SKID RESISTANCE PROPERTIES OF TIRES
AND THEIR INFLUENCE ON VEHICLE CONTROL

Walter Bergman, Ford Motor Company

This paper provides an analysis of skid resistance of tires and vehicles in several operating modes - braking, acceleration, cornering and their combination. It shows the limitations of skid number as a criterion of skid resistance. It describes the tire skid resistance characteristics as a relationship between the braking (driving, cornering) coefficient and wheel slip or slip angle, within the range of coefficient between its peak and slide values. It introduces the concept of peak and slide traction envelopes as boundaries of tire skid resistance characteristics and applies it for analysis of skid resistance characteristics of a tire in cornering with braking or driving torque application. It analyzes tire characteristics in braking-in-a-turn in a complete skid at slip angles up to 90 degrees. It shows that, within a slip angle range between 20 and 90 degrees, the lateral force measured on a free rolling tire decreases with an increase of slip angle; however the force measured on a locked wheel increases. At a 90 degree slip angle, the lateral force measured on a locked wheel reaches the value at the force on the free rolling wheel. It shows effects of proportioning valves and different types of anti-skid systems on braking performance in straight line driving and cornering. Relationships between understeer/oversteer and vehicle skid resistance are shown. Effects of power-to-weight ratio and fore and aft roll stiffness distribution on vehicle skid resistance in cornering with full throttle acceleration are shown.

The volume of literature dealing with skid resistance is probably larger than on any other subject related to motor vehicles and tires. It is quite common in most publications to treat skid resistance as a pavement property. This may be well justified because road surface properties influence the skid resistance more than either tires or automobiles.

Automobiles travel on the roads at different speeds in different environmental and operating conditions. These vehicles perform different maneuvers as they accelerate, brake, and corner. Skid resistance properties of roads built for these vehicles are measured predominately by means of special devices at a specified speed and load, with locked wheels in braking and by using a special straight ribbed tire which is basically different from tires used on motor vehicles. Two fundamental questions will be examined in this paper.

1. How can measurements of road skid resistance (skid number) be related to skid resistance properties of tires used on motor vehicles in a variety of operating conditions?
2. What are some ways in which skid resistance properties of tires and vehicle design factors influence vehicle control in different modes of operation? This paper will attempt to answer these questions.

Tire Skid Resistance

Definition of Tire Skid Resistance

Skid resistance is the shear force developed at the tire-road interface when the tire has lost the grip with the road and slides over its surface. This shear force opposes sliding and acts in a direction opposite to that of sliding. The skid resistance force is synonymous with traction force. Although the skid resistance force results from interaction between tire and the road, it is frequently treated as an indicator of the pavement friction property. The magnitude of this force is influenced by pavement properties and the wheel load. The skid resistance force is normalized by dividing it by the wheel load. The result is called a shear force coefficient. Depending on the mode of operation, this normalized force is called the braking, driving or lateral force coefficient:

$$\mu_x = \frac{F_x}{F_z}$$

$$\mu_y = \frac{F_y}{F_z}$$

where:

μ_x = braking or driving coefficient

μ_y = lateral force coefficient

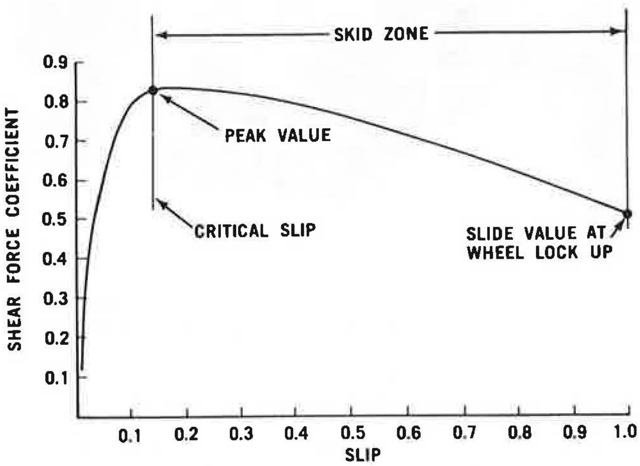
F_x = braking or driving force

F_y = lateral force

F_z = vertical load

The values of these coefficients vary widely with the amount of slip at the tire-road interface (as shown in Figure 1.) With an increase of slip, the shear force coefficient increases steeply to a peak value, and then starts to decrease very gradually until it reaches its slide value at 100 percent wheel slip (locked wheel condition).

Figure 1. The relationship between the shear force coefficient and wheel slip.



It should be noted that these coefficients are not limited to skid conditions but apply over the entire range of tire slip. The decrease of shear force coefficient beyond the peak value is produced by the loss of grip between the tire and road surface. Consequently, the skid resistance property of a tire is a relationship between the shear force coefficient and slip, which occurs in the range beyond the peak value of shear force coefficient. The slip value that corresponds with the peak value of shear force coefficient is defined as the critical slip. The range beyond the critical slip value may be called a skid zone in which the shear force coefficient is controlled primarily by friction between tire and the road. Therefore, in the skid zone, the shear force coefficient may be referred to as a friction factor. In the region below the critical slip, the relationships between the shear coefficient and slip is controlled primarily by elastic tire properties and is essentially independent of friction. This is particularly true at very low values of slip. Therefore, in this region, it is improper to refer to the shear force coefficient as a friction factor.

Skid Number

Since skid resistance is determined by frictional properties of the road surfaces and tires, it is necessary to eliminate the effect of tires in order to determine the skid resistance property of a pavement. This is done by using a special straight ribbed ASTM tire installed on a towed trailer. Measurements are always conducted on the locked wheel, in braking on a wet road surface. The wet surface is used in order to improve discrimination between different road surfaces. Measurements are conducted for strictly controlled test conditions: speed, wheel load, water film thickness, etc., as specified in the ASTM Standard. The skid resistance of a road surface obtained by using the above method is expressed in terms of a Skid Number which is equal to the slide value of braking coefficient multiplied by 100

$$SN = 100\frac{F_x}{F_z}$$

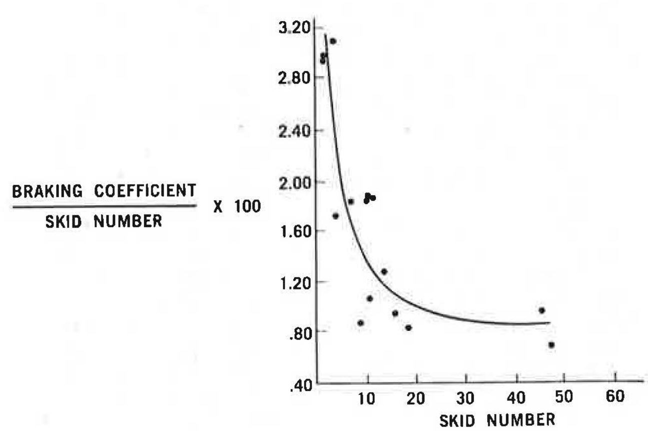
where:

F_x = braking force measured on the locked wheel

The Skid Numbers of different pavements can be used only as approximate indicators of skid resistance for automobiles on these pavements. It is generally understood that vehicle skid resistance improves with an increase of skid number. Quantitatively, however, there may be considerable differences between pavement skid numbers and compatible values measured on automobiles for different modes of operation. These differences exist for the following reasons:

First, the skid resistance properties of tires used on automobiles differ from those of the ASTM tire used for determination of a skid number. For instance, on polished wet surfaces, the skid resistance of an ASTM tire or a similar tire with the same straight rib tread pattern is considerably lower than that of conventional tires used on motor vehicles. However, the trend reverses itself on the high friction surfaces, where the ASTM tire shows a higher Skid Number than the conventional tread pattern tire (see Figure 2).

Figure 2. Effect of road surface friction on the ratio of braking coefficient to skid number for O.E. tires.



Second, tire operating conditions (load, speed, pavement wetness, etc.) prevailing on the ASTM trailer used for measuring skid number may considerably differ from those of a vehicle. The skid resistance usually decreases with an increase of speed and an increase of wheel load. The effects of speed and load are particularly pronounced on very slippery surfaces.

Third, the Skid Number describes only skid resistance in braking on a locked wheel, which represents only the end point of a total skid resistance characteristic in braking. Nor does the Skid Number describe skid resistance in other modes of operation such as driving, cornering, and their combinations.

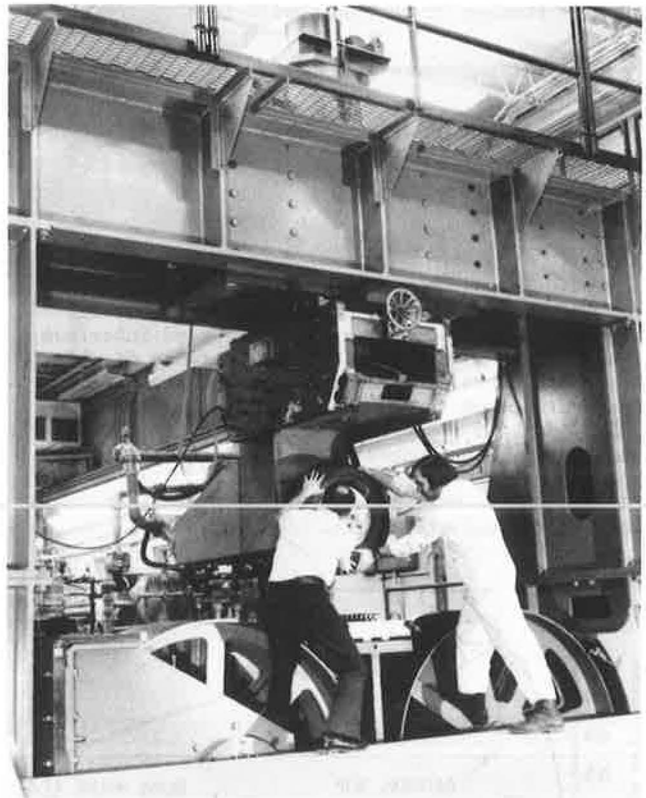
Measurements of Tire Skid Resistance

Measurements of tire skid resistance are generally conducted in a braking mode and very limited test data for the cornering mode appear in the literature (1,2). (Numbers in parentheses designate References at end of paper). No complete tire skid resistance data are available for the braking-in-a-turn mode. Most tire cornering tests, so called force-and-moment properties tests, were conducted at relatively small slip angles up to 10 or 20 degrees. (The slip angle is a measure of slip in cornering, and is defined as an angle between the plane of the wheel and direction of travel of the center of tire contact.) Braking-in-a-turn tests were also limited to relatively small slip angles. Since these values of slip angle were below the critical slip value in most cases, these tests primarily determine elastic tire properties and do not cover skid resistance properties. Limitations in capability of test equipment can be cited as one of the major reasons for lack of tire skid resistance data in cornering and braking-in-a-turn modes. To obtain a complete skid resistance characteristic for tires in cornering and braking-in-a-turn, tests should be conducted through a range of slip angles up to 90 degrees. A 90 degree slip angle produces 100% slip and therefore represents a condition equivalent to that prevailing on a locked wheel in braking.

The purpose of the tests described in this paper was to fill out this gap in our knowledge of tire skid resistance in the cornering and braking-in-a-turn modes. The measurements were performed on the flat belt TIRF machine at Calspan Laboratories (see Figure 3). Tests were conducted by applying brakes up to wheel lock in tires operated at slip angles up to 90 degrees. Measurements were performed at a 48.3 Km/h (30 mph) speed on dry and wet surfaces with Skid Numbers of 85 and 30 respectively. A "Safety Walk" surface manufactured by the 3M Company was used in tests. The original surface produced a Skid Number of 60 for the wet condition. To reduce the wet Skid Number to 30, the surface was honed. The water film thickness employed in testing was 0.5mm (0.02 in.) The Skid Number of the wet surface was measured at 64.4 Km/h (40 mph) and 0.5mm (0.02 in.) of water depth by using the ASTM E-501 Standard Pavement Test. The Skid Number of the dry surface was measured in the same manner except without water application. J78-15 and JR78-15 tires were selected for this test program. Each tire was broken in by running at 9.6 Km/h (6 mph) speed at design load for 61m (200 ft.) at a slip angle of +6° and then -6°. All measurements were conducted with a 6.222 KN (1400 lb) vertical load and 165.5 KPa (24 psi) inflation pressure.

Tests on wet and dry surfaces were conducted by applying steer ramp inputs from 0° to ± 1°, + 2°, + 4°, + 8°, + 16°, + 32°, + 48°, + 60°, + 72°, and + 90 slip angle at a rate of 10 deg/sec. As soon as the specified value of slip angle was reached,

Figure 3. Calspan high speed, moving belt machine.



brakes were applied at a rate necessary to lock the wheel within 1.0 second. The brake application changed the wheel slip from 0 to 1.0 (100%). The values of forces, moments, wheel slip, and slip angles were logged continuously during these tests at a sampling rate of 50 samples per second. In addition to tests described above, tests on the wet surface also included steer of the locked wheel. These tests were conducted by applying brakes to lock the wheel and then steering the wheel through a range of slip angles from -1° to +48° and from +32° to +90° at a rate of 10°deg/sec. Wheel lock up occurred when the slip angle reached 4 to 6 degrees. The sampling rate during this phase of test was 100 samples per second. The total required range of slip angle was divided into two parts because the TIRF machine could not sweep through the whole range from -1° to +90°. The machine head was re-adjusted after completion of the first series of tests between -1 and +48°. Tests at +32° and +48° were conducted in both series of tests in order to establish test repeatability after re-adjusting the machine head.

To reduce wear effects, two tires of each type were tested. One tire in the slip angle range between -1° and +48° and the other between +48° and +90°.

Tire Skid Resistance Characteristic in Braking

Skid resistance characteristics of the J78-15 tires in a braking mode, measured on wet and dry surfaces with skid numbers of 30 and 85 respectively, are shown in Figure 4. The braking coefficient increased steeply in the range of wheel slip between 0 and 0.08. Within this range, the curves for wet and dry surfaces were identical. This indicated that, within this region, road friction has

Figure 4. Braking coefficient versus wheel slip characteristics of J78-15 tire measured on dry and wet surfaces.

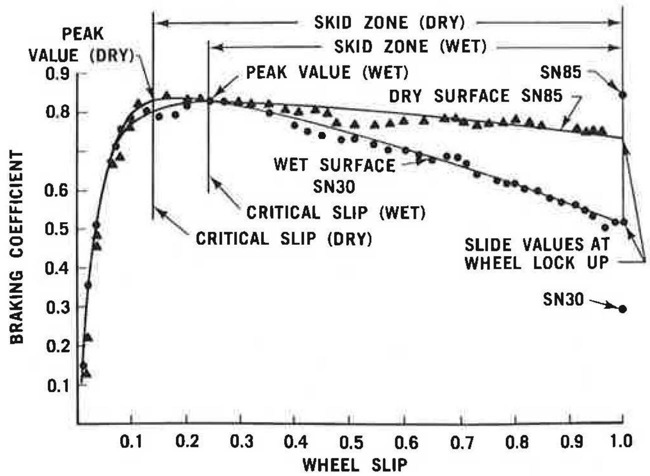
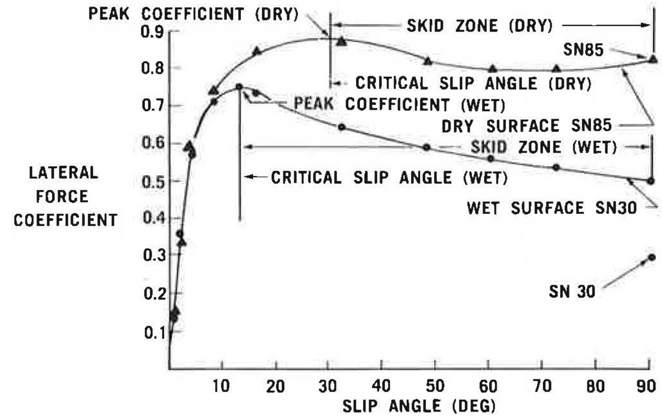


Figure 5. Lateral force coefficient versus slip angle characteristics of J78-15 tire measured on dry and wet surfaces.



practically no effect on the relationship between braking coefficient and wheel slip. The effects of road friction first begin to appear near the critical slip value which is reached at 0.14 on the dry surface and at 0.24 on the wet surface. The peak coefficient on the dry surface was only slightly larger than that on the wet surface. In spite of similarities between tire characteristics obtained on wet and dry surfaces in the region below critical slip, considerable differences are observed in the slid region above critical slip. On the dry surface, the braking coefficient reduces from 0.84 at its peak value to 0.74 at wheel lock up, thus showing a 12% reduction. Contrary to this, on the wet surface, the braking coefficient showed a 37% reduction which is three times larger than that of the dry surface. This indicates that, on low friction road surfaces, the tire experiences a considerable loss in its braking force by locking the wheel. This illustrates the significance of specifying both peak and slide braking coefficients for describing tire skid resistance properties. Figure 4 shows that, on a wet surface, the braking coefficient measured on a locked wheel is higher than the Skid Number divided by 100, however, the opposite is true on a dry surface. This shows the limitation of a Skid Number as a criterion of tire skid resistance in braking for even the most compatible operating conditions.

Tire Skid Resistance Characteristic in Cornering

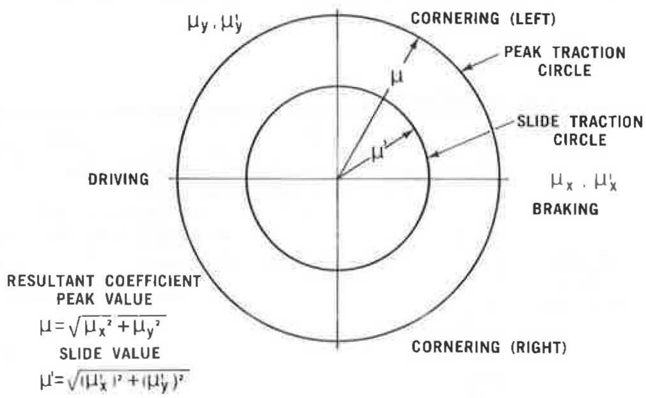
Tire skid resistance characteristics in cornering are described by the relationship between lateral force coefficient and slip angle shown in Figure 5. This plot shows the results of measurements on J78-15 tire for wet and dry road surfaces with skid numbers of 30 and 85 respectively. These curves show a general similarity to the braking coefficient curves for the same tire shown in Figure 4. The lateral force coefficient increases up to its peak value reached at a critical value of slip angle and then shows a gradual decline with further increase of slip angle. In spite of this general similarity between braking and lateral force coefficient curves, some differences can also be noticed.

The lateral force coefficient measured on the wet low friction coefficient surface reaches its peak at a lower value of slip angle than that measured on the dry high friction coefficient surface. An opposite trend was observed in braking. The lateral force coefficient for the dry surface showed a decrease from its peak value to a minimum value reached at about a 70 degree slip angle and then experienced a slight increase between 70 and 90 degrees. This finding qualitatively agrees with the test results published by Rompe (1,2). Contrary to this, the braking coefficient showed a continuous decrease with an increase of wheel slip between its critical value and wheel lock up. On the wet surface, the lateral force coefficient also showed a continuous decrease with an increase of wheel slip. However, the shape of the lateral force coefficient curve differed from that of braking coefficient curve, the former is concave and the later convex. It is interesting to note that, at 100% wheel slip, the values of both coefficients are almost identical. The average difference between lateral force coefficient values measured on dry and wet surfaces is larger than that between dry and wet surface braking coefficients. A "slide" value of lateral force coefficient is reached at a 90 degree slip angle. This condition corresponds to 100% wheel slip on a locked wheel in braking. At a 90 degree slip angle, the tire ceases to rotate and completely loses its grip with the road surface. This illustrates that tire rotation can be stopped by changing its angular orientation with respect to the direction of motion (slip angle), without applying braking torque. At a 90 degree slip angle, the lateral force coefficient measured on the dry surface has the same value as the Skid Number. On the wet surface, however, this coefficient was considerably higher than the Skid Number.

Tire Skid Resistance Characteristic in Braking-In-A-Turn

In braking-in-a-turn, tire skid resistance properties are determined primarily by the peak and slide values of the resultant braking-cornering coefficient. In an imaginary idealized case, the resultant peak and slide coefficients may be assumed constant for all values of their components. In this case, the relationship between lateral and braking coefficients can be described by the two concentric circles shown in Figure 6. The larger circle expresses the peak value of the resultant coefficient and the smaller circle describes the slide value.

Figure 6. Peak and slide traction circles.



$$\mu = \sqrt{\mu_x^2 + \mu_y^2}$$

$$\mu' = \sqrt{(\mu'_x)^2 + (\mu'_y)^2}$$

where:

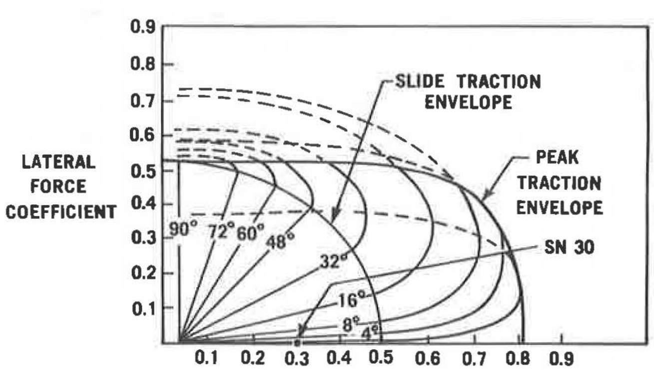
μ, μ' = resultant braking cornering coefficient, peak and slide values respectively

μ_x, μ'_x = braking coefficients, peak and slide respectively.

μ_y, μ'_y = lateral force coefficients, peak and slide respectively.

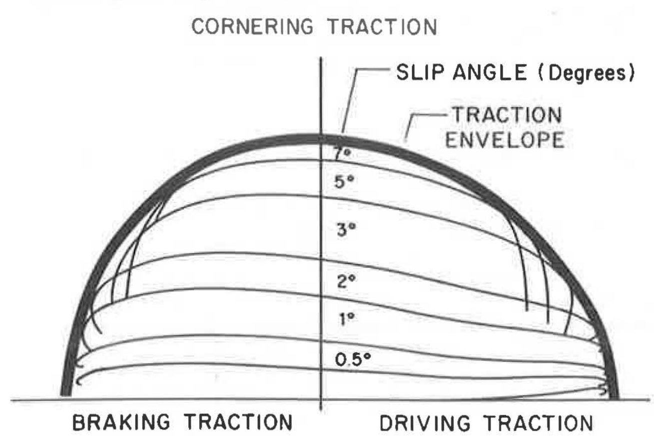
True tire skid resistance properties for braking-in-a-turn can be determined from the XY plots of lateral force and braking coefficients shown in Figure 7. Individual plots are made for different slip angles in the range between 1 and 90 degrees.

Figure 7. Peak and slide traction envelopes of J78-15 tire in braking-in-a-turn determined from measurements within a slip angle range between 1 and 90° on a wet surface with skid number SN30.



These plots were derived from the test data obtained from braking the steered tire up to wheel lock up. A curve fitted to the ends of the curves plotted for the individual slip angles describes the slide values of the resultant cornering-braking coefficient. The same curve can be plotted from the test data obtained from steering a tire locked by braking application. Both test methods - braking of a steered tire and steering of the braked tire produced the same results. By establishing the maximum values of the resultant coefficients for each slip angle and plotting the curve through these points we obtain a curve containing the peak values of the resultant cornering-braking coefficient. These curves can be called peak and slide traction envelopes. For this particular tire, the slide traction envelope has a circular shape. However, the peak traction envelope has the shape of an oval for which the height is considerably shorter than its length. The oval shape of the peak traction envelope shown in Figure 7 can be obtained only by using the values of lateral force coefficient measured within a wide range of slip angle, beyond its critical value. As shown in Figure 6, the lateral force coefficient decreases with an increase of slip angle in the region beyond its critical value. Because of this decrease, the value of lateral force coefficient measured at very large slip angles is smaller than that measured at lower slip angles, consequently, a certain portion of the lateral coefficient versus braking coefficient curves lie outside of the peak traction envelope as shown in Figure 7. Contrary to this, by plotting the peak traction envelope for a smaller range of slip angle, below its critical value, all curves are enclosed by the envelope as is shown in Figure 8 (3). This illustrates that the conventional concept of a friction circle is applicable only to sliding traction measured on a locked wheel and cannot be used for peak traction measured on a rolling wheel. The lateral force versus braking coefficient curves plotted for individual slip angles describe skid resistance as well as elastic properties of tires braking-in-a-turn. The solid portion of these curves located between the peak and slide traction envelope describe skid resistance characteristics of the tire or transition from peak-to-slide at each slip angle. By plotting tangents to the skid resistance curves through the points of intersection with the slide traction envelope, it can be noticed that these tangent lines merge into a common point located on the horizontal coordinate axis. The angle between each of these tangents and the horizontal axis is equal to the slip angle.

Figure 8. Peak traction envelope of 6.00-15 tire in braking- and driving-in-a-turn determined from measurements within a slip angle range between 0.5 and 10°, on dry surface.



The distance between this common point and the origin of the coordinate system is equal to the braking coefficient of a tire at zero wheel slip (free rolling tire) which constitutes the tire rolling resistance normalized by the vertical load. The wheel slip was computed in accordance with the

SAE definition by using the following equation:

$$S = \frac{N R_1}{KV \cos_\alpha} - 1$$

where:

- N = wheel rotations per minute (rpm)
- R_1 = tire loaded radius (in) defined according to the SAE J670c.
- V = road speed (mph)
- α = slip angle (deg)
- K = unit conversion constant. For English system K = 168.07

More recent investigations (4) showed that the translational velocity of a free rolling tire is greater than the product NR_1. The effective rolling radius is only slightly smaller than the undeflected radius. By using an effective rolling radius instead of the loaded radius in computations of wheel slip, it is expected that, for zero wheel slip, the longitudinal force at the tire-road interface would be zero. Therefore, by using this new concept for calculations of wheel slip, the point of convergence of the slip angle lines would be at the coordinate origin.

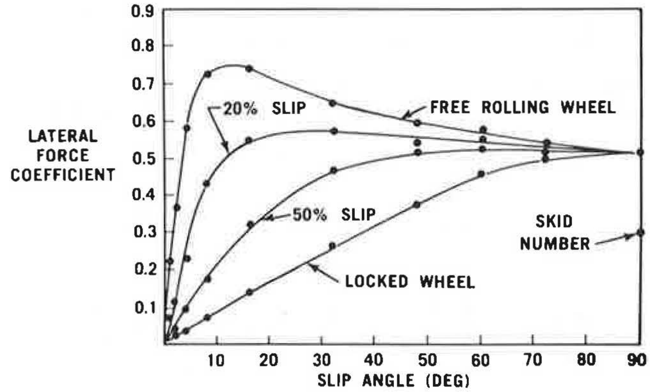
The dotted portions of the lateral force coefficient versus braking coefficient curves describe primarily elastic properties of a tire at a time prior to reaching the peak value of the resultant coefficient.

The lateral force coefficient vs braking coefficient plot represents a complete tire skid resistance characteristic for any possible operating condition of braking-in-a-turn. Similar plots can be used for cornering during driving torque application. The dot shown on the horizontal axis represents the Skid Number. The Skid Number is located completely outside the range of tire-road skid resistance properties. This again demonstrates that Skid Numbers cannot be used for descriptions of tire skid resistance.

Figure 9 shows another way of presenting tire characteristics for braking-in-a-turn. It shows lateral force coefficient vs slip angle relationships for a free rolling tire and for a braked tire at different amounts of slip, including 100% slip in a locked wheel condition. It is interesting to note that, at the 90 degree slip angle, all curves merge into a single point which expresses the slide value of lateral force coefficient. This indicates that, at a 90 degree slip angle, the lateral force capability of a free rolling tire and that of a locked wheel are the same. Contrary to this, at small slip angles, the lateral force coefficient of a braked tire in general and that of a locked wheel in particular is many times lower than that of a free rolling tire. This reduction of the lateral force coefficient produced by locking a wheel in braking can influence vehicle directional control in braking.

The graph shown in Figure 9 can be used to illustrate how tire skid resistance properties influence vehicle skid resistance. Let us assume that a vehicle negotiates a cornering maneuver with front and rear tires operating at 4 and 3 degree slip angles respectively. Let us further assume that by brake application, the front wheels develop 20% slip and the rear wheels become locked. Using these assumptions it can be seen that, as a result of this brake application, the lateral force coefficient would decrease from 0.55 to 0.25 on the front and from 0.45 to 0.03 on the rear. This illustrates that on locked wheels the lateral force coefficient on the front of the vehicle is eight times greater than that on the rear.

Figure 9. The relationship between lateral force coefficient and slip angle determined for J78-15 tire from measurements on free rolling and braked wheel on wet surface with skid number SN30.



Because of sharply reduced lateral forces on the rear tires, they are no longer capable of resisting the vehicle yaw induced by forces on the front tires and the vehicle may spin out. Let us assume that, if spin out occurs, and the brakes continue to be applied, the vehicle will turn 90 degrees which will result in a corresponding increase of slip angles on the front and rear tires. At small slip angles, between 0 and 10 degrees, the lateral force coefficient of the front tires running at 20% slip will increase faster than that of the locked rear wheels as shown in Figure 9. Consequently, the difference between the lateral force coefficients on front and rear tires will continue to increase and intensify vehicle spin out. For the tire shown in Figure 9, the difference between front and rear lateral force coefficients will continue to increase until slip angles reach approximately 15 degrees. Therefore, the yawing moment causing vehicle spin out reaches its maximum value at about 15 degrees slip angle. With further increase of slip angle, the difference between front and rear lateral force coefficients and consequently the yawing moment show a progressive decrease until their values reach zero at a 90 degree slip angle. Similar phenomena can be shown to occur in cornering during full throttle acceleration for a rear wheel drive vehicle.

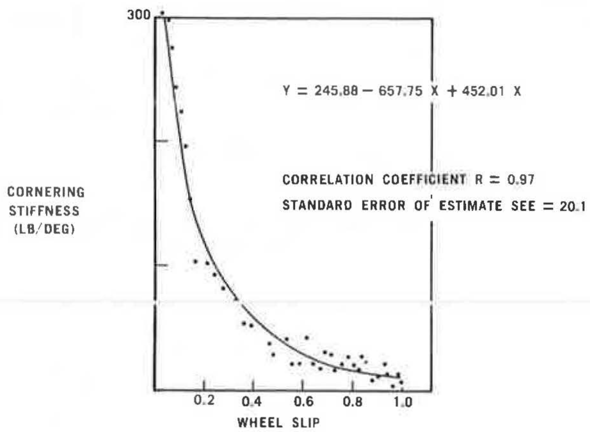
Let us now assume that the front wheels become locked and the rear wheels develop 20% slip. This results in a drastic reduction of lateral force on the front and only a small decrease on the rear. The rear forces will tend to straighten the vehicle's path of travel and the vehicle will leave the curve. Full throttle acceleration in cornering on a front wheel drive vehicle can be shown to produce a similar effect.

Loss of vehicle directional control due to excessive braking or throttle application may occur not only in cornering but also in straight ahead driving. Vehicle directional stability in a straight ahead driving is controlled primarily by the property of a tire which is called cornering stiffness. Cornering stiffness is defined as the rate of change of lateral force coefficient with respect to change of slip angle.

$$C_y = \frac{dF_y}{d\alpha} \qquad \alpha \longrightarrow 0$$

Graphically cornering stiffness is expressed by the slope of the curve lateral force coefficient vs slip angle. Excessive braking or throttle application results in a large amount of wheel slip which causes a dramatic reduction of tire cornering stiffness as shown in Figure 10. The relationship between tire cornering stiffness and wheel slip can be approximated by a second power polynomial. Regression analysis of test data for the J78-15 tire shows a correlation coefficient R = 0.97.

Figure 10. Effect of wheel slip on cornering stiffness of J78-15 tire on wet surface with SN30.



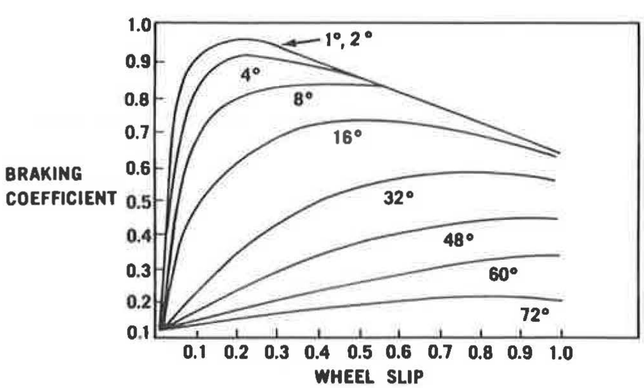
Note: 1 lb/deg = 4.44 N/deg

Cornering stiffness determines the amount of vehicle yaw resulting from side wind or forces due to road irregularities. For small slip angles, lateral forces resisting vehicle yaw are equal to the product of slip angle and cornering stiffness. Therefore, for the tire shows in Figure 10, a ten fold reduction of cornering stiffness produced by the wheel slip due to braking or driving torque will increase vehicle yaw ten fold. Such excessive yaw may cause vehicle spin out and loss of control.

So far, we have shown how braking and throttle applications affect tire lateral forces and consequently vehicle directional stability in a turn and in straight ahead driving. Now we are going to examine how the steer of a tire affects tire force-slip characteristics in a braking mode and therefore vehicle braking performance.

Figure 11 shows the relationship between braking coefficient and wheel slip, in a family of curves plotted for different slip angles within the range between 1 and 90 degrees. The peak value of braking coefficient decreases with an increase of slip angle until it reaches zero at a 90 degree slip angle.

Figure 11. The relationship between braking coefficient and wheel slip at different slip angles, determined from measurements conducted on J78-15 tire on wet surface with skid number SN30.



This illustrates that, when the tire is prevented from rotation by its angular position at a 90 degree steer angle, it completely loses its capability to produce braking forces.

The value of critical slip at which the peak value is reached shifts toward larger values with an increase of slip angle. At slip angles above 32 degrees, a critical slip or peak value of braking coefficient is never reached below 100% slip. The slide value (100% wheel slip or locked wheel condition) of braking coefficient remain the same for slip angles between 0 and 16 degrees. However, it shows a progressive decrease with further increase of slip angle beyond 16 degrees until it reaches zero at a 90 degree slip angle. Full throttle acceleration on a front wheel drive vehicle will produce a similar effect.

Effect of Vehicle Design Factors on Skid Resistance

Effect of Vehicle Design Factors on Skid Resistance in Braking

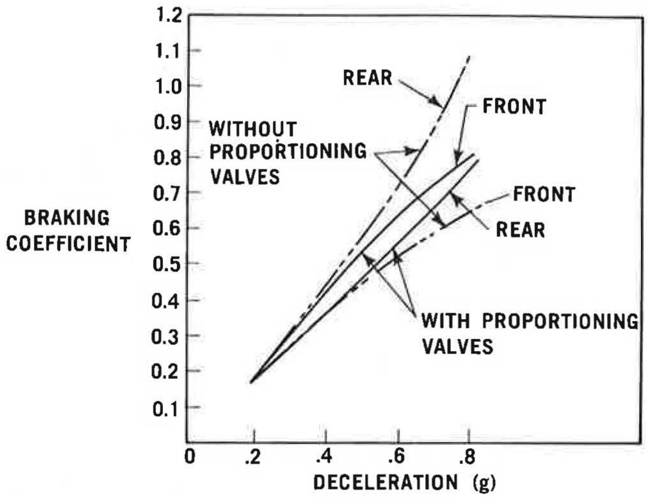
Vehicle skid resistance in braking can be influenced by two major design factors: proportioning valves and antilock systems. Proportioning valves control the distribution of braking torques between front and rear wheels within a wide range of vehicle deceleration. For optimal distribution of braking torques, it is necessary that the braking coefficients on the front wheels are slightly larger than on the rear wheels and the difference between their values remains approximately constant within the entire range of deceleration. This requirement is necessary in order to achieve the maximum effectiveness of braking within the limits of tire-road friction and to avoid lock up of the rear wheels before the front wheels.

To meet the above requirements, it is necessary to vary distribution of braking torques between front and rear wheels with a change of deceleration or change of brake line pressure, in order to counteract the change of braking coefficient due to fore-and-aft weight transfer. The weight transfer increases with an increase of deceleration which results in an increase of the weight on the front wheels and decrease on the rear wheels. Without using proportioning values, the braking coefficient of the rear wheels would experience a more than linear increase with an increase of line pressure while the front braking coefficient would show a less than linear increase as indicated by the dotted curves in Figure 12. Such a characteristic would result in locking the rear wheels first and increase the tendency for the vehicle to spin out. If spin out occurs, the full braking capacity of the vehicle cannot be utilized.

To avoid this undesirable phenomenon, the braking torque on the rear wheels should increase less than linearly with an increase of line pressure while the front braking torque increases linearly. The non-linear characteristic of rear brake torque is accomplished by means of proportioning valves. By using proportioning valves, the front and rear braking coefficients increases almost linearly with an increase of brake pressure and the rear braking coefficient remains always larger than the front. This characteristic is shown with solid lines in Figure 12. Such characteristics improve vehicle braking effectiveness by decreasing the stopping distance and reducing the potential for vehicle spin out resulting from locking of the rear wheels.

In spite of considerable improvement in braking effectiveness offered by proportioning valves the

Figure 12. The effect of proportioning valves on braking coefficient on front and rear wheels at different engine sizes.



theoretical maximum braking efficiency of the vehicle only can be achieved with expert drivers. To eliminate or to greatly reduce the dependence on driver's skill and further improve the braking efficiency, anti-skid devices are used. These devices measure the rate-of-change of wheel spin velocity. An excessive increase in the rate-of-change of spin velocity indicates that a wheel lock up condition is approached. The signal of the wheel spin velocity, after being processed in a micro-computer, actuates the brake pressure control valve by reducing the pressure as the rate-of-change of spin velocity reaches a certain level, thus preventing wheel lock up. Existing anti-skid devices can be divided into two major categories:

- Two rear wheel system
- Four wheel system

A comparative evaluation of the two rear wheel versus four wheel system (5) shows that the most obvious advantage of a four wheel system is the ability to steer the vehicle during panic braking.

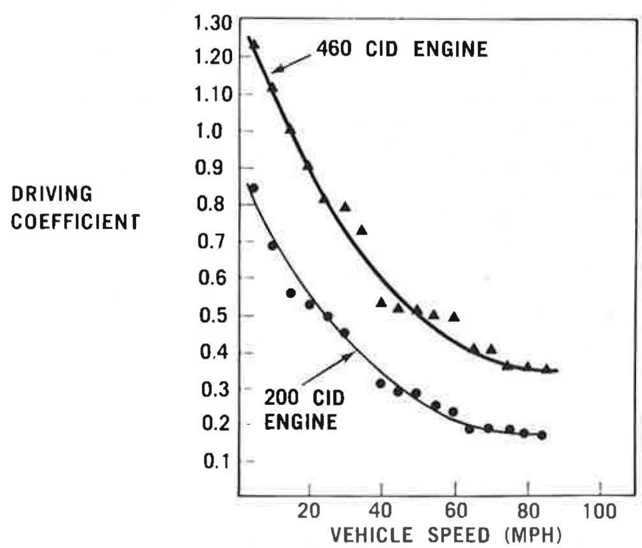
From the standpoint of stopping ability, it would appear that the four wheel system should outperform the two rear wheel system, particularly since the two front wheels normally carry the largest share of the braking load. On some road surfaces and conditions this is the case. In other instances, however, the expected improvement is not realized. On gravel, for example, locked front wheels may plough aside the looser material and allow the rear wheel system to operate on a firmer surface than would be possible with the front wheels rotating. On wet pavements, the locked front wheel has a more effective wiping action than the rotating wheel so that the rear wheel, following in its track, may be operating on a thinner water film or even on virtually dry pavement. The locked front wheel thus can contribute to the efficiency of the two rear-wheel system in some instances.

Effect of Vehicle Design Factors on Skid Resistance in Full Throttle Acceleration

In full throttle acceleration, vehicle skid resistance is determined primarily by the power-to-weight ratio and only at very low speeds on surfaces with low coefficients of friction is the limited by tire-road friction. The power-to-weight ratio can be expressed in terms of driving coefficient, which is defined as the ratio of the driving force to vertical wheel load. The driving coefficient reaches its maximum value in low gear at an engine rpm corresponding to the maximum value of the engine torque. On most vehicles, wheel spin occurs primarily at this condition. When the driving coefficient reaches the friction limit, the tire loses its grip with the road, and engine and wheel speeds increase rapidly until they reach their maximum values. Since the wheels are spinning, an increase in rotational speed is not followed by an increase of vehicle speed. This determines an increase of wheel slip and decrease of driving coefficient. The maximum wheel slip velocity at this condition is equal to the difference between the product of the maximum wheel speed and the effective rolling radius and the vehicle speed. At maximum slip velocity, wheel slip in excess of 100 percent is possible. At most speeds, the value of driving coefficient is limited by the available engine power rather than by tire-road friction.

Since the engine torque decreases with an increase of vehicle speed, the driving coefficient also decreases. The relationship between the driving coefficient and speed calculated for full throttle acceleration for two different engine sizes is shown in Figure 13. For a vehicle equipped with a 7.5 liter (460 CID) engine, the driving coefficient decreases from 1.25 at 4.8 Km/h (3 mph) to 0.35 at 128.7 Km/h (80 mph.) For the same vehicle equipped with a 3.8 liter (200 CID) engine, the driving coefficient decreases from 0.85 at 4.8 Km/h (3 mph) to less than 0.2 at 128.7 Km/h at 80 mph. However, the driving coefficient is limited not only by engine horsepower but also by tire-road friction.

Figure 13. The relationship between driving coefficient and vehicle speed determined for two different engine sizes.



Note: CID = engine displacement in in^3
1 in^3 = 0.016 liter
1 mph = 1.609 km/h

Tire-road friction determines the peak and slide values of the driving coefficient. Obviously, the driving coefficient which can be generated by the engine cannot exceed its peak value obtained on a given road surface. Let us assume, that the peak driving coefficient is equal to 0.8. Then the horizontal reference line drawn through the point equal to 0.8 will enable us to determine the speed, at which skid will occur, and also will show how well the engine torque and tire-road friction are utilized at different speeds. The points of intersection of this horizontal line with the driving coefficient curves will determine the speeds at which skid will occur. For driving coefficient characteristics shown in Figure 13. These speeds are 8 Km/h (5 mph) and 40 Km/h (25 mph), for the 3.3 liter (200 CID) and 7.5 liter (460 CID) engines respectively. This indicates that, because of friction limitations, the driving coefficient produced by the engine torque can be fully utilized only at speeds above these skid speeds. On the contrary, however, the tire-road friction can be fully utilized only at speeds below the skid speeds. Below the skid speeds, the percent of utilization of engine torque can be calculated as the ratio of the peak value of driving coefficient measured in a given road surface (in this example $\mu = 0.8$) to the value of driving coefficient generated by the engine (obtained from the curve). Above the skid speed the percent of utilization of tire-road friction can be determined as the ratio of the value of the driving coefficient generated by the engine to its peak value.

Effect of Vehicle Design Factors on Skid Resistance in Cornering

In cornering without braking or power application, skid resistance is determined primarily by vehicle understeering/oversteering properties. An understeering vehicle has a tendency to drift out of the curve in an extreme maneuver where the inertia forces exceed the cornering capacity of the tires. However, this phenomenon should be considered as a partial skid because the slip angle on the tires stays well below 90 degrees and lateral force coefficient has not reached its slide value.

Complete skid can occur only on an oversteering vehicle which has self-energizing properties resulting in a progressive increase of path curvature until breakaway of the rear tires and spin out occurs. Contrary to this, an understeering vehicle produces a self-energizing effect which tends to resist an increase in path curvature and thus prevents complete skid of the vehicle. Thus, understeer can be considered as a vehicle anti-skid mechanism in cornering.

Understeer/oversteer can be visualized as a vehicle servo-steer or yaw produced by lateral acceleration (7). The vehicle yaw angle produced by understeer/oversteer is called understeer/oversteer angle and can be calculated as the difference between the front and rear slip angles.

$$u = \alpha_f - \alpha_r$$

where:

u = understeer/oversteer angle

α_f, α_r = slip angle front, rear respectively

Understeer/oversteer is defined as the rate of change of understeer angle with respect to change in lateral acceleration:

$$\frac{du}{d\ddot{y}} = \frac{d\alpha_f}{d\ddot{y}} - \frac{d\alpha_r}{d\ddot{y}}$$

where:

$\ddot{y}$ = lateral acceleration

The vehicle understeers if this difference is positive and oversteers if it is negative.

It has been shown (6) that terms $\frac{d\alpha}{d\ddot{y}}$ are inverse values of cornering stiffness coefficients

$$\frac{d\alpha}{d\ddot{y}} = \frac{1}{C}$$

where:

c = cornering stiffness coefficients which is defined as cornering stiffness normalized to the wheel load

$$c = \frac{d\ddot{y}}{d\alpha} = \frac{1}{F_z}\frac{dF_y}{d\alpha}$$

where:

F_y = lateral force

F_z = wheel load

The vehicle understeers if the cornering stiffness coefficient of the rear tires is greater than that of the front tires.

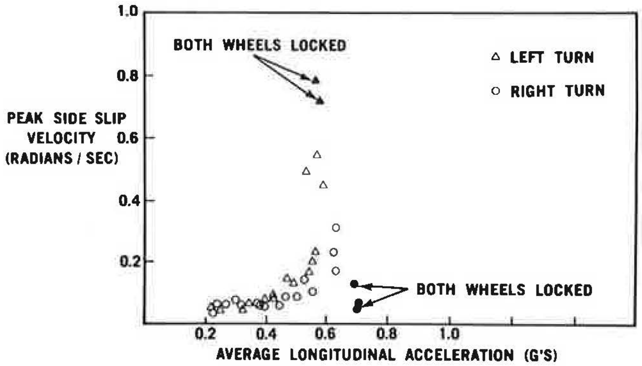
For a given tire, the cornering stiffness coefficient is primarily affected by wheel load, slip angle and tire inflation pressure. Cornering stiffness coefficient decreases with an increase in wheel load, an increase in slip angle or a decrease in inflation pressure.

Effect of Vehicle Design Factors on Skid Resistance in Braking-in-a-Turn

In addition to understeer/oversteer, vehicle skid resistance in cornering is also affected by the distribution of braking torques between left and right wheels. In an ideal case, the braking torques should be equally distributed between the left and right wheels of the front and rear axle. However, due to non-uniformity in frictional characteristic of the brake lining, unequal wear, brake adjustment variations, etc., there may be some difference between these torques.

The effect of unequal torque distribution on vehicle skid resistance in cornering during braking application can be illustrated by citing one of the Highway Safety Research Institute (HSRI) tests (8) as an example. The vehicle used in this example had considerably larger braking torques on the left wheels than on the right wheels. Test results were presented by plotting the peak value of sideslip velocity measured in left and right turns vs vehicle deceleration (Figure 14). (The sideslip velocity may be considered as a measure of vehicle spin-out, and is defined as a time derivative of a vehicle sideslip angle. The sideslip angle is an angle between vehicle's longitudinal axis and the velocity vector at some specified point in the vehicle.) Figure 14 shows that under extreme conditions when the wheels become locked, the sideslip velocity in a left turn was about seven times larger than that in a right turn. In a left turn, the locking of the wheel resulted in an increase of sideslip velocity. Contrary to this in a right turn, the sideslip velocity decreased when the wheels become locked. These results may indicate that left side bias torque distribution contributed to vehicle skid in a left turn but seemed to oppose skid in a right turn. Examination of original recordings revealed that the wheels did not lock simultaneously but in a certain sequence: the inside rear wheel locked first, then the outside

Figure 14. Peak side slip velocity measured on a vehicle in braking in left and right turns at different levels of longitudinal deceleration.



rear, inside front and finally outside front. It was observed that there was a considerable difference between the lock-up times of individual wheels in left and right turns. In a left turn, there was a considerable time lag between lock up of the rear wheels and the outside front wheel. In a right turn, all four wheels were locked almost instantaneously.

The effect of unequal torque distribution on vehicle skid resistance can be explained by the interaction between lateral weight transfer in cornering and braking torque distribution. The lateral weight transfer increases the weight on the outer wheel and decreases it on the inner wheel. Thus the braking coefficients on the inner wheels will become larger than those on the outer wheels. A left biased braking torque distribution increases the braking coefficients on the left side wheels relative to the right side. Therefore, this particular torque distribution will tend to increase the difference between inside and outside wheel braking coefficients in a left turn and decrease it in a right turn. Skid occurs when a braking coefficient reaches its peak value. The large difference between braking coefficients in a left turn resulted in a considerable time lag between locking of the front inner and outer wheels. Since the front outer wheel was still rolling, it was generating a lateral force turning the vehicle inside the curve and thus increasing the sideslip velocity. In a right turn, the difference between braking coefficient was small and thus all four wheels were locked almost instantaneously. Locking of all four wheels resulted in a reduction of lateral tire forces to almost zero. Thus, the vehicle gradually drifted out of the curve as indicated by a reduction in sideslip velocity. It should be noted, that such a drastic effect of unequal torque distribution on vehicle skid resistance in left and right turns could materialize only when the rear wheels lock first. This emphasizes the importance of properly designed proportioning valve system which prevents locking of the rear wheels before the front wheels.

Effect of Vehicle Design Factors on Skid Resistance in Acceleration-in-a-Turn

Acceleration in cornering reduces tire cornering stiffness and therefore causes a reduction of lateral forces on the driven tires (9). This results in an oversteering effect on a rear wheel drive vehicle and an understeering effect on a front wheel drive vehicle. Lateral weight transfer in cornering reduces the load on the inner wheel, which increases the driving coefficient. Because of the increase in driving coefficient, vehicle skid may occur within a much wider range than for a straight ahead acceleration within this speed range. A rear wheel drive vehicle will tend to spin out and a front wheel drive vehicle to plow out. Thus, only a rear wheel drive vehicle can develop spin out during acceleration in cornering.

Vehicle sensitivity to skid will increase primarily with an increase in power-to-weight ratio. Distribution of roll stiffnesses between front and rear also influences vehicle sensitivity to skid. It will usually increase with an increase of rear roll stiffness relative to the front. The distribution of roll stiffness is greatly influenced by rubber bumpers restricting travel of the suspension arms. When a suspension arm reaches the contact with the bumper, the roll stiffness starts to increase very rapidly. The clearance between the bumper and suspension arm varies with vehicle load reacted by the rear springs. On heavily loaded vehicles, this clearance becomes very small and bumper contact is reached at very low levels of lateral acceleration. In this case, the rear roll stiffness will be considerably larger than the front stiffness. Such a vehicle will show a much steeper increase in driving coefficient on the rear tires which will increase its sensitivity to skid.

Conclusions

- Skid Numbers provide only a relative measure of road surface skid resistance and do not describe the skid resistance of tires used on motor vehicles.

- Skid resistance characteristics of tires in a single mode of operation are described by the relationship between force coefficients and slip. Skid Resistance characteristics show a transition from a peak value of force coefficient reached at critical slip to a slide value reached at 100 percent slip.

- In a dual mode of operation such as braking-in-a-turn, skid resistance characteristics of tires are described by the relationship between lateral force coefficient and braking coefficient within boundaries established by the peak and slide values of the resultant coefficient.

- Peak and slide traction envelopes describe peak and slide values of the resultant coefficient in braking-in-a-turn for a range of slip angles between zero and 90 degrees.

- The lateral force coefficient of a locked wheel increases with an increase of slip angle until it reaches the value of this coefficient for a free rolling wheel at a 90 degree slip angle.

- Tire skid resistance characteristics determined in this study provide a foundation for computer simulations of vehicle handling at performance limits.

- Vehicle skid resistance characteristics are influenced by brake features including proportioning valves and anti-skid systems, and also by vehicle design factors affecting vehicle understeer.

References

1. K. Rompe. Zur Wirksamkeit von Blockierreglern bei Kurvenfahrt. Deutsche Kraftfahrtforschung und Strassenverkehrstechnik Heft. 234, 1973.

2. K. Rompe. Zur Schleudern von Kraftfahrzeugen bei zu Schneller Kurvenfahrt. Deutsche Kraftfahrtforschung und Strassenverkehrstechuik Heft. 249, 1975.

3. W. Bergman, H. R. Clemett, N. T. Sheth. Tire Traction Measurement on the Road and in the Laboratory. SAE Paper No. 710630, June 7-11, 1971.

4. W. B. Crum. Road and Dynamometer Tire Power Consumption. SAE Paper No. 750955, October 13-17, 1975.

5. H. R. Clemett and J. W. Moules. Brakes and Skid Resistance. Highway Research Record, No. 477, 1973.

6. W. Bergman and H. R. Clemett. Tire Cornering Properties. Tire Science and Technology, Volume 3, No. 3, August, 1975, pp. 135-163.

7. W. Bergman. The Basic Nature of Understeer/ Oversteer. SAE Paper No. 957-B, January 11-15, 1965.

8. R. D. Ervin, et al. Vehicle Handling Performance Volume 2 Vehicle Response Data. Final Report Prepared by the HSRI for NHTSA under contract DOT-HS-031-1-159, November 1972, Page 34.

9. W. Bergman. Theoretical Prediction of the Effect of Traction on Cornering Force. SAE Paper 186-A, July 1960, SAE Transactions Volume 69, 1961.