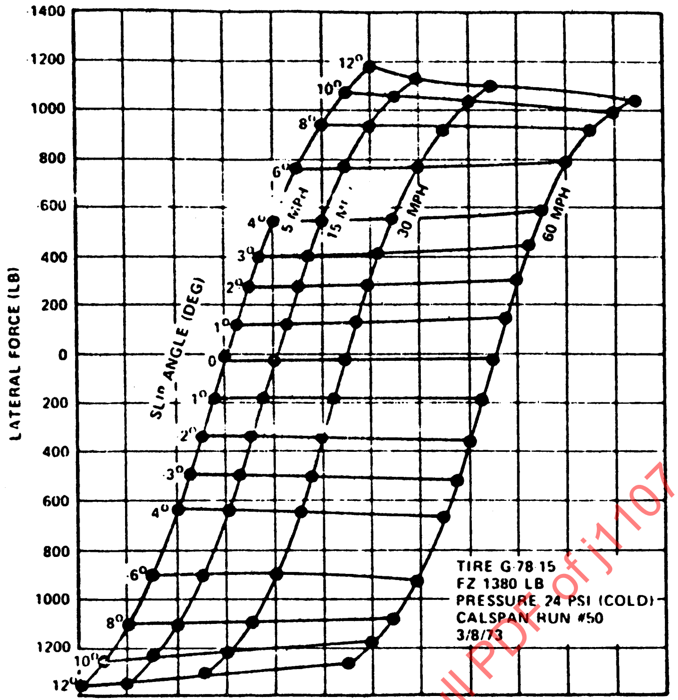


FIGURE 19—EFFECT OF SPEED ON LATERAL FORCE OF A BIAS BELTED TIRE ON DRY ROAD

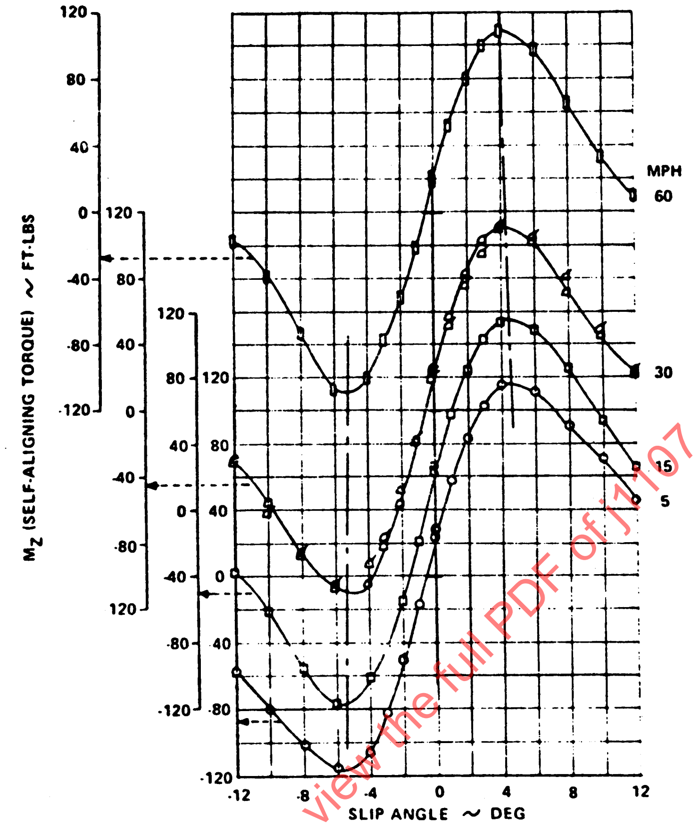


FIGURE 20—EFFECT OF SPEED ON ALIGNING TORQUE OF A FREE ROLLING BIAS BELTED TIRE ON DRY ROAD

7. *Weighing Systems*

7.1 Requirements—The general requirements for weighing system range and accuracy are outlined in J1106, particularly Tables 1-4. Some special comments may be helpful. The normal force range of 1.6 times the largest tire vertical load capacity is a minimum requirement. If the data are to be used for analyses of vehicle overturning behavior, this range may be insufficient. A tire may carry twice its load capacity in sever cornering.

The measurement of longitudinal force is significantly altered with the introduction of axle torque. For large values of axle torque, the longitudinal force may be approximately equal in magnitude to the normal force.

7.2 Orientation—Tire forces and moments must ultimately be expressed in the coordinate system of Figure 1. Therefore, it is mandatory that the geometric relationship between the Tire Axis System and the measured forces and moments be recorded at all times during the testing to guarantee a complete coordinate transformation is made easier if the Tire Axis System and the coordinate system of force and moment measurement are fixed relative to each other or, ideally, are coincident. But with modern data analysis techniques, a fixed relationship between these two coordinate systems does not necessarily shorten data reduction time.

7.3 Type—Two types of weighing systems are common. The first uses six distinct unidirectional load transducers. Three of these are placed parallel, and often coincident, with the x' y' z' axes of the Tire Axis System to primarily measure the three forces; the other three are positioned to measure the tire moments about these axes. This weighing system is generally fixed with respect to the Tire Axis System. The second type employs an integrated multi-directional force transducer which is concentric with the tire spin axis. This weighing system is not fixed relative to the Tire Axis System, changing orientation with changes of tire inclination angle and loaded radius.

The advantages of the first type are related to the use of distinct force transducers. If properly located, each transducer can primarily measure a given tire force or moment. This allows the designer to size each transducer for best accuracy. In addition, if testing conditions change, thereby altering the magnitude of only some of the tire forces and moments, the corresponding force transducers may be replaced for improved accuracy or reliability. Finally, commercially available force transducers may be used.

Certain disadvantages are also inherent with the first type of weighing system. The framework between the tire and the force transducer cannot generally be made rigid enough to eliminate sizeable structural deformations. These displacements result in no-linear interactions between transducers which must be measured during calibration and accounted for from displacements transverse to the load cell are used with each transducer to minimize the interactions. Also, with a large mass between the tire and force transducers, low resonant frequencies may be encountered. These will probably not be excited by tire force and moment nonuniformities if the speed is low (below 5 mph) (8 km/h). These resonances might be a problem for a machine operating at normal road speeds.

With the multi-directional transducer system, it is possible to minimize the mass and obtain a higher stiffness between the center of tire contact and the transducer, thereby raising the natural frequency of the weighing system. Further, the higher stiffness results in reduced interactions. This type of transducer system is frequently used for high-speed testing to obtain the required bandwidth.

But there are disadvantages. Such transducers are not commercially available for the measurement of tire forces and moments. Their design and fabrication must be done with care to obtain the required accuracy in each orthogonal direction. Since the orientation of the transducer changes with respect to the tire axis system, this motion must also be measured during testing and considered in data reduction.

7.4 Bandwidth—The frequency response of the weighing system should be 40 times the tire rotational frequency. This allows the measurement of tire force and moment variations up through the fourth harmonic with insignificant loss of data.

If the machine is to be used for the measurement of tire relaxation characteristics, special requirements must be met. In order to measure relaxation characteristics at the lowest important reduced frequency of 0.015 c/ft, one must test at 23 mph (37 km/h) for a 0.5 Hz absolute frequency (see 11.3). At this speed, a small tire (A70-13) would have a rotational speed of 5.5 rps. A weighing system frequency response of (5.5) (40) = 220 Hz would be required to test this tire at these conditions. The actual frequency response required depends on the tire being tested and the absolute frequency chosen.

7.5 Typical Weighing Systems—Two common types of weighing systems have been described in general terms. The specific layout of some typical systems is important.

The most common system utilizing distinct, uni-directional force transducers employs three nested frames surrounding the tire. These allow changes of tire slip angle, inclination angle and vertical load. The outer-most frame is grounded through six uni-directional force transducers. A schematic diagram of this system is shown in Figure 21. The Z transducer measures tire normal force. The Y_1 and Y_2 transducers, in the y' z' plane, measure lateral force and overturning moment. Similarly, the X_1 and X_2 transducers in the x' z' plane, measure longitudinal force and rolling resistance moment. The X_3 transducer, parallel to the x' z' plane, measures aligning torque. These transducers do not move relative to the x' y' z' system.

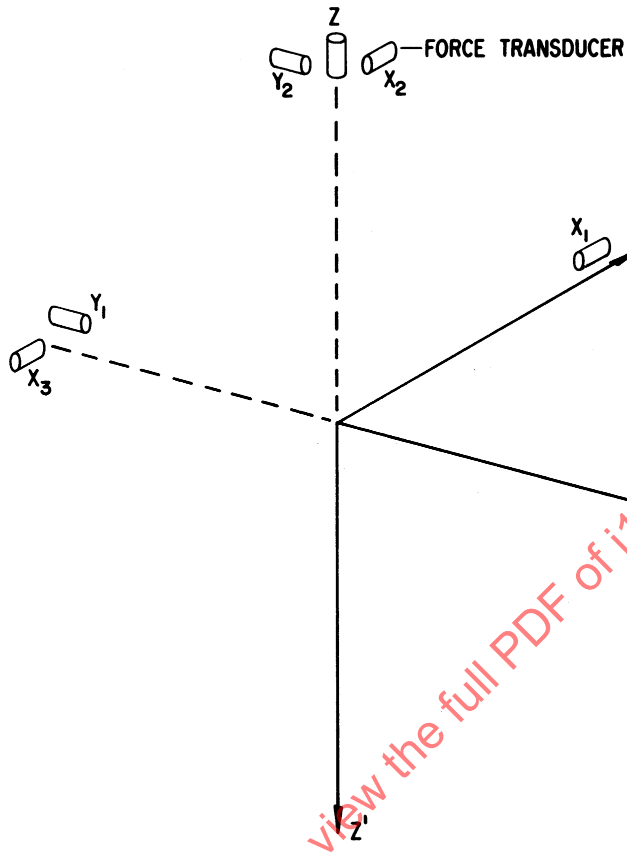


FIGURE 21—TYPICAL MULTI-TRANSDUCER WEIGHING SYSTEM

A second common multi-transducer system is shown in Figure 22. In this system the tire axle is directly supported by five transducers. (The sixth constraint, required for the axle housing, is not shown.) The X_1 and X_2 transducers, parallel to the road plane, measure longitudinal force and aligning torque. Transducers Z_1 and Z_2 measure normal force and over-turning moment. Lateral force is measured with type Y transducer. Rolling resistance moment cannot be measured with this system. Since the transducers are fixed relatively to the tire spin axis, they more with respect to the x' y' z' coordinate system with changes of inclination angle and loaded radius. Therefore, these variables must also be measured.

The common weighing system employing a multi-directional force transducer is arranged with the transducer concentric with the tire spin axis. Unlike the system of Figure 22 the axle is cantilevered.

These transducers are often patterned after those used in wind tunnel studies and employ an array of strain gaged beams which are arranged and sized to measure the tire forces and moments. The transducer is fixed with respect to the tire spin axis, requiring the measurement of inclination angle and loaded radius.

Typical experience in terms of resolution and transducer interaction depends somewhat on weighing system design. Recent designs of system s employing a multi-directional transducer have demonstrated excellent resolution and low interaction characteristics. With any design, the present state-of-the-art should provide resolution of 0.5% of full scale. Transducer interactions should be between 0.5% and 1.0% of the input force.

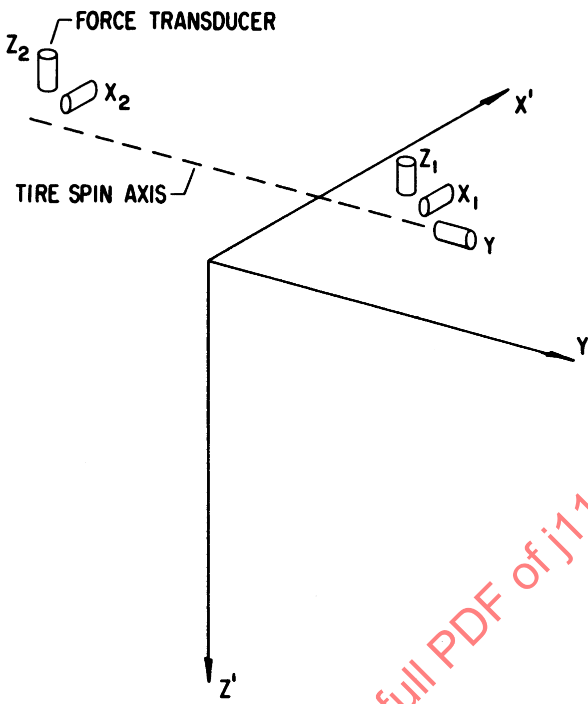


FIGURE 22—ANOTHER TYPICAL MULTI-TRANSDUCER WEIGHING SYSTEM

7.6 Calibration Procedures—One of the most important elements in obtaining an accurate weighing system is a thorough calibration. Attention to detail is very important in this portion of the machine development.

A number of steps are normally involved in a periodic calibration, performed about every six months. the first step is geometric alignment of the weighing system at zero angular positions, before the transducer(s) is (are) installed. After installation, the first calibration run is performed. These data are next analyzed and steps are taken to reduce transducer interactions. Finally, the weighing system is recalibrated. An additional task, not directly a part of the calibration, is performed before testing can begin. This is the interaction compensation, usually part of the data reduction program in a computer.

Calibration should be a consideration early in the machine design phase. A few precautions taken at this time will save time and doubt during the actual calibration. Weighing system alignment is usually very difficult unless each major part of the machine has been previously marked for exact alignment. Parts should have numerous machined surfaces and reference marks which will not change with time.

A further aid is the installation of benchmarks in the floor of the test room. These permanent marks are most conveniently placed in the x' z' and y' x' planes. Other consideration should be given to the hardware needed for the introduction of known forces during calibration. Finally, adjustment in transducer orientation must be provided to facilitate interaction reduction.

Weighing system alignment is normally a repetitive, interactive process. The components of the weighing system must be aligned for zero displacements and angular rotations relative to the x' y' z' axes. (The reference original and axes are defined by the centerline of the roadway and the mounting hub.) Since each alignment change usually affects other alignment components, the process should be repeated at least once. Conventional shop tools are used. These include various types of micrometers, scales, and levels. A transit and plumb line are also very helpful.

The transducer installation is not complete with physical installation. Care should be taken to ensure that the cells are aligned with the x' y' z' axes and contained in the orthogonal planes. When installed, the positions of the transducers in a multi-transducer system must be recorded for use in the data reduction program. To ensure that transducer capacity is not wasted through imbalance, the weighing system should bee balanced so that its center of gravity is near the z' axis. Finally, transducer null balances (voltage reading with no tire force input) should be recorded for future reference.

Some decisions must be made just prior to calibration. While six linearly independent force inputs must be applied for a complete calibration, their actual location is not restricted. Typical practice, with a multi-transducer system, is to apply a force at or near each transducer to exercise primarily that transducer. The range should be large enough to exercise the transducer as much as the largest tire would. Generally, the input forces are parallel to one of the x' y' z' axes. Approximately 10 force levels per channel are required in each direction to define the system response, especially the interactions.

The force input hardware should have been selected during the design stage. Dead weights with wires and pulleys and hydraulic jacks are common tools for introducing the forces. Pulleys suffer from the disadvantage of having unknown, and possibly high, friction. Hydraulic jacks require an additional transducer in series with them. They are sometimes difficult to align accurately due to their short length.

During calibration, each transducer should be loaded in both tension and compression. While force transducers are usually linear through zero, the interaction effects when installed may be bilinear or non-linear. Current practice is to calibrate at zero slip and inclination angle and to apply one force at a time. (The effects of nonzero angle or simultaneous force introduction on system responses have not been investigated. These effects may be significant.) Angular position controls and potentiometers should also be checked during calibration. If the force and moment tester is to be used for relaxation testing, the magnitude and phase response of the weighing system should be measured at three or four frequencies.

Interaction reduction is primarily an art. In a multi-transducer system, the interaction source (an unexpected deformation) is usually hypothesized or measured. Transducer orientation is then altered to reduce the sensitivity to this effect. Recalibration determines the validity of the hypothesis. Care should be taken so that the original slip and inclination angle settings are not altered with transducer adjustments. Transducer movement should be within the measurement error of their original installation.

Interaction compensation is usually achieved in the data reduction program with linear techniques. Matrix algebra is commonly employed. Many observed interactions are nonlinear. Common practice is to treat these as linear interactions. Nonlinear compensation does not seem justified in light of the limited calibration techniques presently used.

Even with a thorough calibration, more frequent checks of weighing system accuracy are required. Most laboratories check transducer null balances daily. Some apply a known force, which loads all the transducers, to check calibration. Common practice is to measure the cornering properties of a given tire once per month. This last technique is usually only useful for indicating large changes of system calibration.

8. Data Processing

8.1 Introduction—The problem of data processing for a tire force and moment machine can be considered as a series of transformations applied to information derived from a system of force and motion transducers to generate data in a form suitable for plotting, storage or other analyses. The nature of these transformations depends on the design of the machine and the completeness of the data set that is ultimately desired. This discussion will consider a most general case where multiple transformations are required by a moving transducer system and there is need for a complete data set. It should be clear from the discussion that certain of these transformations may not be required if other transducer systems are specified or if limited data sets are of primary interest.

8.2 Data Transformation System—The data processing problem can be considered in block diagram form with the system shown in Figure 23. This arrangements is somewhat arbitrary, but useful for conceptual purposes. The force and motion data originating from strain gages, potentiometers, or differential transformer transducers can be viewed as a multi-component vector that must be rescaled and reoriented through a series of transformations for recording in the desired form. Each transformation can be viewed as a matrix manipulation for conceptual purposes. Depending on the nature of the transformations, they may be rearranged or effectively combined into a single transformation.

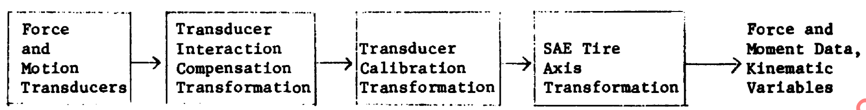


FIGURE 23—DATA TRANSFORMATION STAGES

The hardware required for these transformations will not be discussed in detail since a variety of conventional data processing methods can be applied to yield similar results. Transducer output voltages can be sampled and digitized directly with suitable analog to digital conversion equipment. The transformations can then be implemented as matrix manipulations with a variety of algorithms associated with digital computation. Since some form of digital processor is frequently required for averaging, curve fitting, and plotting of complete data, this approach is frequently used. If there is need for immediate, on-line plotting of limited data, these transformations can be implemented with active, or in some cases, passive analog instrumentation. For machines with moving, spindle oriented force transducer axes, analog implementation of the axis transformation block may prove cumbersome since nonlinear components are required.

8.3 Transducer Interaction Compensation—A tire force and moment machine weighing system may be considered as an array of six or more force transducers. It is rarely possible to produce a system that is free fro mechanical interactions. That is, a force vendor applied in the direction of the principal active axis of one component will, in general, result in some level of output signal from all components. It is important to minimize the degree of mechanical interaction with careful system design. To simplify the implementation of interaction compensation, it is also important that the interaction data be linear with the applied force. Interaction data are acquired by the application of known force vectors in the calibration process. Interaction gradients can be determined from these data and used to determine the elements of the interaction computation matrix. These elements are chosen to eliminate the interaction components so that a force vector applied along a principal axis will only produce a signal at the corresponding transducer output.

8.4 Transducer Calibration—The calibration transformation may be required to convert transducer electrical outputs into engineering units of force and displacement. For an analog system, this stage may be used to achieve convenient units for scaling and plotting. In either case, the interaction and calibration stages can be combined if they are linear since the outputs of the individual stages are not usually of interest.

8.5 Tire Axis Transformation—It is rarely possible to design a force and moment machine with a transducer axis system that is coincident with the reference tire axis system. Since the original of the tire system is in the road plane, the road surface usually precludes location of transducers at this level. A geometric transformation is, therefore, required to produce data in the tire axis system from data in the transducer axis system (24). If the transducer axis system maintains a fixed relationship to the tire system, this transformation is relatively simple and invariant. Many machines employ a spindle axis transducer system, however, and for these machines the axis transformation will vary with test conditions. Spindle motion data must then be included in the geometric transformation. This is an additional complicating factor but need not preclude the acquisition of valid data.

If the three transformations discussed in this section are linear, they may be combined into a single matrix operation. This is often done because the only data of interest are the final force and moment data tabulated in the tire axis system. Conceptual separation of the three operations is important for understanding and error analysis for any proposed system design. Implementation of these transformations on a computer with assumed transducer output data can provide useful insight to component accuracy requirements and error multiplication effects that can result from particular transducer geometries.

8.6 Curve Fitting—All detailed engineering measurements are subject to a degree of variability due to phenomena that cannot be understood or controlled. Tabulation or plotting of raw engineering data is frequently of limited usefulness since statistical variabilities tend to hinder application and interpretation. Tire force and moment tests are run on a very few samples of a particular design under idealized conditions. The data must be treated as a statistical estimate of tire design performance to be used mainly for comparison purposes although test accuracy and test conditions are controlled to the extent possible. Final processing, curve fitting and plotting must be selected to facilitate interpretation and proper application of test results.

Each tire force and moment component can be influenced by vertical deflection (or normal force) slip angle, inclination angle, and applied tractive force (driving or braking). Since each component, in the most general case, can be considered as a nonlinear function of four independent variables, tire data are frequently recorded as surface-like functions of at least two variables. A third variable is sometimes introduced through nesting of two variable plots. No satisfactory technique for expressing the most general four variable plot has yet been developed. Obviously many different combinations of variables could be plotted but attention has been directed mainly to plots of tire force, moment, and geometric components as functions of normal force-slip angle and normal force-inclination angle. These plots are most often used for data taken from a free-rolling test where longitudinal forces are not applied. Since they represent three dimensional surfaces, it is desirable that statistical procedures used to process and fit data be of the surface fitting variety.

Some laboratories apply two dimensional curve fitting procedures to tire data because they are familiar and readily available. If curves obtained with a two dimensional procedure are plotted in a three dimensional carpet plot form, it is clear that the resulting surface is irregular and does not represent a best fit to the available data. Surface fitting can be accomplished manually by plotting raw data in carpet plot (49,50) form and drawing lines of constant angle and normal force with curve templates. This is a tedious procedure that requires substantial judgment and skill. In addition to fitting lines to the data, lines must intersect on certain ordinate axes which is an additional constraint. Once data have been manually fitted with the form of a carpet plot, it is possible to fit each line of constant slip angle with a cubic polynomial to convert the hand-drawn surface into mathematical form.

A more practical and automatic procedure for generating a mathematical model representing these multi-dimensional data surfaces is based on jointed bi-cubic spline polynomial functions. These functions were applied some years ago in computerized auto body drafting work (51). They can be considered as the mathematical equivalent to the drafting template used to generate, in a piece-wise fashion, a smooth and continuous curve with a complete and unique shape. These functions provide a smooth and continuous surfaces rather than curves, however. The form of a bi-cubic polynomial is shown below:

$$\begin{aligned} A_1 &= C_1X^3 + C_2X^2 + C_3X + C_4 \\ A_2 &+ C_5X^3 + C_6X^2 + C_7X + C_8 \\ A_3 &+ C_9X^3 + C_{10}X^2 + C_{11}X + C_{12} \\ A_4 &= C_{13}X^3 + C_{14}X^2 + C_{15}X + C_{16} \\ Z &= A_1Y^3 + A_2Y^2 + A_3Y + A_4 \end{aligned} \qquad \text{(Eq. 5)}$$

The term spline is derived from the fact that a number of these polynomials can be used to represent various sections of a data surface. Polynomial coefficients can be chosen so that the surface represented is smooth and continuous at the joints between two polynomials. Cubic polynomials have proved to be sufficient for representation of most tire data surfaces. Lower order polynomials can be used for some surfaces that are less complicated. Experience has shown that higher order polynomials introduce waviness in the data and do not improve fitting accuracy. Relatively large digital computers are required at present for implementation of this approach.

The polynomial coefficients further provide a convenient medium for storage and retrieval of processed data. Coefficients for a complete set of tire data surfaces can be stored on a few dozen computer cards. Simple programs can then be written for retrieval or further manipulation of the data. While the splined polynomials are conceptually complicated, they need never be manipulated or interpreted by the engineer after the necessary computer software is developed. The computer can quickly evaluate these functions, list or plot the results in any desired format.

Most tire data contain biases and a degree of asymmetry between positive and negative slip angle information as well as left and right rotation mode. Opinions differ on the need for representing these factors in processed data. Laboratories that are m ore tire oriented generally favor presentation of data, as measured, with biases and symmetries clearly indicated so that this information can be related to construction variables. Vehicle oriented laboratories frequently remove these factors or consider them separately as they make data interpretation more difficult. Some tires are constructed with a controlled degree of asymmetry for special reasons. Positive and negative data must be presented separately for these tires. Alternatively, the curves may be forced through the origin with both y-intercepts reported separately. The spline polynomial approach can be applied in either case. With a suitable choice of joints, polynomials can be developed for all quadrants of data. Offset effects can be eliminated by neglecting the constant term (C_{16}). Asymmetries can be handled by reflection of polynomials in one quadrant and averaging of the coefficients.

Tire uniformity characteristics can also be troublesome for low speed tests at small angles where force and moment variations due to nonuniformities are large compared to forces and moments associated with slip and camber angles. Test laboratories vary in their approach to this problem. Filtering and other techniques can be applied but these generally required extended test time and more data. Most agree that it is more desirable to prescreen tires for uniformity and test the best samples of a particular lot.

8.7 **Derived Data Processing—**In addition to basic data surfaces generate directly from routine force and moment tests, a variety of functions can be derived to assist in the interpretation and application of test results. These functions include the friction ellipse used to show the relationship between lateral and longitudinal forces (24), a camber rolloff function (31), used to represent combined inclination and slip angle performance, load sensitivity and, load transfer sensitivity functions (52) used to represent gradient and curvature phenomena of particular significance to vehicle performance. Most of these functions are really approximations used to simplify the complicated tire data space. Calculation of some of these functions requires a degree of technical judgement and has not been formalized at this time. Some of these functions will be discussed further in the section on data interpretation.

9. ***Presentation of Results—***Two problems related to tire force and moment data concern the automotive and tire industries (53). The first is the problem of computer storage of force and moment data; a compact form is needed to aid vehicle directional control simulations. This problem particularly concerns the automobile companies and is somewhat an in-house task since its solution is directly dependent on a particular company's computer capabilities. This section will address the second problem, that of developing a standard format for tire force and moment data to facilitate adequate communication between the tire and automotive companies.

Force and moment characteristics of the free-rolling tire are influenced by vertical load, slip angle and inclination angle. The usual test procedure is to set two of the variables constant and vary the third. Raw test data, the force and moment measurements, and then recorded. When the quantity being varied has run through the desired range of values, the two variables being held constant can be varied and the third variable again run through the desired range of values. In addition to holding two of vertical load, slip angle or inclination angle constant, there are test variables such as inflation pressure and speed of rotation. These can be varied for each combination of values of slip angle, inclination angle and vertical load. It is obvious that unless test variables are clearly identified and a systematic procedure for variation is developed, any data transfer between companies can be confusing and inefficient.

A suggestion has been made that a standard machine-sensible computer format be developed for the transfer of force and moment data (53). The use of data compacting methods on the 80 character field of a computer card or magnetic tape is offered as a form for data transfer. This type of technique would be useful for the transmittal of raw data.

Whether raw test data is compacted in computer format or not, it is still essential that the technique for variation be systematic and uniform. To this end the following method is put forward as a uniform procedure for data presentation.

Tire force and moment data should include the following:

a. Tire identification, to include size, brand and construction features.
b. Inflation pressure.
c. Test speed (depending on machine concept).
d. Tabular printout of test data and graphical presentation (families of curves) as follows:

   Lateral force vs slip angle at five (5) different loads (in the range of 40-160% rated load).

   Aligning torque vs slip angle at five (5) different loads in ranges from 40-160% rated load.

   Lateral force vs vertical load at different slip angles (± 1 deg, 2 deg, 4 deg, 6 deg, 8 deg and 10 deg).

   Aligning torque vs vertical load at different slip angles (± 1 deg –10 deg).

   Lateral force vs aligning torque at different vertical loads from 40-160% rated load and at slip angles ranging from + 1 deg to 10 deg (Gough plots).

   Step d. of the above data presentation method involves more than the transmittal of raw data. It requires that the data be presented graphically.

The inconvenience of multiple plots to show the effect of two independent variables can bea voided by use of a graphical presentation technique known as carpet plots. Here only one of the parameters, slip angle, inclination angle or vertical load, is held constant and the other two are allowed to vary through their desired range of values. At least one of the independent variables must take on nominal values. The force and moment measure of interest is then plotted against values of the two independent variables. A carpet plot showing the effect of slip angle and vertical load on aligning torque is shown in Figure 24 (30). An attractive feature of the carpet plot is that it provides a method for interpolating with respect to both variables.

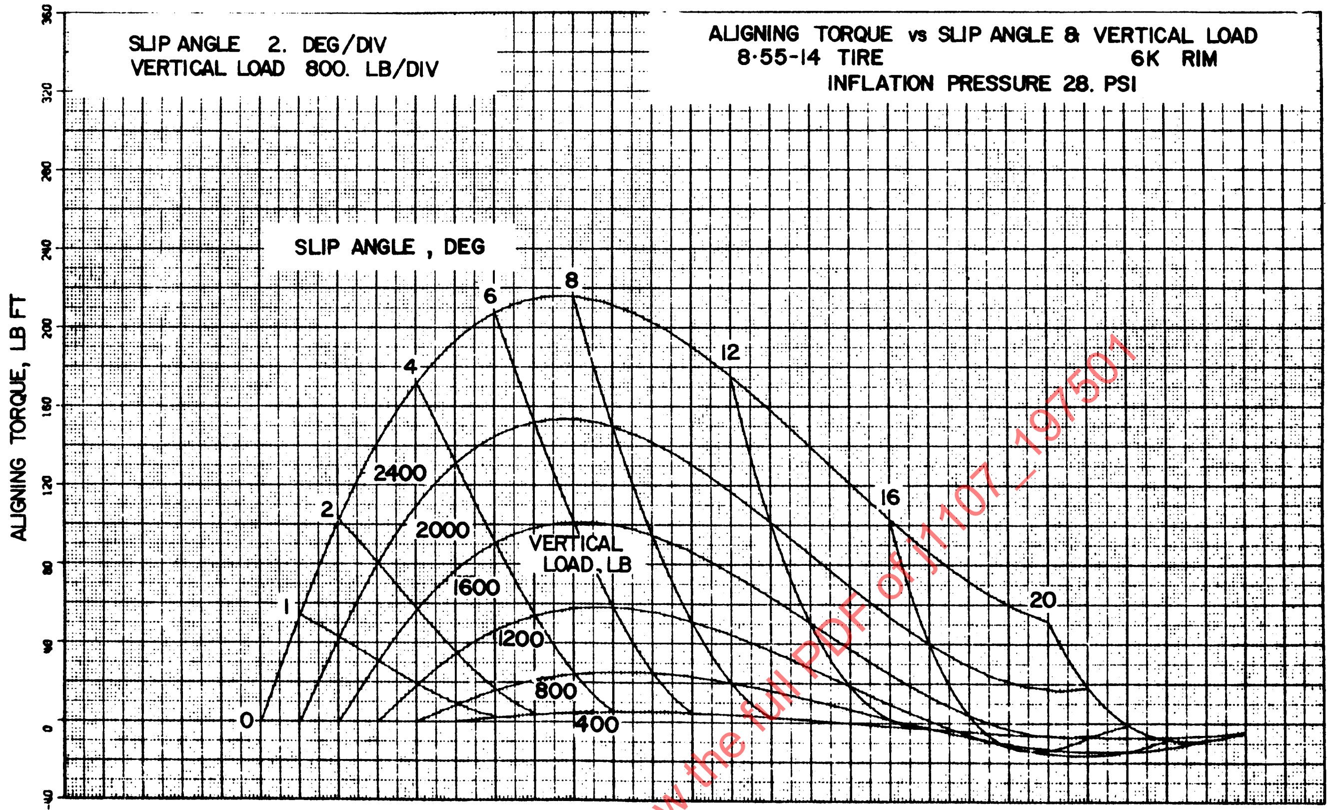


FIGURE 24—ALIGNING TORQUE CARPET PLOT

Another graphical technique that has been used in tire force and moment data presentation is the Gough plot. In his experimental work. V.E. gough points out the sensitivity of the automobile to fluctuations in lateral force and aligning torque (29,54). In his reports he has plotted lateral force versus aligning torque with slip angle and vertical load treated as parameters. The uniqueness of these plots is that one force or moment measurement is plotted against another force or moment measurement; in the case of carpet plots of classical plots, a force or moment measurement is plotted against another force or moment measurement; in the case of carpet plots of classical plots, a force or moment measurement is plotted versus a test parameter. An example is given below of gough plots (Figure 25 and Figure 26) used to compare radial and bias belted tires; vertical load and slip angle are test parameters (55).

Gough plots have proved to be more useful for manual steering applications than for power steering studies.

The two graphical techniques described above have evolved in the history of tire force and moment data presentation. Other less widely used special plots have been developed by individual testers to reflect their test objectives and research orientations. In order to provide uniformity in data presentation, it is recommended that the tire force and moment data be presented in the carpet plot format whenever it is possible to do so.