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SURFACE VEHICLE INFORMATION REPORT

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ANTILOCK BRAKE SYSTEM REVIEW

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

The application of Antilock Brake Systems (ABS) to passenger cars and light trucks has grown in recent years. This has been fueled by advances in automotive electronics, competitive trends, and consumer safety awareness. Although technical literature exists regarding specific systems, hardware, and applications, little exists that addresses the topic from the viewpoint of the industry as a whole. Recognizing this need, the Antilock Brake Standards Committee was formed and began its work by compiling this document.

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1. ***Scope***—This SAE Information Report provides information applicable to production Original Equipment Manufacturer antilock braking systems found on some past and current passenger cars and light trucks. It is intended for readers with a technical background.

It does not include information about aftermarket devices or future antilock brake systems.

Information in this document reflects that which was available to the committee at the time of publication.

2. *References*

2.1 Applicable Publications—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.

1. Zellner, John W., "An Analytical Approach to Antilock Brake System Design," SAE Paper 840249, 1984.
2. Leiber, Heinz., Czinczel, Armin., and Anlauf, Juergen., "Antiskid System (ABS) for Passenger Cars," Bosch Technische Berichte (English Special Edition), Feb. 1982.
3. Ehlbeck, Jim., Moore, Tony., and Young, Warren., "Antilock Brake Systems for the North America Truck Market," SAE Paper 901174.
4. Ellis, J. R., "Vehicle Dynamics," London Business Books, 1969.
5. Flaim, T. A., "Vehicle Brake Balance Using Objective Brake Factors," SAE Paper 890804, February, 1989.
6. Rowell, J. Martin., Gritt, Paul S., editors, "Antilock Braking Systems for Passenger Cars and Light Trucks—A Preview (PT-29);" SAE, 1987.
7. Ward Automotive Reports; Ward's Communications

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8. Lowery, Joseph., "Jensen—Ferguson: 2+2 = 4;" February, 1966; p.u.
9. "Ward's Automotive Antilock Braking Systems In The 1990s," p. 2; Lamm, Michael., Ward's Communications, 1990.
10. ISO 611-1980 "Road vehicles—Braking of automotive vehicles and their trailers—Vocabulary"
11. ECE R13.05, Annex 13 "Requirements Applicable to Tests for Vehicles Equipped with Anti-Lock Devices"
12. SAE J670—Vehicle Dynamics Terminology

3. *Definitions*

3.1 ABS (Acronym for Antilock Brake Systems)—A device which automatically controls the level of slip in the direction of rotation of the wheel on one or more wheels during braking (see ISO 611).

3.2 Accumulator—Low Pressure (Sump)—A low pressure brake fluid storage device not intended as an energy source.

3.3 Accumulator—High Pressure—An energy storage device using pressurized brake fluid as the storage medium.

3.4 Actuation Principle—Pump Back—An ABS system configuration where during modulation control, low pressure brake fluid is restored to high pressure by a pump and made available for a subsequent build cycle. The total amount of fluid available for modulation control for a given stop is limited to the amount of fluid provided by the master cylinder for that particular stop.

3.5 Actuation Principle—Replenishment—An ABS system configuration employing an external source of high pressure fluid in addition to displaced master cylinder fluid for modulation control. This type of system has virtually an unlimited supply of high pressure fluid available during modulation control.

3.6 Add-On ABS System—An ABS configuration in which both the ABS power supply and modulation control functions are independent from the base brake actuation system. The components of this system may be packaged together or separately.

3.7 Control Channel—A portion of the hydraulic brake circuit which can be operated independently from other portions of the hydraulic brake circuit. In ABS braking, it is a hydraulic brake circuit that controls a wheel or wheels independently of other wheels.

3.8 Controller—A component of the antilock braking system which interprets input signals from the sensor(s) and transmits the controlling output signals to the modulator(s) (see SAE J670e).

3.9 Diagonal Split Brake System—A brake system in which separate hydraulic circuits actuate the service brakes for one front wheel and one rear wheel on the opposite side.

3.10 Directly Controlled Wheel—A wheel whose braking force is modulated according to data provided at least by its own sensor (see ECE Regulation 13).

3.11 G-Switch/Accelerometer G-Sensor/(Lateral and Longitudinal)—A device by which acceleration or a change in acceleration of the vehicle is detected or confirmed.

3.12 Indirectly Controlled Wheel—A wheel whose braking force is modulated according to data provided by the sensor(s) of other wheel(s) (see ECE Regulation 13).

3.13 Integrated ABS System—An ABS configuration in which some ABS and base brake actuation functions are shared. Most commonly, both systems may share a hydraulic power supply.

3.14 Lateral Force Coefficient—The ratio of the lateral force to the vertical load (see SAE J670e).

3.15 Longitudinal Force Coefficient—The ratio of the longitudinal force to the vertical load (see SAE J670e).

3.16 Modulation Control—The systematic regulation of braking force resulting from the build, decay, and/or hold of pressure to a given control channel.

3.17 Modulator—The component responsible for modulating the force developed by the brake actuators as a function of the order received from the controller (see ISO 611).

3.18 Nonuniform/Nonhomogeneous Coefficient of Friction—A braking tractive surface in which variable surface conditions exist.

3.19 Pedal Feedback—A tactile sensation felt by the driver's foot on the brake pedal during modulation control.

3.20 Pump Motor—A mechanical pump driven by an electric motor used to pressurize or move brake fluid.

3.21 Select High—Multi-wheel control where the signal of that wheel which is the last to tend to lock controls the system for all the wheels of the group (see ISO 611).

3.22 Select Low—Multi-wheel control where the signal of that wheel which is the first to tend to lock controls the system for all the wheels of the group (see ISO 611).

3.23 Slip—The difference between the angular velocity of a freely rolling wheel (ω) and the angular velocity of the braked wheel (ω_B) divided by the angular velocity of the freely rolling wheel (ω), expressed as a percentage.

$$\% \text{ SLIP} = \frac{\omega - \omega_B}{\omega} \qquad \text{(Eq. 1)}$$

3.24 Slip Angle—The angle between the wheel plane and the direction of travel of the center of the tire contact (see SAE J670e).

3.25 Solenoid—An electromagnetic device in which an electrically energized magnet moves an armature to open or close a hydraulic flow path (see ISO 611).

3.26 Split Coefficient (Split µ)—A braking tractive surface in which two significantly differing coefficients of friction exist at the left and right side of the vehicle.

3.27 Transition Coefficient (Transition µ)—A braking tractive surface in which two significantly differing coefficients of friction exist in the direction of travel of the vehicle.

3.28 Uniform/Homogeneous Coefficient of Friction—A braking tractive surface in which no significantly differing coefficient of friction exist throughout the surface.

3.29 Vertical Split Brake System—A brake system in which separate hydraulic circuits actuate the service brakes, one for both front wheels and one for both rear wheels.

3.30 Wheel Speed Sensor—The component responsible for sensing the condition of rotation of the wheel(s) and for transmitting this information to the controller.

3.31 Yaw Rate (r)—The angular velocity about the (vehicle's) vertical axis (see Figure 1).

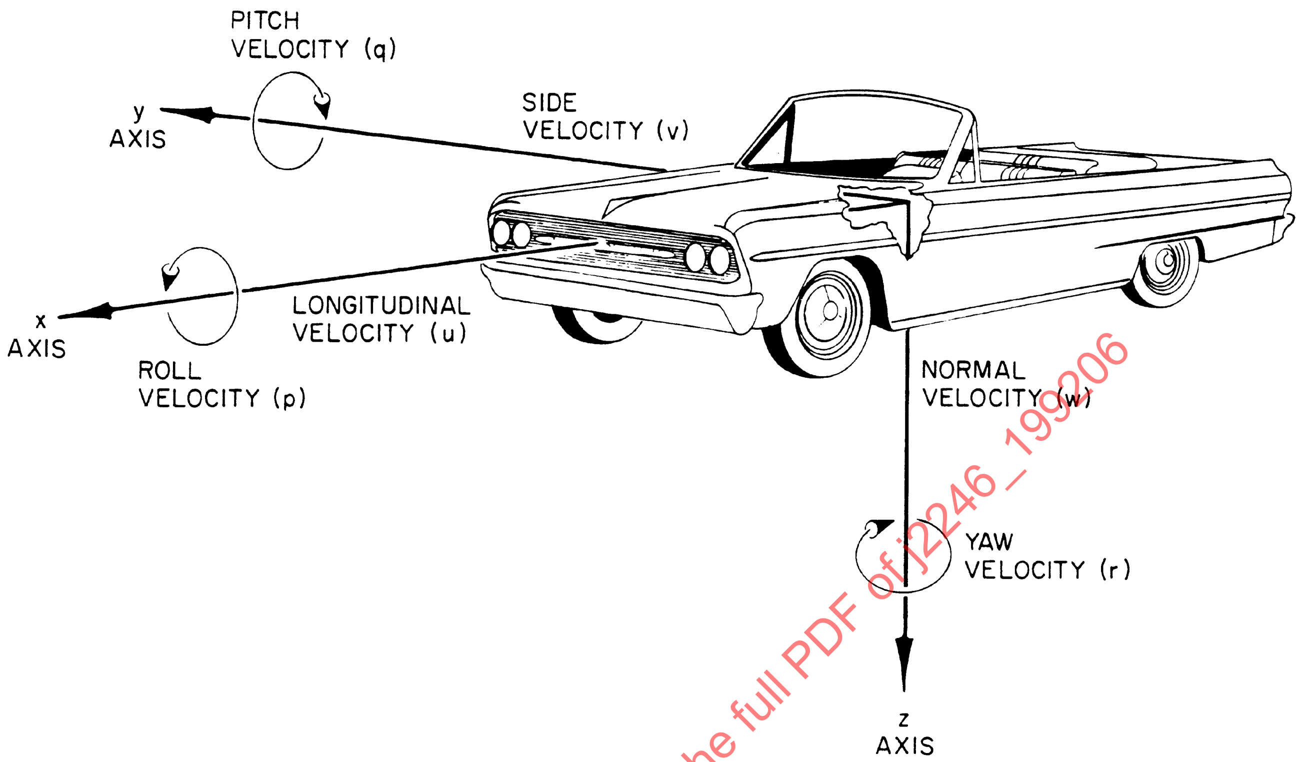


FIGURE 1—DIRECTIONAL CONTROL AXIS SYSTEM

4. *Nomenclature*

a_x — Acceleration along the x (longitudinal) axis of the vehicle
e — Base of Napierian logarithmic system, (2.7182....)
G — Brake specific torque
g — Acceleration due to gravity
J — Wheel inertia
M — Vehicle mass
P — Brake pressure
P_0 — Initial brake pressure
P_1 — Rate of change of brake pressure
R — Radius of tire
S — Stopping distance
s — Slip
s_0 — Initial slip
T — Torque
t — Continuous time
u — Vehicle velocity along its x (longitudinal) axis
V — Peripheral velocity of free straight rolling tire
X — Longitudinal force
Z — Vertical force
μ — Longitudinal force coefficient
α — Slip angle
γ— Slope of longitudinal force coefficient curve
τ— Lumped time constant

ω—Angular velocity

4.1 Subscripts

B Brake
H High limit
i Initial
L Low limit
p Peak
R Road
s Slide

5. ***Introduction*****—**ABS may represent the single greatest advancement in automotive braking since the development of hydraulic brakes. Given the significance, this document has been written to provide the reader with the following;

a. Historical Review of ABS
b. Basic ABS Theory
c. Profile of Current Antilock Brake Systems

This information is intended to provide the reader with an understanding of the fundamentals of ABS and its development. With this knowledge, the reader should have a better understanding of the present, and may have the tools to help understand future ABS developments.

6. Historical Review

6.1 The Evolution Of Passenger Car and Light Truck Antilock Brake Systems—The current hydraulic antilock brake systems were conceived from systems developed for trains in the early 1900's. The development of passenger car and light truck ABS appears to have started around 1936, when Bosch received its first patent for an antilock brake system using electromagnetic wheel speed sensors. When the sensors detected a locked wheel, an electric motor controlled orifice at each brake line was activated, thus regulating the brake pressure.

Several ABS development projects began in the 1950's. The first project began in 1954 at Ford when a Lincoln sedan was fitted with an antilock brake system from a French aircraft. In 1957 Kelsey-Hayes began an "automatic" braking system exploratory development program. The program concluded that the system should prevent the loss of vehicle control and reduce the vehicle's stopping distance. 1957 also saw Chrysler begin research on a "skid control" brake system, however, it was not until 1966 that Chrysler began developing antilock brake systems that were intended for production.

The late 1960's saw the first antilock brake system enter production. Kelsey-Hayes completed development of a rear wheel ABS in 1968. The single channel vacuum powered system was first offered by Ford on its 1969 Thunderbird and Lincoln Continental Mark III, under the trade name "Sure-Track".

Chrysler introduced four-wheel ABS on the 1971 Imperial. The system, developed with Bendix was a 3-channel, 4-wheel vacuum actuated system marketed under the trade name "Sure-Brake".

Jensen Motors became the first automobile manufacturer to offer ABS in conjunction with a viscous coupled 4-wheel drive system. In 1972 the Jensen Interceptor was made available with the Dunlop "Maxaret" antilock brake system. The system used a prop-shaft mounted speed sensor to operate a solenoid, which in turn operated air valves to reduce the brake vacuum servo output force.

Bosch began supplying a hydraulically actuated antilock brake system to Mercedes-Benz in October of 1978. The 3-channel, 4-wheel, add-on system was the first to employ a digital electronic control system to replace the analog electronics.

The use of ABS increased dramatically during the 1980's. In 1984, Teves began volume production of the first "integrated" ABS, in which the hydraulic brake booster, master cylinder, and antilock actuator were combined into a single component. The system, designated MK II, was also the first microprocessor based ABS and was first used on the 1985 Lincoln Mark VII.

Lucas Girling began supplying Ford of Europe with a mechanical ABS called Stop Control System (SCS) in 1986, a derivative of one developed in the early 1980's for motorcycles. The system used two mechanical wheel speed sensors on the front wheels. The rear wheels were valved to prevent lock up.

In 1986 Kelsey-Hayes introduced a single channel rear-wheel ABS on light trucks. The system saw widespread usage in the late 1980s, beginning with Ford in the 1987 model year.

Delco Moraine NDH began production of its ABS VI system in 1990. The system is unique in that the pistons used to control brake pressures are driven by electric motors via gear boxes and ball screws.

A summary of the previous information is shown in Figure 2.

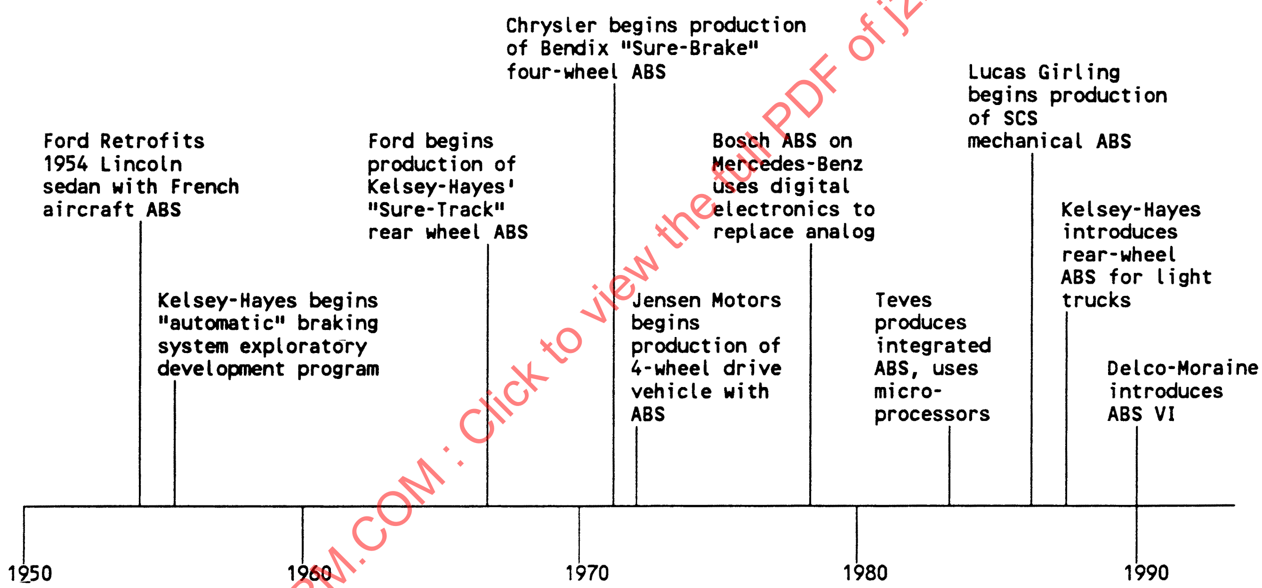
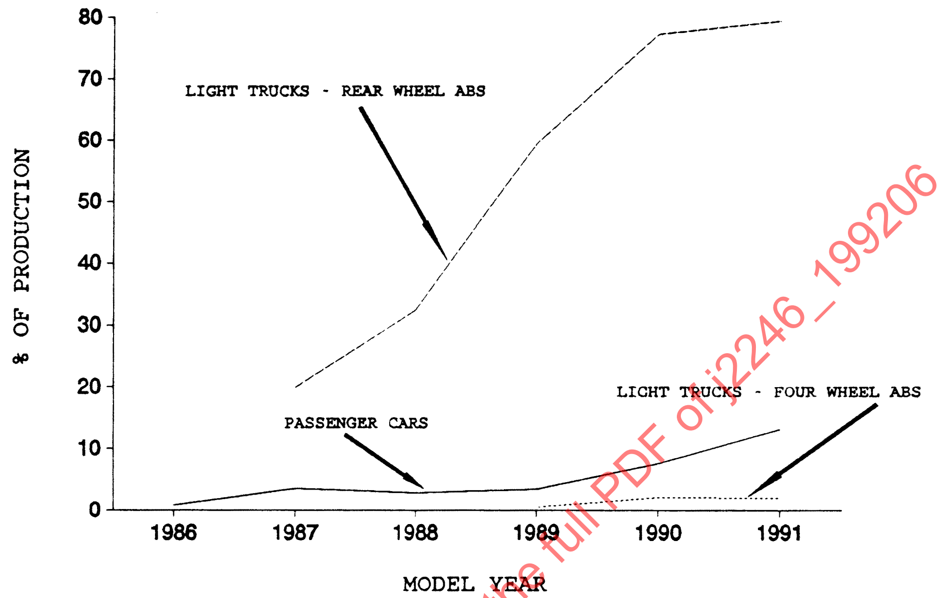


FIGURE 2—ANTILOCK BRAKE SYSTEM SIGNIFICANT EVENTS

6.2 ABS Installation Rates For Passenger Car And Light Trucks—Figure 3 illustrates ABS installations for passenger cars and light trucks as a percent of production. The information shown represents both rear wheel and four wheel ABS.



NOTE—ABS installation rates are based on passenger cars and light trucks assembled in the U.S., Canada, and Mexico, for the U.S. market. 1991 model year figures are estimates derived from production figures through March 31, 1991. Ward's Automotive Reports were used as a source for this data.

FIGURE 3—NORTH AMERICAN ABS INSTALLATIONS

7. Basic ABS Theory

7.1 Goals Of ABS Application—The application of ABS to a vehicle can provide improvements in the vehicle performance under braking compared to a conventional brake system. Improvement is typically sought in the areas of stability, steerability, and stopping distance. In the following sections, some basic approaches to achieving these goals are examined.

7.2 Inherent Limitations And Compromises—It must be remembered that the addition of ABS to a vehicle does not release it from compliance to the basic laws of physics. The interface between the road and tire still defines the maximum braking force that can be applied, the vehicle chassis will still determine steer responses and load transfer patterns, and the vehicle geometric and inertial properties will still be a major factor in these responses. In addition, the vehicle brake system still must provide for brake application up to the time that ABS begins to control the braking. Even then, the ABS must work through the foundation brakes.

7.3 **Overview Of ABS—**When a driver applies the brake pedal on a vehicle moving initially at a uniform rate of speed, the wheels tend to slow down relative to the ground, causing slip at the wheels. This slip between the tire and the road results in the generation of horizontal tire-road forces, which then govern the longitudinal, lateral, and yaw motion of the vehicle. As the brake apply is increased, the slip at each wheel increases, thus increasing the braking forces on the vehicle, in response to driver commands. This continues until the maximum braking capability of the particular tire and road surface is reached. Upon further application of the brake input, wheel slip increases uncontrollably, and the ability to follow driver commands is reduced considerably. In particular, the ability to steer the vehicle or counteract disturbing lateral forces is diminished.

ABS is a feedback control system that attempts to maintain controlled braking under all operating conditions. This is accomplished by controlling the slip at each wheel so as to obtain optimum forces within the limits of the tire-road combination.

In the following sections, the various aspects of braking dynamics are explained. The first part of the discussion, which centers on the control of the slip at a given wheel, is based on a single wheel model of a vehicle. Some of the important characteristics including capabilities and limitations are discussed. Subsequent discussion covers the coordination of slip at the four wheels in an automobile. A four wheel model is used for this purpose. The effect of ABS control on the stability, steerability, and stopping ability of the vehicle are outlined. The characteristics of some of the more common wheel lock control strategies are examined.

7.4 **Braking Dynamics—Single Wheel Model—**The various forces and torques acting on a single wheel vehicle are shown in Figure 4. The vehicle is modeled to have only longitudinal motion. In a hydraulic brake, torque is developed at the wheel brake by means of brake pressure. The brake torque per unit pressure is called the specific torque. The normal load, Z, corresponds to the weight of the vehicle. The longitudinal road force, X, is the product of the longitudinal force coefficient and the normal load. This force retards the motion of the vehicle. The rotational motion of the wheel is governed by the torques resulting from the brake pressure (brake torque) and the road force (road torque).

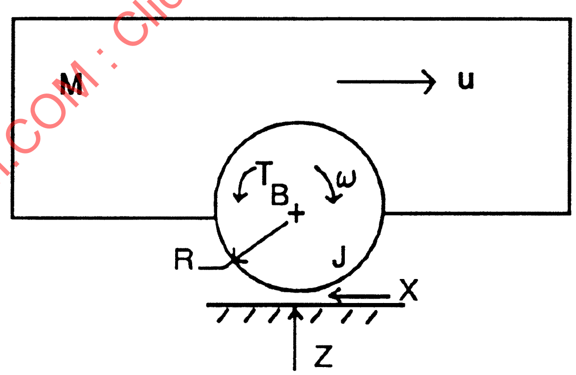


FIGURE 4—SINGLE WHEEL VEHICLE MODEL

The equations of motion can then be written as:

$$Z = M*g \quad \text{(Eq. 2)}$$

$$X = \mu*Z \quad \text{(Eq. 3)}$$

$$T_R = R*X \quad \text{(Eq. 4)}$$

$$T_B = G*P \quad \text{(Eq. 5)}$$

$$J*(d\omega/dt) = (T_R - T_B) \quad \text{(Eq. 6)}$$

$$M*(du/dt) = -(X) \quad \text{(Eq. 7)}$$

7.4.1 TIRE-TO-ROAD INTERFACE DESCRIPTION—The horizontal force generated at the tire-road interface has two components. The longitudinal force is along the length of the tire, while the lateral force is perpendicular to it. The normalized longitudinal force for a rubber tire is primarily a function of the longitudinal slip between the tire and the road. Wheel slip is defined as the difference of the angular velocity of a freely rolling wheel and that of the braked wheel, divided by the former (Equation 8). The road forces result from deformation of the tire and are transmitted via the contact patch at the tire-road interface.

$$s = \frac{u - \omega R}{u} \quad \text{(Eq. 8)}$$

A typical relationship between the longitudinal force coefficient and the wheel slip is shown in Figure 5. Note that the coefficient increases with small slip values. At high slips, values decrease as slip increases. The maximum coefficient, μ_p, is the peak traction capability of the tire-road interface. The coefficient of friction for a locked wheel (slip = 100%) is called the sliding coefficient of friction.

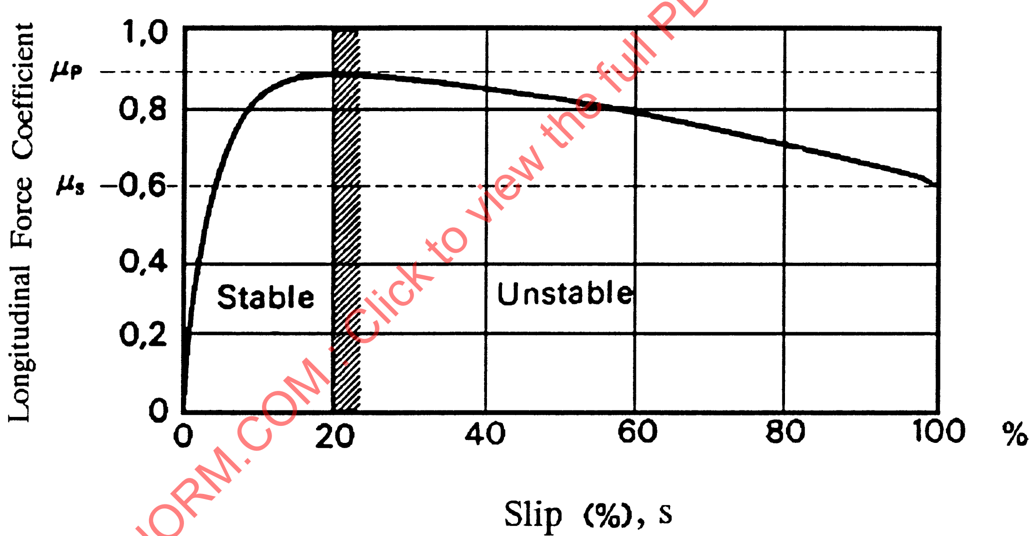


FIGURE 5—LONGITUDINAL FORCE COEFFICIENT VERSUS SLIP

The lateral force coefficient for an unbraked wheel is primarily a function of the slip angle. A typical curve is shown in Figure 6.

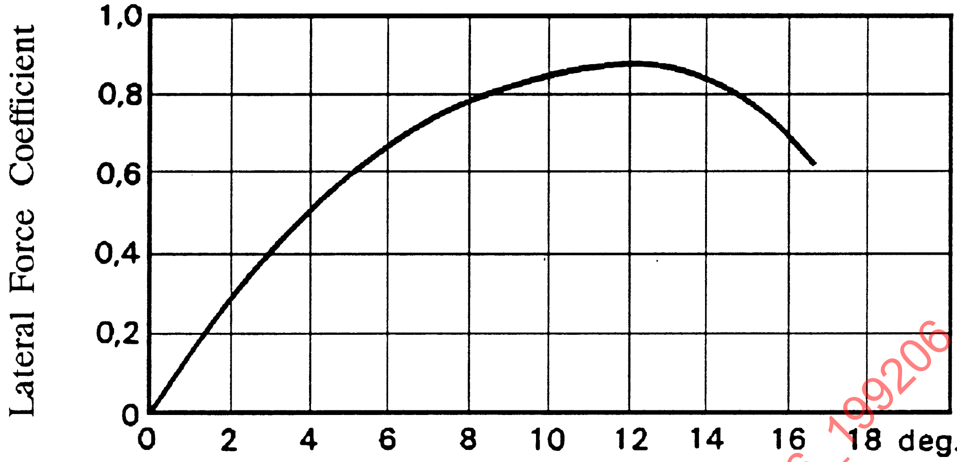


FIGURE 6—LATERAL FORCE COEFFICIENT VERSUS SLIP ANGLE FOR AN UNBRAKED WHEEL

The force coefficient is dependent on a number of parameters. These include road surface condition (dry, wet, ice), tire construction, tire wear, surface roughness, normal force and tire pressure. Typical characteristics for different surfaces are shown in Figure 7. The curves for deformable surfaces, such as unpacked snow and loose gravel are of particular interest.

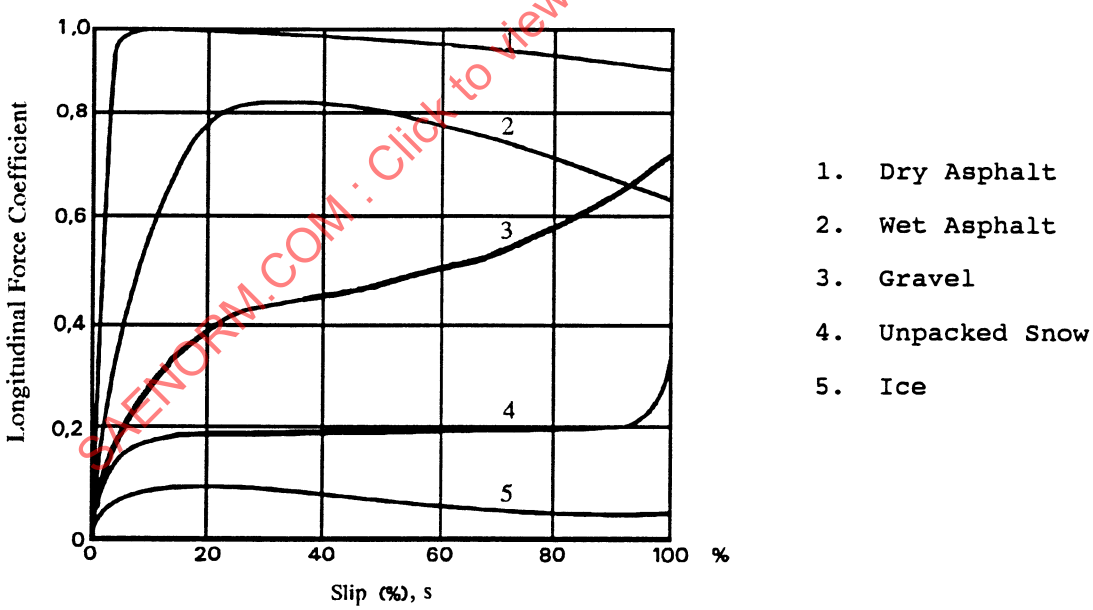


FIGURE 7—TYPICAL LONGITUDINAL FORCE COEFFICIENT VERSUS SLIP

On these surfaces, the longitudinal force increases continuously with increasing slip, until wheel lock occurs. This is typical of a deformable surface, where the ploughing action affects the braking significantly.

The friction characteristics are also a function of the operating conditions, including longitudinal and lateral slip, as shown in Figure 8. Vehicle velocity can also be an important parameter, especially on wet surfaces where hydroplaning can occur.

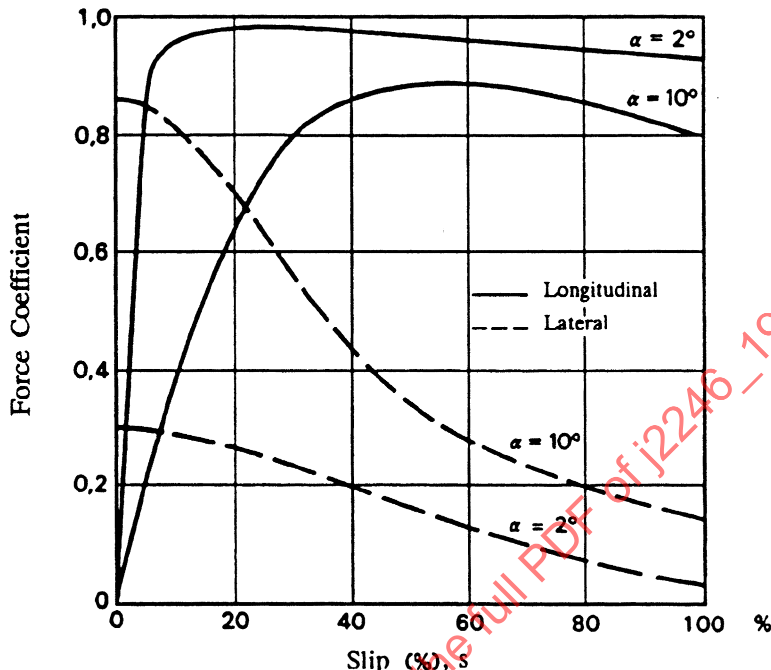


FIGURE 8—FRICTION COEFFICIENT AS A FUNCTION OF LONGITUDINAL AND LATERAL SLIP

In order to develop the equations for longitudinal motion for the single wheel vehicle, the curve of Figure 5 will be idealized to that shown in Figure 9. Hereafter, unless otherwise mentioned, the term friction coefficient will refer to the longitudinal value. Hence,

$$\mu = ({}_{\mu}\rho{*}(s/s_{\rho}) \qquad s \le s_{\rho} \qquad \text{(Eq. 9)}$$

$$\mu_{\rho} - (\mu_{\rho} - \mu_{s})\ \frac{(s - s_{\rho})}{1 - s_{\rho}}$$

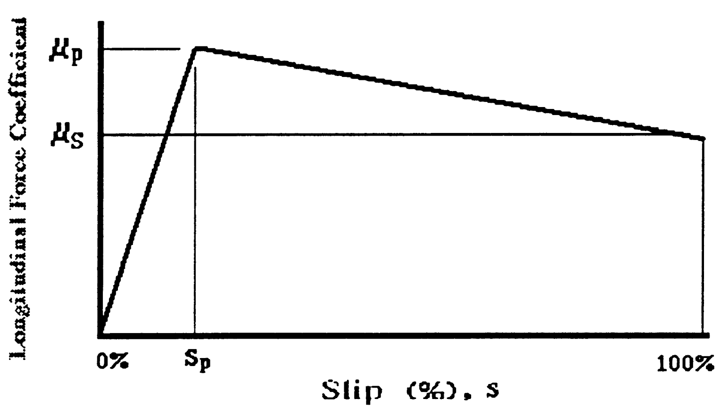


FIGURE 9—IDEALIZATION OF μ-SLIP CURVE

7.4.2 BRAKING WITHOUT ABS—Let us consider an application of brakes without ABS action, where the brake pressure is ramped up continuously from an initial pressure, as in Equation 10.

$$P = P_0 + P_1 t \qquad \text{(Eq. 10)}$$

For $s \le s\rho$;

Further, let the vehicle velocity be a constant. Then, Equations 2 through 10 can be solved to give the time trajectory for the slip as:

$$s(t) = s_o e^{-t/r} + \frac{s_\rho}{\mu_\rho}\left\{\frac{KP_1}{ZR}[t + \tau(1-e^{-t/r})] - \frac{(KP_o)}{ZR}(1-e^{-t/r})\right\} \qquad \text{(Eq. 11)}$$

where:

$$\tau = \frac{JVs_\rho}{ZR\mu_\rho}$$

For $s > s_p$;

$$s(t) = (s_o - s_\rho)e^{-t/r} + \frac{1}{\gamma}\left\{\frac{KP_1}{ZR}[-t + \tau(1-e^{-t/r})] - \frac{(KP_o)}{ZR} - \mu_\rho(1-e^{-t/r})\right\} \qquad \text{(Eq. 12)}$$

where:

$$\gamma = \frac{\mu_\rho - \mu_s}{1 - S_\rho}$$

and

$$\tau = \frac{-JV}{ZR_\gamma}$$

The previous equations show all the parameters that govern the transient of slip. Of particular interest is the time constant in the two cases. When the slip is lower than the peak slip, the time constant is positive. This implies that the exponential terms in Equation 11 decay to zero, implying a stable condition. However, when the slip exceeds the peak slip, the time constant is negative and the exponential terms increase without bound, resulting in an unstable condition.

The time responses for various terminated ramp inputs of brake pressure are shown in Figure 10. For inputs 1, 2, and 3, the wheel stabilizes at a constant slip on the stable side of the μ-slip curve. With input 4, the wheel passes gradually into the unstable side, and then proceeds to a locked state although the pressure is held constant for the latter part of the curve. With input 5, the wheel approaches lock very rapidly because of the excessive brake pressure.

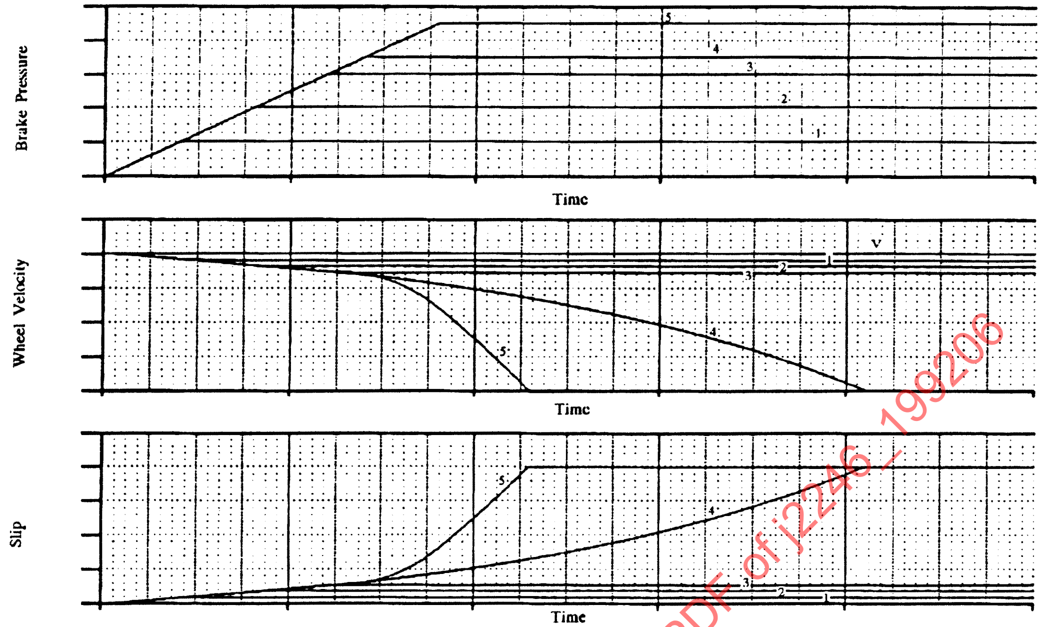


FIGURE 10—SYSTEM RESPONSE TO TERMINATED RAMP INPUTS

7.4.3 ROAD SURFACE FRICTION UTILIZATION

7.4.3.1 *Longitudinal Force Utilization*—To obtain the maximum deceleration possible during braking, the force coefficient of friction between the road and the wheel should be at its peak value, μ_p. This is obtained when the longitudinal wheel slip is maintained constant at s_p and results in the minimum stopping distance. Then,

$$\text{vehicle deceleration} = a_x = \mu_p * g \qquad \text{(Eq. 13)}$$

$$\text{stopping distance} = S = u_i^2 / 2\mu_p g \qquad \text{(Eq. 14)}$$

The relationship between the peak force coefficient of friction and the stopping distance is shown graphically in Figure 11.

It must be noted that the physics of the tire-to-road interface dictates the minimum stopping distance. Further, if the wheel slip strays from the optimum value, s_p, vehicle deceleration and stopping distance will be degraded somewhat. Theoretical values for an initial speed of 60 km/h for different surfaces are included in Figure 12.

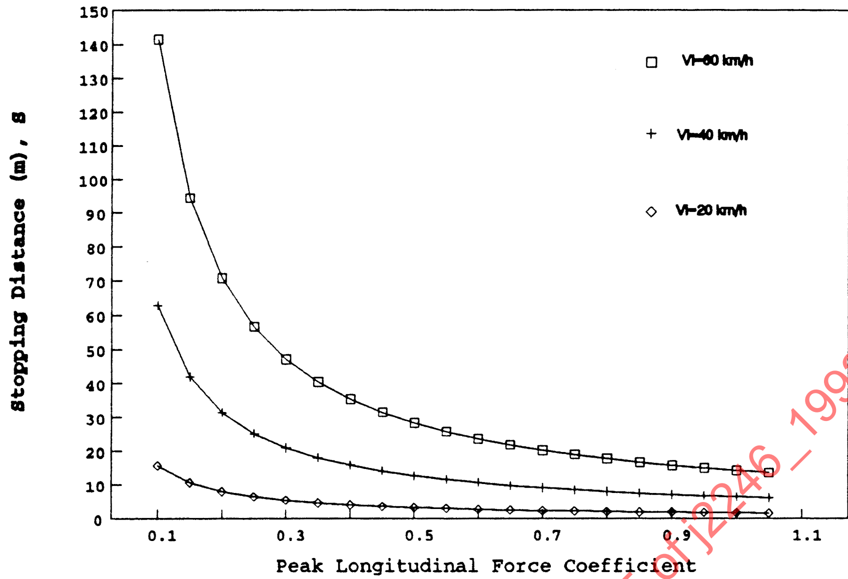


FIGURE 11—STOPPING DISTANCE VERSUS PEAK LONGITUDINAL FORCE COEFFICIENT

Peak Longitudinal Force Coefficient	Representative Surface	Stop Distance (m)	Comment
1.00	Dry Asphalt	14.2	
0.82	Wet Asphalt	17.3	
0.30	Unpacked Snow	47.2	Locked Wheel
0.10	Ice	141.6	
0.65	Gravel	21.8	Locked Wheel

FIGURE 12—THEORETICAL MINIMUM ACHIEVABLE STOPPING DISTANCE FOR REPRESENTATIVE SURFACES AT 60 km/h

7.4.3.2 Lateral vs. Longitudinal Force Utilization

7.4.3.2.1 Nondeformable Surface—Let us first consider a typical nondeformable surface. In reference to the curves of Figures 5 through 8, maximum braking is obtained when the wheel slip is maintained to reach the peak coefficient.

However, in order to obtain the maximum lateral force, the wheel slip should be maintained at zero, or in a free rolling condition. It is not possible to simultaneously obtain both maximum longitudinal force and maximum lateral force.

7.4.3.2.2 Deformable Surface—To obtain maximum longitudinal friction on a deformable surface, the slip will have to be controlled to 100% (locked wheel). From Figure 9, the lateral force that is available in this situation will be very small. Once again, if maximum lateral force is desired, the longitudinal force capability will have to be severely compromised.

7.4.4 ABS OBJECTIVE—ABS attempts to regulate the tire-road forces during braking to follow the driver's steering and braking commands within the constraints of the tire-road traction capability. This is accomplished by controlling the wheel slips to obtain a suitable balance between the longitudinal and lateral tire-road forces.

7.4.5 SIMPLE ABS MODULATION LOGIC—A driver normally brakes a vehicle by modulating the wheel brake pressure through the brake pedal to obtain the desired deceleration. If the braking capability of the tire and road surface is exceeded, the wheels tend to lock. It is at this time that the antilock brake system's control logic takes over the pressure regulation at the wheel in order to obtain optimum braking.

An example of a simple control logic to control the braking is shown in Figure 13a. To illustrate the concept, the system is greatly simplified. It is assumed that the brake pressure at the wheel can be regulated directly by the control logic. Further, this regulation can be through one of two modes; (1) pressure increase or "build" and (2) pressure decrease or "decay" modes. The desired operating slip region, from s_L to s_H, has been determined by the various considerations discussed earlier.

```
IF    "APPLY"   THEN

      IF SLIP  ≥  S_H   THEN

            MODE  =  "RELEASE"     ;   SWITCH TO RELEASE

      ELSE

            MODE  =  "APPLY        ;   REMAIN IN APPLY

      ENDIF

ELSE  ("RELEASE")

      IF SLIP  ≤  S_L   THEN

            MODE  =  "APPLY"       ;   SWITCH TO APPLY

      ELSE

            MODE  =  "RELEASE"     ;   REMAIN IN RELEASE

      ENDIF

ENDIF
```

FIGURE 13A—SIMPLE ABS MODULATION LOGIC

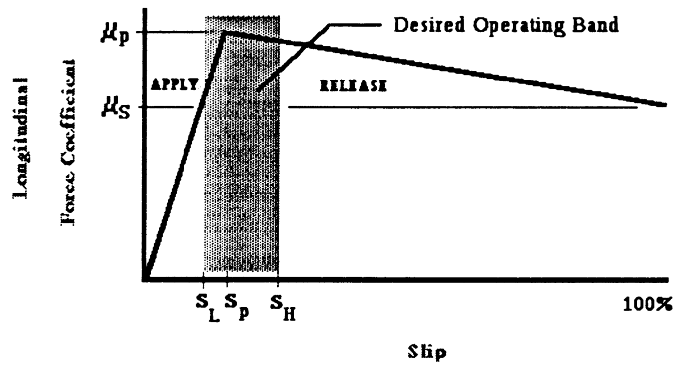


FIGURE 13B—OPERATING MODES FOR ABS MODULATION CONTROL

7.4.6 The control logic shown in Figure 13b is implemented when optimum braking is desired. As mentioned previously, this occurs only after impending wheel lock is first sensed. The control mode is initially set to "build". In the "build" mode, when the slip exceeds the higher limit of the desired operating slip band, the control mode is changed to "decay," so that the slip may return to the desired range. When the slip drops below the lower threshold, the control mode is changed to "build." This allows the brake pressure to increase, thereby building slip once again. The cycle is repeated continuously, until the vehicle stops or the driver takes his foot off the brake.

7.4.7 EXAMPLE RUN ON SIMULATION—The control logic described previously, results in frequent switching between the "build" and "decay" modes to regulate the braking. This results in continuous "cycling" of the wheel slip, the brake pressure, and the road force, all varying in nominal bands around their operating values.

Per the idealized μ-slip characteristics of Figure 9, as the pressure changes the transient response of the wheel is governed by Equations 11 and 12. Hence, the actual range of the wheel slip and the brake pressure will depend on the various terms in these equations, including wheel inertia, the pressure rates in the "build" and "decay" modes, the specific torque, and so on.

The time responses of a typical run, on a surface with a peak longitudinal force coefficient of 1.0, are shown in Figure 14.

7.5 Braking Dynamics—Four Wheel Model—Several simplifications were made in the development of the single wheel model. A more complete description will include the generation of tire-road forces at each of the four wheels. Further, both longitudinal and lateral forces and motions need to be considered. In addition, suspension and drivetrain interactions and the longitudinal and lateral load transfer also affect the response of the vehicle.

Without going into a rigorous development of the directional response of a vehicle undergoing braking, an overview of the application of ABS to a vehicle is presented here. Three aspects of vehicle performance under braking conditions are considered; stability, controllability, and stopping distance. Straight line braking is considered first, the influence of ABS control on stopping distance and stability to external disturbances will be the focus. Controllability and stability in response to steering commands will then be considered. Braking in a turn and split coefficient braking issues are also examined.

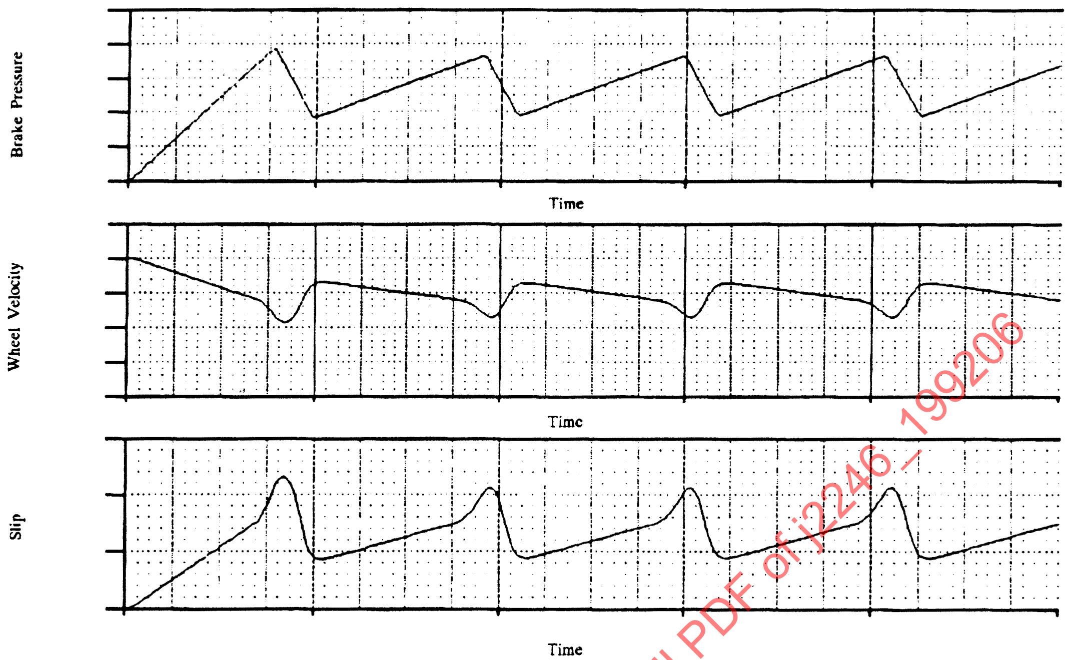


FIGURE 14—ABS TRANSIENT RESPONSES FOR A SINGLE WHEEL SYSTEM

The analysis presented here makes several simplifying assumptions to make the problem tractable. The quasi static response of the vehicle is considered to adequately characterize the more complex response of the vehicle. The vehicle is analyzed at trim conditions defined by lateral and longitudinal accelerations with wheel loads corresponding to the steady-state values dictated by load transfer. Additionally the influence of varying speed is neglected and results are presented for a range of speeds to demonstrate changes with speed. The influence of the ABS cycling is neglected, variations in slip level are assumed to average out to a value determined by a slip setpoint or operating level. The potentially nonlinear behavior of the tires for conditions near the limit of adhesion has been avoided by linearizing the tire properties about the trim condition. This linear representation is not intended to provide a complete description of the vehicle dynamic behavior at these conditions, but rather to illustrate the influence of ABS on vehicle directional response.

The focus of this discussion is on systems providing control over all four vehicle wheels. The control of slip level is considered paramount and the transients involved in arriving at the selected slip will be neglected. Features of single axle systems (rear axle only) are noted separately when significantly different performance is achieved. Various two channel systems, select high / select low, and diagonal control for instance, will not be dealt with in detail.

7.5.1 STRAIGHT LINE BRAKING—A vehicle undergoing deceleration must develop retarding forces at the tire-road interface through the development of relative longitudinal slip between the tire and road surface as discussed previously. This deceleration generates a longitudinal load transfer from the rear axle to the front axle. This load transfer causes the maximum level of braking force available at each axle to vary as a function of deceleration. Utilization of this available force is a fundamental aspect of brake system design.

In a vehicle equipped with a conventional brake system, tailoring of the front and rear brake balance as a function of deceleration is accomplished through the relative magnitudes of front and rear specific torques, and the characteristics of proportioning valves. With ABS cycling, the balance of the front and rear braking forces is accomplished through slip regulation. To achieve minimum stopping distance, maintaining both front and rear slip at levels that correspond to the peak force production is desirable. With this strategy, the load transfer effect on longitudinal force availability is compensated for and the vehicle braking efficiency could theoretically approach 100%. In practice this is not accomplished due to the system transients, the need to search for the peak longitudinal force and the uncertainties of peak force and slip values.

There exists a continuum of slip setpoints that will produce the force necessary for a deceleration of 0.8g. Figure 15 illustrates the combinations of front and rear slip that will produce a certain deceleration. The lowest value of front slip occurs when the rear slip corresponds to the peak value for longitudinal force. Sweeping through front slip results in decreasing rear slip requirements up to the slip for peak front force. As the front force coefficient curve passes over the peak, increased rear slip is demanded to compensate for the lost front force. This set of combinations represent the range of slip setpoints that can be considered when tailoring the vehicle response at this given trim.

Using the definition of brake efficiency applied to conventional brake systems (see SAE Paper 890804), an efficiency for ABS operation can be defined. Defining efficiency as the ratio of deceleration to the highest longitudinal force coefficient of the front or rear axle, the efficiency ascribed to the potential slip combinations of Figure 14 is shown in Figure 16. As mentioned previously, this measure of efficiency does not take into account any effects other than slip setpoint.

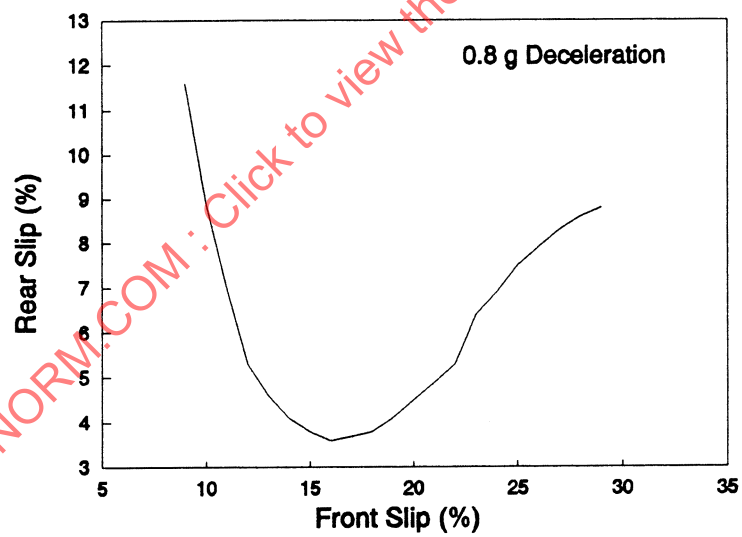


FIGURE 15—SLIP COMBINATIONS FOR 0.8 g DECELERATION

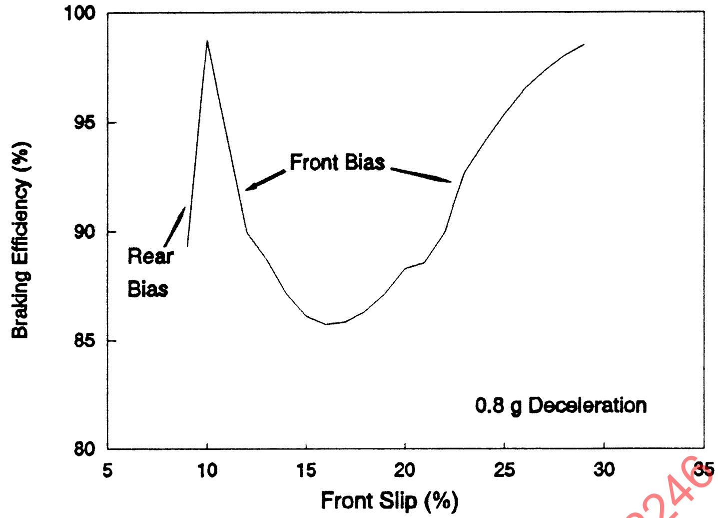


FIGURE 16—BRAKING EFFICIENCY AS A FUNCTION OF FRONT SLIP SETPOINT

The determination of true vehicle velocity is one of the major obstacles to controlling slip at the individual wheels. This must be known accurately to establish the slip level. A reference velocity must be generated by the ABS to estimate the vehicle velocity from the available inputs. The exact algorithm used to calculate a reference velocity is generally a closely guarded secret. Often the wheel speed data collected from an undriven axle, or a diagonal pair of wheels are used as initial indications of the vehicle velocity. Once braking is underway, the wheel speeds are monitored, typically to provide a lower limit on the reference velocity, and maximum allowable levels of deceleration are imposed to arrive at reference velocities. Many variations in vehicle velocity calculations are used. Several levels of maximum deceleration can be used to more accurately estimate speed on different surfaces. Switching between levels is controlled by the recovery rates of the wheels in some algorithms. Auxiliary transducers are sometimes employed, such as accelerometers, to determine the actual vehicle deceleration and appropriately adjust the reference velocity. The accuracy of reference velocity estimation is fundamental to achieving good control with the ABS.

In addition to minimizing stopping distance, vehicle stability is another aspect to straight line braking that must be considered. For the case of straight line braking with the steering held fixed in the straight ahead position, the vehicle should brake in a straight line in the presence of external disturbances. One example of such a disturbance is a lateral force applied at the vehicle center of gravity. This is chosen as it accurately represents the disturbance caused by road crossgrade, and approximates the response to a crosswind. The latter results in a force applied somewhere other than the vehicle center of gravity, but the general trends are the same.

To describe the vehicle response to this external disturbance the vehicle equations of motion are linearized about a specific trim condition and classic linear disturbance responses (see Vehicle Dynamics, reference 4) are used as an indication of the more complex vehicle response. The specific trim is defined by the vehicle deceleration, lateral acceleration (zero for this case), vehicle speed, and the selected front and rear slips. The yaw response to an external disturbance is shown for a range of vehicle speeds in Figure 17. Yaw responses with a positive sign indicate that the vehicle is turning away from the disturbing force, always a stable response. Negative responses indicate the existence of a speed above which the vehicle could have an unstable or increasing response to the input.

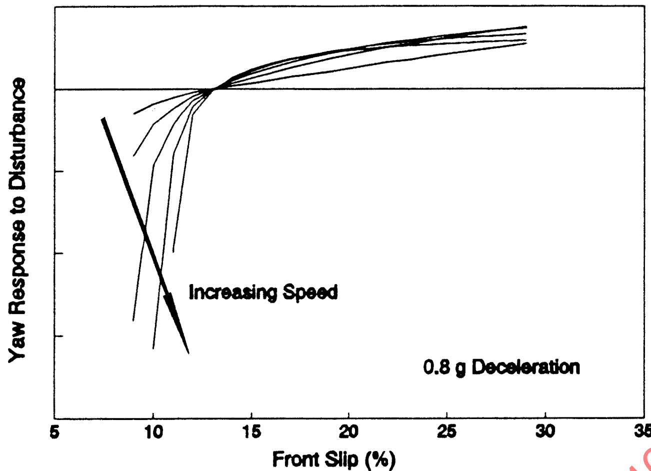


FIGURE 17—DISTURBANCE RESPONSE AS A FUNCTION OF FRONT SLIP SETPOINT

Figure 17 shows that the vehicle response becomes and remains positive with front slips above some value. This is due to the reduction in cornering stiffness of the tire with increasing slip as illustrated in Figure 8. Decreases in front cornering stiffness and increases in rear cornering stiffness cause the vehicle to have a more stable response to the disturbance. For high values of front slip, those in excess of the slip at peak force generation, the rear slip has to be increased to compensate for the loss in front longitudinal force. As a result, both front and rear cornering stiffnesses are reduced and the change in response is not as dramatic as for the lower slip levels.

For the vehicle not equipped with ABS there are three possible limiting behaviors for this disturbance response dictated by three brake balance conditions at the traction limit. In the case of a vehicle that is front biased at the limit, the front wheels will lock and the response will be a positive, stable response. This case would also describe the limiting condition for a vehicle equipped with a rear wheel only ABS. If the vehicle is rear biased at the limit, the response will be a large negative response and the vehicle response will be unstable for any nonzero speed. Neutral balance will result in a vehicle that develops a lateral velocity but no yaw velocity, a neutrally stable system.

7.5.2 STABILITY AND CONTROLLABILITY IN RESPONSE TO STEERING INPUTS—Preservation of steering control and stability during braking are prime goals of the application of ABS to vehicles. The stability of the straight-running vehicle in response to disturbance inputs has been discussed. This section will address the stability and response of the vehicle to steering inputs. Again the linearization of the vehicle equations of motion about the straight running, braking condition is used to demonstrate the influence of ABS control on vehicle performance.

The yaw velocity response to steering is used as a measure of both stability and steerability. The linearized yaw response to steering inputs is shown in Figure 18 for a range of vehicle speeds as a function of front slip.

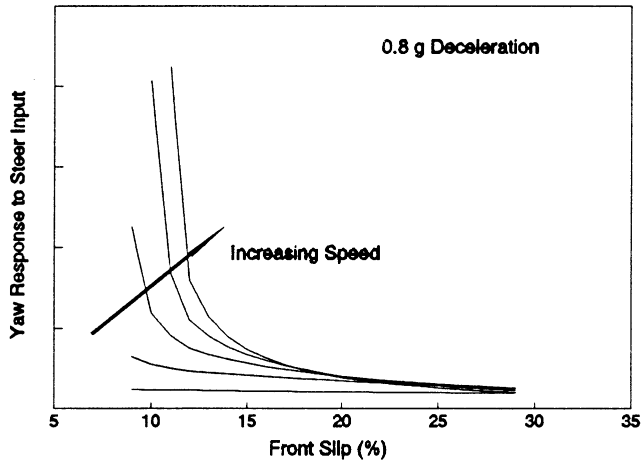


FIGURE 18—RESPONSE TO STEERING AS A FUNCTION OF FRONT SLIP SETPOINT

With low levels of front slip, the yaw response to steering inputs is very high, particularly at high speed. For the two higher speed curves the response is unstable for the very low front slip cases. Increases in front slip reduce the yaw response observed at higher speeds, and the response is stable for all speeds. Figure 19 translates the responses of Figure 18 into a path radius response. The range of path radii achievable for a given steering input with changes in the front slip setpoint is large at high speeds. Variations in control sensitivity can present the operator with a considerable driving challenge.

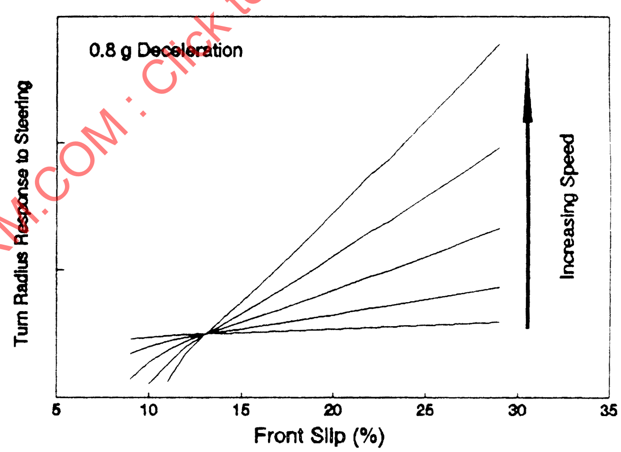


FIGURE 19—TURN RADIUS RESPONSE TO STEERING INPUT

Vehicles not equipped with a four wheel ABS are subject to the same limiting conditions as mentioned previously: front lock, rear lock, and four wheel lock. In terms of steering response, these vehicles represent the extremes. For the front lock case, the vehicle will not respond to steering input, the locked wheel always generating its force in a direction opposing its motion. This response, or lack of it, is also attributable to the four wheel lock and rear wheel ABS only cases. In the case of rear wheel lock the response to steering input is unstable, the rear tires providing virtually zero resistance to the buildup of yaw velocity.

7.5.3 BRAKING IN A TURN—This aspect of performance addresses the response of a vehicle negotiating a steady-state turn subjected to a braking input. In this maneuver the vehicle has already established a constant turn and the associated slip angles and loads have been developed at the tires. The presence of a lateral acceleration causes load to be transferred from the inside wheels to the outside. The distribution of this load transfer is controlled mainly by the suspension roll stiffness distribution.

The introduction of a deceleration to the vehicle will cause a longitudinal load transfer to be imposed upon the lateral load transfer. This will cause the front axle to be loaded and the rear to be unloaded. This load transfer is a destabilizing influence, increasing the front lateral force production and decreasing the rear. This causes the vehicle to increase its yaw velocity and decrease its turn radius. This scenario is true for maneuvers that do not challenge the traction available at any of the tires.

The lateral load transfer associated with the steady turn will cause the first wheel to lock to be on the inside of the turn. When this lockup occurs on a vehicle without ABS, the vehicle will tend to increase its turn radius with front wheel lock, and decrease its turn radius with rear wheel lock. With a limit defined by the point (deceleration) that both wheels on a given axle lock, the front and rear lock scenarios diverge widely. In the case of front axle lock, the vehicle will leave its curved path and proceed in a straight, tangential path. Rear axle lock causes the vehicle to increase its yaw velocity, decrease its turn radius and become unstable. These responses are illustrated in Figure 20.

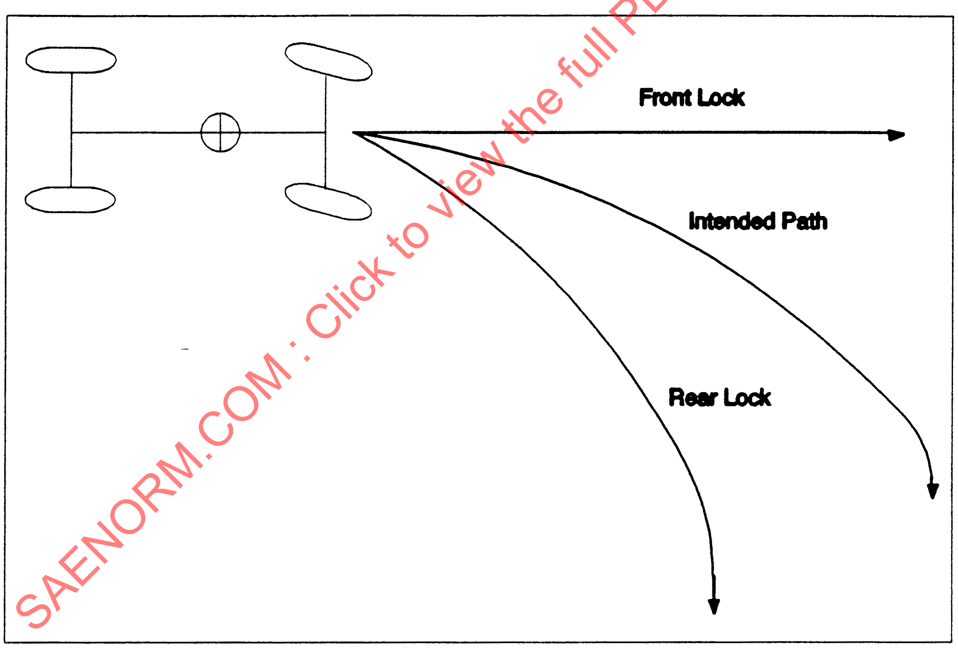


FIGURE 20—EXTREMES OF PATH DEVIATIONS WITH WHEEL LOCK

ABS control of the slip at the front and rear axles eliminates the lockup scenarios described previously. Additionally, the control of slip level instead of torque, causes a moment to be generated opposing that created by the load transfer effect. This occurs only at deceleration levels that challenge the traction of the inside wheel of an axle. Once the ABS takes control of the applied brake pressure, it attempts to control the slip operating point. With a heavily loaded outside wheel and a lightly loaded inside wheel operating at similar slips, the imbalance in forces will create a rigid body moment that tends to increase the turn radius.

Additional steps are often taken to preserve the rear cornering force available using a select low logic. With this type of logic, the brake pressure command to the two rear wheel brakes is controlled by the wheel most disposed to lock. In this manner, the cornering force of the inside tire is maintained at some level, and the outside wheel is operated at a slip well below its peak capability, leaving it with substantial cornering force available.

The slip setpoints of the front and rear axle can be used in a similar fashion to the previous discussion to modify the response of the vehicle once it is under ABS control. Increasing the slip level at the front will decrease the tendency for the vehicle to tighten its radius, while higher rear slip levels will increase this tendency.

With rear wheel only ABS the rear wheel lockup scenario is avoided, but front lock is still a possibility. This implies that unstable yaw response is avoided, but with front wheel lock, the curved path cannot be maintained.

7.5.4 SPLIT COEFFICIENT BRAKING—It is not uncommon to encounter a road surface with differing coefficients under the left and right tires. Brake applications that exceed the friction available on the low coefficient side cause imbalanced, side-to-side, longitudinal forces. The resultant rigid body moment tends to steer the vehicle to the higher coefficient surface.

A vehicle not equipped with ABS is prone to spin once the wheels on the low coefficient side have locked up. The rigid body moment will tend to turn the vehicle in such a way that the front tire on the high coefficient will develop forces sympathetic to the spin, and the load transfer will reduce the load on the rear axle and reduce its stabilizing influence. It is not unheard of for a non-ABS vehicle to spin along the split coefficient surface, while its center of gravity travels in a straight line along the split.

Again, ABS can assist in stabilizing the vehicle in this situation. Select low logic at the rear will preserve some level of cornering stiffness at the rear of the vehicle, aiding in stabilization of the yaw response. Another form of control, known colloquially as "yaw control", is also helpful. The main source of disturbance in this maneuver is the imbalance of front brake forces. Using select low logic on the front axle would eliminate this imbalance and the associated disturbance. However, this is not practical, as stopping distances may become unacceptably long. As a compromise, the yaw control logic recognizes the locking of the low coefficient wheel, and as it acts to recover that wheel, it also modifies the pressure to the high coefficient wheel. This can be accomplished by holding or releasing the pressure to the high coefficient wheel. The system then allows the high coefficient wheel to increase its pressure at a reduced rate until it reaches its limit. The onset of the disturbing moment and its subsequent increase is delayed, thus giving the driver time to react. The subsequent increase in brake force on this wheel allows the stopping distance to be decreased over that possible with a select low logic. This control in illustrated in Figure 21.

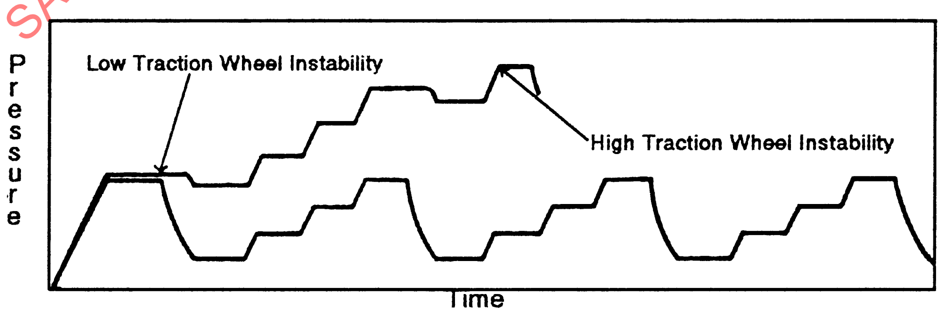


FIGURE 21—EXAMPLE YAW CONTROL STRATEGY