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# Brake Force Requirement Study: Driver-Vehicle Braking Performance as a Function of Brake System Design Variables

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16. Abstract The objective of this study was to define those brake characteristics, within the space bounded by the relationship between brake pedal force and vehicle deceleration, which lead to acceptable driver-vehicle performance. A driver-vehicle braking test was performed in which the deceleration/pedal force ratio, the pedal displacement, the surface-tire friction, and driver characteristics (age, weight) were systematically varied in order to determine the influence of these variables upon minimum stopping distance and other performance variables. The tests that were performed on a low coefficient of friction surface showed that high values of deceleration/pedal force gain result in large number of wheel lockups and lower mean deceleration in bringing the vehicle to a stop, compared to intermediate or low deceleration/pedal force gain levels. Tests conducted on intermediate and high coefficient of friction surfaces showed that high and intermediate deceleration/pedal force gains produced greater mean decelerations and greater frequencies of wheel lockups than lower gain systems. The frequency of loss of lateral control was significantly greater with the high deceleration/pedal force gain brakes on all surfaces than with lower gains. There were minor benefits of 2.5 inch pedal displacement compared to zero inches. Potential brake failures and their effects upon pedal force requirements were analyzed. The implications of the findings for a vehicle braking standard were shown in terms of deceleration/pedal force gain and pedal force.

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This research program in driver and vehicle braking performance was conducted by staff of the Human Factors and Physical Factors Departments at the Highway Safety Research Institute.

The program was under the direction of Dr. R.G. Mortimer, Human Factors Department, and Mr. L. Segel, Physical Factors Department, who were responsible for the overall planning and execution of the research tasks and the final report.

Mr. H. Dugoff provided assistance in overall project planning and in many specific project tasks. The design of the electro-hydraulic servo control system in the braking test vehicle and the data recording systems were carried out by Mr. J. Campbell. The rear disc brake conversion and other aspects of mechanical engineering were the responsibility of Mr. J. Wirth. Instrumentation was constructed and vehicle modifications and maintenance were carried out by Messrs. J. Boissonneault, G. Popp and R. Stein all of the Physical Factors Department. Mr. R. Murphy and Mr. R. Limpert were responsible for conducting the engineering segments of the Failure Analysis Section. The Review of the Literature was carried out by Mr. D. Fisher.

Mr. C. Jorgeson directed the operations at the testing site and was responsible for the data collection phase in the braking test. He was assisted by Dr. J. Lower, Dr. D. Damkot, Mr. H. Schick, Mr. G. Popp, and Mr. R. Stein. Analyses of the data were carried out by Dr. J. Lower and Mr. C. Jorgeson. Mr. S. Sturgis measured the brake pedal configurations and analyzed the pedal force and peak deceleration data.

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# INTRODUCTION

One of the most desirable and important characteristics of motor vehicles is that they should have good handling characteristics in terms of directional response to steering inputs and performance as affected by accelerator and brake application. A previous report by the Highway Safety Research Institute (HSRI) was concerned with evaluating the present status of vehicle handling properties and of the potential role of these characteristics upon collisions (HSRI, 1967). In that study, various aspects of the man-vehicle interface were identified as having safety significance.

For example, the ability of the driving population to exert the pedal forces required to brake a car is one facet of this interface. Further, this facet has both a static and dynamic component. From a static viewpoint, it appears desirable to build motor vehicles such that a specific percentile of the driving population can exert the maximum pedal forces associated with peak decelerations on dry, high-friction surfaces. From a dynamic standpoint, it appears that pedal-force characteristics should also enable the driving population to attain maximum braking performance irrespective of the friction conditions prevailing at the tire-road interface. By "maximum braking performance" we mean the shortest distance to slow or stop that can be obtained without excessive locking of the wheels in order that sufficient control and stability prevail for holding the vehicle in the desired lane of travel.

It is primarily the dynamic aspect of the man-vehicle interface with which this study is concerned. The braking process is viewed as a task in which the driver must control and modulate his pedal force such that he achieves the shortest braking distance possible under the prevailing road conditions, while further satisfying the requirement that the trajectory of his vehicle be under reasonable control. The investigation, as conceived, is concerned with both the static and dynamic component of the ergonomics of braking. The major question addressed is: "In the distribution of anthropometric characteristics and perceptual-motor skills possessed by the driving population, how do the relationship of pedal force and pedal displacement to vehicle deceleration influence the braking performance of the man-vehicle combination?". From the standpoint of generating a braking standard, this question can be rephrased to: "What are the bounds on brake pedal force/vehicle deceleration space wherein the bulk of the driving population shall find it possible to maximize deceleration while making controlled (i.e., well modulated) braking maneuvers on both dry pavement and surfaces with a reduced coefficient of friction?".

## SUMMARY OF TASKS

The study was conceived as having six major phases which will be briefly discussed here so as to provide the reader with a general orientation of the overall approach.

### 1. LITERATURE REVIEW

A review of the literature was carried out pertinent to an analysis of the pedal force/vehicle deceleration characteristics of an automotive vehicle. The factors considered important in the review were brake system design, brake usage, skidding, brake testing, and driver characteristics. The review was submitted earlier as an interim report.

### 2. FOOT FORCE CAPABILITY OF DRIVERS

The vehicle braking system is actuated with the feet of the driver and, therefore, it is essential to learn more of the foot force capabilities of individuals comprising the driving population. A procedure was developed by which left and right foot maximum force exertion could be measured for a large sample of female and male drivers. The purpose was to determine a suitable upper force limit for the operation of vehicle service brakes.

3. DRIVER BRAKING PERFORMANCE AS A FUNCTION OF PEDAL-FORCE AND PEDAL-DISPLACEMENT LEVELS

The major emphasis in this study was to learn more of the dynamic relationships between the driver, as a controller of the brake system, and various characteristics of that system. Major variables to be considered were the relationship between the force applied to the pedal and the resulting deceleration of the vehicle, and the pedal displacement, in affecting the stopping distance in a braking task requiring vehicle directional control. The core of the experiment was the development of a special test vehicle in which variations in brake response characteristics could be readily obtained. A driver-vehicle braking test was developed and measurements taken to determine the effect of the variables of interest on braking performance.

4. DRIVER BRAKING PRACTICE

It was necessary to obtain empirical data describing the levels of deceleration that drivers employ under normal driving conditions. Such data were needed as a part of another phase of this program, the failure analysis, since brake deceleration levels can form one criterion measure of required performance both under normal and failed conditions.

5. FAILURE ANALYSIS

An analysis was conducted to ascertain the effect upon vehicle performance of various failures in the braking system. Conditions under which such failures are likely to occur were considered, and the consequences estimated from the required deceleration level probability and the ability of the driver to exert the needed pedal force.

## 6. RECOMMENDATIONS

Finally, based upon the preceding work, recommendations for a modified brake performance standard were made utilizing the information that had been gathered in the analytic and experimental phases of the project.

# TASKS

## 1. LITERATURE REVIEW

### INTRODUCTION

Braking is a complex energy conversion process whereby the kinetic energy of the vehicle is converted into thermal energy at the brakes and at the road-tire interface. During the braking process, the pedal force acts through a mechanical-hydraulic system to apply a retarding torque to each of the four wheels of the vehicle. The braking torque is opposed by the inertia of the wheel and the frictional force between the tire and the road, with the net result being the deceleration of the vehicle. The characteristics of the mechanical-hydraulic system determine the braking torque available at each wheel, while the road-tire friction coefficient and the mechanics of the vehicle determine the decelerating forces. Consequently, the control of the deceleration of a vehicle through brake pedal force depends on the static and dynamic characteristics of the entire system, including the driver.

In the review that follows, the terminology established by the Society of Automotive Engineers (SAE) for the automotive vehicle braking process (SAE J656c, 1968; SAE J657a, 1968) will be used. By definition, braking is initiated in a motor vehicle by a driver applying a force to the foot-actuated lever termed the brake pedal. The magnitude of the actuating force at any instant is called the pedal force. The variation of this force by the operator with time is defined here as pedal force modulation.

A set of simplified equations describing the operation of a typical brake system is presented below. The relationships will be helpful in evaluating the literature pertinent to the braking process, and will serve to define the variables that are involved.

Figure 1.1 illustrates a typical brake system (Crouse, 1965).

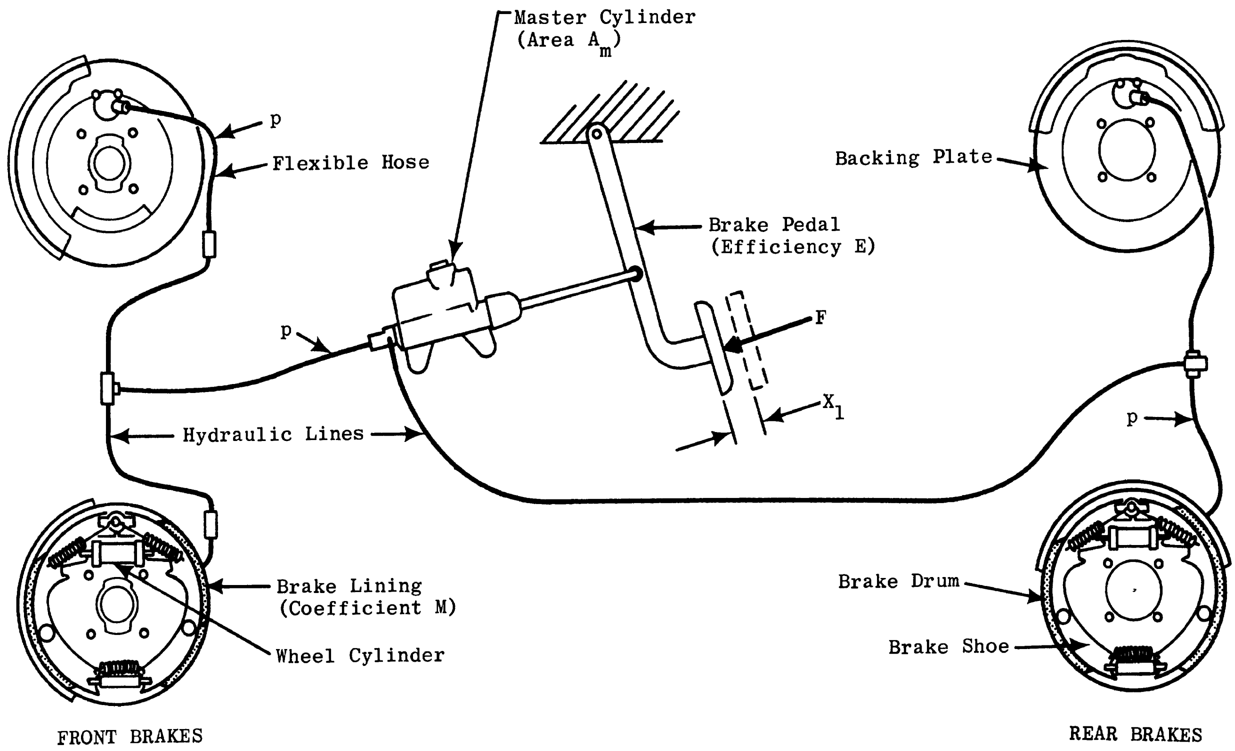



Figure 1.1. Typical hydraulic brake system (drum type).

Application of a pedal force, F, causes the brake pedal to be displaced through a distance $X_1$. The pedal linkage is designed to yield a mechanical advantage of force, r, between the pedal and the master cylinder piston, resulting in a displacement of the piston, $X_2$, that is less than the displacement $X_1$. The master cylinder, with an area $A_m$, traps the oil in the brake line, thereby developing a hydraulic pressure, p. Since there are frictional losses, it is common to assume a pedal efficiency, E, that is less than unity. The relationship between line pressure and pedal force is actually nonlinear but may be simplified for illustration purposes (Brown, 1965).

$$p = \frac{rEF}{A_m} \tag{1}$$

where

$$r = \frac{X_1}{X_2} \tag{2}$$

In power brakes, a vacuum-powered device assists the driver by multiplying the force F by a factor K. This power boost modifies the above relationship to:

$$p = \frac{rEFK}{A_m} \tag{3}$$

It should be noted that the factor K may be a nonlinear function of the pedal force.

In the brake proper, a friction material having a coefficient $\mu_L$ is pressed against the brake drum (or disc), resulting in a brake torque, T. Measurements (Brown, 1965, Shigley, 1963, Stroh, 1968) have shown that the relationship between brake torque and line pressure is not necessarily linear and is very much dependent upon the type of brake employed. Thus, we have:

$$T = kp \tag{4}$$

where

$$k = \text{fn}\left(\mu_L, \text{ brake type, brake geometry, } A_{wc}\right)$$

and

$$A_{wc} = \text{area of wheel cylinder.}$$

The development of vehicle deceleration by means of brake torque follows directly when wheel inertia is neglected and it is assumed that no skidding occurs. The deceleration of a vehicle due to applied brake torque can be expressed as:

$$a = \frac{g}{W} \sum_{i=1}^{4} \frac{T_i}{R_{E_i}} \tag{5}$$

where

$g$ = acceleration of gravity

$R_{E_i}$ = effective radius of wheel i

$W$ = vehicle weight

$T_i$ = brake torque at wheel i

On combining Equations 3, 4, and 5, the deceleration of a vehicle can be expressed as a function of pedal force, viz:

$$a = \frac{g}{W} \sum_{i=1}^{4} \frac{k_i}{R_{E_i}} \frac{rEK}{A_m} F \tag{6}$$

where

$$k_i = \chi \left( \mu_{L_i}, \text{ brake type } \big|_i, \text{ brake geometry } \big|_i, A_{wc}\big|_i \right)$$

On the other hand, the maximum deceleration that can be produced in a locked-wheel stop can be reduced to the simple expression:

$$a_{max} = \mu_t g \tag{7}$$

where

$\mu_t$ = measured road-tire friction coefficient under locked-wheel conditions.

It should be noted that neither Equation 6 nor 7 holds for the braking regime in which there is substantial slipping between tire and road with the wheel still rotating.

PEDAL FORCE AS A FUNCTION OF DESIGN PARAMETERS

BRAKE SYSTEM DESIGN.

Pedal Force Design Goals. A successful brake system design should embody the characteristics of reliability, preciseness of control, and the ability to withstand short periods of extreme overload (Vallin, 1968). The selection of components for the brake system (Robson, 1967) establishes the nominal characteristics of the pedal force/brake torque relationship. The final produce, however, is the result of many assumptions, compromises, and design choices.

Pedal Force/Brake Torque Relationship. One measure of brake performance is the plot of the brake torque versus line pressure (Winge, 1961); namely, a graphical representation of Equation 4 for a constant lining coefficient of friction. Typical performance curves obtained experimentally with disc and drum brakes are shown in Figure 1.2.

Several features of the performance curve for the drum brake may be noted. The offset of the curve near the origin is termed the pushout pressure and is the brake line pressure necessary to overcome the brake shoe return springs. The concavity of the performance curve results from the distortion of the shoes and the drum (Winge, 1961) and therefore may vary considerably from one brake design to another. Although drum distortion has been calculated analytically (Winge, 1961), the influence of drum distortion remains to be incorporated into theoretical brake torque analysis. Consequently, the concave feature of the drum brake performance curve is not present in theoretically derived brake torque/line pressure relationships (Shigley, 1963; Steeds, 1960; Stroh, 1968).

With the aid of Equation 5, it can be shown that the vehicle deceleration is approximately proportional to the sum of the individual brake torques when no skidding occurs. Consequently, any design feature affecting the linearity of the pedal force/ vehicle deceleration relationship. If the drum brakes are utilized,

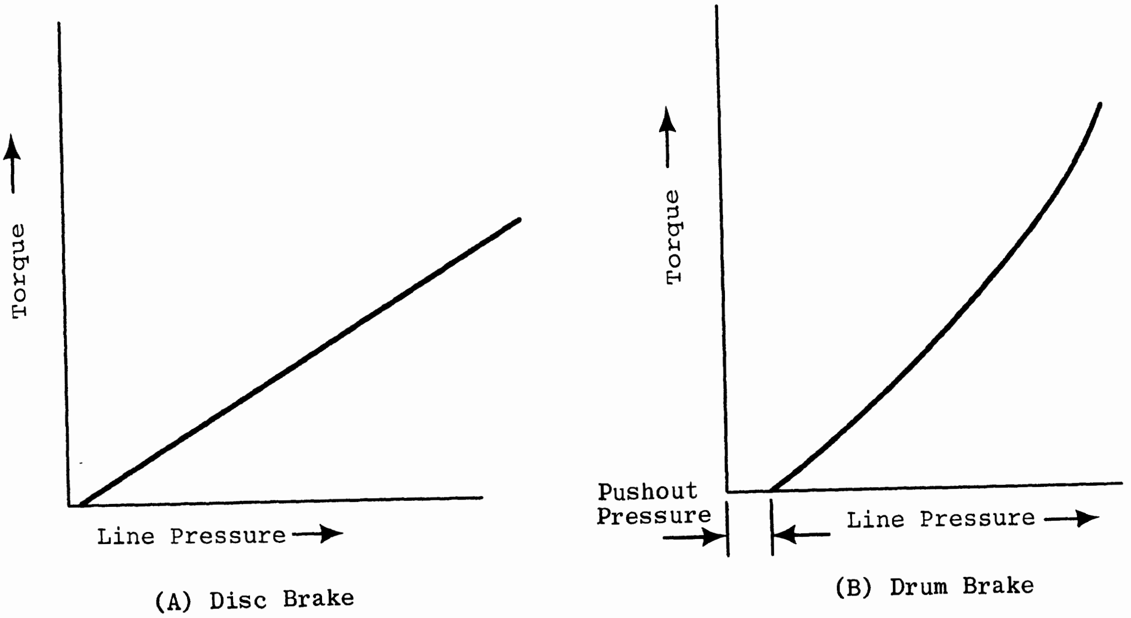



Figure 1.2. Typical brake performance curves.

the nonlinearities associated with the brake torque/line pressure curve are introduced. There is some evidence (Leah, 1964) to indicate that the brake line pressure itself may not be exactly proportional to the pedal effort. In the case of power assisted brakes, the line pressure has been reported (Spurr, 1965) as being directly proportional to the pedal effort up to a line pressure corresponding to saturation of the vacuum assist component.

Disc brake systems are generally acknowledged as having a linear modulation characteristic; i.e., the vehicle deceleration is approximately proportional to the pedal effort. It has been reported (Brown, 1965) that this feature of disc brakes allows the driver to avoid unintentional wheel lockup and permits high deceleration rates under adverse road conditions. It should be noted, however, that this last statement resulted from qualitative driver evaluations rather than an extensive study.

It was stated earlier that the brake torque is dependent on the lining coefficient of friction as well as the brake line pressure. Although the torque outputs of both drum and disc brakes are, in the main, proportional to the brake line pressure, these brakes differ substantially in their behavior with respect to changes in the lining coefficient of friction. The torque output of a disc brake is directly proportional to the coefficient of friction for a given line pressure (The Bendix Corp., 1964). Most drum brake designs have a self-energizing feature (i.e., the brake assists in its own actuation) which results in the brake torque being very sensitive to changes in the coefficient of friction of the brake lining (Lueck, 1965; Kinchin, 1961; Furia et al., 1967; Farobin, 1968). Thus, both fade and the normal day-to-day variations in the friction coefficient (Winge, 1961) of the lining can have an exaggerated effect on the pedal effort of a drum brake system. Further, there is an increased possibility of side to side variations in braking torque when a drum brake system is employed. This phenomenon may result in an undesirable directional response of the vehicle (Furia et al., 1967; Lister, 1965).

One factor affecting the choice between drum and disc brakes is the self-energizing design of most drum brakes. In effect, this self-actuation feature assists the driver in applying the brakes, thus allowing relatively low pedal effort for a given brake torque. Compared to the self energizing drum brake (Farobin, 1968; Winge, 1963) a typical disc brake requires four to five times the wheel cylinder area and twice the hydraulic line pressure to produce the same brake torque. In terms of brake pedal actuation this implies higher pedal efforts and greater pedal displacement (Shaw, 1965; SAE, 1963; Burke & Prather, 1965). This problem is often solved by using a power assist.

Brake Proportioning. When a four-wheeled vehicle decelerates there is a transference of load (Parker, 1960; Taborek, 1957) onto the front wheels because the body's center of gravity is above the ground plane. To achieve optimum braking the proportioning of the braking effort between the front and rear axles should match the instantaneous load distribution (Taborek, 1957; Alexander, 1967). Manufacturers build in a front to rear proportioning (Automotive Industries, 1968), but a brake system having a fixed front to rear braking ratio can only achieve optimum performance for a given rate of deceleration (Chase, 1949; Hofelt, 1959; Parker & Newcomb, 1964). Typically, brake proportioning is fixed with 60 percent to 70 percent of the braking occurring at the front wheels (Automotive Industries, 1968).

A series of tests on vehicles having different weight distributions and different brake proportioning has shown (Alexander, 1967) that a front/rear brake ratio equalling the wheel load distribution of the vehicle at a deceleration of 1.0 g is desirable if the car is to remain directionally stable when heavily braked on all surfaces. On low coefficient surfaces, however, this would result (Parker, 1960) in premature front wheel lockup and a deceleration less than maximum for that surface. The overall effect of proportioning on the driver-vehicle performance during braking is that under certain conditions the driver may be able to apply higher pedal efforts and achieve higher maximum decelerations without incurring wheel lockup.

One solution to the problem of proper proportioning is to vary the brake proportioning with the deceleration of the car. This can be done by controlling the pressure at each axle in correspondence to the axle loading (Furia et al., 1967; Eaton & Schreur, 1966; Engineering, 1964), but the usual procedure is to limit or proportion the rear brake hydraulic pressure above some fixed upper limit (Furia et al., 1967). At best, however, these latter methods are only compromises as a result of variations in vehicle loading and brake performance.

DESIGN PRACTICE AND TRENDS. Vehicle braking systems may be classified as being either four-wheel drum, front disc and rear drum ('hybrid'), or four-wheel disc. Disc brakes have been common in Europe for several years (Furia et al., 1967; Huntington, 1964; Strien, 1961), but have been introduced only recently in the United States (Burke et al., 1965; Tignor, 1966; Thomas, 1967). Only one American-built car has four-wheel disc brakes as standard equipment, and one manufacturer offers them as an extra cost option. Several automakers offer front disc brakes as standard equipment while most list them as an option (Automotive Industries, 1968; King, 1968). Disc brakes were first made available on U.S. cars in 1965, and 2.19 percent of the cars sold were so equipped. By 1967 this percentage had risen to 6.22 percent, and it is expected that this trend will continue.

In recent years there has been a trend towards lower pedal efforts to achieve a given deceleration. In 1935 a design bogey (a standard of performance) was adopted (Chase, 1949) for the rate of deceleration for a given pedal pressure, specifying that a brake system should require between 100 lb and 130 lb of pedal effort to achieve a deceleration of 20 ft/sec$^2$. This bogey was extended in 1949 (Chase, 1949) to reduce the required pedal effort to as low as 77 lb for the same deceleration. The purpose of this new bogey was to set an upper limit of brake sensitivity that would still exclude brakes capable of locking the wheels on dry pavement (e.g. $\mu \simeq 1.0$) with less than 100 lb of pedal effort. It was noted that a number of the cars tested obtained 20 ft/sec$^2$ decelerations with pedal forces lower than those allowed by the new design bogey. There has been no literature published in the interim dealing with the desirability of various deceleration versus pedal effort curves.

Some recently published design summaries have indicated satisfactory performance with vehicle brake systems that achieved 20 ft/sec$^2$ decelerations with 50 lb (Brown, 1965), 44 lb (Winge, 1963), and 31 lb (Vansteenkiste, 1963) of pedal effort. Data presented in a braking test of four different British built cars (Mackenzie et al., 1966) allowed the calculation of comparative data: for a 20 ft/sec$^2$ deceleration, the required pedal efforts ranged from 48 lb to 73 lb. All four cars were equipped with hybrid brake systems. Comparative data derived from other published performance curves include seven foreign sports cars (Motoring Which?, 1968) requiring 40 lb to 79 lb of pedal effort, and six foreign sedans (Motoring Which?, 1968) requiring 39 lb to 50 lb, all for a 20 ft/sec$^2$ deceleration. These cars were equipped with all three types of brake systems. All but one of these cars represent performance in excess of the limit suggested in 1949, indicating that the introduction of power brakes or advances in brake stability have resulted in a considerable shift in the design bogey toward lower pedal efforts. Motor Vehicle safety Standard No. 105 (1968) specifies that the service brake performance of motor vehicle passenger cars must not be less than that described in Section D of SAE Recommended Practice J937 (1968), when tested in accordance with SAE Recommended Practice J843a (1968). The SAE standards permit a very broad range of braking performance (SAE Publication SP-299, 1967). For example, on dry Portland cement concrete they only require that the brake pedal effort fall in the range of 15 to 120 lb in order to achieve a deceleration of 20 ft/sec$^2$ from 60 mph while maintaining the vehicle within a 12 foot lane.

BRAKE USAGE

BRAKE USAGE MEASUREMENTS. Several studies (Carpenter & Lees, 1956) have been made investigating the use brakes receive during normal driving. All but one of the studies were European and involved a variety of driving conditions. During a test involving four drivers over a distance of 1400 miles, it was found that only 5 percent of the stops exceeded an average deceleration of .3 g and that 50 percent of all stops were made at .09 g or lower (Carpenter, 1955). In a test involving 23 drivers over 300 miles of European driving, the average of the maximum decelerations observed on a number of different test routes varied between .21 g and .34 g, the mean being .26 g. The single maximum deceleration recorded was .6 g. A British study (Livsey, 1960-61) involving 16 vehicles covering a wide selection of vehicle types was conducted over four different routes, including fast mainroad, cross country, winding roads, and an alpine descent. It was found that decelerations on all the routes were usually in the range of .2 g to .3 g, and rarely exceeded .4 g. This is in agreement with an American study which obtained deceleration frequency data on an "in town" driving course (Kummer & Meyer, 1965). It was felt that the braking levels experienced were the maximum that would be generated by the general public. Additionally it was found that the root mean square of the speed at which braking was initiated for all the routes increased in direct proportion to the vehicles' maximum speed, but the proportionality constant was different for the four routes.

Another British study (McKenzie et al., 1962-63) was made attempting to establish mathematical relationships which would enable the prediction of brake usage. By defining two parameters, $V_o$, a characteristic speed associated with the route, and M, describing the manner of driving, the average deceleration could be expressed as

$$a = (.039 \frac{\bar{V}}{V_o} + .093)g \tag{8}$$

where

$$\bar{V} = M\left(V_{max}\right)^{.45}\left(V_o\right)^{.62} \tag{9}$$

$\bar{V}$ is the average route speed for the car, and $V_{max}$ is the top speed of which the car is capable. For the cars tested, the decelerations ranged from .15 g to .28 g on all routes, and the root mean square of the speed at which braking was initiated could be related to $V_o$ and $\bar{V}$ for each route.

In conclusion, then, it appears that the decelerations experienced by a particular car during routine driving depends on the driver and the top speed performance of the car, and that these decelerations will rarely exceed .3 g. Since none of the studies dealt with emergency braking, the frequency of occurrence and the magnitude of the decelerations in such a situation are still undetermined.

FADE AND FADE TESTING.

The Phenomenon of Fade. Brake fade (SAE J657a, 1968) is the general term used to describe any one of several conditions which result in reduced brake torque/line pressure gain for a given vehicle. Heat fade results from the change in brake parameters caused by the energy dissipated at the lining-drum interface; water fade results from the reduction in the lining coefficient of friction due to water contamination; and washout describes fade due to any other cause (Fleet Owner, 1966; Percy, 1952).

Heat fade has received by far the greatest attention in the literature. As noted earlier, the brake torque depends on the lining friction coefficient, the line pressure, and the geometry of the brake. Heat fade results (Herring, 1967) in two ways--thermal distortion of the brake geometry and changes in the apparent coefficient of friction of the lining due to high temperatures. The latter mode of brake fade is apparently the most influential, however no comparisons have been reported. Considerable effort is being directed towards controlling the temperature rise in the brake and towards developing fade resistant lining materials (Weintraub & Bernard, 1968; Jacko et al., 1968).

Thermal Analysis. Theoretical investigations have concerned themselves with the temperature rise expected during braking with both drum brakes (Fazekas, 1953; Newcomb, 1958-59; Bannister, 1957; Noon et al., 1964) and disc brakes (Noon, et al., 1964; Newcomb, 1959; Newcomb, 1960; Richardson & Saunders, 1963). An analysis of drum and disc brakes (Newcomb, 1960; Newcomb & Millner, 1965-66; Petrof, 1965) indicates that of the heat generated during a single stop, 95 percent is dissipated at the drum while 99 percent is dissipated at the disc. During a single stop the temperatures achieved are approximately the same for both drum and disc brakes. The increased convection cooling capacity of disc brakes (Vansteenkiste, 1963; Newcomb & Millner, 1965-66; Newcomb, 1960), however, results in lower average temperatures during repeated braking. This fact, combined with the disc brake's lower sensitivity to lining coefficient changes, has been the reason for its adoption on high performance vehicles (Lueck, 1965; Huntington, 1964; Ihnacik, Jr. & Meek, 1967; Kemp, 1961).

Other design variations such as bimetallic brake drums (Automobile Engineer, 1959; Engineering, 1959) and ventilated discs (Koffman, 1956) have also been used to reduce temperature rise (SAE J971, 1968). Various papers have presented procedures for calculating the appropriate brake size (Newcomb, 1964; Newcomb, 1964; Rabinowicz, 1964) for vehicles on the basis of energy absorption considerations, and for establishing test schedules for evaluating brake fade performance by driving tests (Mackenzie et al., 1962-63; Livsey et al., 1960-61).

Lining Materials. Several theories have been advanced (Herring, 1967; Rabins & Harker, 1960; Garg & Rabins, 1965) dealing with the mechanism of fade caused by an increase in the lining temperature. The most recent of these (Herring, 1967) has proposed that the phenomenon is due to an evolution of gases from the lining material which tends to separate the rubbing surfaces. Additional work has been done investigating the effects of composition (Weintraub & Bernard, 1968; Jacko et al., 1968) on the thermal stability and fade characteristics of friction materials.

Investigators have also been concerned with developing appropriate test equipment (Wilson et al., 1968; Anderson et al., 1967; Clayton Manufacturing Co., 1967; Percy, 1951) for the evaluation of lining materials. Choosing an appropriate friction material is a trade-off (Fleet Owner, 1966; Autocar, 1965; Burkman & Highley, 1967; Mulvogue, 1966) between good and bad characteristics of the available products. Many Society of Automotive Engineers (SAE J661a, 1968; SAE J840a, 1968; SAE J667, 1968) and Federal (Federal Specification No. KKK-L-370c, 1961; Interim Federal Specification HH-L-00361d, 1965; Virginia Equipment Safety Commission, 1966; Federal Specification No. HH-L-361b, 1952) standards have been generated to provide guidelines in brake lining evaluation.

Fade and Pedal Effort. The driver will sense any decrease in the lining coefficient of friction as a decreased gain in the system, i.e., greater pedal effort will be required to produce the same vehicle deceleration.

Heat fade is not a problem in normal driving conditions (Vallin, 1968; MacKenzie, 1966) as the brake temperatures are generally below 300° F. The performance standard recommended by SAE specifies (SAE J843a, 1968; SAE J937, 1968) that in four successive stops from 60 mph not more than 200 lb pedal effort shall be necessary to achieve a deceleration of 15 ft/sec$^2$.

Normal brake effectiveness by the same standards is specified as being 15 to 120 lb to achieve a 20 ft/sec$^2$ deceleration.

Most drivers are more likely to encounter fade through water contamination of the lining than through heat fade. The mechanism is basically the same in that the effective lining coefficient is reduced, thereby requiring higher pedal efforts. SAE Recommended Practice (SAE Publication SP-299, 1967; SAE J937, 1968) specifies that 8 ft/sec$^2$ decelerations should be obtainable from 25 mph with less than 200 lb pedal effort after a two minute soaking of the vehicle's brakes.

BRAKE SYSTEM DEGRADATION. The performance of a braking system may be decreased by wear or 'failure' of any one of its components. 'Failure' here refers to both catastrophic failure such as a ruptured brake line or to marginal performance caused by the deterioration of a single component. A comprehensive brake system failure analysis of motor vehicles, such as that normally performed in the aircraft industry (Glasenapp & Gaffney, 1967), has never appeared in the literature. A qualitative discussion of many of the factors affecting brake performance is available (White, 1963), but it is written from the viewpoint of vehicle inspection. Some work has been done in determining the effectiveness of different dual braking arrangements (Vallin, 1968). Federal Standards (Federal Standard No. 515/9, 1965; MVSS No. 105, 1968) specify that following a pressure loss in a portion of a hydraulic braking system, the remaining portion of the system must be capable of stopping the vehicle from 60 mph in less than 646 feet on dry Portland cement concrete while maintaining the vehicle within a 12 foot lane. Degradation of the braking system will generally result in increased pedal forces and/or greater pedal travel, or a total loss of braking.

Hydraulic System. Hydraulic brake fluid performance is affected by its boiling point, water avidity, freezing point, viscosity, and corrosive action on rubber parts (Markey, 1956; SAE J664, 1968; Ker, 1968; Shiffler et al., 1968; Hanson & Coryell, 1960; Sharrard & Hanson, 1956). Considerable legislation, based mainly on SAE Recommended Practice (SAE J70b, 1968; Wright, 1965), has been passed (Richards, 1954; Lederer, 1955; Federal Specification No. VV-B-680a, 1967) to control the quality of brake fluid reaching the consumer. Motor Vehicle Safety Standard No. 116 (1969) sets federal specifications for hydraulic brake fluid using the testing procedures set forth in SAE Standard J70b (SAE J70b, 1968; Wright, 1965). Other SAE publications include information on the storage and handling (Niehaus & Shiffler, 1966; Shiffler, 1966; SAE J75, 1968) of brake fluid, brake line hoses (SAE J40d, 1968), and hydraulic cylinder seals (SAE J60, 1968; SAE J65, 1968). Motor Vehicle Standard No. 106 (1968) sets federal specifications for hydraulic brake hoses.

Wear of Friction Elements. The wear characteristics of the lining and drum (or disc) depend on the particular combination of the materials used (Willer, 1967; Lang, 1961). Lining wear is compensated for on all current U.S. production vehicles by automatic adjusters (Automotive Industries, 1968) and is therefore only a problem when this device fails. Procedures for determining lining wear performance are specified in various SAE publications (SAE J661a, 1968; SAE J667, 1968) yet no specific level of performance is suggested. Other SAE documents cover the lining bonds (SAE J840a, 1968) and rivets (Csathy, 1964) for attaching the brake lining or pad to the shoe.

## SKIDDING AS RELATED TO BRAKING

IMPORTANCE OF BRAKING CONTROL IN ACCIDENTS. Skidding, as used here, is any situation in which there is gross slippage between one or all of the vehicle's tires and the road surface. Skidding of a vehicle occurs when the limits of adhesion between the tire and the road are exceeded, and frequently results in a loss of directional control of the vehicle. The relationship between braking, skidding, and highway accidents should not be underestimated.

In a study (Grime, 1963) of 453 accidents involving one or more vehicles, more than three-fourths of the accidents involved skidding, and loss of control occurred following application of the brakes in more than half of these. It was not made clear in this study whether or not there was a cause and effect relationship between braking and skidding or whether skidding was simply symptomatic of the situation. A similar trend was indicated in a survey of commercial vehicle accidents (Starks, 1963). Although two-thirds of all accidents occur on dry roads, the incidence of accidents involving skidding is two to seven times higher when the roads are wet (Grime & Giles, 1954-55; Bulmer, 1962; Normann, 1953). It is felt that improved braking control (Grime, 1963) in many of these skidding accidents would have had a beneficial effect.

BRAKING DYNAMICS. During braking there is a dynamic transfer of weight onto the front wheels of a vehicle with a corresponding decrease at the rear wheels. Consequently there is a redistribution of usable braking torque between the front and rear axles for every different vehicle deceleration (Parker, 1960; Newcomb & Spurr, 1967; Lister, 1963; Ellis, 1963; Chandler, 1960). Studies of vehicle dynamics during braking (Lister, 1965; Odier, 1960; Radt & Milliken, 1960; Jones, 1962-63) have shown that the initial speed, braking characteristics, road surface friction coefficient, and vehicle parameters are important factors determining the directional response characteristics of the car. If a brake application should result in 100 percent longitudinal slip of a tire, the lateral force capability of that tire is reduced essentially to zero (Francia, 1963). This means that locking the front wheels in a braking maneuver results in nearly total steering loss. Locking the rear wheels causes the car to slew around if the vehicle encounters a yawing moment disturbance. If all wheels lock, the car can sideslip as well as rotate, depending on whether or not there is an external disturbance consisting of a side force and a yawing moment.

ROAD-TIRE FRICTION COEFFICIENTS. As indicated by Equation 7, the deceleration produced in a locked-wheel braking maneuver is determined by the locked-wheel coefficient of friction between the tire and the road. It is possible, however, for the friction coefficient achieved by a rolling, braked tire to allow decelerations in excess of those produced under locked-wheel conditions. Considerable research has shown (Csathy, 1964; Frood, 1962; Virginia Council of Highway Investigation and Research, 1959; Giles, 1963; ASTM Special Technical Publication No. 326, 1962; Texas Transportation Institute, 1962) that the effective coefficient of friction achieved by a rolling tire is dependent on many variables, the primary one being the longitudinal slip of the tire.

Longitudinal slip is the ratio of the equivalent ground speed of the tire to the actual vehicle speed. Measurements have shown that the coefficient of friction reaches a maximum at a longitudinal slip of 10 percent to 30 percent and then decreases gradually to a value termed the sliding coefficient at 100 percent slip. For a given tire, the shape of this curve and its magnitude varies (Hofelt, 1959; Kulberg, 1962) with both the surface condition and the vehicle speed. It has been found that both the peak and sliding coefficients generally decrease with increasing speed, but the peak value decreases at a lower rate (Kulberg, 1962; Schulze & Beckman, 1962). As the speed increases, the value of slip at which the coefficient is maximum tends to decrease (Goodenow et al., 1968). The measured coefficient of friction for a public road is by no means a constant and varies seasonally (Csathy, 1964; Kummer & Meyer, 1967 and from lane to lane (Mahone, 1962) on the highway. The tread pattern, construction, and composition of the tire influence the tire-road friction coefficient (Easton, 1960; Mechanical Engineering, 1968; Kelley, Jr. & Allbert, 1968) whereas tire inflation pressure appears to have negligible effect (DeVinney, 1967). Wet road surfaces change the frictional characteristics of the road (Kelley, Jr. & Allbert, 1968; Maycock, 1965-66; Obertop, 1962; Hoefs, 1961) and introduce the additional factor of hydroplaning (Allbert, 1968). It appears that vehicle speed (DeVinney, 1967) is the single most important variable in wet surface conditions, the coefficient decreasing with increasing vehicle speed. Investigations have also been made on winter driving conditions (NSC, 1966; Sapp, 1968; NSC, 1962) and tread wear (Leland & Taylor, 1964).

Although the problem of accurately measuring the coefficient of friction (Texas Transportation Institute, 1962; Goodwin & Whitehurst, 1962; Davisson, 1968) of a road surface has been dealt with in many ways, basically three methods have been used: skid trailers, vehicle stopping distance measurements (Whitehurst, 1965), and portable testers (ASTM Special Technical Publication No. 366, 1965). Skid trailers of various designs (Kulberg, 1962; Goodenow et al., 1968) have been employed to measure both the sliding and peak friction coefficients. Comparisons between the British Portable Tester and automobile-stopping distance measurements (Rizenbergs & Ward, 1967) have shown a useful correlation (ASTM Special Technical Publication No.366, 1965) between their results when patterned tires are used. In all skid resistance measurements (Frood et al., 1962) the conditions and methods must be carefully controlled to obtain consistent results.

SKID CONTROL BY BRAKING MODULATION. An examination of the frictional characteristics of the tire-road interface indicates that maximum braking is obtained when tire slip is maintained near the peak of the curve. Braking beyond this point (Bulmer, 1962) can result in significantly longer stopping distances (Lister & Kemp, 1961) and the loss of directional control. Modulating the pedal force so as to control the wheel slip at the peak of the curve is a difficult task for the driver, so simpler methods of pedal modulation have been suggested.

Pumping and cadence braking (Lister & Kemp, 1961) are two ways of improving braking performance and maintaining steering control. The philosophy of these methods is that repeated on-off braking will cause the tire slip to pass repeatedly through the peak of the coefficient/slip curve. Pumping consists of applying and releasing the brakes rapidly, while cadence braking is pumping the brakes in resonance with the pitch motion of the car. Comparisons (Lister & Kemp, 1961) indicate that these two methods can achieve shorter stopping distances than those of locked-wheel stops on low coefficient surfaces. On high coefficient surfaces the stopping distances were the same, although pumping or cadence braking also enabled the driver to maintain directional control.

Anti-wheel-lock systems have been employed for several years on large aircraft (Collier, 1958) and a few systems have been developed for automotive use (SAE J840a, 1968; Autocar, 1962; Automobile Engineer, 1958; Lister & Kemp, 1958; SAE J660, 1968). When using these systems, the driver operates as usual under normal conditions; however, as the driver causes the wheels to approach lockup in a panic stop, the device takes over. Using an inertia switch (Design News, 1959; Design News, 1957; Machine Design, 1959) to sense impending wheel lockup, the anti-skid device automatically pumps the brakes to maintain a slip rate close to that value producing maximum adhesion. Control is returned to the driver when he reduces the pedal effort. Other anti-skid systems operate similarly but derive the wheel acceleration signal from wheel velocity. It is claimed that several of these systems (Lister & Kemp, 1958; Scafer & Howard, 1968) provide performance superior to locked wheel stops by improving driver steering control and shortening stopping distances, particularly in adverse driving conditions.

## TESTING OF THE VEHICLE-TIRE-BRAKE SYSTEM

DECELERATION PERFORMANCE. The deceleration performance (Tignor, 1966; Harding, 1961) achieved during a stop can be expressed as a function of (a) the stopping distance, (b) the average deceleration produced during the stop, and (c) the degree of driver control.

Stopping Distance. The total distance covered during a braking maneuver includes the distance traveled during the driver's response time and the actual deceleration of the vehicle. During the driver's response period the vehicle would be continuing at nearly the initial velocity. The deceleration experienced during the braking interval depends on the rolling resistance, aerodynamic drag, and engine drag as well as the braking torque and the coefficient of friction existing at the tire-road interface.

Stopping distance may be measured experimentally by integration of a fifth wheel velocity signal or by direct distance measurement. The latter method can be made reasonably accurate by employing an explosively-fired chalk pellet (Lister, 1959) to indicate the point of brake application. Minimum driver response times have been measured by recording the time period between the flashing of a signal and the start of brake application (Normann, 1953, Konz & Daccarett, 1967).

Deceleration Measurement. An average value of vehicle deceleration can be computed from a measurement of the initial speed and stopping distance. An instantaneous value can be obtained by differentiation of a fifth wheel velocity signal (Carpenter, 1956) or by direct measurement with an accelerometer on board the vehicle (Harding, 1961). The computation of average deceleration method is not useful, however, in correlating pedal force with deceleration. Accelerometers on board the vehicle have the problem (Harding, 1961) of being sensitive to the vibrations caused by normal road roughness, the pitch motion during braking, and the ascent and descent of grades. The latter two problems may be eliminated by mounting the accelerometer on a stabilized platform within the vehicle. The velocity signal differentiation method avoids the difficulties encountered with the on-board accelerometer and has been successfully used in brake usage tests (Carpenter, 1955). This method does, however, suffer from a loss in frequency response because of the necessary electrical filtering circuits.

Directional Control. Studies in which measurements have been made of the steering control required during braking have not appeared in the literature. The few quantitative measurements of directional response that have been made consist of a determination of the final angular deviation of the vehicle from its intended path after a straight line braking maneuver (Lister, 1963). Although tests and evaluations of anti-skid systems (Traffic Institute, Northwestern University, 1960) have included cornering maneuvers during severe braking, quantitative evaluations of improvements in directional stability and control with respect to conventional braking schemes have not been made. SAE Recommended Practice (SAE J937, 1968) for brake evaluation tests requires only that the vehicle remain within a straight 12 foot roadway.

CONTROL OF BRAKING TESTS. The variables influencing stopping distance measurements (Goodwin & Whitehurst, 1962) are largely the same as those affecting the measurement of the surface friction coefficient. A method for statistically analyzing deceleration data (Leah, 1964) has been published claiming that decelerations may be measured accurately to within 2 1/2 percent. Repeatability of the measurements, however, depends on controlling other variables such as the surface coefficient, tire wear, and pedal actuation. In an attempt to remove the human element from pedal actuation, programmed servo-controlled brake pedal actuators have been employed in some brake tests (Automotive News, 1968). Despite careful control of the variables, high speed braking tests, using the same car and driver, and conducted on the same day and surface, have shown considerable scatter with respect to stopping distance (Normann, 1953) and directional stability (Odier, 1960).

DRIVER RESPONSE

STATIC DRIVER-VEHICLE RELATIONSHIPS. Several investigators have dealt with the problem of defining driver-vehicle relationships from the standpoint of applied anthropometry. In a study of the knee heights of 2,376 civilian drivers (McFarland, 1954), a 95th percentile knee height of 23 1/2 inches was established, with the recommendation that there be a minimum distance of 24 1/2 inches between the pedal and the steering wheel of a vehicle. Data accumulated for 12 different brake pedal designs indicated a wide range of pedal heights, sizes, and locations.

A similar study of 10 truck cabs indicated (McFarland, 1958) that many designs were far below the minimum standards essential to ease of operation and driver efficiency. An example cited was the physical interference of the cab interior with leg movement during brake pedal actuation. The anatomical variables considered important for proper pedal design were foot breadth, foot length, leg length, knee height, buttock-popliteal length, and the range of angles formed by the leg foot articulation. In addition to providing sufficient seat adjustment, a proper design should (McFarland, 1958) also include appropriate clearances forward of the pedals and lateral clearances between the pedals as required by a 95th percentile driver. Useful anthropometric data have been compiled (Drillis & Contini, 1966; Product Engineering, 1967) providing information on the dimensions, masses, volumes, densities, centers of gravity, and moments of inertia of many body segments for various population samples.

A study of the driver's position relative to the brake pedal indicates that the pedal force which a subject can exert (Aoki, 1960) is maximized at a particular knee angle and posture angle. In 1953 researchers at the Harvard School of Public Health (Regis,

1953) found that the maximum foot power for a downward motion could be generated when the initial included angle between the foot and tibia was 78 degrees. This is based on a horizontal femur and an included angle of 114 degrees between the femur and the tibia. Data presented for a study of Japanese drivers (Aoki, 1960) indicated that the 95th percentile driver could exert at least 25 pounds and recommended that the force necessary to operate the brakes should not exceed 20 kg (44 lb). If the pedal is designed for operation with the driver's heel placed on the floor, the required force should be further reduced.

DRIVER TRANSIENT RESPONSE CHARACTERISTICS. During normal braking maneuvers the driver and vehicle operate as a closed loop system, but in maneuvers approaching emergency conditions, it is likely that the braking is performed in a completely open loop manner. In the former instance it is postulated that the driver observes the current rate of deceleration and increases or decreases the brake pedal effort according to the deceleration error sensed. It is possible, therefore, to represent the driver as a servo-system element operating within a complex man-machine system (see Figure 1.3). In using this representation, the dynamics of the 'error' sensing operation are included in the driver's transfer function.

The response characteristics of the driver as a control element are discussed in this section. The next section deals with a study of the dynamic behavior of a man-pedal force system.

An examination of the literature indicates that the total human response time in braking is considered to consist of three periods: a reaction time (time period from stimulus until the foot is removed from the accelerator), a transfer time (period from removal of the foot from the accelerator to start of brake application), and a force transient (time to apply the full pedal force). The brake-pedal configuration has been shown to have a considerable effect on the driver's performance (Barnes, 1942;

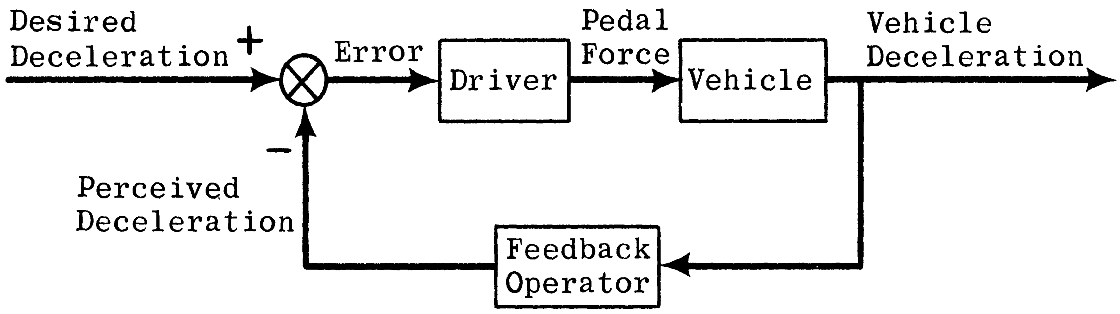



Figure 1.3. The braking process represented as a feedback control system.

Ensdorf, 1964). Overall response time measurements on 12 university students using an unspecified stationary 1964 auto and a light stimulus resulted in an average response time of .59 seconds (Kontz & Daccarett, 1967). In a similar Japanese study (Aoki, 1960) using a fixed brake pedal, 80 percent of the male and female drivers tested showed response times less than 1.2 and 1.4 seconds respectively. Laboratory experiments with a combined brake-accelerator pedal (Konz et al., 1968; Motor Vehicle Research Inc., 1959) have resulted in a savings in the overall response time of .1 to .2 seconds over that obtained using a conventional brake-pedal configuration. Actual road tests employing a conventional brake pedal and a light stimulus have resulted in a response time of .73 seconds (Normann, 1953).

In a Japanese study (Aoki, 1960) the magnitude of the brake force was reported to have little effect on the reaction time, and 50 percent of the subjects tested had a reaction time of .30 seconds or less. An American study (Ayoub, 1967), however, indicated that the reaction time increased in proportion to the required force. Furthermore, the reaction time was minimized for a foot-tibia angle of 78 degrees, which coincides with the angle that maximizes (Rejis, 1953) the power output of a human operating a foot pedal.

The transfer time (Aoki, 1960) appears to be a function of pedal angulation and the vertical and lateral heights between the pedals. The transfer time for 50 percent of the Japanese subjects tested was approximately .25 seconds, and empirical equations for transfer times were derived for two cases; namely, the driver's foot being on or off the floor.

The pedal force transient is a function of the required final force and the posture of the driver (Aoki, 1960; Ayoub, 1967). This force response can be described by a first order lag transfer function (Aoki, 1960; Aoki, 1964) in which the time constant (see Equation 10) decreases as the maximum pedal force decreases and as the driver's position approaches that corresponding to his maximum force output.

$$F = \frac{F_o}{(1 + TS)} \tag{10}$$

where

$F_o$ = final pedal force achieved

$F$ = driver's response

$T$ = time constant

$S$ = Laplace operator

In the case of a fixed brake pedal, values of the time constant, T, were observed to range from .04 seconds to .2 seconds for $F_o$ = 10 kg (22 lb). As the commanded force increased, a trend towards slightly higher values of the time constant was observed.

DRIVER-BRAKE PEDAL SYSTEM DYNAMICS. A laboratory investigation (Aoki, 1964) of the effects of force and displacement feedback on the performance of a subject actuating a foot pedal has been reported in the Japanese literature. The experimental apparatus consisted of a simulated driver's seat (stationary) with a brake pedal having controlled force and displacement characteristics. The subject was shown a display representing a commanded force signal. His resulting pedal force effort was compared to the command signal on an oscilloscope, thus providing feedback. Feedback was not begun, however, until the driver's pedal force exceeded the command force. The experiments showed that the overall system response was similar to that of a second order underdamped system. It was found that the amount of overshoot increased as (a) the time to the first overshoot decreased, (b) the commanded brake force decreased, and (c) the pedal displacement decreased. The measured overshoots were approximately 35 percent at 10 kg and 10 percent at 20 kg. On the basis of these results and because of the desire to avoid fatiguing high pedal forces, the author speculated that the best operator performance would be achieved when the required pedal force was in the region of 20 kg (44 lb).

## 2. FOOT-FORCE CAPABILITY OF DRIVERS

### INTRODUCTION

Many measurements have been taken of the human's ability to exert pressure in pushing movements with the feet located at various lateral positions with respect to the midline of the body and at various horizontal angles and distances from the body (Damon et al., 1966). In most instances, these data have been collected for design applications other than those of concern in the present study. Moreover, almost all of these studies were carried out with subjects selected from military populations. In the few studies in which data were obtained for civilians, the samples were small and, with one exception, did not involve an American population.

For example, meager data for Japanese males and a selected group of young Japanese females showed that the 5th percentile young Japanese female could exert a maximum pedal force of only 37 lbs (Aoki, 1960). Thus, on a dry surface, this female would be unable to obtain the maximum braking capability of any American car that does not possess power-assisted brakes. On the other hand, studies (involving male military personnel) have resulted in much higher 5th percentile values, e.g., 407 lb (Elbel, 1949) and 484 lb (Haigh-Jones, 1947).

It is apparent that pedal-force capabilities are highly variable and very much a function of the population sample. Although major differences exist between certain population groups, measurements have shown that the force capabilities (and indeed the anthropometric measurements) of German, Russian, Australian, and certain other populations are quite similar to the American population (Australian Army Op. Res. Group, 1958; Kroemer, 1966).

Recently, maximum force capability, in depressing a brake pedal, was measured on a representative sample of 50 U.S. females

(Stoudt et al., 1969). A mock-up of an automobile was used and measurements were made of the average force maintained by subjects in depressing a brake pedal over a ten second period. Measurements were made for five consecutive trials, resulting in a 5th percentile force of 86, 110, 122, 131, and 140 pounds being recorded in trials one through five, respectively.

Studies made of the interaction between pedal-force capability and limb orientation and geometry have shown that the driver's knee angle should be between 160 and 170 degrees when the brake and clutch are in the undeflected position (HSRI, 1967). With this geometry, maximum force can be attained; in addition, there is sufficient allowance for leg extension to depress the clutch and brake. It should be noted that the driver is placed in an awkward and uncomfortable position if the knee angle is less than 90 degrees.

Relative locations and dimensions of throttle, clutch and brake pedals that have been demonstrated to be a preferred arrangement have been summarized in a previous HSRI report (1967). Another ergonomic study dealing with the location of driver controls has since been reported (Woodson et al., 1969).

It should be noted that the Harvard study (Stoudt et al., 1969) was not completed until after this project got underway. At the very beginning of this project, a decision was made to acquire pedal-force capability data similar to that being sought by the Harvard group, but using a larger sample of both male and female subjects. It was also decided to use a hard seat to collect these data, in contrast to the Harvard effort which employed a soft seat.

Prior to finalizing the design of HSRI's test apparatus, a sample of vehicles were surveyed to obtain information on current practice in dimensioning and locating brake and accelerator pedals. The survey was restricted to 1968 models with measurements being made at new- and used-car dealerships.

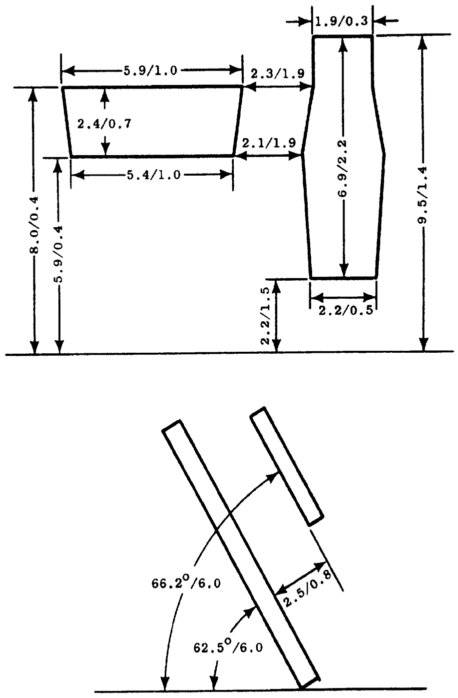
The sample consisted of 10 intermediate and 13 full-size cars with power brakes.

The mean and standard deviation of (1) the dimensions of the accelerator and brake pedals, (2) the pedal separation distance and (3) pedal angular inclination are shown in Figure 2.1 for the four vehicle groupings. Brake pedal angles of 33 degrees to 39 degrees were found. Accordingly, the simulated pedal used to measure foot force in this study was adjusted to fall within this range. It is of interest to note that the accelerator and brake pedal dimensions found in this survey generally met the recommended minimum requirements derived by HSRI in a previous review of anthropometric data (HSRI, 1967).

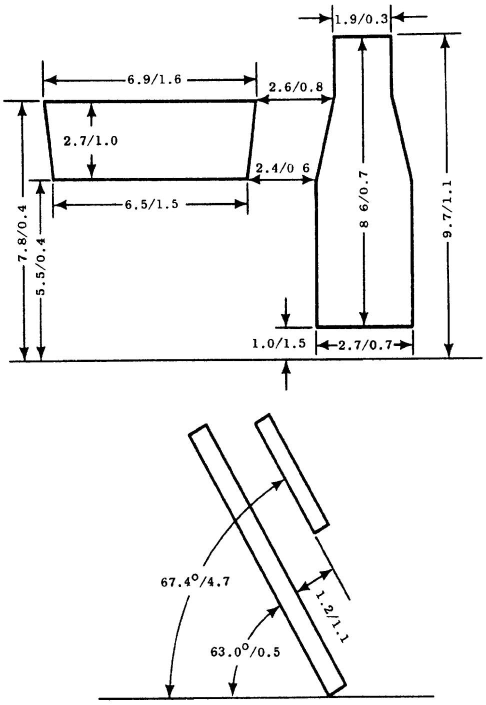
## METHOD

APPARATUS. Figure 2.2 shows the device used to measure the maximum foot-force capability of subjects. A chair, 28 inches wide and 16 inches deep, was covered with a nonslip vinyl surface and was raised/lowered by means of a hydraulic lift. The chair back was 17 inches high, mounted at an angle of 25 degrees from the vertical. A hydraulic force gauge, 300 lbs full scale, equipped with a ribbed circular steel pad (1.75 inches in diameter), was mounted at an angle of 35 degrees and was horizontally and vertically adjustable (Figure 2.3). Body weight and foot weight were measured with a general utility scale.

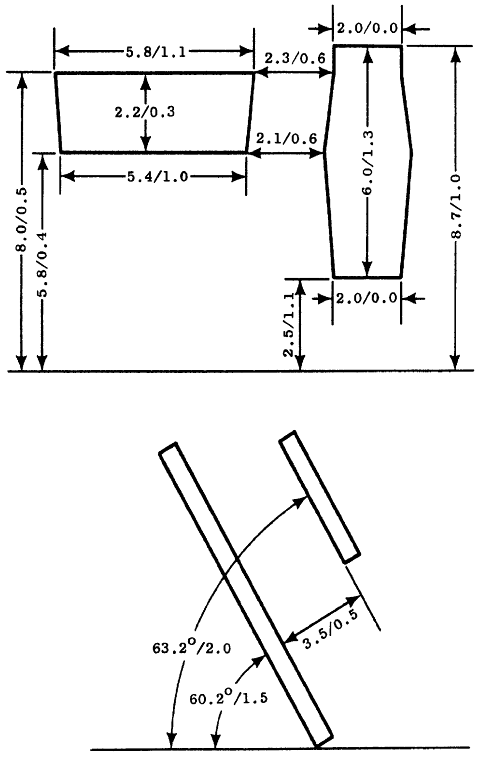
In using the pictured apparatus, the pedal height and distance from the subject were adjusted to yield a thigh angle of zero degrees and a knee angle of 160 degrees. This adjustment was facilitated by computing these settings in advance as a function of all likely combinations of driver foot length and lower leg length that might be encountered.



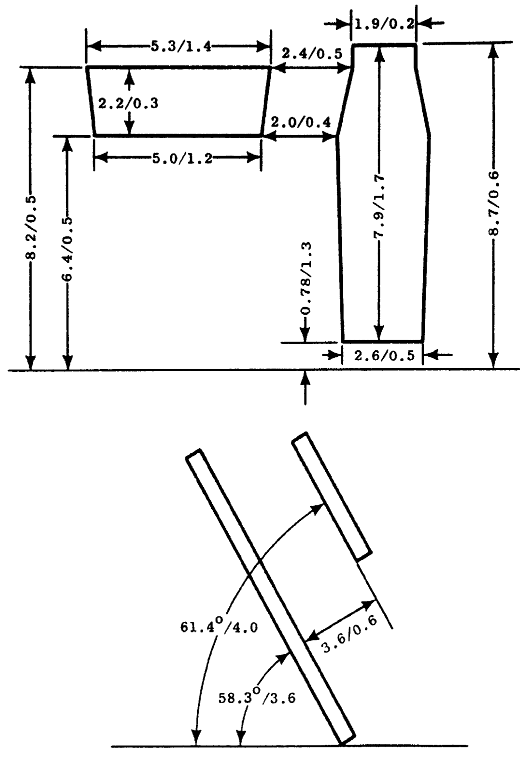

(a) 10 Intermediates, Size Cars, Power Brake




(b) 13 Full-Size Cars, Power Brake




(c) 6 Intermediate Size Cars, Manual Brake




(d) 7 Full-Size Cars, Manual Brake

Figure 2.1. Brake and Accelerator Dimensions, Mean/ Standard Deviation, 1968 Cars.

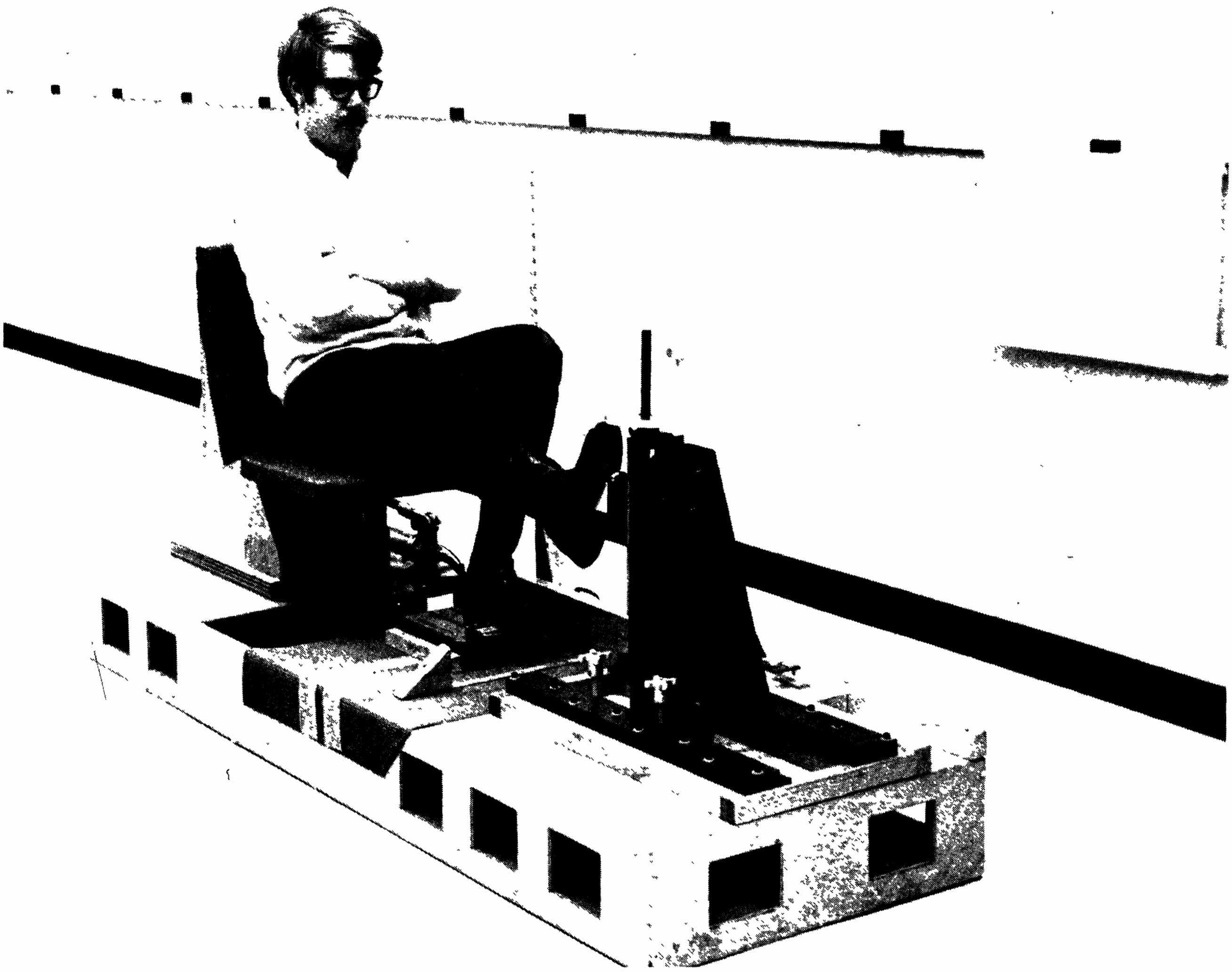


Figure 2.2. Foot pedal force measurement buck.

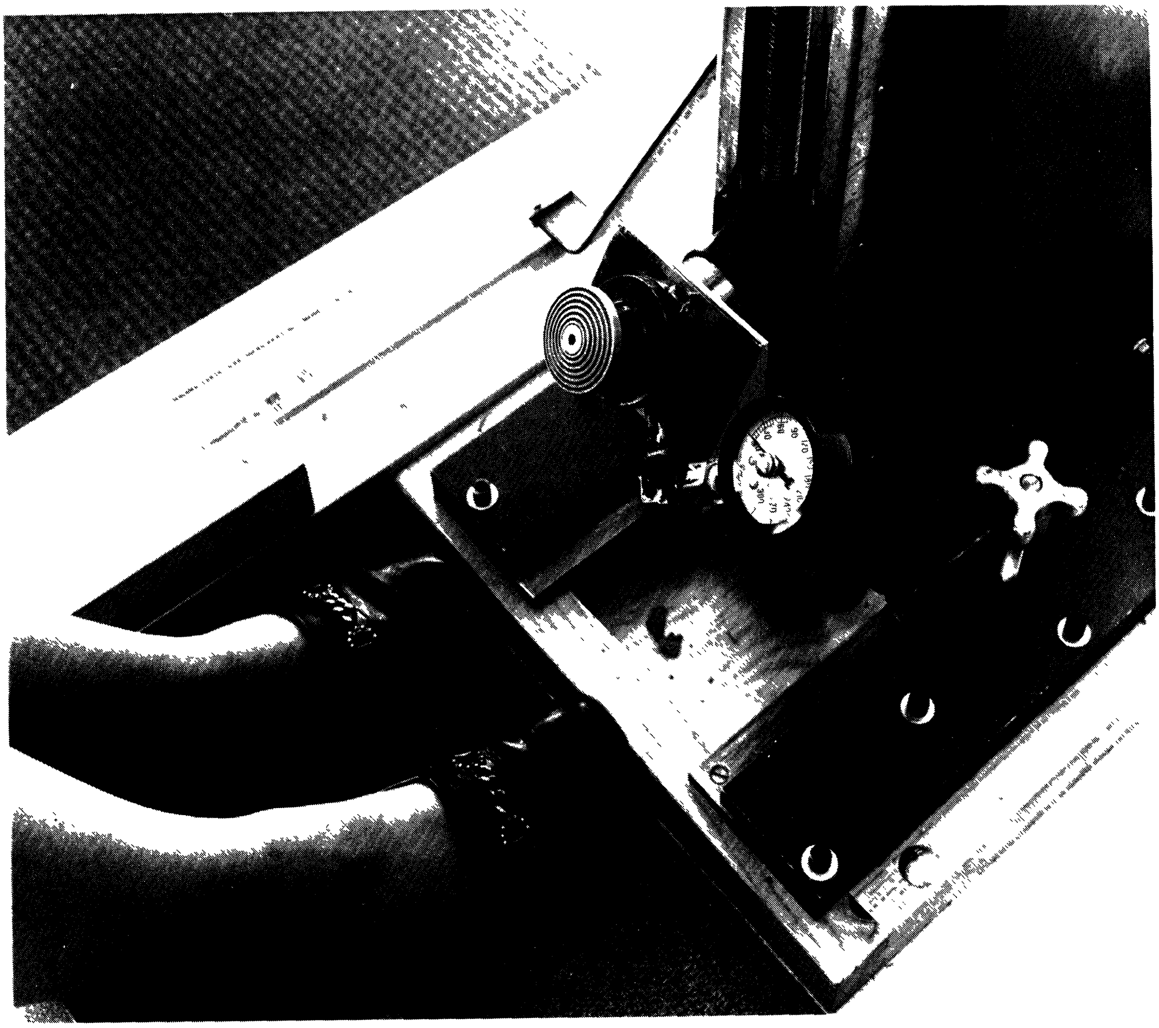


Figure 2.3. Hydraulic force gauge and foot pad.

PILOT STUDY.

Procedure. A pilot study was conducted to determine the consistency of pedal forces as measured under various conditions: (1) high and low motivation instructions, (2) right- and left-foot force, and (3) subject able to see the gauge while applying a force.

Participating in the pilot study were eight males weighing from 142 to 250 lbs with a mean weight of 180 lbs, and 28 females weighing 105 to 168 lbs with a mean of 133 lbs.

Subjects were tested on two consecutive days. On the first day right-foot pedal forces were recorded for each subject with standard (low) motivation instruction--"push the pedal as hard as you can and hold it for three seconds." In all cases, the force gauge was visible to the subjects. On the second day all subjects were retested. The 8 males and 16 (Group A) of the 28 females were able to see the force gauge as on the first day. For the remaining 12 females (Group B) visibility of the gauge was occluded. Also, on the second day each subject was tested for right- and left-foot force using the standard instruction (see above) on the first trial and then immediately retested with the following instruction: "This time really push as hard as you can--like you are driving a car and have to stop to avoid a serious accident."

Results. A comparison of the mean forces exerted (Table 2.1) indicates that all three groups improved from the first to the second day. The mean force of the females that were allowed to see the gauge in creased 34.3 percent compared to an increase of less than two percent for the females not allowed to see the gauge. For the females allowed to see the gauge, "induced" motivation further increased the mean force 20.7 percent over the standard instruction. The mean force for the females seeing the gauge and given "induced" motivation instruc-

TABLE 2.1. PILOT STUDY MEDIAN RIGHT AND LEFT FOOT FORCE BY THREE GROUPS OF SUBJECTS FOR "STANDARD" AND "INDUCED" MOTIVATION INSTRUCTIONS. DATA ARE IN POUNDS

| | | "STANDARD" MOTIVATION | | "INDUCED" MOTIVATION | |
|---|---|---|---|---|---|
| DAY | SUBJECTS | RIGHT | LEFT | RIGHT | LEFT |
| 1 | Males (N=8) | 249 | - | - | - |
| | Females ($N_A$=16) | 119 | - | - | - |
| | Females ($N_B$=12) | 113 | - | - | - |
| 2 | Male (N=8) | 254 | 268 | 276 | 283 |
| | Females ($N_A$=16) | 160 | 158 | 194 | 191 |
| | Females ($N_B$=12) | 116 | 128 | 175 | 183 |

$N_{(A)}$: Force gauge visible both day 1 and 2

$N_{(B)}$: Force gauge visible on day 1, occluded on day 2 tions was 71.3 percent greater than that for the group not seeing the gauge and given the standard instructions. High correlations were found between right- and left-foot force in both motivational conditions. For all subjects (N=36) on the second day of testing in the "standard" motivation condition, $r^*_{R.L} = .96$; in the "induced" motivation condition, $r_{R.L} = .93$.

Right foot forces were also highly correlated across the two ("standard" and "induced") motivational conditions ($r_{S.I} = 0.94$) for those subjects allowed to see the gauge on the second day. For those subjects who did not see the gauge on the second day, the correlation in foot force between "standard" and "induced" motivation was $r_{S.I} = 0.66$. Thus, variability is reduced when the subject is able to see the gauge. Rank-order correlations between right-foot forces obtained in the standard motivation condition on the first and second days seemed to reflect the positive reinforcement effect of seeing the gauge. Subjects able to see the gauge on both days produced a correlation of $r_{1.2} = 0.39$.

In view of the above results, it was decided to make the gauge visible to the subjects in the final survey, as well as to use both levels of motivational instructions and to measure forces produced by each foot in order to obtain the most comprehensive and reliable results.

MAIN STUDY.

Procedure. The test equipment was taken to a large shoe store in a local shopping center and subjects were recruited from patrons and passers-by. The equipment was later moved to the Driver License Bureau of the Michigan Department of State

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$^*r_{R.L}$ is the correlation between right and left foot maximum force.

where, with the cooperation of officials, a greater age range of subjects could be tested. Experimenters followed a procedure identical to that used in the second day of the pilot study. The force gauge was visible. Subjects were given the "standard" instructions and data recorded for the right and left foot. Foot order was alternated across subjects. Right- and left-foot force measurements were then taken with the "induced" instructions. In addition, foot length, body weight, lower leg weight (with subject seated and legs resting on the scale), and lower-leg height were measured.

Subjects. The study sample consisted of 276 female and 323 male drivers. The females were 16 to 79 years of age with a mean age of 32.5 years. Their weights ranged from 89 to 225 lbs with a mean of 135.9 lbs. The males were 16 to 89 years of age with a mean age of 31.8 years. Their weights ranged from 119 to 285 lbs with a mean of 178.1 lbs. The age and weight distribution of subjects is shown in Tables 2.2 and 2.3. It should be noted that younger drivers (16-24 years) are over represented in the sample. Accordingly, the measured distribution is not likely to be an underestimate of the pedal-force capability of the driver population.

## RESULTS

Tables 2.4 and 2.5 show the cumulative frequency distributions of maximum force achieved by female and male subjects, respectively, with the right foot. (Left-foot data are not shown because of the high correlation that was found for the two feet.) For the standard motivation instruction, the 5th and 50th percentiles of maximum force achieved by the 276 females (Table 2.4) are respectively, 70.3 lbs and 152.7 lbs. For the induced motivation instruction, the 5th and 50th percentiles are equivalent to 102.3 lbs and 193.7 lbs. Males, on being given the standard instruction, attained a 5th percentile force of 133.1 lbs and a 50th percentile level of 279.1

lbs (Table 2.5). Note that these results are very similar to those obtained in the pilot study for the analogous test conditions (see Table 2.1). Performance at the 50th percentile could not be determined for highly motivated males since the majority of male subjects exceeded the 300 lb limit of the force gauge. Figures 2.4 and 2.5 show the cumulative percent of foot-force capability as achieved by females and males in the two trials.

Person Product-Moment correlations performed on a random sample of 100 subjects (57 males and 43 females) showed that foot weight ($W_F$) is highly correlated with total body weight ($W_B$): $r_{W_F \cdot W_B} = .83$. However, for this sample of subjects, body weight and right foot force (F) with standard motivation had a low correlation of $r_{W_B \cdot F} = .24$. A sample of 46 females produced a correlation of $r_{W_B \cdot F} = 0.26$ between body weight and right-foot force produced with a standard instruction. The same sample attained a correlation of $r_{W_B \cdot F} = .18$ when body weight was compared with right-foot force produced under an induced motivation.

DISCUSSION

A comparison of the above results with those obtained in the Harvard study produces the following findings. The 50 female subjects tested by Stoudt et al., attained a mean force of 201 lbs and a 5th percentile force of 126 lbs, averaged over all five trials. The Harvard subjects were tested five times (all on the right foot) under conditions corresponding to the "induced" motivation condition of this study. The measured forces increased with each successive trial, suggesting that both a learning and motivational effect were present. The subjects in the HSRI study were tested four times. Only two of the tests were on the right foot, the first in the "standard" and the second in the "induced motivation condition.

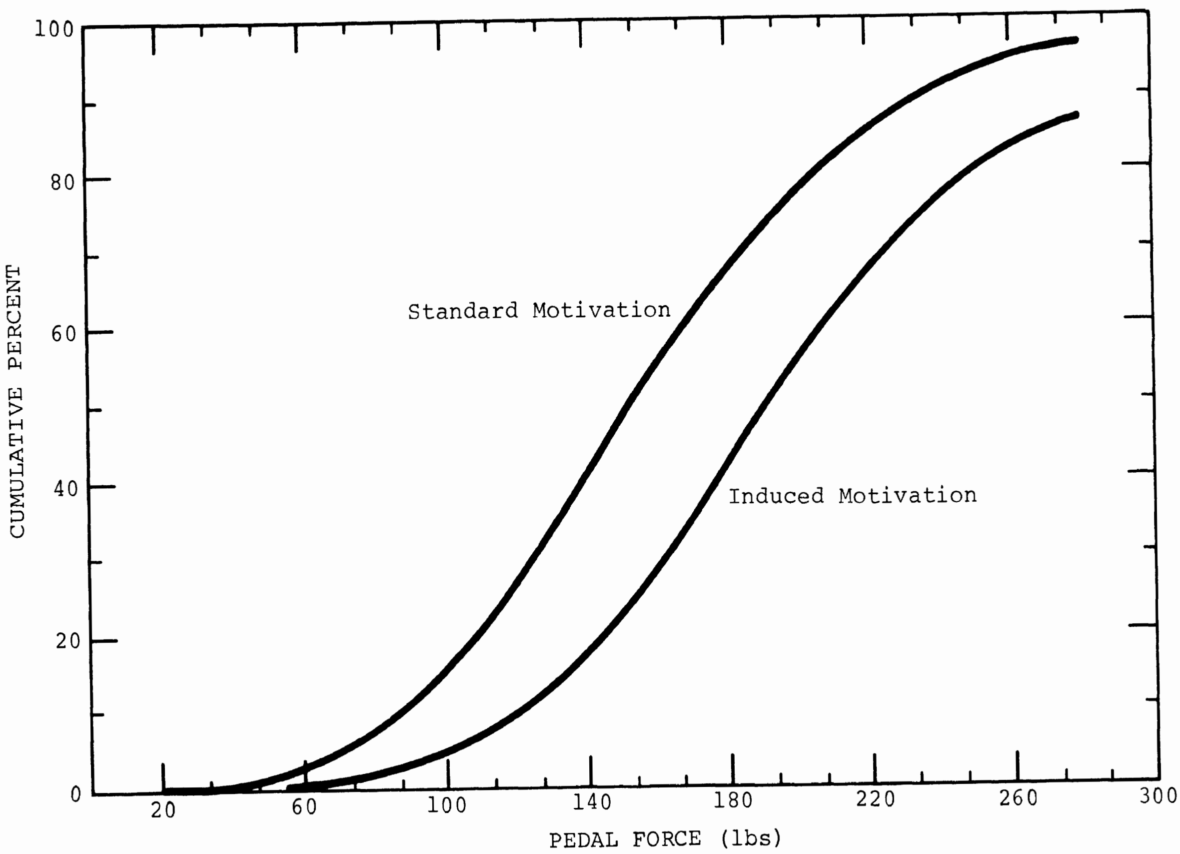


Figure 2.4. Cumulative percent pedal force for 276 females.

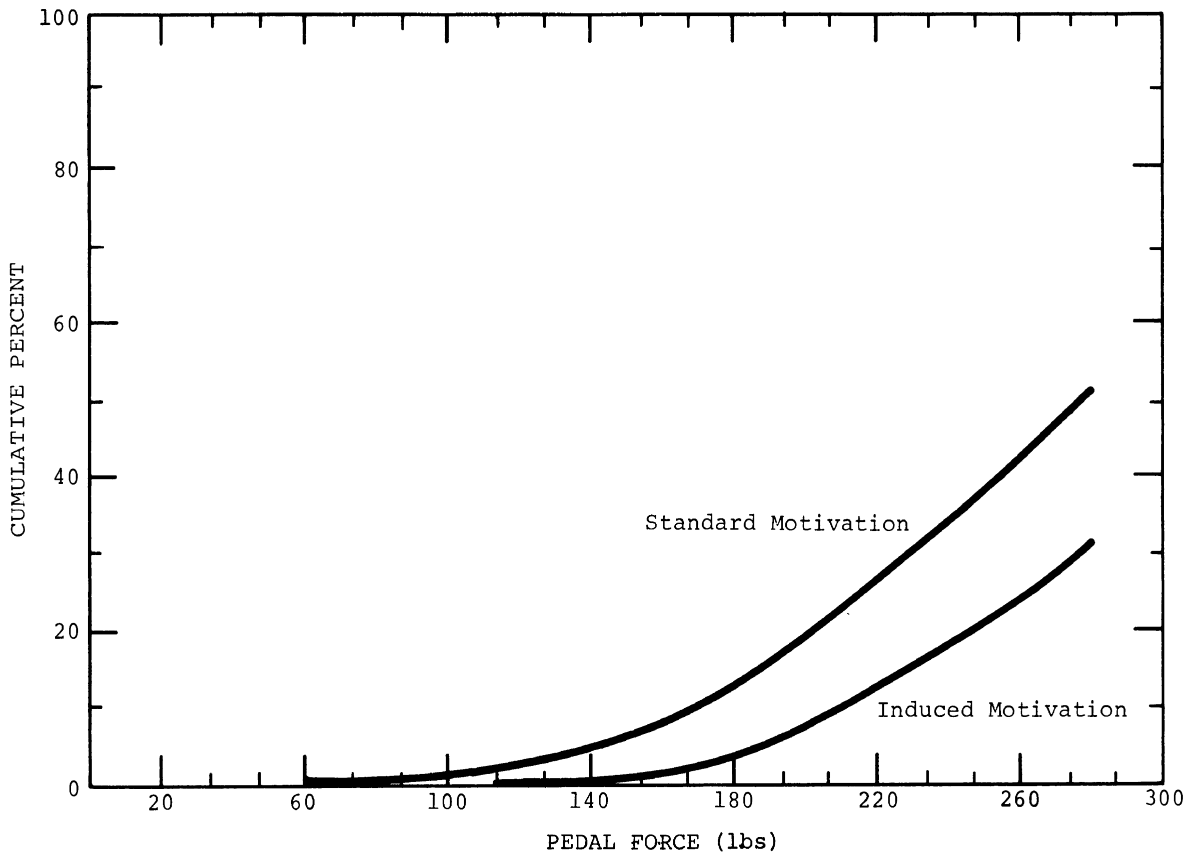


Figure 2.5. Cumulative percent pedal force for 323 males.

TABLE 2.2. AGE DISTRIBUTION OF FEMALE AND MALE SUBJECTS

FEMALES

| Age | Frequency | Percent | National Estimate (%)** |
|---|---|---|---|
| 65+ | 4 | 1.45 | 5.2 |
| 55-64 | 17 | 6.18 | 11.1 |
| 45-54 | 35 | 12.73 | 18.1 |
| 35-44 | 48 | 17.45 | 22.2 |
| 25-34 | 63 | 22.91 | 21.7 |
| 16-24 | 108 | 39.28 | 21.7 |
| Mean Age= | | | |
| 32.5 | N=275* | 100.00 | 100.0 |

MALES

| Age | Frequency | Percent | National Estimate (%)** |
|---|---|---|---|
| 65+ | 19 | 5.90 | 9.6 |
| 55-64 | 16 | 4.97 | 12.8 |
| 45-54 | 23 | 7.14 | 17.5 |
| 34-44 | 37 | 11.49 | 20.0 |
| 25-34 | 90 | 27.95 | 19.8 |
| 16-24 | 137 | 42.55 | 20.3 |
| Mean Age= | | | |
| 31.8 | N-322* | 100.00 | 100.0 |

* Age not given for one subject

**From: Automobile Facts and Figures (1968)

TABLE 2.3. WEIGHT DISTRIBUTION OF FEMALE AND MALE SUBJECTS

FEMALES

| Weight (lbs) | Frequency | Percent | Cumulative Percent |
|---|---|---|---|
| 281-300 | - | - | - |
| 261-280 | - | - | - |
| 241-260 | - | - | - |
| 221-240 | 1 | .36 | 100.00 |
| 201-220 | 3 | 1.09 | 99.64 |
| 181-200 | 5 | 1.81 | 98.55 |
| 161-180 | 25 | 9.06 | 96.74 |
| 141-160 | 58 | 21.01 | 87.68 |
| 121-140 | 116 | 42.03 | 66.67 |
| 101-120 | 59 | 21.38 | 24.64 |
| 81-100 | 4 | 1.45 | 3.26 |
| no weight taken | 5 | 1.81 | 1.81 |
| | 276 | 100.00 | |

Range: 89-225 lbs

Mean: 135.9 lbs

MALES

| Weight (lbs) | Frequency | Percent | Cumulative Percent |
|---|---|---|---|
| 281-300 | 1 | .31 | 100.000 |
| 261-280 | 1 | .31 | 99.69 |
| 241-260 | 6 | 1.86 | 99.38 |
| 221-240 | 15 | 4.64 | 97.52 |
| 201-220 | 39 | 12.07 | 92.88 |
| 181-200 | 76 | 23.53 | 80.80 |
| 161-180 | 97 | 30.03 | 57.28 |
| 141-160 | 64 | 19.81 | 27.24 |
| 121-140 | 21 | 6.50 | 7.43 |
| 101-120 | 2 | .62 | .93 |
| 81-100 | 0 | - | - |
| no weight taken | 1 | .31 | .31 |
| | 323 | 100.00 | |

Range: 119-285 lbs

Mean: 178.1 lbs

TABLE 2.4. CUMULATIVE PERCENT RIGHT FOOT FORCE DISTRIBUTION: 276 FEMALE DRIVERS

| | STANDARD MOTIVATION | | INDUCED MOTIVATION | |
|---|---|---|---|---|
| Pressure (lbs) | Frequency | Cumulative Percent | Frequency | Cumulative Percent |
| 281+ | 9 | 100.00 | 32 | 100.00 |
| 261-280 | 14 | 96.74 | 21 | 88.41 |
| 241-260 | 2 | 91.67 | 11 | 80.80 |
| 221-240 | 13 | 90.94 | 27 | 76.81 |
| 201-220 | 27 | 86.23 | 32 | 67.03 |
| 181-200 | 31 | 76.45 | 44 | 55.43 |
| 161-180 | 28 | 65.22 | 29 | 39.49 |
| 141-160 | 36 | 55.07 | 32 | 28.99 |
| 121-140 | 46 | 42.03 | 26 | 17.39 |
| 101-120 | 27 | 25.36 | 9 | 7.97 |
| 81-100 | 21 | 15.58 | 5 | 4.71 |
| 61-80 | 16 | 7.97 | 6 | 2.90 |
| 41-60 | 4 | 2.17 | 1 | .72 |
| 21-40 | 2 | .72 | 1 | .36 |
| | Σ=276 | | Σ=276 | |

50th percentile=152.7
5th percentile= 70.3

50th percentile=193.7
5th percentile=102.3

TABLE 2.5. CUMULATIVE PERCENT RIGHT FOOT FORCE DISTRIBUTION: 323 MALE DRIVERS

| Pressure (lbs) | STANDARD MOTIVATION Frequency | STANDARD MOTIVATION Cumulative Percent | INDUCED MOTIVATION Frequency | INDUCED MOTIVATION Cumulative Percent |
|---|---|---|---|---|
| 281+ | 160 | 100.00 | 221 | 100.00 |
| 261-280 | 22 | 50.46 | 25 | 31.58 |
| 241-260 | 28 | 43.65 | 16 | 23.84 |
| 221-240 | 27 | 34.98 | 18 | 18.89 |
| 201-220 | 24 | 26.63 | 19 | 13.31 |
| 181-200 | 25 | 19.20 | 15 | 7.43 |
| 161-180 | 12 | 11.46 | 4 | 2.79 |
| 141-160 | 7 | 7.74 | 2 | 1.55 |
| 121-140 | 5 | 5.57 | 1 | .93 |
| 101-120 | 8 | 4.02 | 0 | .62 |
| 81-100 | 3 | 1.55 | 1 | .62 |
| 61-80 | 1 | .62 | 1 | .31 |
| 41-60 | 1 | .31 | 0 | - |
| 21-40 | 0 | - | 0 | - |
| | $\Sigma$=323 | | $\Sigma$=323 | |

50th percentile=279.1
5th percentile=133.1

50th percentile=N.A.
5th percentile=190.1

On the first trial the mean and 5th percentile force values were 153 lbs and 70 lbs, respectively, which compare quite well with the 164 lbs and 86 lbs obtained at Harvard. On the second trial, the Ann Arbor females produced mean and 5th percentile levels of 194 lbs and 102 lbs, as compared to Harvard's 194 lbs and 110 lbs. The lower values obtained in these tests in the first trial, as compared to the results obtained at Harvard, may be due to the less emphatic instructions. However, the correspondence between these two sets of results seems to be surprisingly good, particularly when it is recognized that different seat configurations were used.

The conclusion that "almost all" drivers can exert 100 lbs of force on a pedal for ten seconds is not supported by the measurements obtained in this study. Even the values of 80 and 90 lbs which Stoudt et al. claim could be reached by "all but the most aberrant or pathologically weak" seem high when compared with the 5th percentile female force of 85 lbs that is obtained by averaging the result produced by both instructional sets, or trials. In the Harvard study, the first percentile did not surpass 100 lbs until the fifth trial. A more appropriate conclusion from that study would be that 99 percent of the female driving population might be able to make a ten second pedal press of over 100 lbs after several attempts.

It is clear that these data do not say what a driver will be able to do in real braking emergencies. Presumably, he might be more motivated than was the case in these experiments. However, the motivation produced by a stressful driving situation is, as yet, unknown and unexplored.

# 3. DRIVER BRAKING PERFORMANCE AS A FUNCTION OF PEDAL-FORCE AND PEDAL-DISPLACEMENT LEVELS

## INTRODUCTION

An examination of the literature reveals that the research seeking to define the role of the human operator as a dynamic, vehicle brake controller is indeed sparse.

Feedback variables in the braking process are those which provide information to the driver directly, namely, the force and displacement applied to the brake pedal, or indirectly, namely, the visual, auditory, kinesthetic, vestibular, or proprioceptive sensations produced by the response of the vehicle to the braking input. Some earlier studies have attempted to determine the manner in which these direct feedback processes influence the ability of a driver to achieve minimum braking distances (Kontz et al., 1969; Ayoub and Trombley, 1967; Aoki, 1960; Barnes et al., 1942; Trumbo and Schneider, 1963; Hindle, 1964; Dupuis, 1957). These laboratory studies were necessarily carried out in an open-loop manner (i.e., without vehicle motion cues) and were concerned with evaluating the influence of pedal geometry together with the feedback that comes from operation of the pedal itself. Spurr (1965) reported an on-the-road study in which he was able to demonstrate that passengers were able to estimate deceleration levels quite well without visual feedback being provided, indicating that the vestibular, kinesthetic and proprioceptive stimuli derived from deceleration provide the major cues in the braking process. Alexander (1967) performed a study primarily to investigate the influence of both brake-torque and weight distribution on braking performance. Of particular interest to this study was his finding that human operators could, on the average, attain a maximum braking performance that was only 60 to 85 percent of the braking efficiency* built into the vehicle.

With respect to this investigation, the most pertinent study has been that conducted by Brigham (1968). Using a vehicle in which it was possible to vary the deceleration/pedal-force gain** and the pedal-displacement level, he found that a relatively high gain (g's/lb) plus a low value of pedal compliance produced the best braking performance and the highest driver ratings. In Brigham's study, the slopes of the linear portion of the pedal force versus deceleration curves were 48, 72, 76, and 100 pounds per g (the forces required to produce 1.0 g deceleration were 55, 80, 90, and 130 pounds, respectively). Test data show that the highest deceleration/pedal force gain (.0208 g/lb) used in Brigham's study is well below the maximum gains designed into U.S. automobiles with power-assist brake systems. Notwithstanding this restricted variation in deceleration/pedal force gain, Brigham's study should be considered a pioneering effort. Unfortunately, it was not funded to permit tests which included friction coefficient as an experimental variable. (This work has yet to be reported in the open literature and was discovered after the NHSB study was initiated.)

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*Braking efficiency is defined as the deceleration in g units that can be achieved, prior to wheel locking, ratioed to the coefficient of friction existing at the tire-road interface.

**Gain is defined either as a ratio of an input to output variable or as a ratio of an output to input variable. Thus, we may have a pedal force/deceleration gain in pounds per g units of deceleration or a deceleration/pedal force gain in g's per pound.

The existing U.S. Motor Vehicle Safety Standard No. 105 (1968) for braking system effectiveness for passenger vehicles is derived from a performance requirement developed by the Society of Automotive Engineers (SAE J937, 1969; SAE J843a, 1966). Briefly stated, it is required that the pedal force, under non-degraded conditions of the brake system, be not less than 15 nor more than 100 pounds from 30 mph and 120 pounds from 60 mph, for a deceleration of 20 feet per second per second.

The question remains as to whether the deceleration/pedal force gains associated with the current U.S. standard represent a match with driver modulation skills and force capabilities or that the standard is in need of revision and, if such is the case, what should be the nature of this revision. The purpose of this experiment was to investigate this problem.

METHOD

SUBJECTS. Originally fifteen men and fifteen women were to be test subjects to fill the cells of a 3 x 5 matrix (3 weight and 5 age categories) for each sex. The three weight categories (lower, middle, upper third) were derived from data obtained by Stoudt et al. (1965). The five age categories (18-24, 25-34, 35-44, 45-54, and 55-60 years) were limited by a maximum age of 60 years. This was done as a safety precaution because of the demands placed on the drivers by the experiment. Because of adverse weather conditions it was impossible to complete the study with all desired subjects. Sixteen men and twelve women were used. Their ages and weights are shown in Table 3.1.

THE TEST VEHICLE.

General Description. The test vehicle was a 1969 Chevrolet Townsman station wagon which was extensively modified to achieve the characteristics required in the driver-vehicle test. A

TABLE 3.1. CHARACTERISTICS OF THE TEST SUBJECTS (DRIVERS)

| | Age | | | | |
|---|---|---|---|---|---|
| Weight | 18-24 | 25-34 | 35-44 | 45-54 | 55-60 |
| | | | Male | | |
| Lower Third | 21/136 | 25/148 | 41/151 | 51/154 | 59/148 |
| Middle Third | 22/162 | 26/168 | 37/165 | 46/162<br>51/170 | 60/175 |
| Upper Third | 23/190 | 25/250 | 35/188 | 47/231 | 56/188 |
| | | | Female | | |
| Lower Third | 24/112 | 30/118 | 37/119 | 52/110<br>45/115 | 55/130 |
| Middle Third | 21/133 | | 40/130 | | |
| Upper Third | 21/163 | | 43/193 | 47/175 | 57/165 |

The first number in each cell is the age of the subject; the second number is the weight of the subject.

special electrohydraulic brake control system was installed (Figure 3.1) which provided a simple and rapid method of selecting braking characteristics from a fixed set of deceleration/pedal force gains and pedal displacements. A two-fluid system was used to insure compatibility with the seal materials used in the hydraulic components.

In order to minimize problems of brake fade during the test and to obtain a linear relation between brake line pressure and deceleration, disc brakes were used on all four wheels. For the front wheels the vehicle was equipped with standard factory installed disc brakes. After delivery the rear wheel drum brakes were removed and the axle and axle tube modified for the instal-

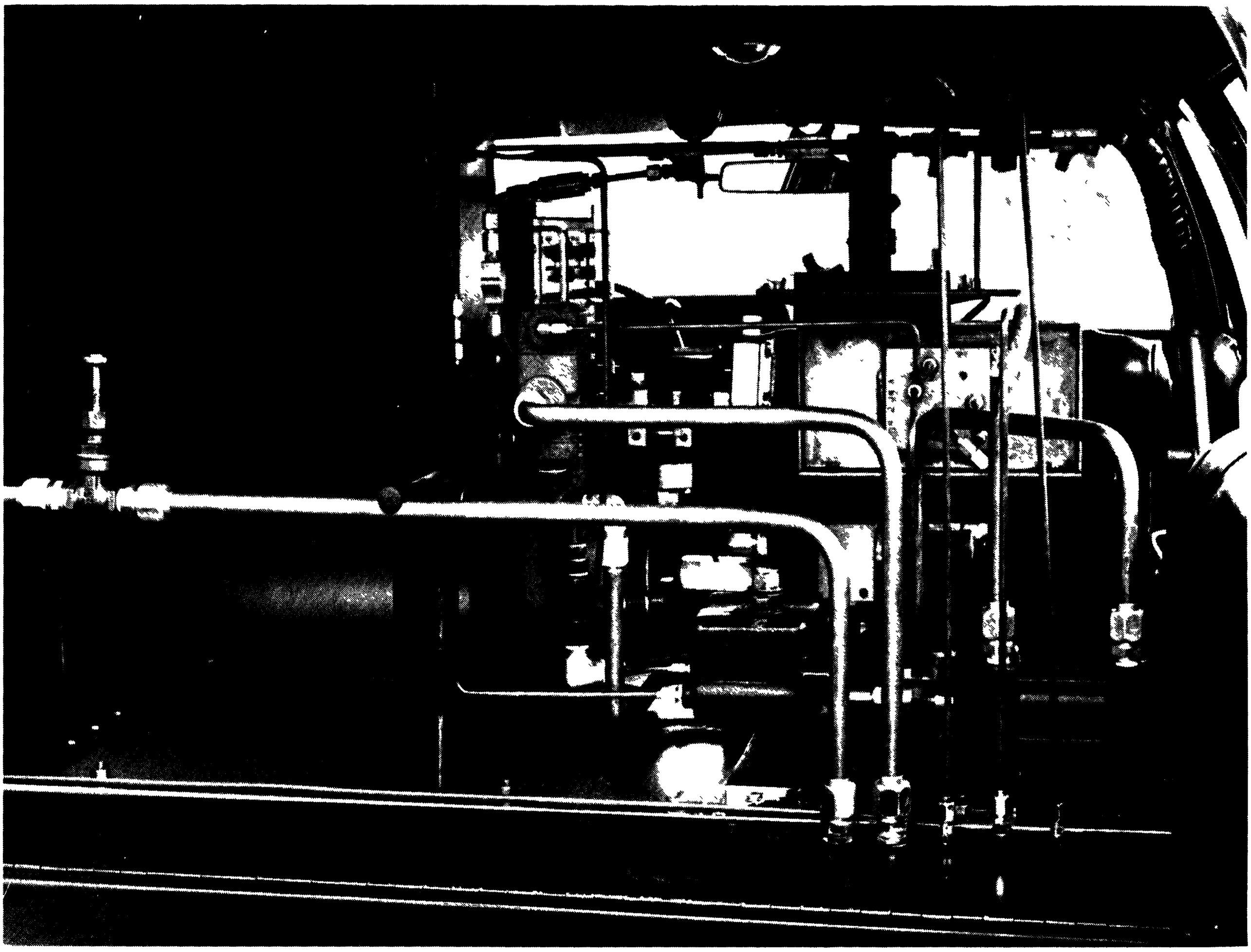


Figure 3.1. The hydraulic brake control system.

lation of calipers and brake discs. Two separate calipers were installed at each rear wheel, one operated by the electrohydraulic brake control system and the other operated by a separate brake pedal and conventional hydraulic system to provide emergency braking. The calipers and discs used were identical to those on the front wheels to insure similar braking characteristics front and rear. Standard factory equipment friction material was used at all wheels. The SAE (J843a) prescribed burnishing procedure was followed each time new friction pads were installed. This amounted to several stops from 40 mph and 60 mph at defined g levels with intervals between to allow the brakes to cool. Thermocouples were installed in one brake pad in each wheel, with a read-out in the car. The pad temperature during burnishing was not allowed to exceed 300° F.

Permanent magnet DC tachometer generators were located at each wheel and driven directly by the wheel to indicate wheel lockup.

Heavy duty shock absorbers were installed on the front of the vehicle and air-adjusted, car-leveling shock absorbers were installed on the rear to compensate for the additional load of the hydraulic equipment and reduce rear-end drag.

A roll bar and competition type seat belts and shoulder straps for the driver and experimenter were installed to protect the occupants in the event of roll-over during violent maneuvers.

The vehicle was equipped with 8.55 x 15, polyester cord, 4-ply (load range-D) tires. Inner tubes were used to prevent air loss during hard turns and stops. The tires were replaced when tread wear reached 50 percent.

The curb weight of the vehicle during the test was 5945 lbs. This was distributed 2563 lbs on the front and 3382 lbs on the rear wheels.

The Brake System. The brake system required quick and simple selection of six levels of deceleration/pedal force gain and two levels of pedal displacement. The latter were a zero displacement pedal and a nonlinear displacement characteristic with a displacement of 2.5 inches producing 1000 psi in the brake line.

Figure 3.2 is a diagram of the brake control system. Brake pedal force/displacement was controlled by six, quick-change, nonlinear spring canisters through a hydraulic line and master cylinders 1 and 2. Cylinder 2 and the spring canisters were located in the rear of the car near the experimenter and only a few seconds were required to change canisters. Zero[1] pedal displacement was obtained by mechanically locking the push rod of master cylinder 1 at a point after the pedal force load cell.

Deceleration/pedal force gain control was obtained by controlling brake line pressure through a closed-loop electrohydraulic servo. The difference, or error, between the brake line pressure transducer output and the pressure command signal from the pedal force load cell was amplified by the servo amplifier and applied to the servo valve which controlled the activating cylinder and master cylinder 3, as required, to minimize the error. By adjusting the electronic amplification of the pedal force load cell output with the pedal force-gain potentiometer the ratio of brake line pressure/pedal force was variable from 0 psi/lb to 80 psi/lb. Except for the hydraulic pump the hydraulic components were mounted on an aluminum plate on the deck behind the second seat. This assembly also included lines and valves, not shown in Figure 3.2 for switching from servo controlled brakes to normal brakes.

The pump, mounted in the engine compartment, was driven through a magnetic clutch and pulley by the engine. During

[1]There were about 1/16 inches of pedal travel.

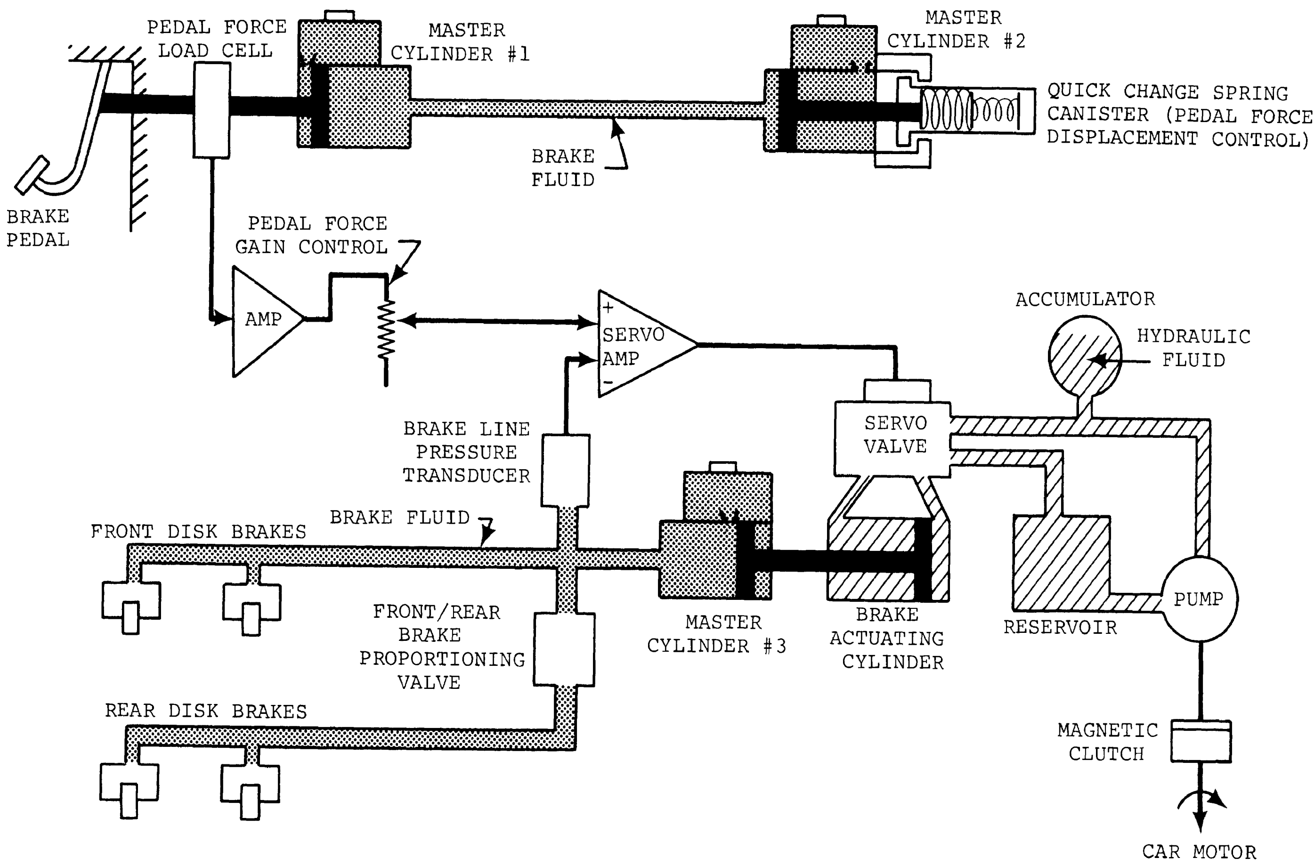



Figure 3.2. Brake control system.

test runs the pump was disengaged to unload the engine and to eliminate excessive pump noise transmitted through the hydraulic lines to the inside of the car. Peak hydraulic supply pressure was 1500 psi. During test runs the supply pressure was maintained about 1000 psi by the accumulated charge. Brake line pressure regulation was about one percent for a hydraulic supply pressure variation from 1000 psi to 1500 psi.

Brake Proportioning. The test vehicle, as obtained from the manufacturer, was equipped with front wheel disc brakes and rear wheel drum brakes. In order to minimize brake fade, and to provide a more nearly linear relationship between pedal force and deceleration, disc brakes were installed on the rear wheels which were identical to those on the front wheels. This provided equal brake force capability front and rear, which is generally not desirable in a passenger car. The braking efficiency diagram for the test vehicle is given in Figure 3.3.

Braking efficiency is a quantitative measure of how well the vehicle utilizes the friction forces available at the tire-road interface. On the horizontal axis is plotted the friction coefficient. The vertical axis shows braking efficiency defined as: the deceleration capability of the vehicle on a given surface without wheel lockup divided by the friction coefficient of that surface. Above the horizontal line in the figure, front wheel lockup occurs first, while below the horizontal line, rear wheels lock first.

As received from the manufacturer the vehicle had disc brakes front and drum brakes rear, with a front to rear brake force distribution of 60:40, yielding a brake efficiency characteristic, indicated in the figure, which is typical for passenger cars. On low coefficient surfaces, ice and wet pavement, the front wheels lock first. On higher coefficient surfaces, the rear wheels lock first. Since rear wheel lockup (on low coefficient surfaces especially) generally renders the vehicle

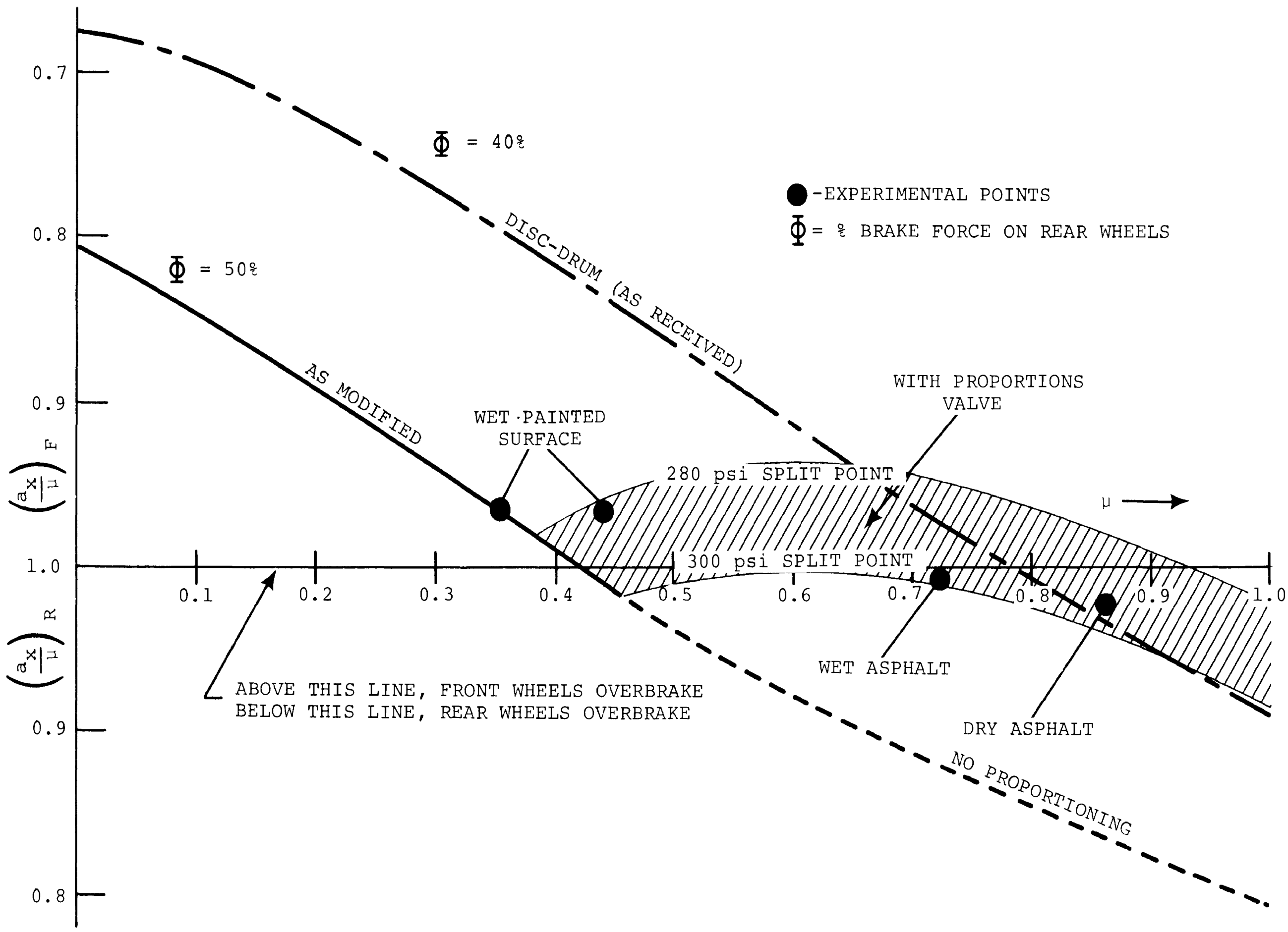



Figure 3.3. Braking efficiency of the test vehicle.

directionally unstable, vehicles are designed such that the 100 percent efficiency point (where all wheels lockup simultaneously) occurs at about 0.75 friction coefficient for normal vehicle loading. However, when disc brakes, identical to those on the front wheels, were installed on the rear wheels of the test vehicle, giving a front to rear brake force distribution of 50:50, the braking efficiency line was moved downward. Such proportioning would cause dangerous rear wheel lockup to occur on low coefficient surfaces. A vehicle with its brake system so proportioned would not be satisfactory for the intensive testing of subjects required by this program. Therefore, a Kelsey-Hayes model D74801 proportioning valve, with a "split-point" at about 300 psi, was installed in the hydraulic fluid line to the rear brakes. Up to about 300 psi (280-320 psi), flow of brake fluid to the rear brakes is not impeded, giving equal pressure front and rear. However, at the "split-point" and above flow to the rear brakes is restricted, causing the pressure to be increased in the rear brakes by only 2 psi for every 5 psi increase in front brake line pressure. As shown in the figure, the braking efficiency lines for the system with the proportioning valve indicate that rear wheel lockup on low coefficient surfaces is prevented.

Several test runs were made to verify the analysis, and four experimental points are indicated in the figure. For these runs the test vehicle was equipped with a decelerometer, and the wheel lockup indicator was used. On the wet painted surface, front wheels locked first, but on the wet and dry asphalt the rears locked first. On all three surfaces, braking efficiencies in excess of 95 percent were achieved.

Brake System Parameters. Dynamic measurements of deceleration vs. brake line pressure were made on the dry blacktop area of the test track. The curve showed a linear relationship with a slope of $0.883 \times 10^{-3}$ g/psi.

Static measurements were made of brake line pressure vs.

pedal force and pedal displacement for the six pedal force gains and the six corresponding spring canisters. The six gain values used are shown in Table 3.2 in terms of deceleration/pedal force in g/lb and the inverse, lb/g.

TABLE 3.2. PEDAL FORCE GAINS

| Level | lb/g | g/lb |
|---|---|---|
| 1 | 15.5 | 0.065 |
| 2 | 27.2 | 0.037 |
| 3 | 47.4 | 0.021 |
| 4 | 83.0 | 0.012 |
| 5 | 146.0 | 0.007 |
| 6 | 254.0 | 0.004 |

Figure 3.4 shows the pedal force/pedal displacement for each of the six spring canisters corresponding to the six deceleration/pedal force gains. The wide range of force-displacement characteristics is readily seen from these curves. Ideally the six spring canisters should provide a constant pedal displacement/deceleration gain. The range of pedal displacement/deceleration gain (Figure 3.5) that was obtained was considered reasonably constant. The means by which this was accomplished is described in Appendix I.

Speed Control System. On the approach to the test track the experimenter pushed the "counter clear" switch, thereby clearing all counters and clocks and starting the Brush recorder paper drive. Speed control lock-in was indicated by a green light in front of the driver, so that he could release the accelerator. Upon application of the brake the speed control was released and all counters and timers were enabled. When the vehicle velo-

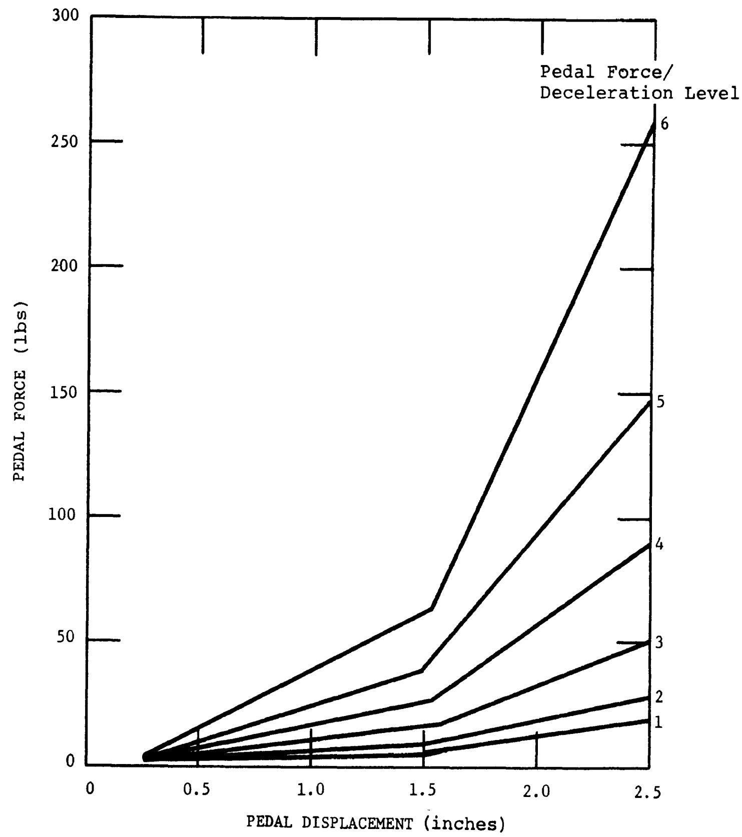



Figure 3.4. Pedal force and displacement for each deceleration/pedal force level.

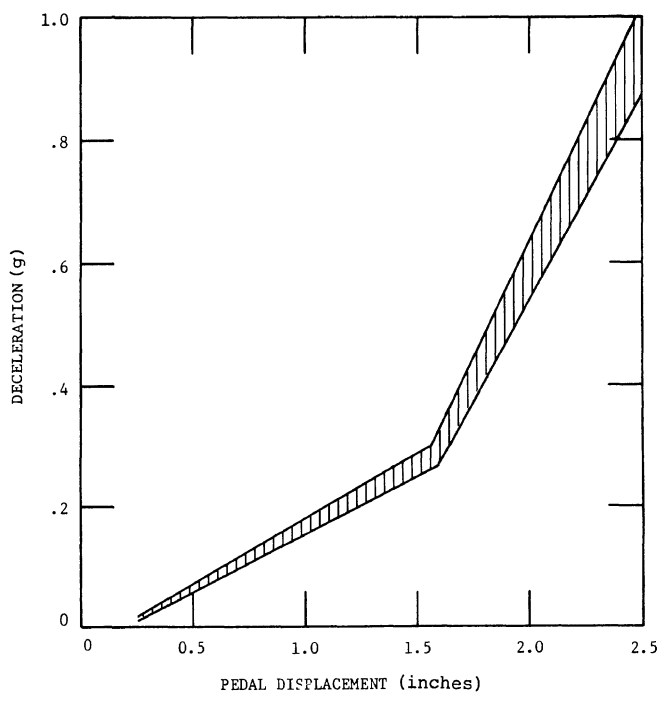



Figure 3.5. Deceleration/pedal displacement range.

city dropped below 1 mph all counters and timers stopped, holding their readings until the experimenter again activated the "clear" switch.

Data Collection Instrumentation. Data collection instrumentation was installed (Figure 3.6) to provide a readout or recording of the following:

1. Vehicle velocity
2. Vehicle deceleration
3. Braking distance
4. Braking time
5. Wheel lockup count (each wheel and total all wheels)
6. Wheel lockup time (total time one or more wheels locked)
7. Brake pedal force
8. Brake pedal displacement
9. Brake line pressure
10. Brake pad temperature

Brake pad temperature was monitored during the test to determine test repetition rates which would keep pad temperatures low and minimize brake fade.

A block diagram of the performance data collection instrumentation is shown in Figure 3.7. Wheel lock events for each wheel were totaled on four electromechanical counters, and the time one or more wheels were locked was totaled on an electronic digital timer in 0.01 seconds. Wheel locks were detected by four hysteresis, threshold detectors operating on the outputs from a DC tachometer generator located at each wheel. The criterion for wheel lockup was wheel velocity less than 0.5 mph while the actual vehicle velocity was greater than 1 mph. The latter was detected by the threshold detector on the output of the fifth wheel tachometer. Thus, wheel lock counts were not recorded when the vehicle actually stopped. Hysteresis of 2 mph was designed into the wheel lock threshold detectors to prevent rotational vibration of the sliding wheel from causing extra counts.

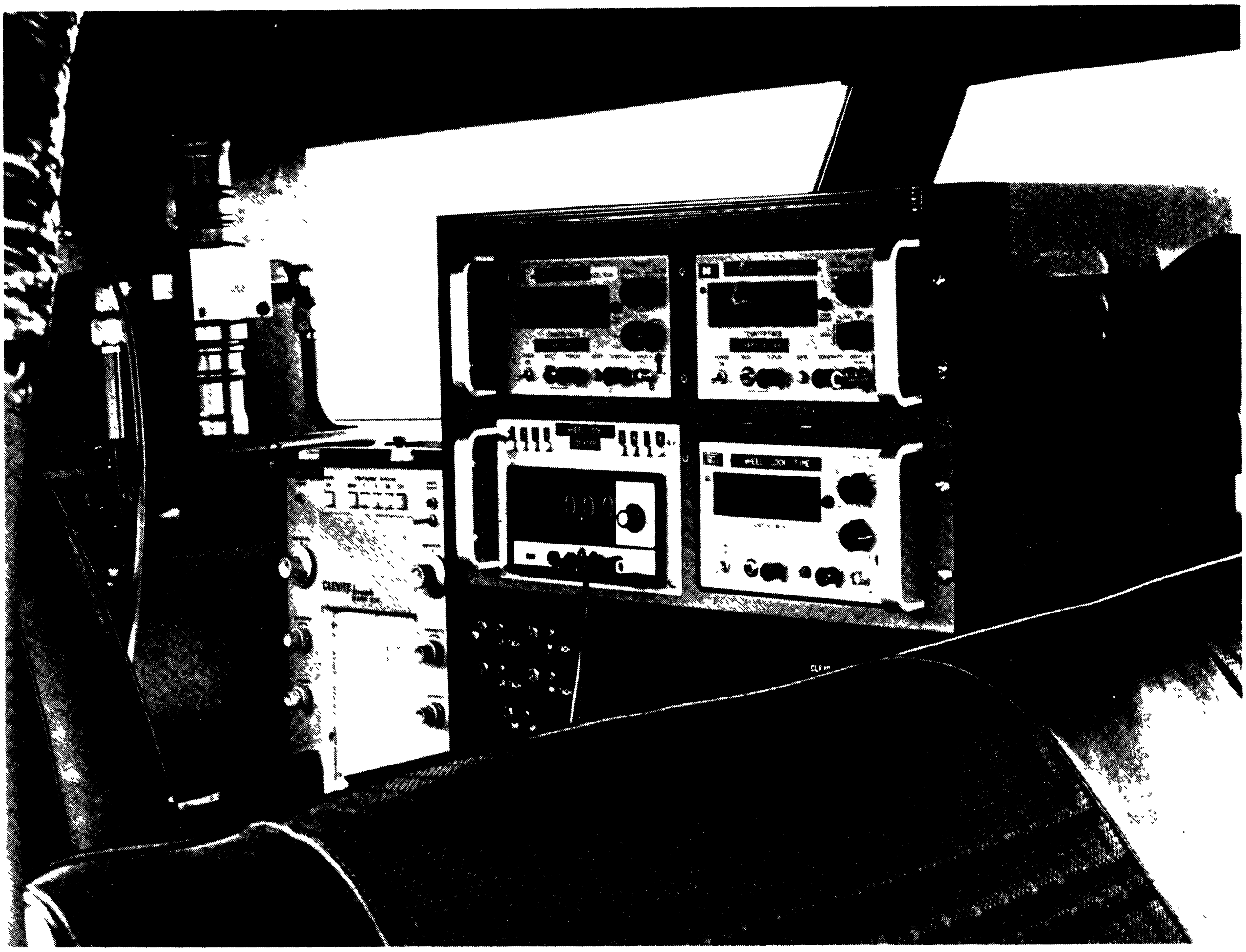


Figure 3.6. Performance recording displays in the test vehicle.

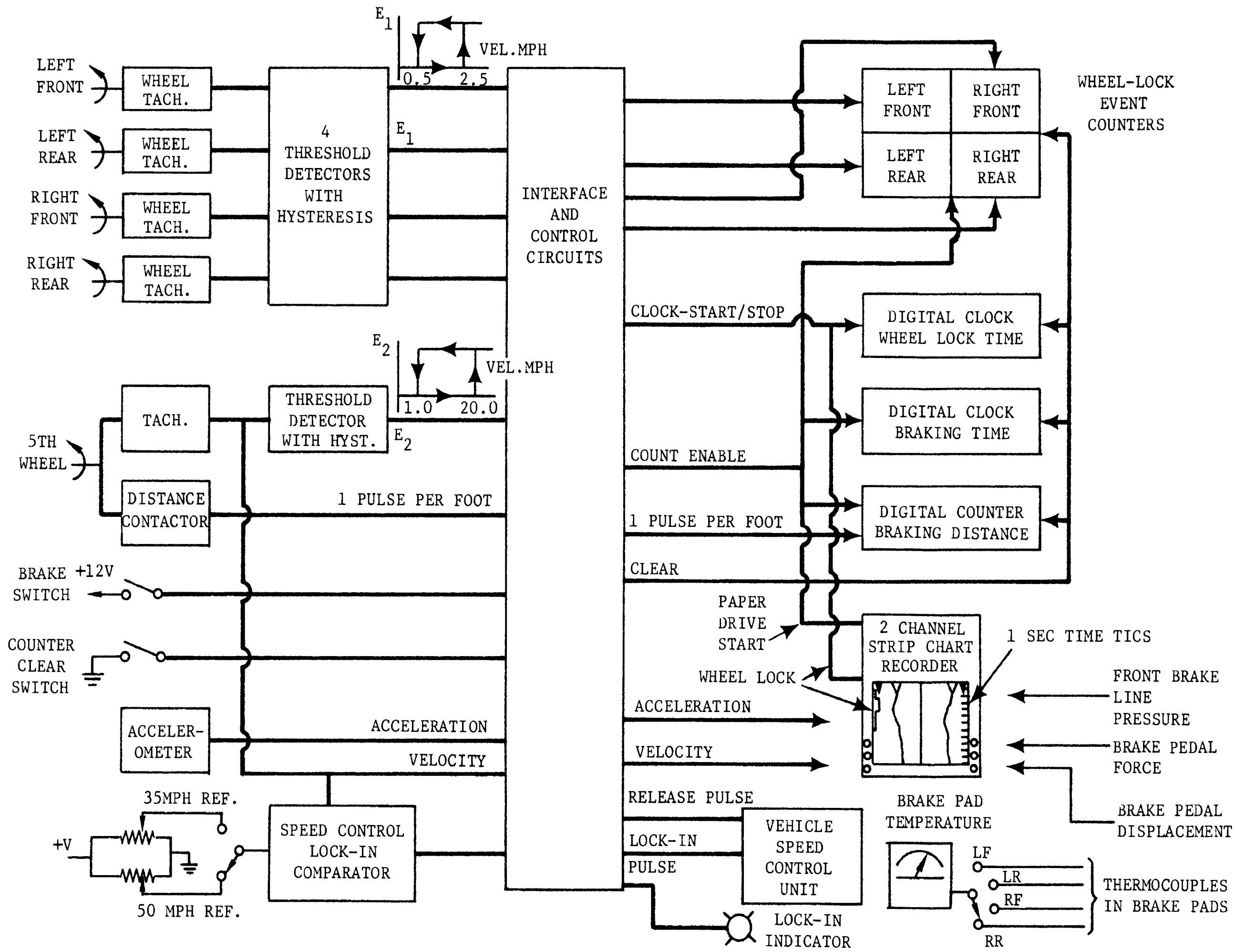



Figure 3.7. Performance data collection instrumentation block diagram.

Braking time was measured in 0.01 seconds on a digital timer gated-on by the initial brake application and gated-off by the fifth wheel threshold detector output when vehicle velocity dropped below 1.0 mph. The fifth wheel contactor output pulses (one pulse per foot) were counted on an electronic counter, which was enabled during the braking time, to obtain braking distance. At the end of each test run the experimenter recorded the read out of the counter and timers.

The instrumentation also included a Brush strip chart recorder with two event channels and two analog channels. One second time lapses were recorded on the right event channel and frequency of occurrence and duration of locked wheels were recorded on the left event channel. Any two of the following could be simultaneously recorded on the two analog channels for system calibration and/or data recording: velocity, from the fifth wheel tachometer; acceleration, from an accelerometer mounted on the fore/aft axis near the vehicle center of gravity; front brake line pressure and brake pedal force, from the brake control system pressure and force transducers; and brake pedal displacement, from a linear potentiometer connected to the brake pedal arm. During the braking test velocity and pedal force were recorded.

TEST SITE. A taxi ramp at the University of Michigan Willow Run Airport was used as a test site. An asphalt surface 100 feet by 700 feet was laid for the study, and the area was divided lengthwise into three test lanes 33 x 700 feet each.

Each lane provided a different road surface. One lane remained dry, one was watered to simulate a road on a rainy day, and the third was painted with yellow traffic paint and watered to simulate a slippery surface.

Measurements were made of the sliding coefficient of friction of these surfaces on twelve days of the test program by redording the deceleration of the test vehicle when all four wheels were locked. The data were highly variable and the average and spread of these measurements are given in Figure 3.8 as a function of sliding velocity. (These measurements were confirmed by tests made with the Highway Safety Research Institute's mobile tire tester and with a portable friction-measurement device.) It should be observed that the sliding friction level is velocity sensitive, particularly for the wet asphalt and the wet-painted asphalt. Consequently it is not truly meaningful to characterize these surfaces by a single numeric representing the friction couple produced at the tire-road interface. Further, the peak coefficients of friction as achieved by a rolling tire on these surfaces are also velocity dependent. During the braking efficiency tests, peak coefficients of 0.86, 0.71 and 0.40 were obtained on the dry, wet, and wet-painted surfaces, respectively (see Figure 3.3). These peak coefficients can be taken as generally representative of the friction level of the surfaces prepared for this program. It is clear that the wet-painted surface yields a significantly higher friction coefficient when a tire is partially slipping than when it is fully looked. Accordingly, we should anticipate that this surface would make the greatest demands on test subjects as they endeavor to minimize their stopping distance.

Traffic cones were used to delineate a 10-foot wide driving lane within each of the three test areas. Cones were placed at 15-foot intervals for 300 feet on the dry surface, 400 feet on the wet surface, and 700 feet on the wet-painted surface. Each driving lane was in the form of a shallow cosine wave (3-feet peak-to-peak amplitude and 400-feet wavelength) so that some steering was necessary.

Three lamps were placed at 30-foot intervals near the end of each test lane (Figure 3.9). These lamps were used as signals to initiate braking and as approximate stopping points. Onset of the lamp was triggered by a tapeswitch over which the

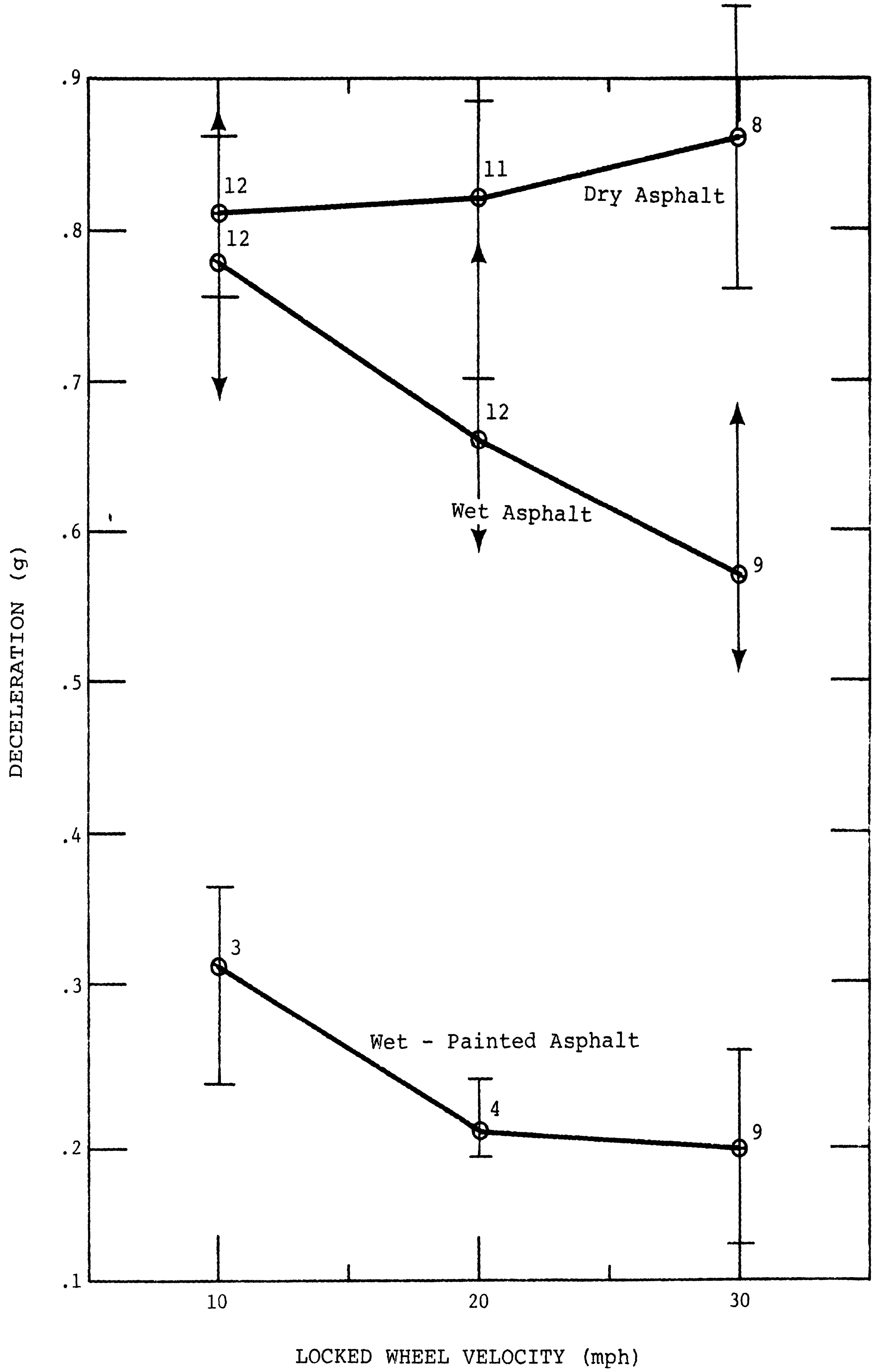



Figure 3.8. Deceleration as a function of locked wheel velocity and surface.

Figure 3.9. Test car in the track, showing lane marker cones and stimulus/goal lamps.

vehicle passed before entering the test lane. One experimenter determined which lamp came on in the test lane and operated a control box which controlled the delay between tapeswitch impulse and onset of the lamp. The delays, based on trial subjects, were timed so that subjects would stop beyond the lights approximately 75 percent of the time. This was done so that the goal of stopping before passing the light, which was to represent a truck or child in the vehicle's path, was challenging and occasionally feasible. (This was confirmed by the test.) The same experimenter also shut off the sprinklers when a run was being made in the wetted lanes.

INDEPENDENT VARIABLES. Five independent variables were studied: speed, deceleration/pedal force, pedal displacement, tire-road friction coefficient, and driver physical characteristics.

1. Two speeds, 35 and 50 mph, were used in the test. The determination of these speeds was based on the desire to have a moderately low velocity such as would occur in suburban driving, and a moderately high velocity such as would occur in rural driving. Initially 60 mph had been selected as the latter speed, but trial runs indicated that this speed was potentially dangerous on the lowest friction surface.

2. Six linear deceleration/pedal force gains were investigated, shown in Table 3.2.

3. The two pedal displacement levels were essentially 0 inches and 2.5 inches at 1000 psi.

4. Three road surfaces were used with sliding wheel coefficients of friction of about .82, .66, and .20, and rolling wheel coefficients of friction of about .86, .71, and .40.

5. The subjects were systematically selected by sex, age, and weight to represent a wide cross section of drivers.

DEPENDENT VARIABLES. All data output was displayed to the experimenter in the back seat (Figure 3.6). The performance measures were:

1. Stopping distance, measured to the nearest 1.0 foot.
2. Stopping time, in 0.01 seconds.
3. Total number of successive wheel lockups.
4. Total wheel lockup time, to the nearest 0.01 seconds.
5. Number of wheel lockups for each wheel.
6. Speed and pedal force time history.

PROCEDURE: PILOT STUDIES. During the development of the braking test a considerable effort was devoted to pilot testing. Initial tests, before the brake test car was available, were carried out using conventional vehicles.

One such test involved two Mercury Montego, 1968, two-door sedans having deceleration/pedal force functions shown in Figure 3.10. A sine wave course was laid out with traffic cones. A fifth wheel was used on each car to measure speed and braking distance. The surface was used dry and wet. Stops were made from 60 mph. The results are given in Tables 3.3-3.5 in terms of braking distance, mean deceleration and time to reduce speed by 10 mph, and show that the power brake provides better performance on the dry and the manual on the wet surface. From Table 3.4 and Figure 3.10 it would be inferred that pedal force levels should be not less than 30 lbs nor more than 80 lbs at about 20 $ft/sec^2$.

A large number of shake down tests were conducted with the brake test vehicle by which the procedure was refined and vehicle and test site problems identified and remedied. During this period about 500 test runs were made.

<u>Hydroplaning</u>. During the pilot tests the critical importance of tire tread depth in affecting directional stability of the car during braking in the wet was confirmed. In hard braking on low coefficients of friction it was almost impossible to

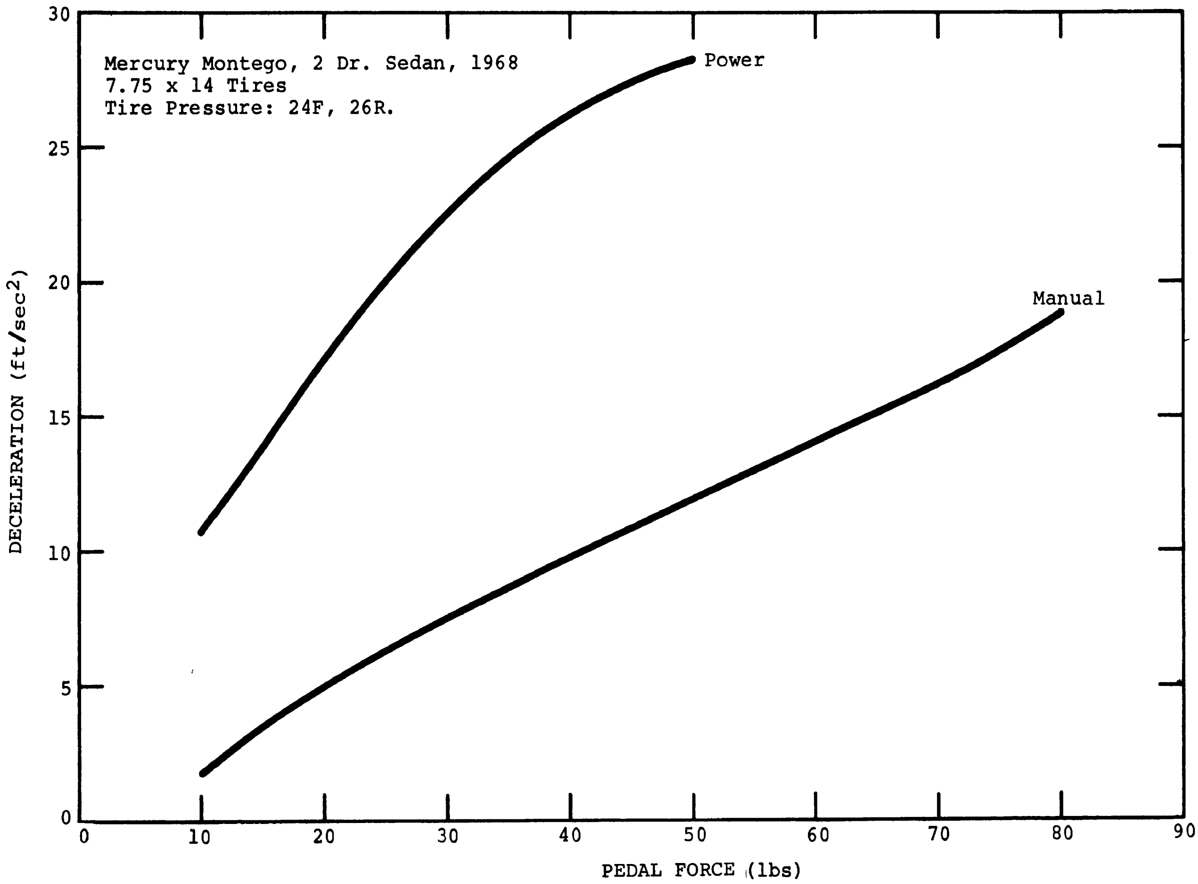



Figure 3.10. Deceleration/pedal force for pilot test cars.

TABLE 3.3. PILOT TEST: MEAN BRAKING DISTANCE (FEET) ON DRY AND WET FOR POWER AND MANUAL BRAKE

| | Power | Manual |
|---|---|---|
| Dry | 169.98 | 207.11 |
| Wet | 228.39 | 211.86 |
| Mean | 199.19 | 209.49 |

TABLE 3.4. PILOT TEST: MEAN DECELERATION (ft/sec$^2$) ON DRY AND WET FOR POWER AND MANUAL BRAKE

| | Power | Manual |
|---|---|---|
| Dry | 23.38 | 20.44 |
| Wet | 19.34 | 19.74 |
| Mean | 21.36 | 20.09 |

TABLE 3.5. PILOT TEST: MEAN TIME (SECONDS) TO DECREASE SPEED BY 10 MPH FROM START OF BRAKING ON DRY AND WET FOR POWER AND MANUAL BRAKE

| | Power | Manual |
|---|---|---|
| Dry | 1.03 | 1.31 |
| Wet | 1.50 | 1.41 |
| Mean | 1.27 | 1.36 |

keep the vehicle within the ten-foot wide test strip. It was noted that the tread depth for the front tires was about 40 percent and the rear tires about 30 percent of new tire depth. When new tires were placed on the front wheels of the vehicle there was a great improvement in control. Placing new tires on the rear wheels also improved control, but the increment was small. As a result, tread depth was checked daily, and tires were changed whenever tread depth became less than 50 percent of new tire depth (11/32 inches). This necessitated changing tires after approximately each 5 subjects during the brake test.

PROCEDURE: BRAKING TEST. Before each run tread depth and air pressure in each tire were measured (tire pressure was based on SAE minimum recommendations based on the weight on each wheel). A nitrogen accumulator, which was part of the braking system, was also checked for proper pressure. Anthropometric data were collected on each subject. This information included total weight, foot length, leg weight, leg height, and maximum foot force with the right and the left foot under "normal" and "induced" motivational conditions using the foot force measuring device shown in Figure 2.2.

The subject, experimenters, and test vehicle were then driven to the test site. The braking system was calibrated and instructions were given to the subject. The subject was told that the purpose of the study was to learn of his ability to bring the car to a safe stop in as short a distance as possible after initiating braking. A safe stop was one in which none of the traffic cones were knocked down. Instructions on the operation of the vehicle and the layout of the test lanes were given. The subject was then told to bring the car up to a speed until the speed-control device was actuated and then to keep his foot resting lightly on the accelerator until one of the three lamps near the end of the test lane was turned on. This was the signal to begin braking and also acted as a reference mark for the subject who was told to try to stop before reaching the lamp.

The subject was seated in the vehicle and attached the shoulder harness and seat belt. The experimenter rode in the back seat to record the data. After testing the brakes for familiarization, the driver was given a minimum of two practice runs on each of the three surfaces. Practice runs were used to familiarize the driver with the procedure, the automobile and the test lanes. Because the data gathering runs were made at 35 and 50 mph, subjects practiced until they were able to brake at these speeds in reasonable distances without knocking down traffic cones. Minor to extensive practice was necessary to perform the task at 50 mph, particularly on the wet-painted surface. When the experimenter in the vehicle felt that the subject was capable of performing the task successfully the instructions were summarized again. This time the very best, safe braking performance of the subject was emphasized.

If any cones were knocked down this was noted; the run was considered invalid, and was repeated. When necessary, additional instructions were given on how to control and brake the vehicle in a skid.

Performance And Subjective Data Recording. After each successful run, the data were recorded by the experimenter, and the subject was told his stopping distance in feet. After the completion of the six runs for a particular force gain the subject was asked two questions:

1. "Disregarding your stopping distances, how would you rate the braking system you have just used in terms of your ability to control the car during braking?" The response was made on a five-point rating scale which ranged from "very poor" to "very good."

2. "Was the force level you had to exert on the brake pedal to stop the car too low, somewhat low, just right, somewhat high, or too high?"

After these questions were answered the deceleration/pedal force gain was changed, the new braking system was tried by the subject, and the next runs were made.

Subjects usually had to drive for a total of four to six hours in a day. A lunch break was given approximately midway through the experiment, and short rest breaks were taken in the morning and afternoon.

EXPERIMENTAL DESIGN. Either the 0 or the 2.5 inch pedal displacement condition was selected to be used first. Then, within a displacement condition, the six deceleration/pedal force gain levels were randomly ordered. For a given deceleration/pedal force gain a run was made at 35 mph followed by one at 50 mph on the dry surface, then at 35 mph and 50 mph on the wet surface, and finally at 35 mph and 50 mph on the wet-painted surface. The procedure was repeated for the other displacement.

The design was a complete factorial with the deceleration/pedal force gain randomly ordered in the pedal displacement factor, and with speed and road surface systematically ordered in each deceleration/pedal force gain condition.

## RESULTS

A sample data sheet for one subject is shown in Appendix II. The results for each dependent variable are considered below.

BRAKING DISTANCE. Table 3.6 and Figure 3.11 show the mean distance to stop as a function of speed, deceleration/pedal force gain and surface. Overall means due to deceleration/pedal force gain and surface are also shown. The speed and surfaces had an obvious effect on braking distance. Within each surface-speed combination there are noticeable differences of up to about 20 percent braking distance between levels of the deceleration/pedal force factor. The effect of pedal displacement is shown in Figure 3.12. Mean braking distances for both displacement levels on each surface are very similar.

TABLE 3.6. MEAN BRAKING DISTANCE (FEET) AS A FUNCTION OF DECELERATION/PEDAL FORCE GAIN, SURFACE AND SPEED

| Road Surface | Speed | DECELERATION/PEDAL FORCE GAIN (g/lb) .065 | .037 | .021 | .012 | .007 | .004 | Mean |
|---|---|---|---|---|---|---|---|---|
| Dry | 35 | 78.9 | 78.9 | 76.6 | 80.0 | 87.0 | 93.7 | 116.1 |
| | 50 | 160.7 | 156.6 | 155.8 | 157.5 | 169.7 | 188.2 | |
| Wet | 35 | 98.2 | 92.2 | 93.8 | 97.3 | 102.9 | 117.6 | 105.9 |
| | 50 | 222.2 | 223.1 | 210.2 | 220.2 | 238.1 | 254.7 | |
| Wet Painted | 35 | 247.2 | 233.9 | 221.5 | 211.1 | 211.9 | 220.6 | 338.7 |
| | 50 | 554.6 | 521.3 | 511.9 | 492.2 | 503.7 | 492.2 | |
| | Mean | 183.2 | 176.7 | 173.0 | 174.6 | 183.9 | 196.6 | |

DECELERATION. Each braking distance was converted to an equivalent average deceleration computed from:

$$\frac{a_x}{g} = \frac{1}{2\,g}\,\frac{v^2}{s}$$

where

$a_{x/g}$ = mean deceleration in g units

v = initial velocity in ft/sec

s = braking distance to stop in feet

$g = 32.2$ ft/sec$^2$

The mean deceleration values were transformed to $\log_e\left(\frac{a_x}{g} + 1\right)$ to normalize the data and were treated by an analysis of variance, shown in Table 3.7. The main effects of speed, deceleration/pedal force gain and surface friction were statistically significant. The effect of pedal displacement and its interactions with the other variables were not significant at the 0.01 level. The mean deceleration values for the significant

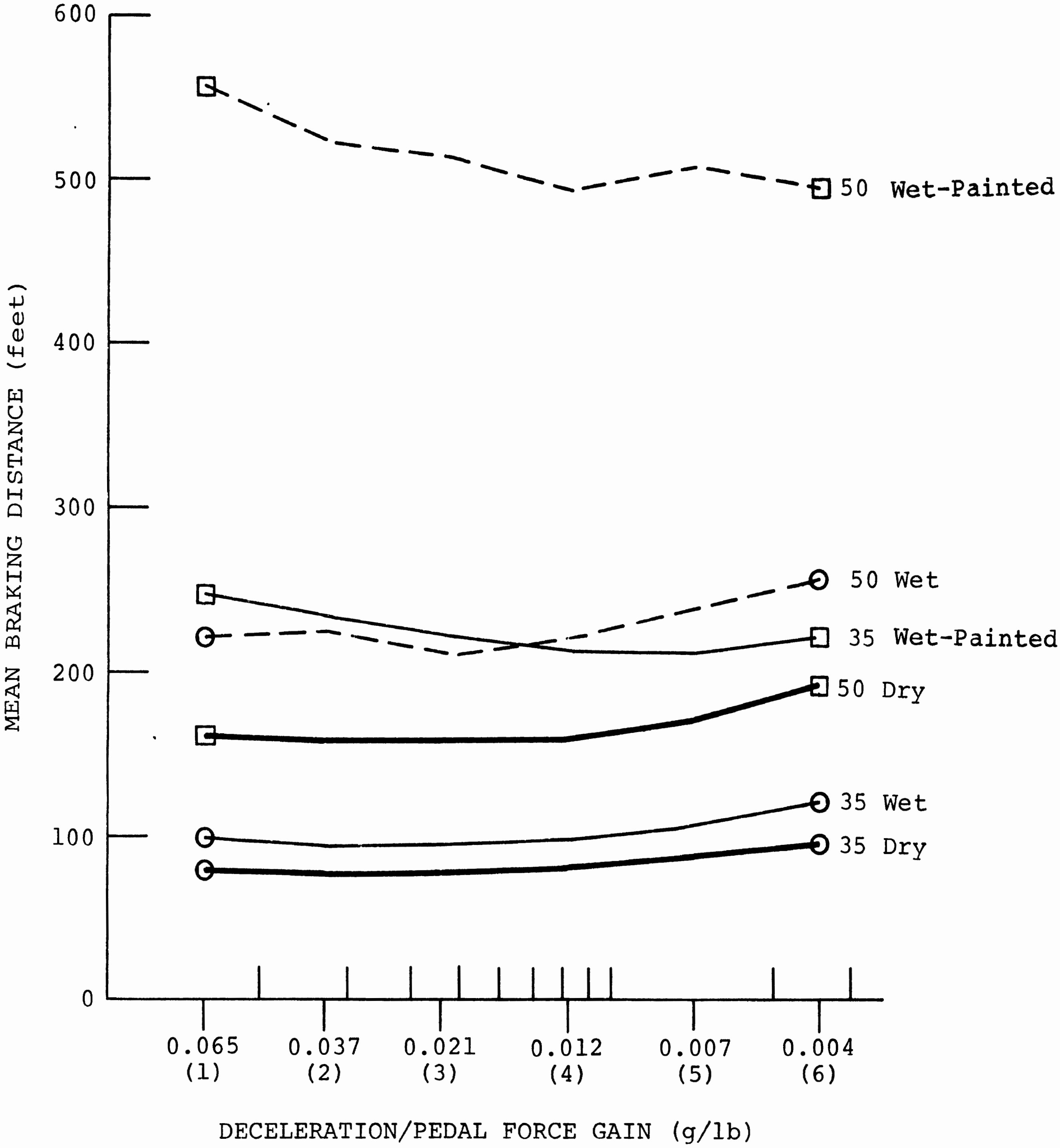



Figure 3.11. Mean braking distance as a function of deceleration/ pedal force gain, speed and surface.

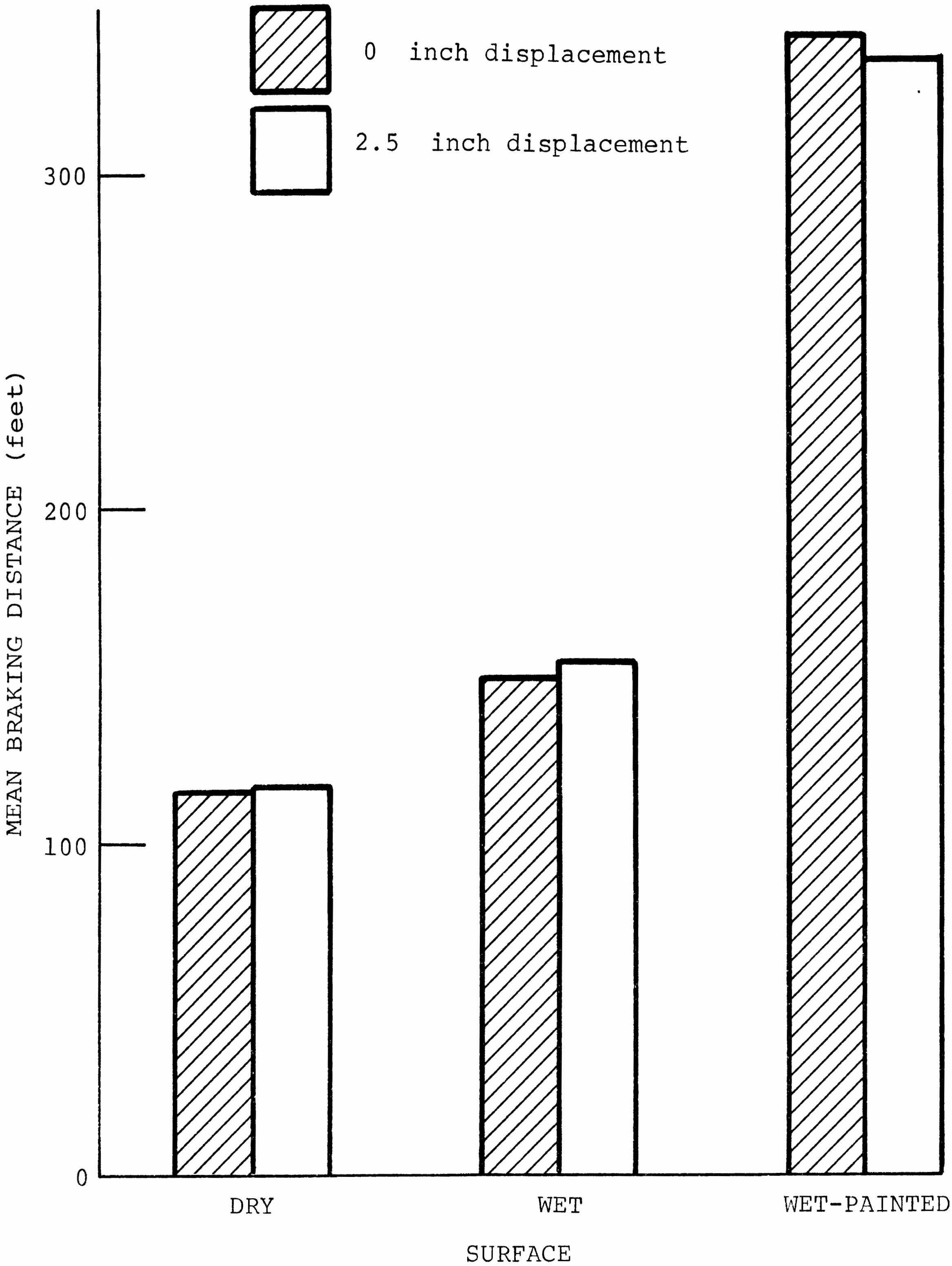



Figure 3.12. Mean braking distance as a function of surface and pedal displacement.

TABLE 3.7. ANALYSIS OF VARIANCE OF DECELERATION $\left(\frac{a_x}{g}\right)$ PERFORMANCE

| Source of Variation | df | MS | F |
|---|---|---|---|
| Speed (S) | 1 | 0.1103 | 23.5* |
| Decel./Pedal Force Gain (F) | 5 | 0.0674 | 31.2* |
| S x F | 5 | 0.0012 | -- |
| Pedal Displacement (D) | 1 | 0.0095 | 1.7 |
| S x D | 1 | 0.0001 | -- |
| F x D | 5 | 0.0007 | -- |
| S x F x D | 5 | 0.0003 | -- |
| Surface (μ) | 2 | 11.0576 | 744.1* |
| S x μ | 2 | 0.0534 | 30.5* |
| F x μ | 10 | 0.0263 | 27.0* |
| S x F x μ | 10 | 0.0019 | 3.0* |
| D x μ | 2 | 0.0061 | 3.2 |
| S x D x μ | 2 | 0.0005 | -- |
| F x D x μ | 10 | 0.0009 | -- |
| S x F x D x μ | 10 | 0.0004 | -- |
| Subjects (E) | 27 | 0.0553 | -- |
| S x E | 27 | 0.0047 | -- |
| F x E | 135 | 0.0022 | -- |
| S x F x E | 135 | 0.0007 | -- |
| D x E | 27 | 0.0056 | -- |
| S x D x E | 27 | 0.0007 | -- |
| F x D x E | 135 | 0.0016 | -- |
| S x F x D x E | 135 | 0.0006 | -- |
| μ x E | 54 | 0.0149 | -- |
| S x μ x E | 54 | 0.0018 | -- |
| F x μ x E | 270 | 0.0010 | -- |
| S x F x μ x E | 270 | 0.0006 | -- |
| D x μ x E | 54 | 0.0019 | -- |
| S x D x μ x E | 54 | 0.0008 | -- |
| F x D x μ x E | 270 | 0.0009 | -- |
| S x F x D x μ x E | 270 | 0.0007 | -- |
| TOTAL | 2015 | | |

*Significant at $P \leq 0.01$

three factor interaction between speed, deceleration/pedal force gain and surface is shown in Table 3.8 and Figure 3.13.

TABLE 3.8. GEOMETRIC MEAN DECELERATION, IN g, FOR THE INTERACTION OF SPEED, DECELERATION/PEDAL FORCE GAIN AND SURFACE

| Road Surface | Speed | DECELERATION/PEDAL FORCE GAIN (g/lb) .065 | .037 | .021 | .012 | .007 | .004 | Mean |
|---|---|---|---|---|---|---|---|---|
| Dry | 35 | .526 | .538 | .540 | .515 | .478 | .444 | .51 |
| | 50 | .525 | .538 | .540 | .534 | .496 | .449 | |
| Wet | 35 | .420 | .447 | .440 | .424 | .402 | .353 | .39 |
| | 50 | .379 | .378 | .400 | .384 | .353 | .332 | |
| Wet Painted | 35 | .168 | .175 | .186 | .197 | .194 | .186 | .17 |
| | 50 | .151 | .161 | .164 | .171 | .166 | .171 | |
| | Mean | .35 | .36 | .37 | .36 | .34 | .32 | |

The figure shows that on the dry surface somewhat greater mean decelerations were obtained at 50 mph than 35 mph, whereas the reverse was true on the wet and wet-painted surface. The trends across the deceleration/pedal force gains are concave showing that there are bandwidths of this variable which provided good performance which was degraded at higher or lower gains.

Table 3.9 shows the results of a Newman Keuls test which compares the mean decelerations achieved with the deceleration/pedal force gains in each surface and speed condition. For example, on the dry at 35 mph, subjects performed significantly poorer with gain 6 (.004 g/lb) than others. Gain 5 was next poorest, followed by 4 and 1. Levels 2 and 3 were superior to the others. At 50 mph the findings were similar.

Table 3.10 shows the levels of deceleration/pedal force gain (PFG) rank ordered according to significant differences

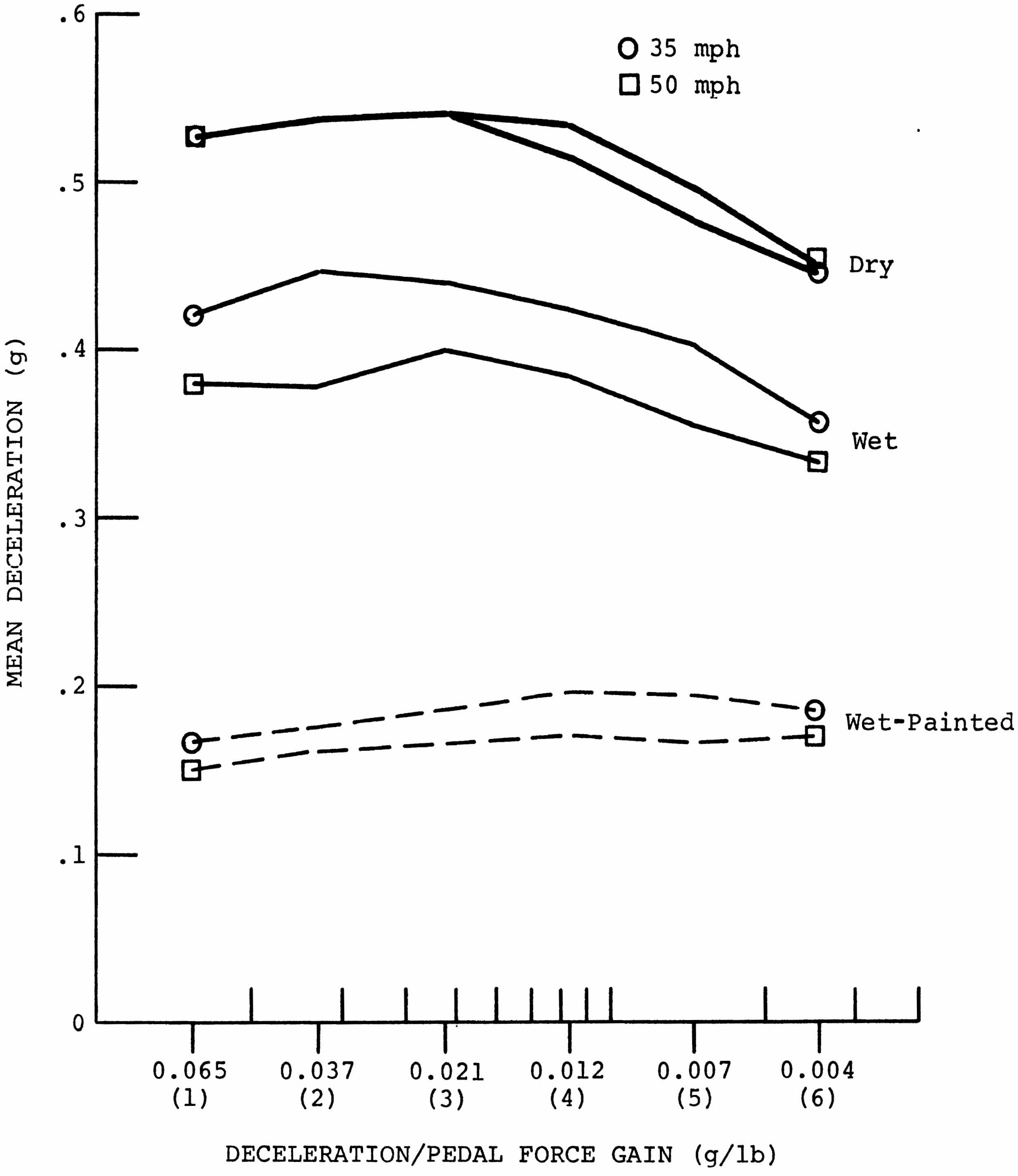



Figure 3.13. Geometric mean deceleration as a function of deceleration/pedal force gain, speed and surface.

TABLE 3.9. NEWMAN-KEULS TEST OF MEAN DECELERATION FOR DECELERATION/PEDAL FORCE GAINS AT EACH SURFACE AND SPEED

| Surface | Speed | Levels | Have Significantly* Higher Mean Deceleration Than | PF Gain Levels |
|---|---|---|---|---|
| Dry | 35 | 1, 2, 3, 4, 5 | | 6 |
| | | 1, 2, 3, 4 | | 5 |
| | | 1, 2, 3 | | 4 |
| | | 2, 3 | | 1 |
| Dry | 50 | 1, 2, 3, 4, 5 | | 6 |
| | | 1, 2, 3, 4 | | 5 |
| | | 2, 3, 4 | | 1 |
| Wet | 35 | 1, 2, 3, 4, 5 | | 6 |
| | | 1, 2, 3, 4 | | 5 |
| | | 2, 3 | | 1, 4 |
| Wet | 50 | 1, 2, 3, 4, 5 | | 6 |
| | | 1, 2, 3, 4 | | 5 |
| | | 3 | | 1, 2, 4 |
| Wet-Painted | 35 | 2, 3, 4, 5, 6 | | 1 |
| | | 3, 4, 5, 6 | | 2 |
| | | 4, 5 | | 3, 6 |
| Wet-Painted | 50 | 2, 3, 4, 5, 6 | | 1 |
| | | 4, 6 | | 2 |

*$P \leq .01$

in mean deceleration. Those levels that are in brackets are ones with which subjects achieved significantly greater deceleration in a surface-speed condition compared to non-bracketed gain levels. For example, on the dry at 35 mph, PFG levels 2 and 3 produced significantly better performance than other gains, and are ranked equally and bracketed. Level 1 was significantly superior to 4, 5 and 6 and hence, has a rank of 3; level 4 was significantly superior to 5 and 6 and has a rank of 4; level 5 was significantly superior to 6 and is ranked 5; and level 6 is ranked 6--the poorest configuration for that surface-speed combination.

The sum of the ranks across the surface-speed conditions is shown in Table 3.10. A low rank denotes good performance. Thus,

TABLE 3.10. RANK ORDER OF DECELERATION/PEDAL FORCE GAINS DIFFERING SIGNIFICANTLY IN DRIVER VEHICLE BRAKING DECELERATION

| Level | PFG (g/lb) | DRY | | WET | | WET-PAINTED | | RANK SUM |
|---|---|---|---|---|---|---|---|---|
| | | 35 MPH | 50 MPH | 35 MPH | 50 MPH | 35 MPH | 50 MPH | |
| (1) | 0.065 | 3 | 4 | 3.5 | 3 | 6 | 6 | 25.5 |
| (2) | 0.037 | [1.5 | [2 | [1.5 | 3 | 5 | 5 | 17.5 |
| (3) | 0.021 | 1.5] | 2 | 1.5] | [1] | 3.5 | 3.5 | 14.0 |
| (4) | 0.012 | 4 | 2] | 3.5 | 3 | [1.5 | [1.5] | 15.5 |
| (5) | 0.007 | 5 | 5 | 5 | 5 | 1.5] | 3.5 | 24.5 |
| (6) | 0.004 | 6 | 6 | 6 | 6 | 3.5 | [1.5] | 29.0 |
| | | 21 | 21 | 21 | 21 | 21 | 21 | 126.0 |

[Those values blocked off indicate the pedal force gains providing significantly greatest mean deceleration under each surface-speed condition].

overall, PFG levels 3 and 4 were most conducive to effective modulation by drivers in their attempts to achieve a maximum deceleration.

BRAKING TIME. Braking time was measured from the onset of braking. The results were quite similar to those reported for braking distance and are, therefore, not shown.

FORTY PERCENT DECREASE IN SPEED. The velocity during braking, recorded on a chart recorder, was examined to derive the time required to reduce speed by 40 percent. The 40 percent decrease in speed was arbitrarily selected and results in velocities of 21 and 30 mph, respectively, from initial velocities of 35 and 50 mph. Table 3.11 shows the mean time to stop and

TABLE 3.11. MEAN TIME TO REDUCE SPEED BY 40 PERCENT FOR MAIN EFFECTS OF SPEED, PFG AND SURFACE

| | A | B | C |
|---|---|---|---|
| | Stopping Time | 40% Decrease in Speed | [C=B/A] |
| Speed (mph) | | | |
| 35 | 4.12 | 1.78 | 43.3 |
| 50 | 6.28 | 2.82 | 44.9 |
| PFG (lb/g) | | | |
| 15.5 | 5.19 | 2.32 | 44.7 |
| 27.2 | 4.99 | 2.21 | 44.3 |
| 47.4 | 4.93 | 2.17 | 44.0 |
| 83.0 | 4.94 | 2.18 | 44.2 |
| 146.0 | 5.11 | 2.27 | 44.5 |
| 254.0 | 5.49 | 2.44 | 44.4 |
| Surface | | | |
| Dry | 3.28 | 1.32 | 40.3 |
| Wet | 4.09 | 1.85 | 45.2 |
| Paint | 9.45 | 4.25 | 45.0 |

to decrease speed by 40 percent for the main effects of speed, PFG and surface. The time to decrease speed by 40 percent is more than 40 percent of the total stopping time. Hence, deceleration was less initially than later on during braking. Figure 3.14 shows mean time for a 40 percent decrease in speed as a function of speed, surface, and deceleration/pedal force gain. The time required is greatest for levels 1 and 2 on the lowest coefficient surface, whereas levels 5 and 6 give poorest performance on the wet and dry asphalt.

WHEEL LOCKUP FREQUENCY. A square root transformation was carried out on the wheel lockup frequency data prior to the analysis of variance. The analysis of variance (not shown here) indicated significant ($p \leq .01$) differences due to speed, deceleration/pedal force gain, displacement, PFG x displacement, surface, speed x surface, and PFG x surface. Table 3.12 shows the mean wheel lockup frequency for the main effects. The frequency of wheel lockups decreased with decreasing speed, decreasing values of PFG, increasing pedal displacement and increasing friction coefficient.

The interaction of speed and pedal force gain (Figure 3.15) shows that the number of wheel lockups decreased as the deceleration/pedal force gain was reduced at each speed, but the difference between the two speeds was reduced at low PFG values causing the significant interaction. There were minor effects due to pedal displacement across PFG, but the zero displacement pedal was poorer overall (Figure 3.16). Figure 3.17 shows the interaction of surface and PFG upon wheel lockup frequency. High deceleration/pedal force gains resulted in high frequencies of wheel lockup particularly on the wet-painted surface.

WHEEL LOCKUP DURATION. The total time (T) during which one or more wheels were locked up was recorded on each trial. The data were transformed to $\log_e (T+1)$ and treated by an

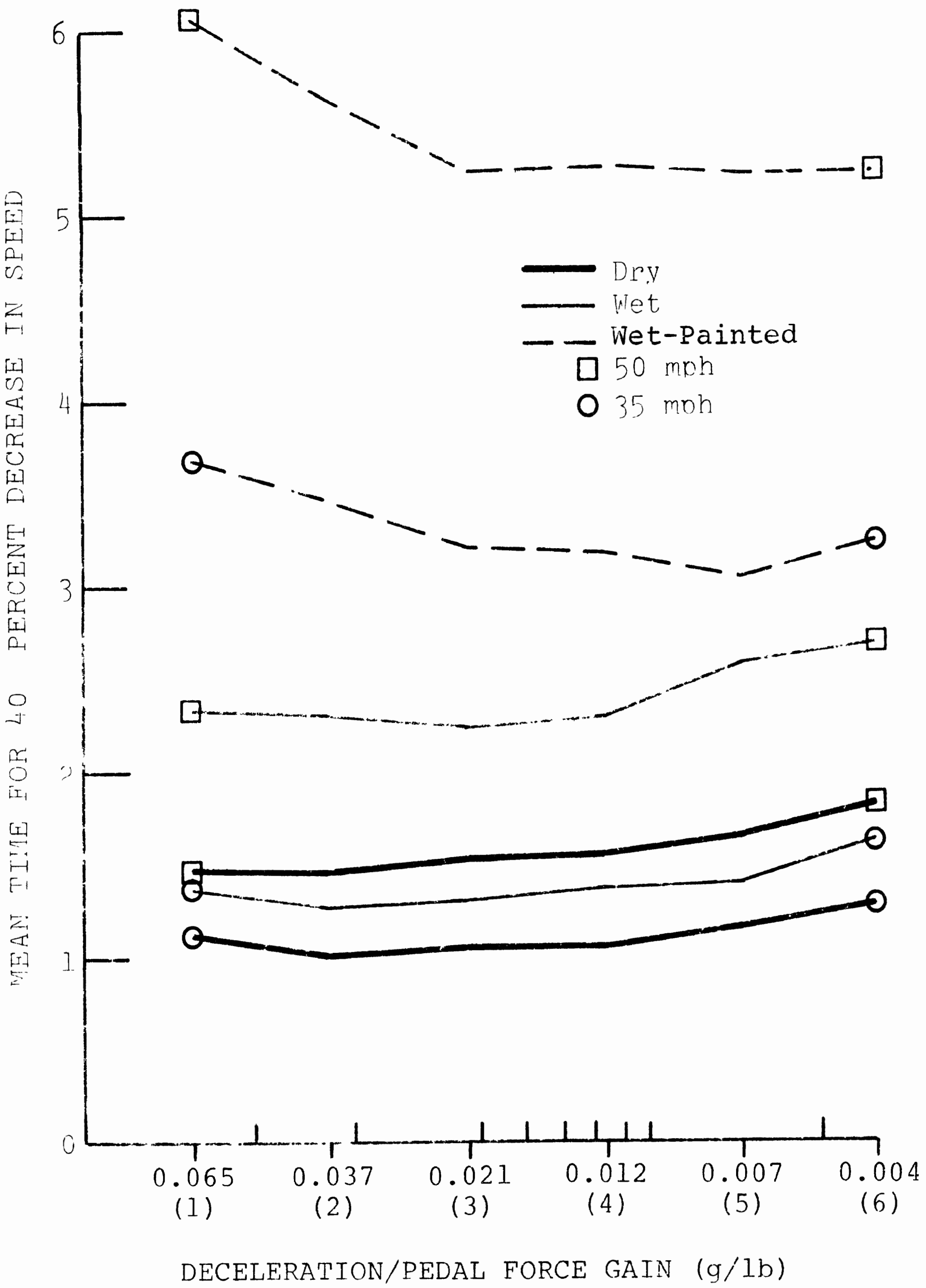



Figure 3.14. Mean time to reduce speed by 40% as a function of deceleration/pedal force gain, speed and surface.

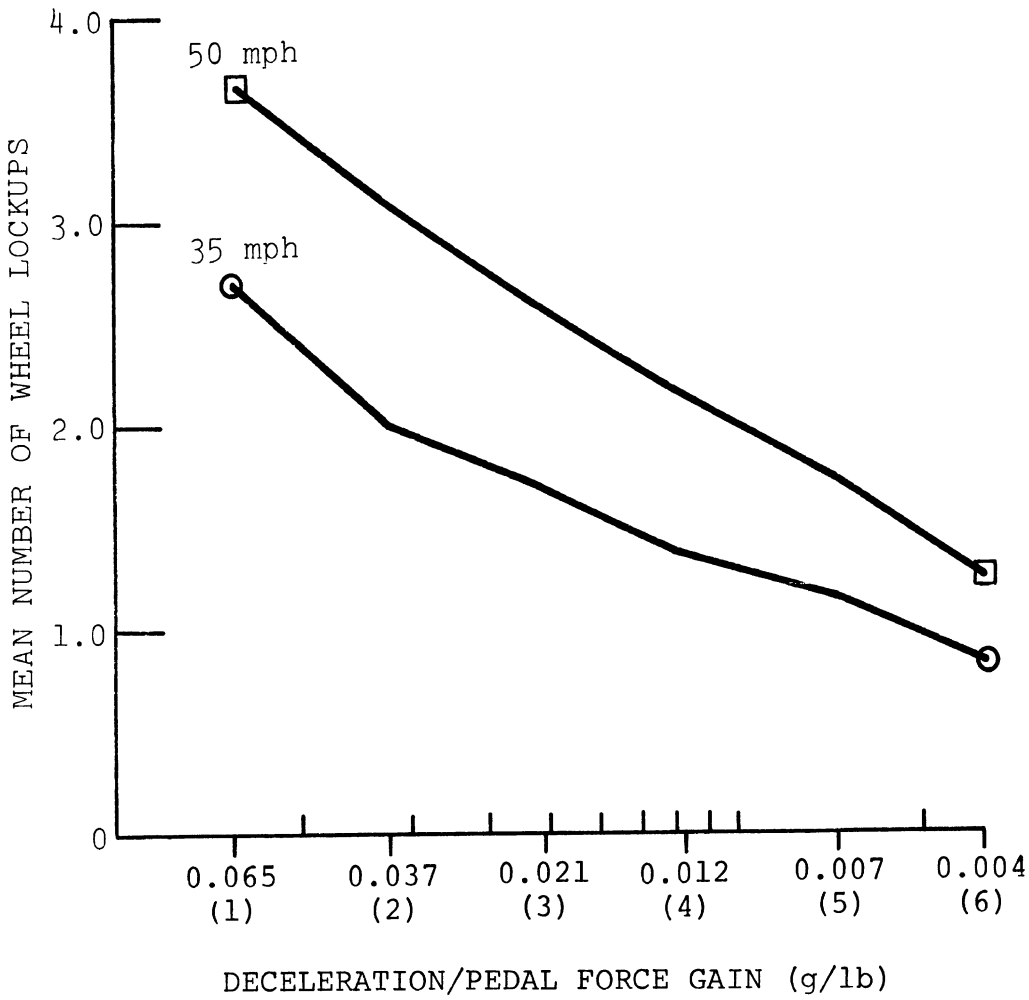



Figure 3.15. Mean number of wheel lockups as a function of deceleration/pedal force gain and speed.

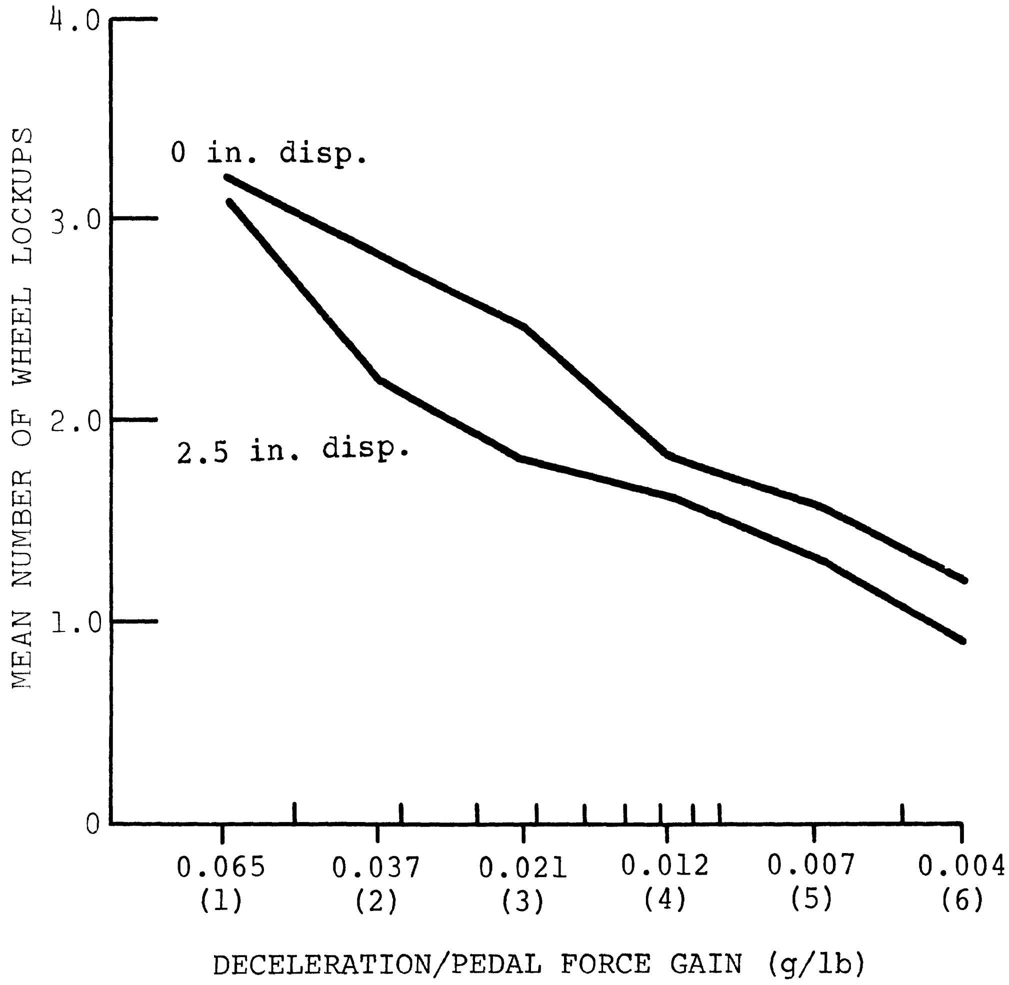



Figure 3.16. Mean number of wheel lockups as a function of deceleration/pedal force gain and displacement.

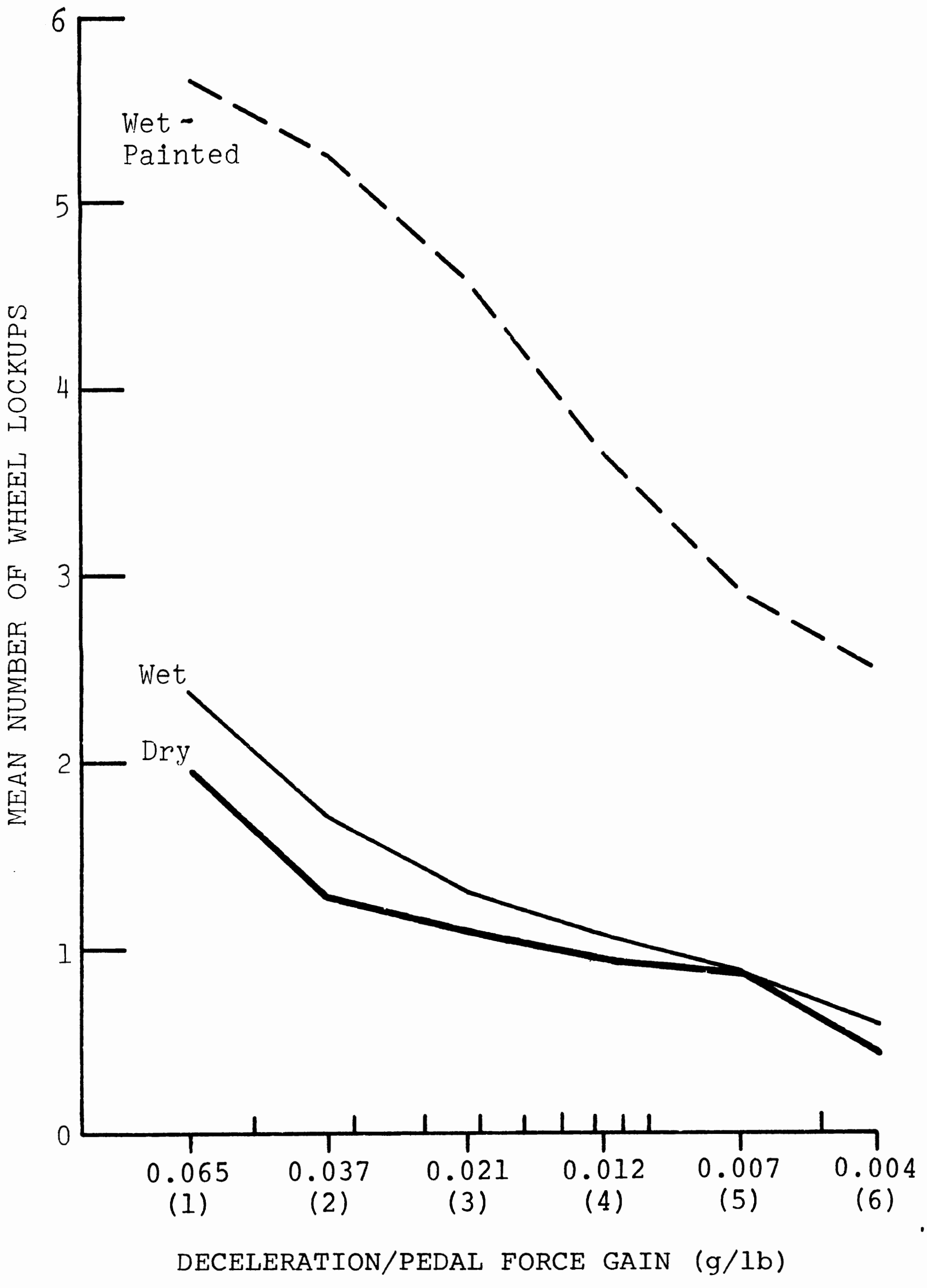



Figure 3.17. Mean number of wheel lockups as a function of deceleration/pedal force gain and surface.

TABLE 3.12. MEAN NUMBER OF WHEEL LOCK-UPS PER TRIAL FOR MAIN EFFECTS

| | Mean Number |
|---|---|
| 1) Speed (mph) | |
| 35 | 1.57 |
| 50 | 2.35 |
| 2) PFG (lb/g) | |
| a) 15.5 | 3.15 |
| b) 27.2 | 2.51 |
| c) 47.4 | 2.12 |
| d) 83.0 | 1.72 |
| e) 146.0 | 1.43 |
| f) 254.0 | 1.03 |
| 3) Displacement (inches) | |
| 0 | 2.12 |
| 2.5 | 1.77 |
| 4) Surface | |
| Dry | 1.05 |
| Wet | 1.26 |
| Wet-Painted | 4.00 |

analysis of variance. Significant ($p \leq .01$) differences in wheel lockup duration were due to speed, PFG, pedal displacement, surface, speed x surface, and PFG x surface. The mean lockup durations for the independent variables are shown in Table 3.13

The mean lockup time was slightly greater, overall, for the zero displacement pedal. The speed x surface interaction (Figure 3.18) shows the small effect of speed on the dry surface with increasing effects on the wet and wet-painted surfaces. Lockup

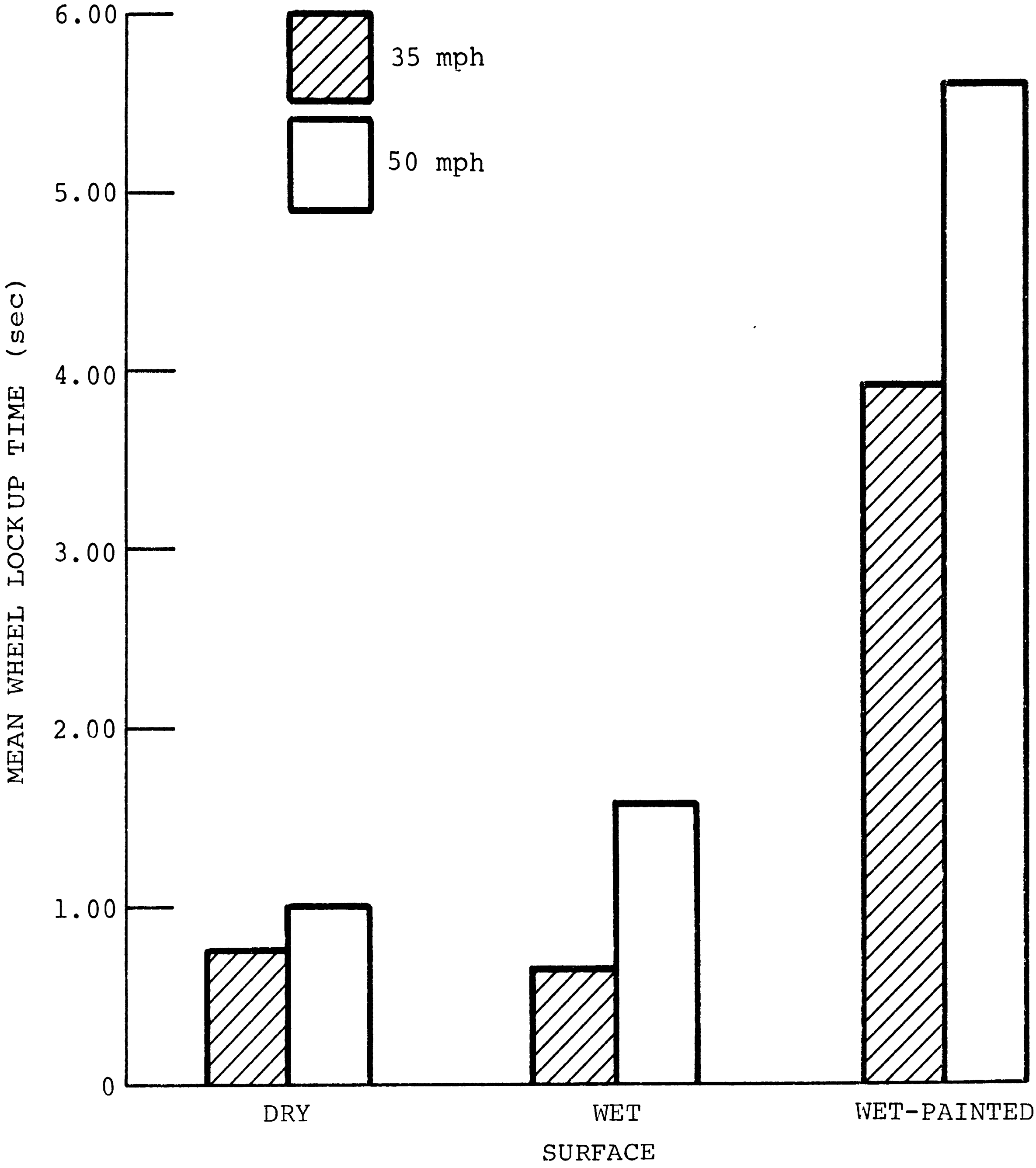



Figure 3.18. Mean wheel lockup time as a function of speed and surface.

TABLE 3.13. MEAN WHEEL LOCK-UP TIME PER TRIAL FOR MAIN EFFECTS

| | Mean (Sec) |
|---|---|
| 1) Speed (mph) | |
| 35 | 1.44 |
| 50 | 2.24 |
| 2) PFG (lb/g) | |
| a) 15.5 | 2.73 |
| b) 27.2 | 2.45 |
| c) 47.4 | 2.07 |
| d) 83.0 | 1.66 |
| e) 126.0 | 1.43 |
| f) 254.0 | 0.93 |
| 3) Displacement (inches) | |
| 0 | 1.91 |
| 2.5 | 1.72 |
| 4) Surface | |
| Dry | 0.89 |
| Wet | 1.06 |
| Wet-Painted | 4.71 |

time was not affected adversely at 35 mph on the wet surface, but there was an increase at 50 mph, compared to the dry condition.

The interaction of PFG and surface is shown in Figure 3.19. The consistent reduction of locked-wheel time across PFG is evident, particularly on the wet-painted surface. The negligible differences between dry and wet surface performance with PFG's 4, 5, and 6 will be noted.

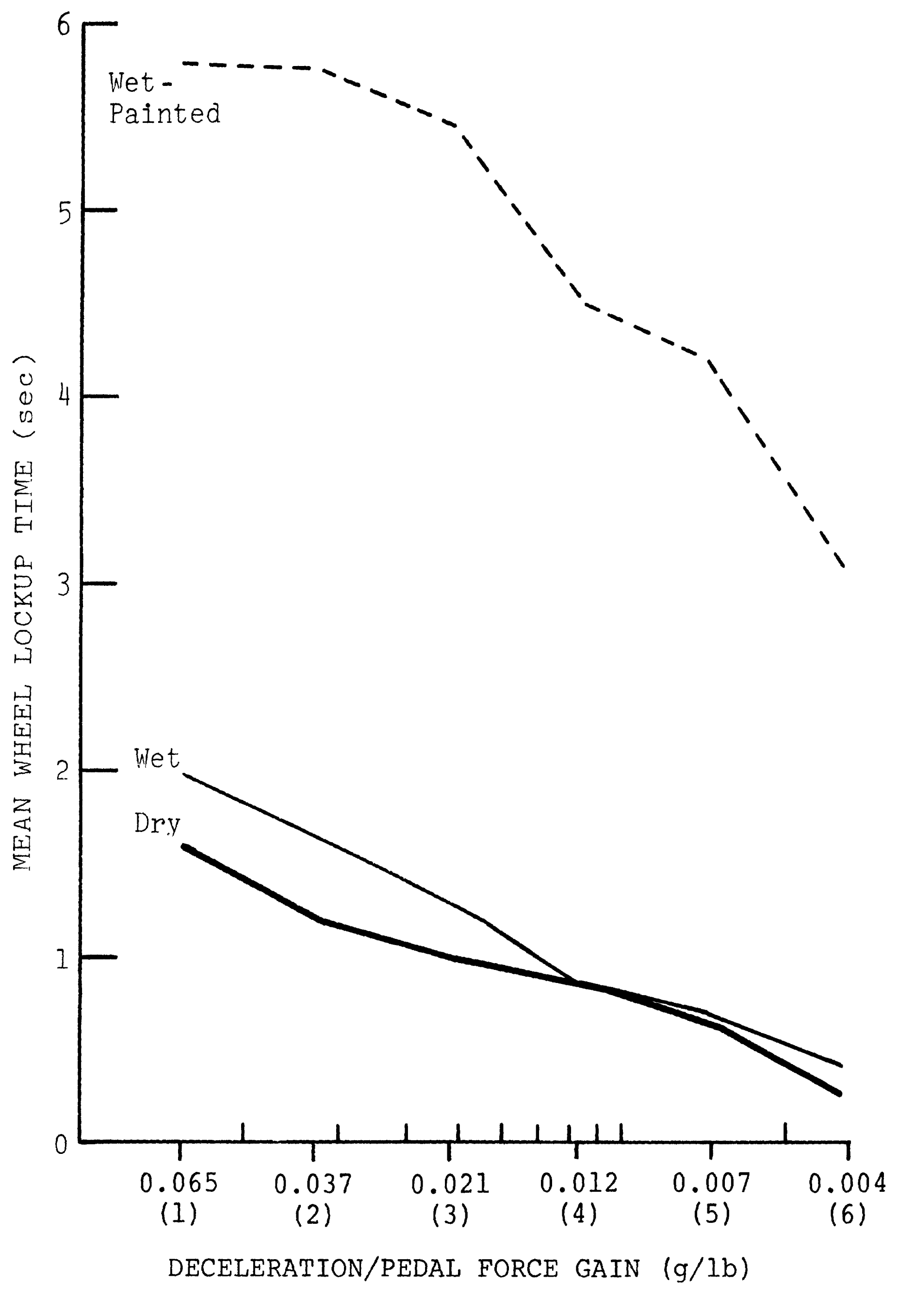



Figure 3.19. Mean wheel lockup time as a function of deceleration/pedal force gain and surface.

PROPORTION OF WHEEL LOCKUP TIME TO TOTAL BRAKING TIME. The wheel lockup time was divided by the total braking time in a trial to obtain the percent of locked-wheel time/braking time. Since pedal displacement did not interact with other factors affecting lockup time, it would not do so in this analysis. The effect of speed and surface is shown in Figure 3.20, indicating little difference at 35 and 50 mph on the dry surface, with an improvement at 35 mph over 50 mph on the wet which is reversed on the wet-painted surface. One or more wheels were locked up from 20 to 55 percent of the braking time (Figure 3.20) when averaged over PFG levels.

Figure 3.21 shows the percent of braking time for which wheels were locked up across PFG levels and surfaces. It will be noted that dry and wet surface results are almost identical, while there is a considerable increase on the wet-painted surface. On the dry and wet surfaces, in particular, there was a large reduction in percent of wheel lockup time to total braking time as the pedal sensitivity decreased.

It will also be noted that, with the most sensitive pedals (high deceleration/pedal force gain), drivers incurred close to the same percent of wheel lockup time on all three surfaces.

LOSS OF LATERAL CONTROL. Those trials in which the driver lost steering control of the vehicle, defined as touching one or more traffic pylons marking the lane, were repeated. The percent of trials in which the driver lost control, in each test condition, were recorded and are shown in Table 3.14 for all 28 subjects. These data are shown in Figure 3.22 for the zero displacement pedal and in Figure 3.23 for the 2.5 inch displacement pedal. It is apparent that subjects lost control of the test vehicle frequently when they braked from an initial speed of 50 mph on the wet-painted surface. The worst condition was the highest setting of deceleration/pedal force gain with loss of control occurring in 48 and 39 percent of the runs, with the

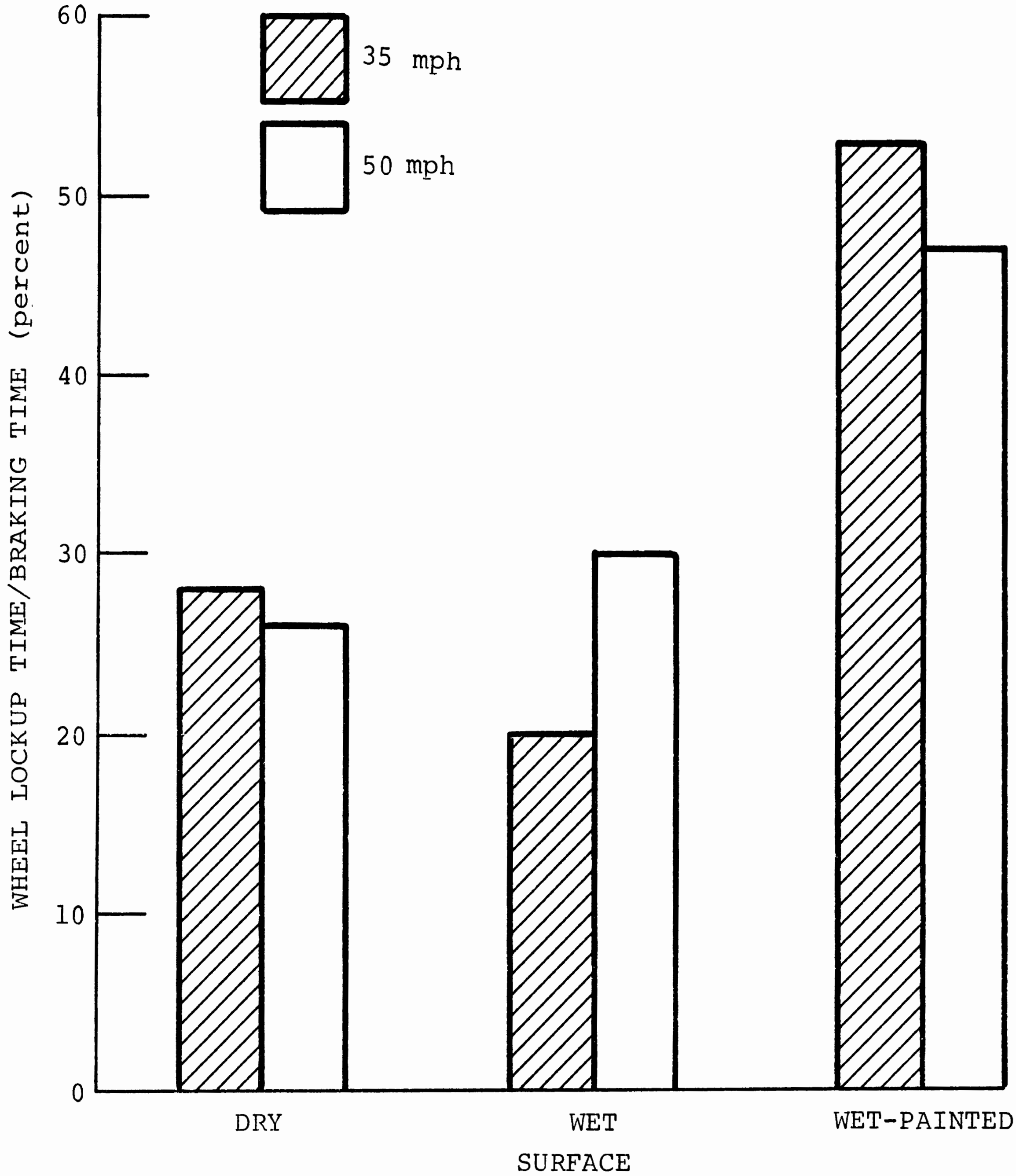



Figure 3.20. Percent wheel lockup time/total braking time as a function of surface and speed.

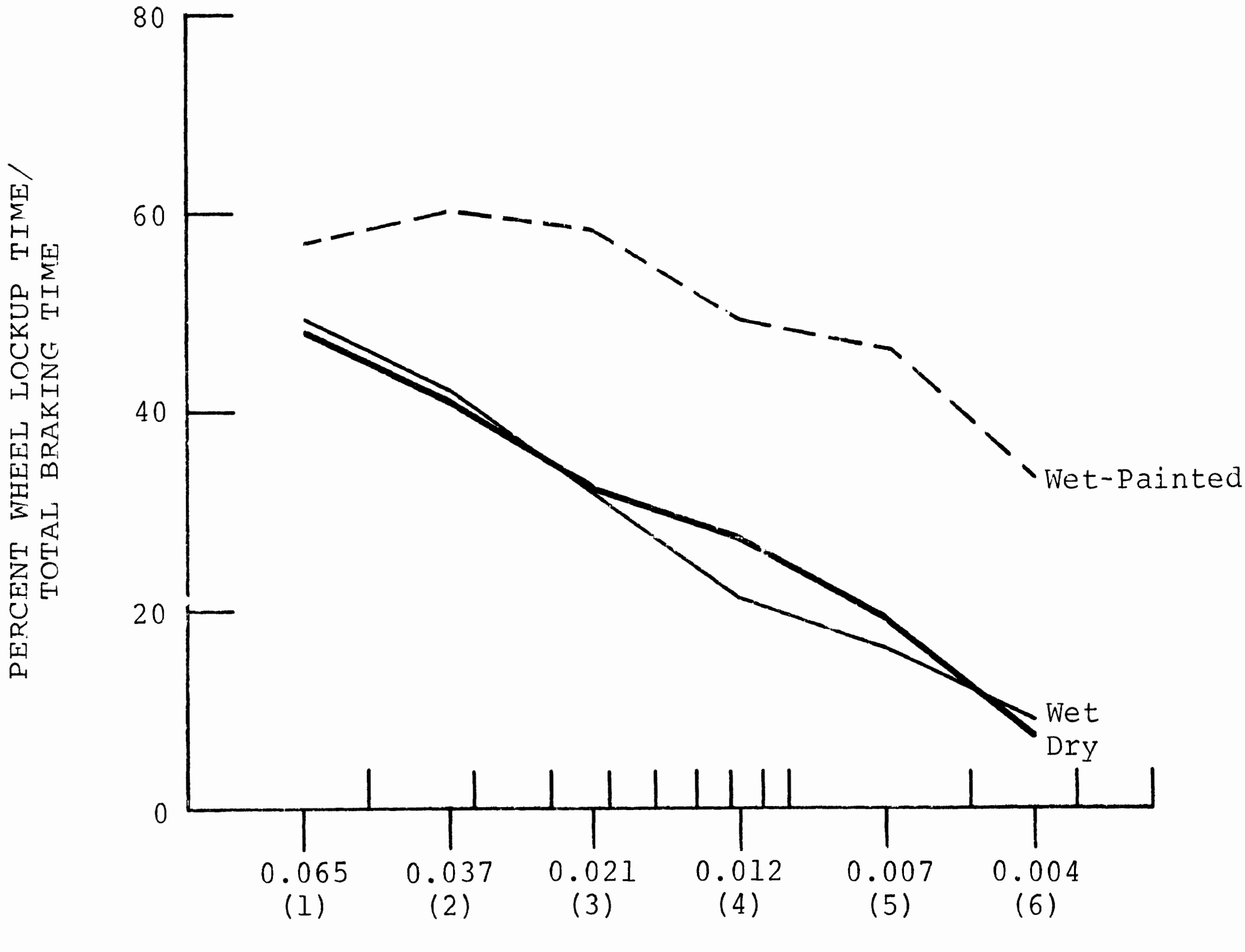



Figure 3.21. Percent wheel lockup time/total braking time as a function of deceleration/pedal force gain and surface.

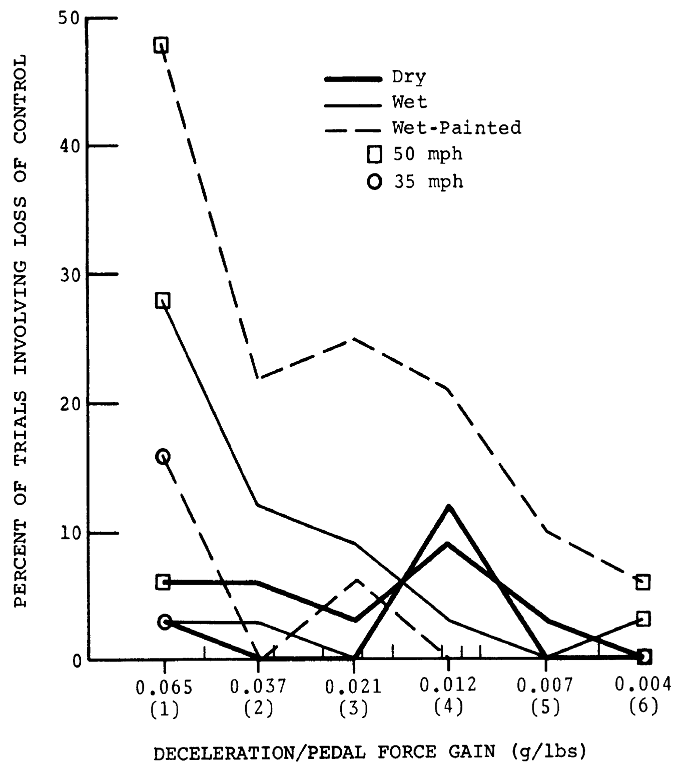



Figure 3.22. Percent of trials involving loss of lateral control as a function of deceleration/pedal force gain, surface and speed: 0 inch displacement.

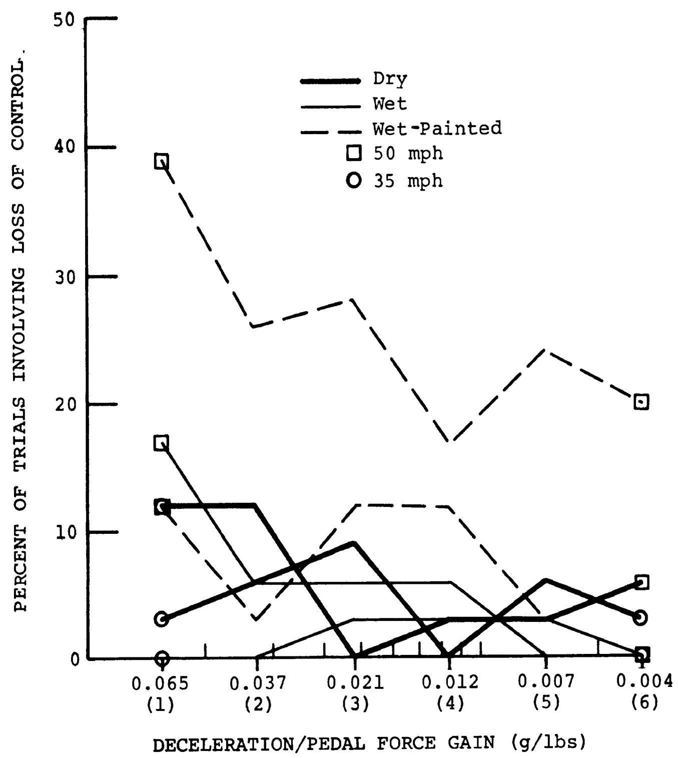



Figure 3.23. Percent of trials involving loss of lateral control as a function of deceleration/pedal force gain, surface and speed: 2.5 inches displacement.

TABLE 3.14. PERCENT[1] OF TRIALS INVOLVING LOSS OF LATERAL CONTROL AS A FUNCTION OF BRAKE SYSTEM, SPEED AND SURFACE

| Pedal Displacement | Surface | MPH | DECELERATION/PEDAL FORCE GAIN 0.065 | 0.037 | 0.021 | 0.012 | 0.007 | 0.004 |
|---|---|---|---|---|---|---|---|---|
| 0 | Dry | 35 | 3 | 0 | 0 | 12 | 0 | 0 |
| | | 50 | 6 | 6 | 3 | 9 | 3 | 0 |
| | Wet | 35 | 3 | 3 | 0 | 0 | 0 | 0 |
| | | 50 | 28 | 12 | 9 | 3 | 0 | 3 |
| | Wet-Painted | 35 | 16 | 0 | 6 | 0 | 0 | 0 |
| | | 50 | 48 | 22 | 25 | 21 | 10 | 6 |
| 2.5 | Dry | 35 | 3 | 6 | 9 | 0 | 6 | 3 |
| | | 50 | 12 | 12 | 0 | 3 | 3 | 6 |
| | Wet | 35 | 12 | 3 | 12 | 12 | 3 | 0 |
| | | 50 | 17 | 6 | 6 | 6 | 0 | 0 |
| | Wet-Painted | 35 | 12 | 3 | 12 | 12 | 3 | 0 |
| | | 50 | 39 | 26 | 28 | 17 | 24 | 20 |
| | MEAN | | 19.0 | 9.2 | 9.7 | 8.0 | 5.1 | 3.7 |

[1]Percent = $\frac{\text{Loss of Control Trials in a Test Condition}}{\text{Total (Successful \& Loss of Control) Trials in a Test Condition}} \times 100$

0 and 2.5 inch displacement pedal, respectively. Other than at 50 mph on the wet-painted surface, loss of control with PFG level 1 occurred in less than 10 percent of the runs.

RATINGS OF CONTROLLABILITY. The "controllability" ratings, averaged over all subjects, are shown in Figure 3.24 as a function of deceleration/pedal force gain and pedal displacement level. The influence of pedal displacement level on the "controllability" rating is seen to be quite small. It is clear that the highest level of deceleration/pedal force gain is rated significantly higher than the other gain levels. Gain levels 3 and 4 are preferred above all other gain settings.

RATINGS OF PEDAL FORCE. Subjective ratings as to the level of pedal force required to brake are shown in Figure 3.25, averaged over all subjects. Again it appears that pedal displacement level has a minor influence on driver opinion of the level of required force. Drivers judged deceleration/pedal force gain levels 1 and 2 require force levels that are too low, levels 5 and 5 as requiring force levels that are too high, and levels 3 and 4 as requiring forces that are "just right".

BETWEEN-SUBJECT PERFORMANCE COMPARISON. The two subjects producing the highest and lowest mean deceleration (over all test conditions) are compared with each other and with the mean performance of all subjects in Figure 3.26. It is clear that the influence of deceleration/pedal force gain, as derived for all subjects, holds, in general, for the two extreme cases. It is also clear that there were differences in braking modulation skill among subjects, that these differences were consistent over all three test surfaces, and that between-subject performance dispersion was least on the wet-painted surface. Sample time histories of pedal force application in the test are shown in Appendix II (Figure A.II. 1-3).

CORRELATION BETWEEN MAXIMUM PEDAL FORCES MEASURED IN THE VEHICLE AND THE BUCK. The highest values of pedal force produced by subjects during braking runs on the dry surface at the lowest deceleration/pedal force gain setting was measured and recorded on a strip-chart recorder. Since 260 pounds constituted the upper limit on the read-out instrumentation, the data were classified in terms of pedal forces being above or below 260 pounds. Table 3.15 shows that 14 subjects exerted more than 260 pounds both in the vehicle and on the static buck. Of the 14 subjects who had less than a 260 pound maximum pedal force capability, as measured on the static buck, two exerted greater than 260 pounds in the test vehicle. It was also observed that 10 of the 14 subjects, rated by the static buck as not being able to produce 260 pounds of pedal force, did, in fact, apply a greater force in the test vehicle. There were four subjects who produced the same pedal force on the buck and in the test vehicle. By arbitrarily assigning a maximum force of 260 pounds

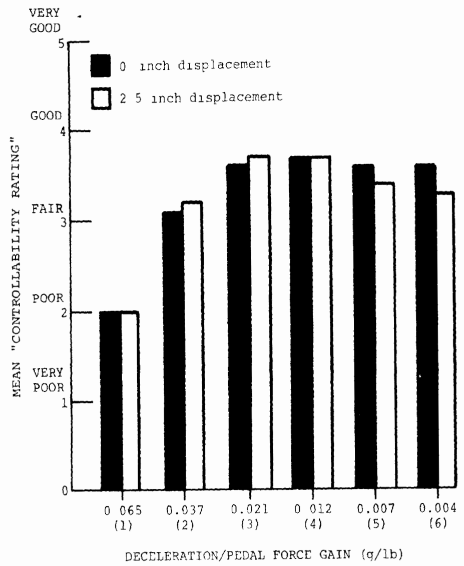



Figure 3.24. Mean controllability rating for 28 subjects as a function of deceleration/pedal force gain and pedal displacement.

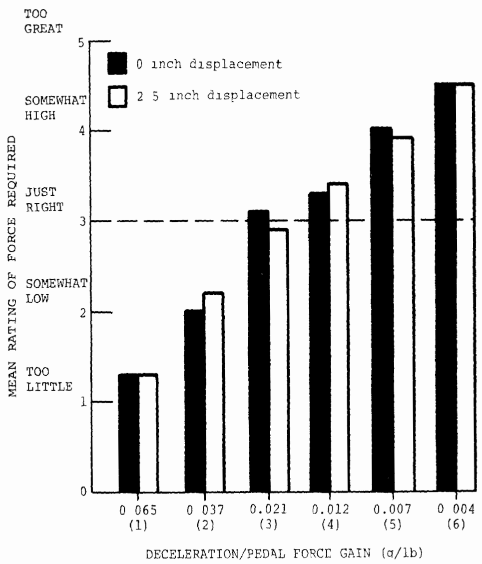



Figure 3.25. Mean rating of force required for 28 subjects as a function of deceleration/pedal force gain and displacement.

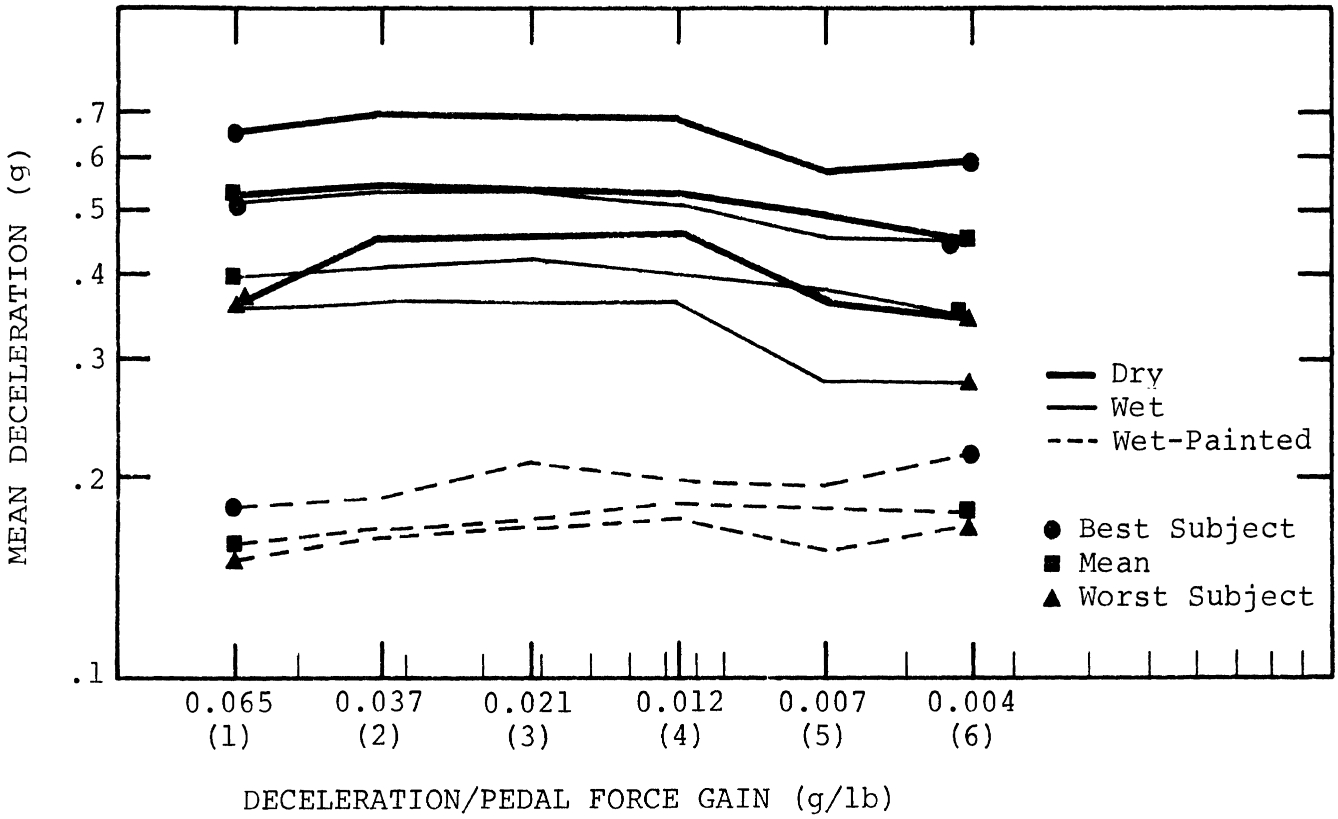



Figure 3.26. Braking performance of the best subject, group mean and poorest subject as a function of deceleration/pedal force gain and surface.

TABLE 3.15. MAXIMUM PEDAL FORCES IN THE STATIC TEST AND IN THE TEST VEHICLE. CELL VALUES INDICATE NUMBER OF SUBJECTS

| | | PEDAL FORCE IN TEST VEHICLE | | |
|---|---|---|---|---|
| | | ≤260 lbs | ≥260 lbs | TOTAL |
| STATIC PEDAL FORCE | ≤260 lbs | 12 | 2 | 14 |
| | ≥260 | 0 | 14 | 14 |
| | TOTAL | 12 | 16 | 28 |

to subjects who exceeded this value in either test, a Pearson Product-Moment correlation coefficient of $r_{s \cdot v} = 0.78$ was obtained between the maximum pedal force produced on the static buck and in the test vehicle.

SUBJECT AGE AND WEIGHT. A complete matrix of age and weight categories existed only for the male subjects (see Table 3.1). An analysis of variance for braking distance and wheel lockup frequency indicated that there were no significant effects attributable to either age or weight of the subjects. There was, however, a significant four-factor interaction of wheel lockup duration involving speed, deceleration/pedal force gain, pedal displacement level, and driver age. Since such a high-order interaction has little useful information, the analysis was not carried further.

## DISCUSSION

DECELERATION MEASURES. The study has shown that deceleration/pedal force gain influences driver-vehicle braking performance and that this influence is, in turn, affected by the friction coefficient of the road surface. The mean deceleration achieved by the 28 subjects also proved to be dependent upon the initial velocity particularly when the road surface was wet. Since the mean decelerations achieved in 35 mph stops was greater, when braking on the wet and wet-painted surface, than at 50 mph it appears that the braking task is less difficult at the lower speed when friction levels are reduced from dry-road values (Figure 3.13).

Pedal-displacement level (0 and 2.5 inches) did not have a significant influence upon mean deceleration, which result indicates that the brake is modulated largely by force feedback, rather than by displacement.

Table 3.7 shows the significant differences that were found in performance within a given combination of speed and tire-road friction level. It is seen that deceleration/pedal force gain needs to be reduced to optimize performance as friction levels are reduced. The findings show that the range of deceleration/ pedal force gains employed in the experiment was sufficient to show those values that lead to peak man-machine performance. When averages are taken across all variables other than deceleration/pedal force gain, it is found that the intermediate gains (levels 3 and 4) produced the shortest braking distances, i.e., the greatest mean decelerations.

LOSS OF CONTROL MEASURES. The frequency and duration of wheel lockups constitute data that indicate the extent to which the driver-subjects are able to control the path of the vehicle. Front-wheel lockup results in the vehicle not responding to steering inputs while rear-wheel lockup constitutes an unstable condition, particularly on low friction surfaces. The test results show that the frequency of wheel lockup was less when the pedal had a finite displacement, with the difference between the two displacement levels being small at most deceleration/pedal force gains (Figure 3.16). As expected, there were more lockups in stops made from 50 mph. A consistent decrease in frequency of wheel lockup is obtained as deceleration/pedal force gain was reduced. Figure 3.17 shows that wheel-lockup frequency is much greater on the wet-painted surface than on the wet or dry surface with the influence of deceleration/pedal force gain being very marked. The data show quite clearly that the highest level of deceleration/pedal force gain used in these tests causes high frequencies of wheel lockup.

Lockup durations were significantly longer for the zero-displacement pedal, but the mean difference between the two displacement levels was less than 0.2 seconds (Table 3.13). This result is minor compared to the influence of the other independent variables. As deceleration/pedal force gain was reduced, there was a consistent decrease in lockup duration. Deceleration/pedal force gain levels 1 and 2 produced significantly longer durations of wheel lockup than levels 4,5 and 6 on all surfaces. When measured durations were ratioed to the total braking times achieved at each gain level, it was found (see Figure 3.19) that the wheels were locked up on the dry and wet surfaces the same percentage of time. A consistent reduction occurred in the percentage of time the wheels were locked up as deceleration/pedal force gain was decreased, though this trend was less marked on the wet-painted surface, in which case the wheel lockup time was high (35% - 60% of total braking time).

Although wheel-lockup frequency and duration can be taken as indicators of potential (or actual momentary) loss of control, loss of control events did occur (defined as the inability to hold the car within a 10 foot wide lane) in the test program. Note that all of the performance measures considered thus far were taken on runs in which the car was held in the lane and, therefore, the loss of control indicators (wheel lockup frequency and duration) are conservative predictors. Table 3.14 shows the percentage of runs terminated because the driver left the lane. It is seen that these results are related to the independent variables in a manner similar to that observed for wheel lockup frequency and duration. Consequently, the latter measures appear to be good predictors of possible loss of control. Note that loss of control occurred most frequently with the highest deceleration/pedal force gain, particularly in 50 mph runs.

SUBJECTIVE MEASURES. Before considering the implications of the findings with respect to objective measures of performance, the subjective ratings should be considered. Driver ratings of brake system controllability showed that the highest deceleration/pedal force gain was viewed as not providing adequate control. Levels 3-6 were clearly preferred. Driver ratings of the force levels required by each brake configuration showed that gain levels 1 and 2 were viewed as too sensitive (i.e., not requiring enough force) while levels 5 and 6 were viewed as requiring too much force. In the aggregate, the subjective data indicate that gain levels 3 and 4 were preferred by the 28 driver subjects. Further, the ratings produced by these subjects were not significantly influenced by pedal displacement level. In general, the subjective ratings support the objective performance measures rather well.

DRIVER-VEHICLE BRAKING EFFICIENCY. The ability of a driver to modulate his brakes to achieve minimum stopping distances while maintaining adequate directional control is measured, in part, by the braking efficiency attained by the driver-vehicle system. For this reason, it appeared logical to examine the driver-vehicle braking efficiencies achieved in the test program.

To compute this efficiency, it is first necessary to know or determine the braking efficiency designed into the vehicle. With this information, it is possible to calculate the efficiency with which the driver utilizes the available road friction in stopping without losing directional control as:

$$\eta_{d-v} = \frac{ax/g \text{ driver-vehicle}}{\eta_v \mu}$$

where

$ax/g$ driver-vehicle = mean deceleration produced by a subject in a given trial

$\eta_v$ = vehicle braking efficiency

$\mu$ = friction coefficient produced at the tire road interface

In applying the above formulation, there is a question as to the numeric that should be used to characterize the friction coefficient, $\mu$, of the roadway. It can be argued that braking efficiency calculations should be based on the peak value of friction that can be attained by a rolling tire on the grounds that this is the deceleration that the vehicle would attain if the driver were able to perform as an ideal controller. Accordingly, braking efficiencies have been computed using coefficients of peak friction that were established for each of the three test surfaces (Note that a single numeric has been used to describe each of the test surfaces even though it is realized that friction coefficients are velocity dependent). The braking efficiency, $\eta$, of the test vehicle was obtained in tests described earlier (See Figure 3.3).

As a result of these tests and measurements made with HSRI's on-the-road tire test device, the peak friction coefficients established for the wet-painted, wet, and dry surfaces were 0.40, 0.71, and 0.86, respectively. Using these numbers and the efficiency data produced in actual tests with the instrumented vehicle, the combined driver-vehicle efficiencies plotted in Figure 3.27 were obtained. It is seen that the highest values of combined efficiency were attained when drivers braked on the dry

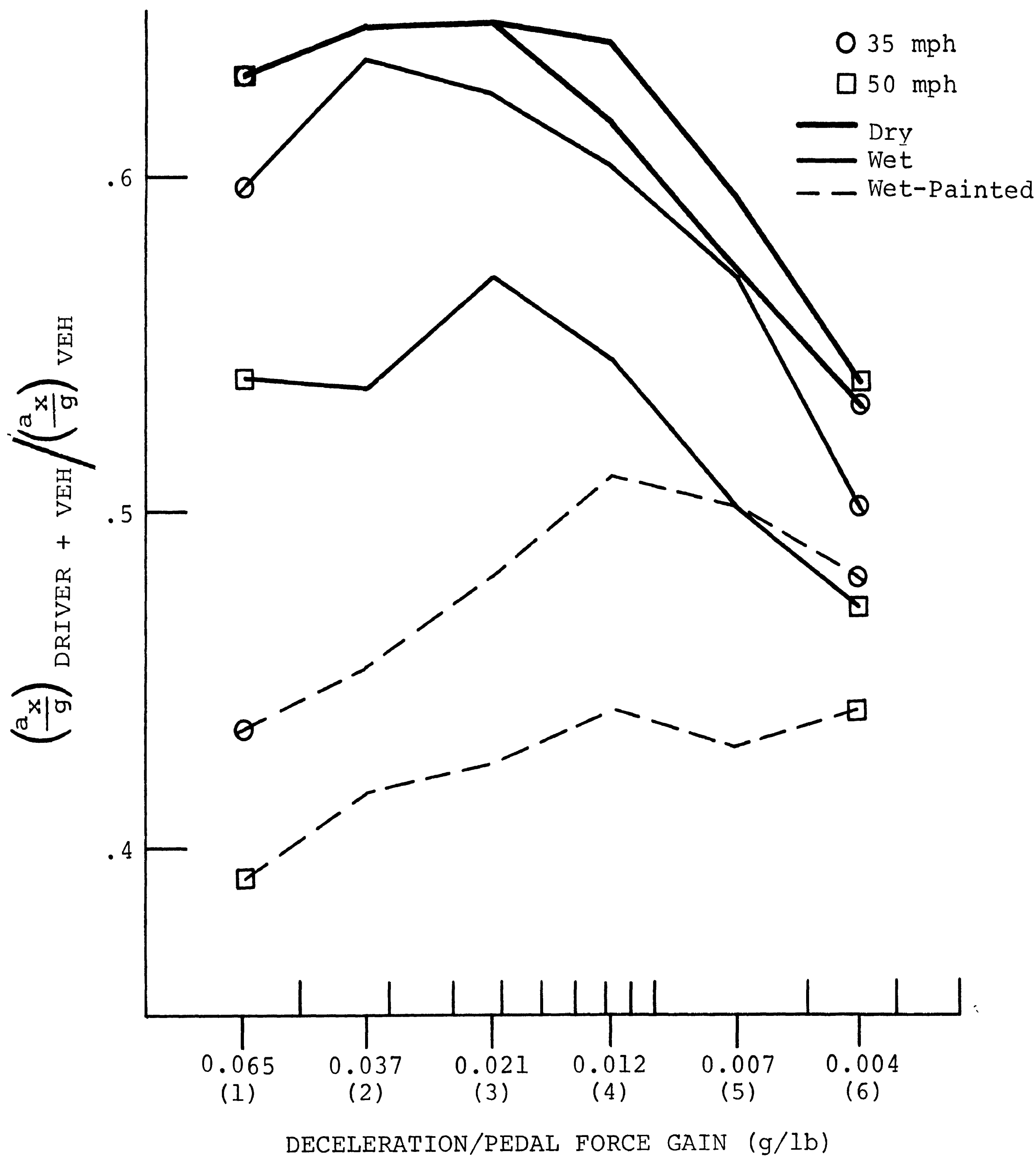



Figure 3.27. Mean braking efficiency as a function of deceleration/pedal force gain, speed and surface.

surface, where $\mu$ = 0.86. Braking on the wet and wet-painted surfaces resulted in lower values of efficiency. On these latter surfaces, efficiency was further reduced when the initial velocity was 50 mph compared to 35 mph.

These results indicate that drivers, by and large, are poor modulators of brake systems when they attempt to make minimum distance stops and hold the vehicle within a slightly curved lane.

It is not known whether this poor modulation performance should be attributed primarily to a lack of driver skill or training or whether this decrement in closed-loop performance can be attributed, in part, to the dynamics of the brake-tire system. Since braking efficiency is lowest on the surface with minimum friction, it might appear that the potential for improvement by driver training is greatest for this operational condition. However, the comparison made earlier between the performance of the best and poorest drivers (Figure 3.26) on the lowest friction surface suggest that the task is sufficiently difficult that driver training and/or skill is of little avail. Driver skill does seem to make a difference, however, as the task becomes less demanding, that is, as the friction coefficient is increased above that produced by the wet-painted surface.

DERIVATION OF THE PFG ENVELOPE. The results of the braking test can be used to suggest bounds on PFG. This was the major objective of this research. The rationale is to consider those PFG levels within each of the surface conditions which resulted in impaired performance. For example, Table 3.10 shows that when attempting to achieve maximum deceleration on the dry surface ($\mu_{peak}$ = 0.86) performance fell off at PFG greater than level 2, namely at PFG level 1 (.065 g/lb), at both test speeds. Therefore, maximum PFG when braking on a road having a surface-tire friction coefficient the same as the dry asphalt should be less than 0.065 g/lb. This gain value can be taken as a boundary condition, and is shown as point $A^1$ in Figure 3.28. Similarly, PFG level 3 provided significantly greater mean deceleration than level 2, at 50 mph on the wet surface. Therefore, PFG level 2 (0.037 g/lb) can be taken as a boundary condition for that surface ($\mu_{peak}$ = 0.71), and is shown as point B in Figure 3.27. In an analogous manner PFG level 3 (0.021 g/lb) is the boundary gain condition for braking on the wet-painted surface ($\mu_{peak}$ = 0.40), and is shown as point C in Figure 3.27. These points have been derived only from the deceleration performance data to select maximum PFG levels. The measurement of wheel lockup frequency and duration, and the loss of control measures strongly argue against the use of the highest PFG used in this test. PFG level 1 had significantly greater frequency and duration of wheel lockups than other levels. Therefore, it is proposed that, for the dry pavement case ($\mu_{peak}$ = 0.86), the maximum PFG should be level 2 (0.037 g/lb), which actually produced slightly better deceleration performance than level 1, and considerable improvement in loss of control measures. Therefore, the cut-off maximum PFG for braking at about 0.86 g is shown as A in Figure 3.28. This strategy is also supported by the subjective "controllability" and "force" ratings (Figures 3.24, 3.25).

Points A, B and C define maximum gain values at the indicated deceleration values.

Table 3.10 can also be used to set minimum PFG levels in terms of deceleration performance. For example, level 4 is a cut-off point for the dry and the wet pavement, and level 5 for the wet-painted surface. These cut-off values are shown as points D, E and F in Figure 3.28. They define minimum PFG levels to maximize driver braking performance at the respective deceleration values. Thus, PFG values between the maximum and minimum cut-off points at each deceleration define desirable brake characteristics.

Based on these considerations, it could be recommended that PFG values be limited by the boundaries set at A, B, C, D, E, and

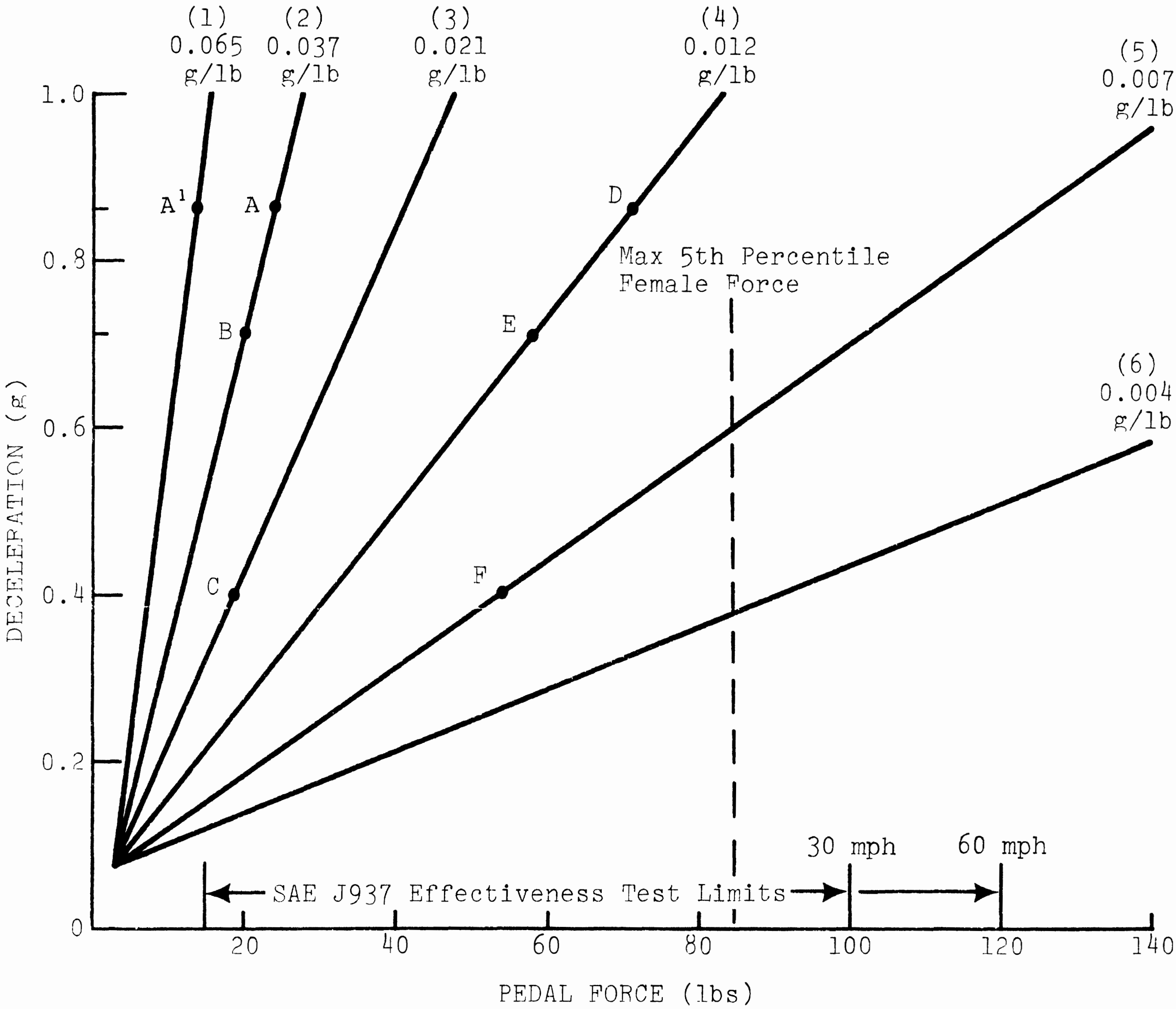



Figure 3.28. Cut-off PFG values for satisfactory driver-vehicle braking performance.

F. This means that:

1. PFG values should not exceed those found at A, B and C for the indicated vehicle deceleration levels, i.e., the slopes of deceleration/pedal force should not exceed 0.037 g/lb at 0.86 g and 0.71 g, and 0.021 g/lb at 0.40 g.
2. Pedal force values should not be less than those at A, B and C to obtain the indicated vehicle deceleration levels.
3. PFG values should not be less than those found at D, E and F for the indicated vehicle deceleration levels.
4. Pedal force should not exceed 85 lbs at 0.75 g (based upon an approximate maximum vehicle deceleration of 0.75 g and female, 5th percentile, pedal force data obtained in Task 2).

DEVELOPMENT OF A REVISION TO MVSS-105. Using the data shown in Figure 3.28 it is possible to develop a modification of this Figure that more aptly can be used to describe a revision to MVSS-105. Such a revision should be practicable, and meaningful with respect to safety objectives.

In order to provide a brake control that allows efficient modulation of vehicle deceleration on low friction surfaces, to minimize stopping distance, pedal force should not be too low and the deceleration/pedal force gain should not be too high. This condition is fulfilled at point C in Figure 3.28, where the slope of the deceleration/pedal force gain is 0.021 g/lb. Higher deceleration/pedal force gains did not provide significantly improved performance in any test condition compared to 0.021 g/lb; but they resulted in relatively impaired performance as measured by a number of the dependent variables. Therefore, a deceleration/pedal force gain of 0.021 g/lb can be taken as the maximum gain, and a line of this slope, passing through the origin in the deceleration-pedal force space shown in Figure 3.29, defines the maximum gain-minimum pedal force boundary.

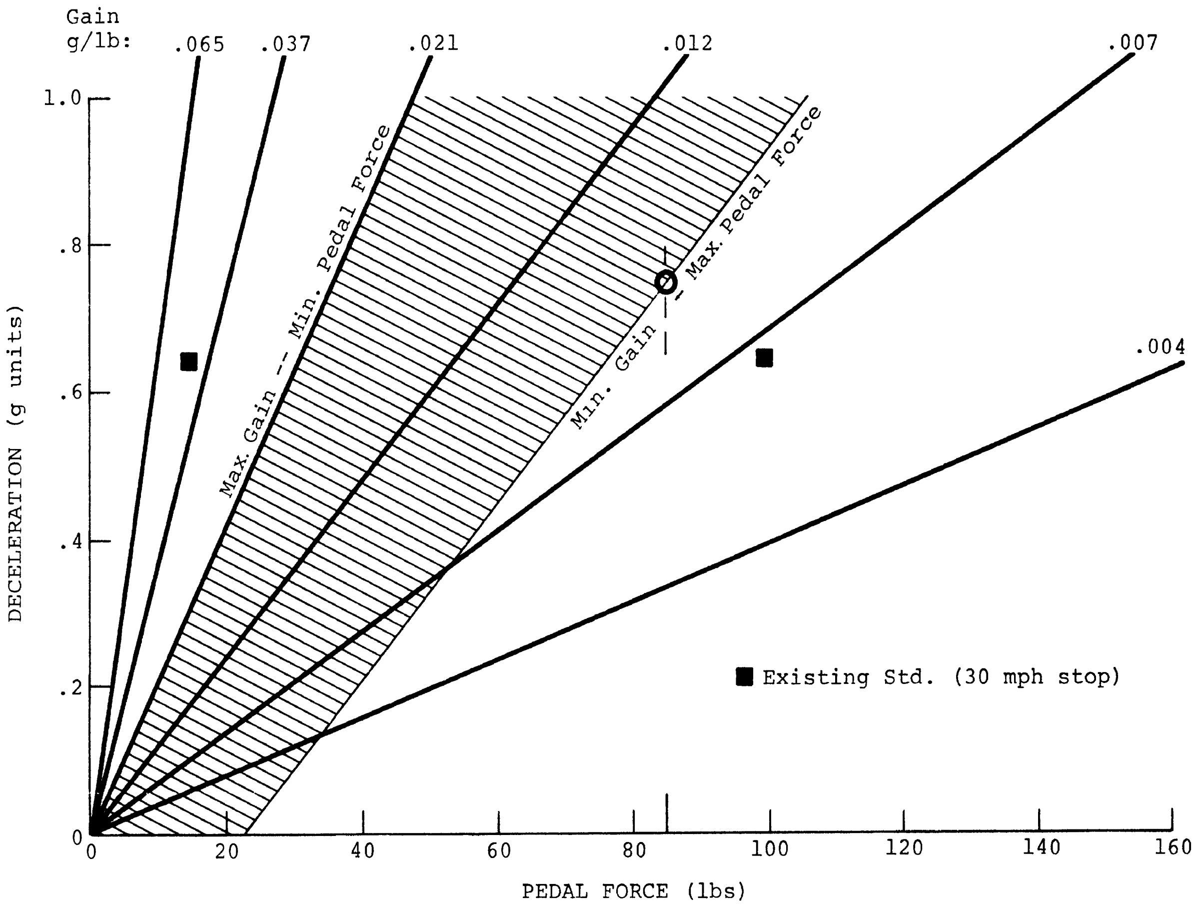



Figure 3.29. The recommended deceleration/pedal force space.

Figure 3.28 showed that at high decelerations the minimum gain should be 0.012 g/lb, and this minimum also gave rise to good performance at low decelerations. In addition, at 0.75 g pedal force should not exceed 85 lbs.

Therefore, a line of slope 0.012 g/lb passing through the point 0.75 g: 85 lbs defines the minimum gain boundary.

# 4. DRIVER BRAKING PRACTICE

## INTRODUCTION

An evaluation of the consequences of partial failure in a brake system, in terms of the requirements imposed on the fully operational system, constitutes a secondary objective of this study. This evaluation requires, however, substantive information on how drivers typically brake their vehicles on our street and highway network.

It is clear that the majority of braking performed by drivers does not involve traffic conflicts of an emergency nature. On the other hand, a routine braking activity can develop into an emergency if a loss of braking effectiveness occurs due to a partial failure in the service brake. For example, the stop that could readily be performed by a woman with limited pedal force capability may prove to be a deceleration level that she cannot achieve with a partially failed system. Some statistics on braking practice are obviously needed to draw firm conclusions relevant to the safety consequences of partial failure.

An examination of the literature showed that a few attempts have been made to record the frequency of brake applications and deceleration levels. Carpenter (1955) found that the average duration of brake applications decreased with increased speed of travel. He also found that there is an increase in the number of brake applications with increasing speed, with total braking time per mile being independent of speed. During 1400 miles of driving in hilly country, a total of 2800 brake applications were recorded. Only 5 percent of all brake applications involved decelerations exceeding 0.30 g, and 30 percent involved accelerations exceeding 0.17 g. On only 20 applications was a deceleration of 0.40 g exceeded.

Another study showed that on a random course through a business district a deceleration of 0.15 g was used most frequently although decelerations up to 0.40 g did occur (Kummer and

Meyer, 1967). It was observed that the deceleration and, hence, the frictional requirements for preventing a skid, usually increased toward the end of a stop.

On a 276 mile cross-country trip, braking took place 122 times (0.42 applications/mile) although the pedal was slightly depressed 288 times (1 application/mile). In six of these 122 applications, the traffic situation required decelerations of 0.40 g or above. Two of these situations resulted from driver inattention, and four from unexpected acts by other traffic. Speeds were below 20 mph in every case (Kummer and Meyer, 1967).

One major drawback of the studies cited consisted of driver awareness of the study purpose. This deficiency was eliminated in the data collection program conducted for this project. Drivers were only aware that the vehicle was on a test of some kind, since they were asked to fill out a trip sheet. Otherwise, they were not involved and, therefore, could be and were expected to drive in a normal manner.

## METHOD

APPARATUS. A 1968 Plymouth sedan was equipped with instrumentation to record the deceleration of the vehicle whenever the brake pedal was depressed. Except for a tachometer and a brakeline pressure transducer, the instrumentation package was located in the trunk (Figure 4.1). A complete description of the apparatus, calibration procedure, and sample data is given in Appendix III.

In the first phase of the study, the vehicle was equipped with standard brakes. For the second phase, the standard brakes were converted to power brakes, using a conversion kit obtained from the manufacturer. In this manner it was possible to obtain data on braking practice as influenced only by changes in brake force/deceleration gain (Figure 4.2), displacement/deceleration gain, and pedal location. Figures 4.3 and 4.4 show the pedal

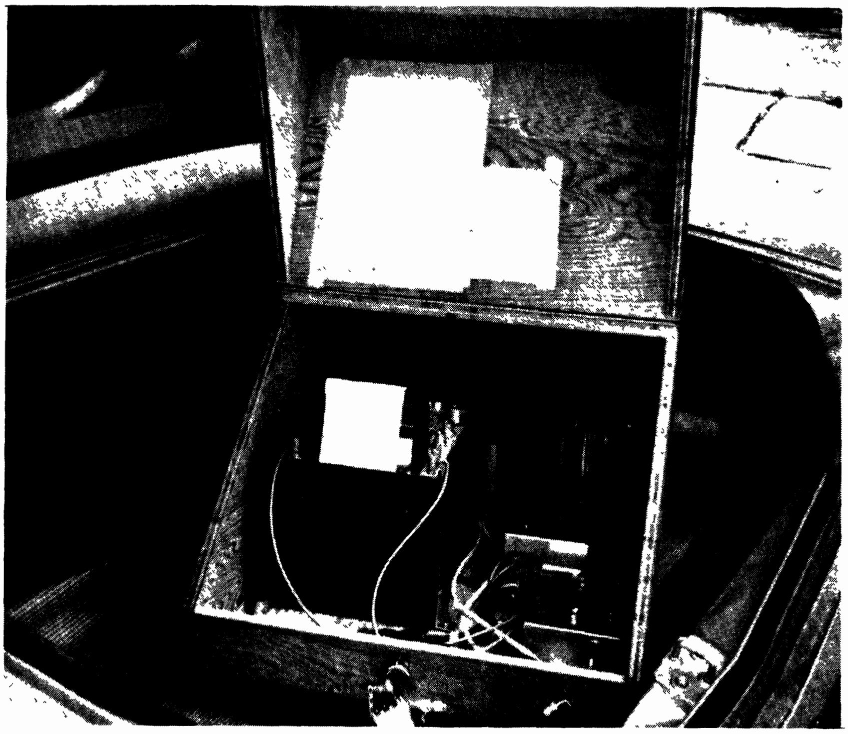


Figure 4.1. Data recording instrumentation in trunk of test car.

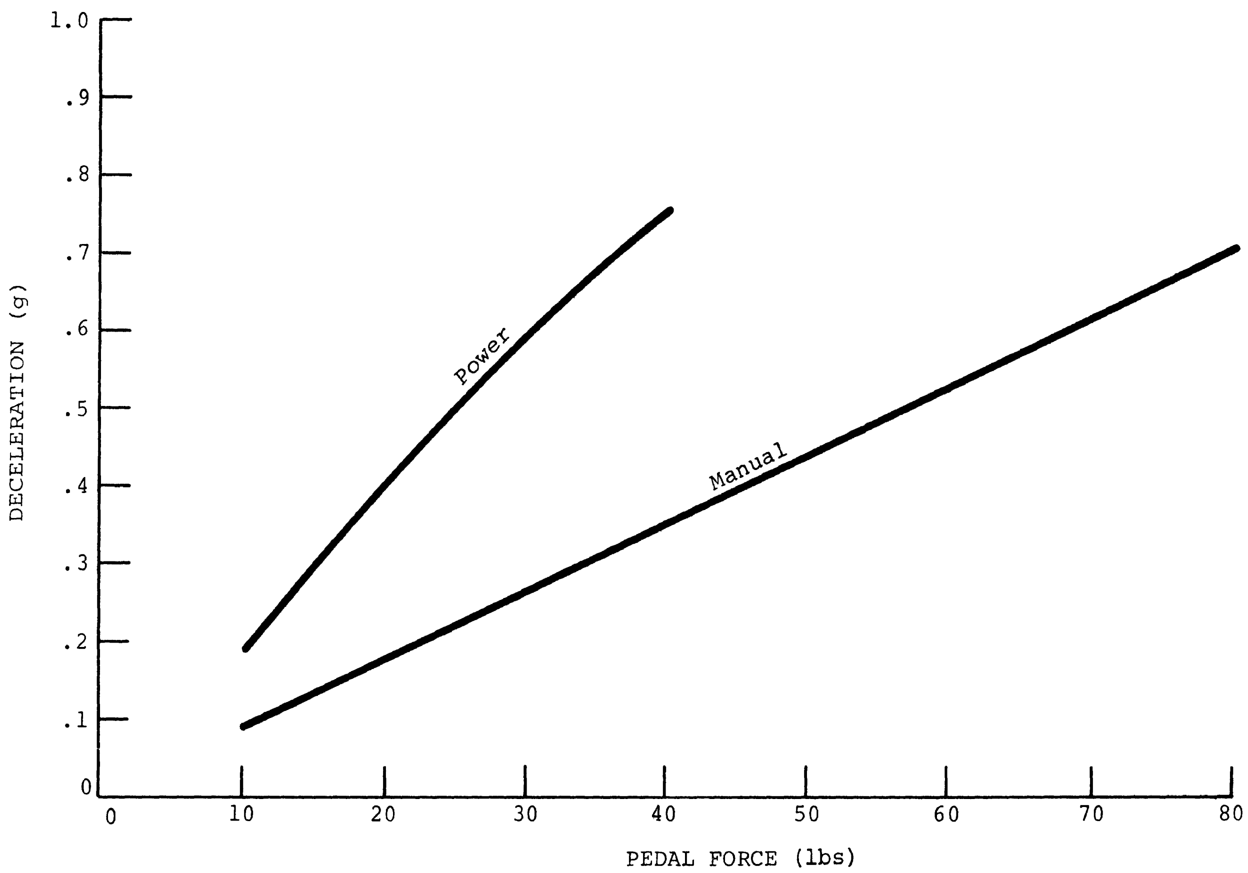



Figure 4.2. Deceleration/pedal force for manual and power brake mode. 1968 Plymouth 4 door sedan.

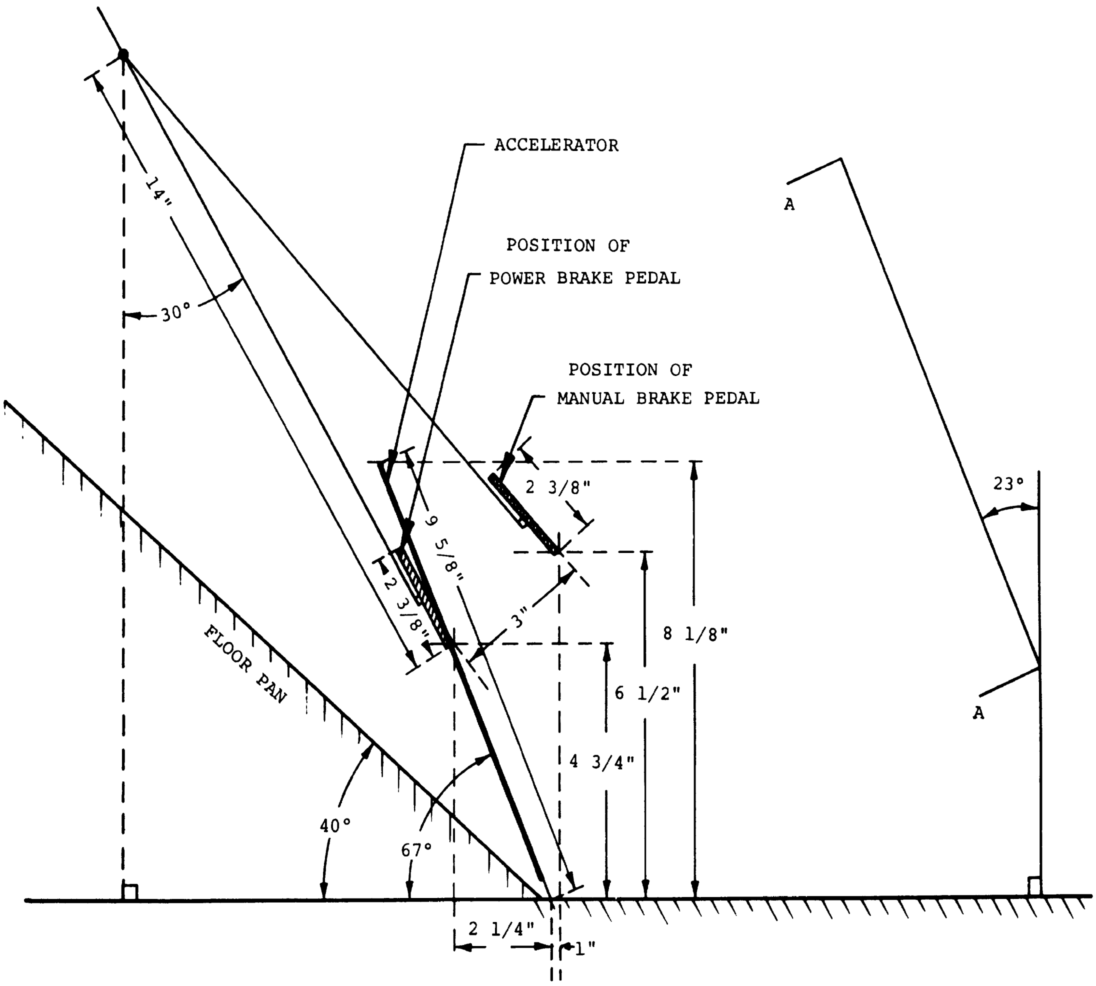



Figure 4.3. Longitudinal location of manual and power brake pedals and accelerator.

VIEW AA

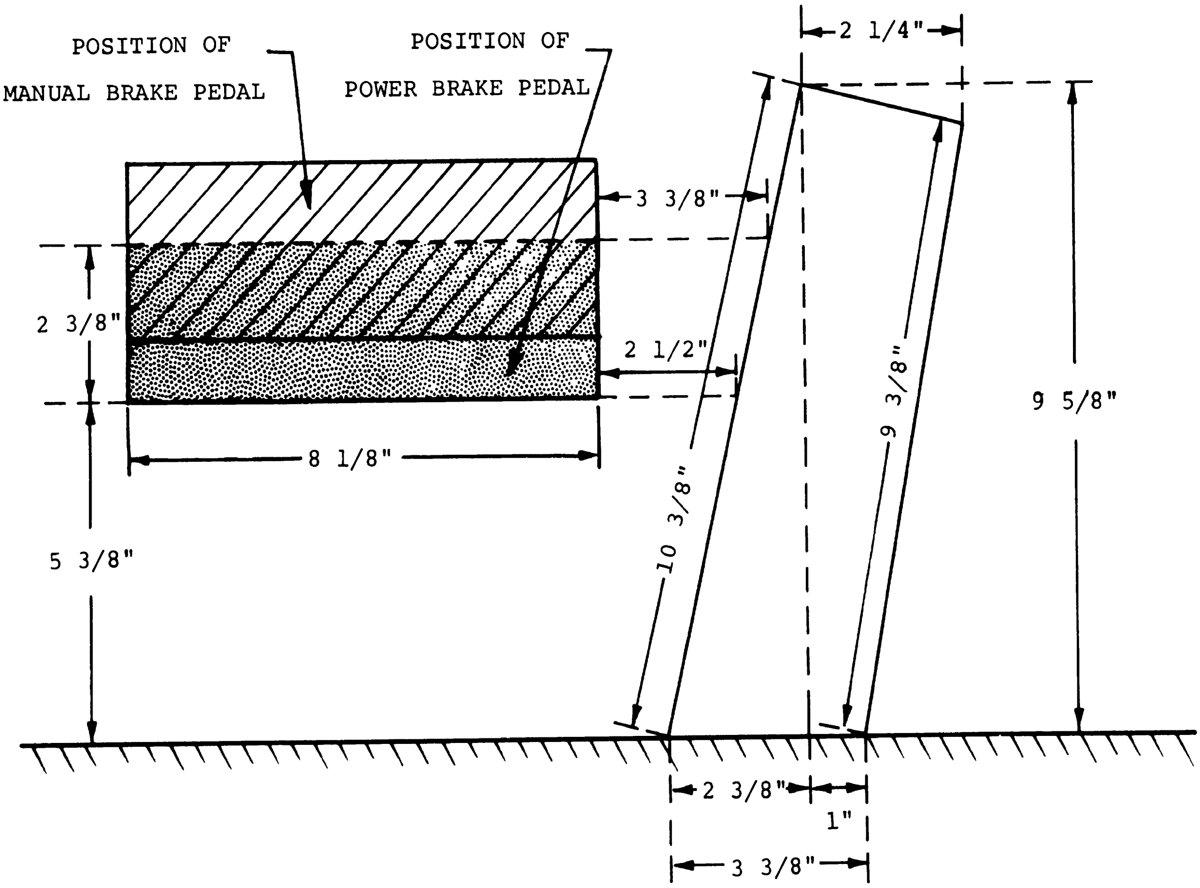



Figure 4.4. Height and location of power and manual brake pedal and accelerator.

configuration existing in the manual and power-brake mode.

PROCEDURE. The test vehicle was assigned as a pool car to the University of Michigan Transporation Department. This procedure assured a variety of drivers and driving conditions. Drivers who obtained this vehicle were asked to record on a log: name; date of trip; approximate miles driven in city, country, and expressway conditions; odometer readings at start and end of trip; approximate times of start and end of trip; and foot used for braking. The specific instructions that were given appear in Appendix III.

SUBJECTS. Forty-four subjects, all University of Michigan employees, drove the instrumented vehicle. Twenty-eight people drove the vehicle with standard brakes and 16 people drove the vehicle with power brakes installed.

## RESULTS

A total of 6255 miles were logged during which 8934 brake applications were made. The distributions of measured peak decelerations are shown in Tables 4.1 and 4.2 as obtained with standard brakes and power brakes respectively. The cumulative percent distributions of peak deceleration are shown in Figure 4.5 for both brake configurations. Note that the curves have a similar shape and overlap. With either brake configuration, a 0.3 g deceleration, or greater, occurred about 3.80 percent of the time; a 0.4 g deceleration, or greater, occurred 0.57 percent of the time, and a deceleration of 0.5 g, or greater, occurred about 0.10 percent of the time. Approximately 80 percent of the driving (miles) was done on expressways, 16 percent in the city and about 3 percent on rural roads.

These results agree quite well with those found both by Carpenter (1955) and by Kummer and Meyer (1967). Whereas the latter investigators found the brake to be depressed about once per mile on a 276 mile cross-country trip, these data as

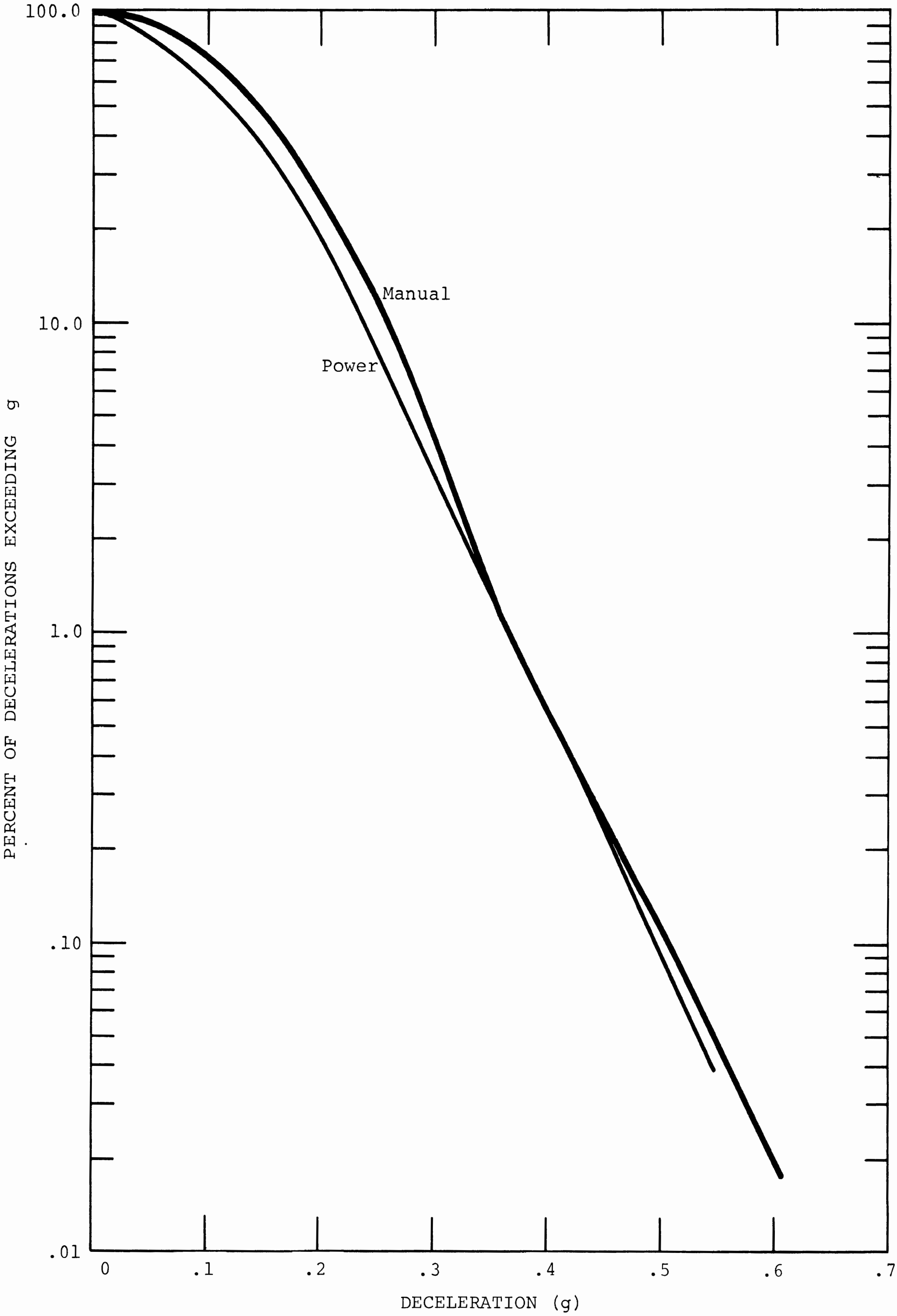



Figure 4.5. Cumulative percent of decelerations for manual and power brakes.

TABLE 4.1. CUMULATIVE FREQUENCY DISTRIBUTION OF PEAK DECELERATIONS (MANUAL BRAKES)

| Interval (g) | Frequency | Percent | Cumulative Percent | 100-Cumulative Percent |
|---|---|---|---|---|
| .60 - .64 | 1 | .016 | 100.000 | .0 |
| .55 - .59 | 4 | .063 | 99.984 | .016 |
| .50 - .54 | 3 | .047 | 99.921 | .079 |
| .45 - .49 | 6 | .095 | 99.874 | .126 |
| .40 - .44 | 22 | .347 | 99.779 | .221 |
| .35 - .39 | 46 | .725 | 99.432 | .568 |
| .30 - .34 | 202 | 3.183 | 98.707 | 1.293 |
| .25 - .29 | 497 | 7.832 | 95.524 | 4.476 |
| .20 - .24 | 1,034 | 16.294 | 87.692 | 12.308 |
| .15 - .19 | 1,395 | 21.982 | 71.398 | 28.602 |
| .10 - .14 | 1,544 | 24.330 | 49.416 | 50.584 |
| .05 - .09 | 1,255 | 19.776 | 25.086 | 74.914 |
| .00 - .04 | 337 | 5.310 | 5.310 | 94.690 |
| .00 | N=6,346 | | | 100.000 |

| | Miles | Percent |
|---|---|---|
| City Driving (40 MPH - ) | 633 | 14.88 |
| X-Way Driving | 3,332 | 78.33 |
| Country Driving (40 MPH + ) | 289 | 6.79 |
| | 4,254 | 100.00 |

TABLE 4.2. CUMULATIVE FREQUENCY DISTRIBUTION OF PEAK DECELERATIONS (POWER BRAKES)

| Interval (g) | Frequency | Percent | Cumulative Percent | 100-Cumulative Percent |
|---|---|---|---|---|
| .55 - .59 | 1 | .039 | 100.000 | .0 |
| .50 - .54 | 3 | .116 | 99.961 | .039 |
| .45 - .49 | 1 | .039 | 99.845 | .155 |
| .40 - .44 | 10 | .386 | 99.806 | .194 |
| .35 - .39 | 20 | .773 | 99.420 | .580 |
| .30 - .34 | 70 | 2.705 | 98.647 | 1.353 |
| .25 - .29 | 140 | 5.409 | 95.942 | 4.058 |
| .20 - .24 | 297 | 11.476 | 90.533 | 9.467 |
| .15 - .19 | 451 | 17.427 | 79.057 | 20.943 |
| .10 - .14 | 628 | 24.266 | 61.630 | 38.370 |
| .05 - .09 | 665 | 25.695 | 37.364 | 62.636 |
| .00 - .04 | 302 | 11.669 | 11.669 | 88.331 |
| .00 | N=2,588 | | | 100.000 |

| | Miles | Percent |
|---|---|---|
| City Driving (40 MPH - ) | 361 | 17.96 |
| X-Way Driving | 1,650 | 82.04 |
| Country Driving (40 MPH + ) | 0 | 0 |
| | 2,011 | 100.00 |

accumulated from a cross-section of drivers show 1.4 depressions of the pedal per mile.

DISCUSSION

Notwithstanding the significant difference in deceleration/ pedal force gain, the frequency distributions of decelerations obtained for the two brake configurations are almost identical. This result suggests that drivers adapt very well to different braking systems and that braking levels adopted by drivers are independent of the design parameters of the brake system.

It should be noted that the circumstances under which a car from the motor pool is requested affect the choice of roads traveled. Most of the miles put on pool cars represent business trips to other cities in Michigan and adjoining states. This usage results in more freeway driving than is probably done with the normal family car. It seems reasonable to expect that in city driving there would be a greater frequency of high decelerations and more brake applications per mile.

In view of their consistency, these data will be considered to be characteristic of the peak braking deceleration levels that can be expected to occur in the driving conditions represented in the survey. Accordingly, it appears reasonable to utilize the curves presented in Figure 4.5 in the Failure Analysis phase of this study.

# 5. FAILURE ANALYSIS

## INTRODUCTION

The pedal force required to decelerate a motor vehicle at a given rate is a function of a number of design parameters whose final selection and implementation are governed by a variety of design compromises. It is not our purpose here to review the process by which these compromises are reached but rather to consider how the effectiveness of the brake system (i.e., the deceleration/pedal force relationship) is modified if a partial failure should occur within the system.

Three categories of failures are considered in this failure analysis:

1. Loss of line pressure in one-half of a split or dual braking system.
2. Loss of vacuum boost in a power boost element.
3. Loss of effectiveness exhibited by an overheated brake (fade).

Each of these partial failure modes are considered and evaluated with respect to their influence on vehicle braking performance and with respect to the resulting consequences for safety, namely the ability of drivers to achieve their desired levels of deceleration.

## FAILURE MODES

LINE PRESSURE FAILURE. A standard dual braking system. with or without power boost, shall be analyzed. A tandem or dual master cylinder with a front- and rear-axle split (in conformance with MVSS 105) is assumed.

Given a loss of pressure in either the front-brake line or in the rear line, the mechanics of the braking process yields that

$$PF_R = a \frac{PF_o}{a_o} \frac{W}{W_o} \frac{1}{\Phi}$$

$$PF_F = a \frac{PF_o}{a_o} \frac{W}{W_o} \frac{1}{1-\Phi}$$

where

$PF_R$ = pedal force with rear system only operative

$PF_F$ = pedal force with front system only operative

$\Phi$ = rear axle brake force divided by total brake force for both front and rear systems operational

$a$ = deceleration, g units

$\frac{PF_o}{a_o}$ = pedal force/deceleration ratio for the unloaded vehicle condition (curb weight plus driver)

$\frac{W}{W_o}$ = loaded to unloaded vehicle weight ratio

A typical value for the ratio of $PF_o/a_o$ for cars without vacuum assist is 134, (Strien, 1968) while $W/W_o$ for domestic cars ranges from 1.13 to 1.18 (Automotive Industries, 1969). The brake-force distribution ranges from $\Phi = 0.30$ (e.g., the Lincoln) to $\Phi = 0.55$ (e.g., the Corvair). However, more than 90 percent of American cars have a brake-force distribution of $\Phi = 0.40$ (Automotive Industries, 1969). Table 5.1 summarizes the pedal force to deceleration ratios computed for various loading and failure conditions using these typical values. The highest values of deceleration/pedal force ratio obtain when the front hydraulic line fails in the loaded vehicle. A typical result is plotted in Figure 5.1, showing the large influence of brake-force distribution on the pedal force required to achieve a given deceleration when the front brakes are inoperative.

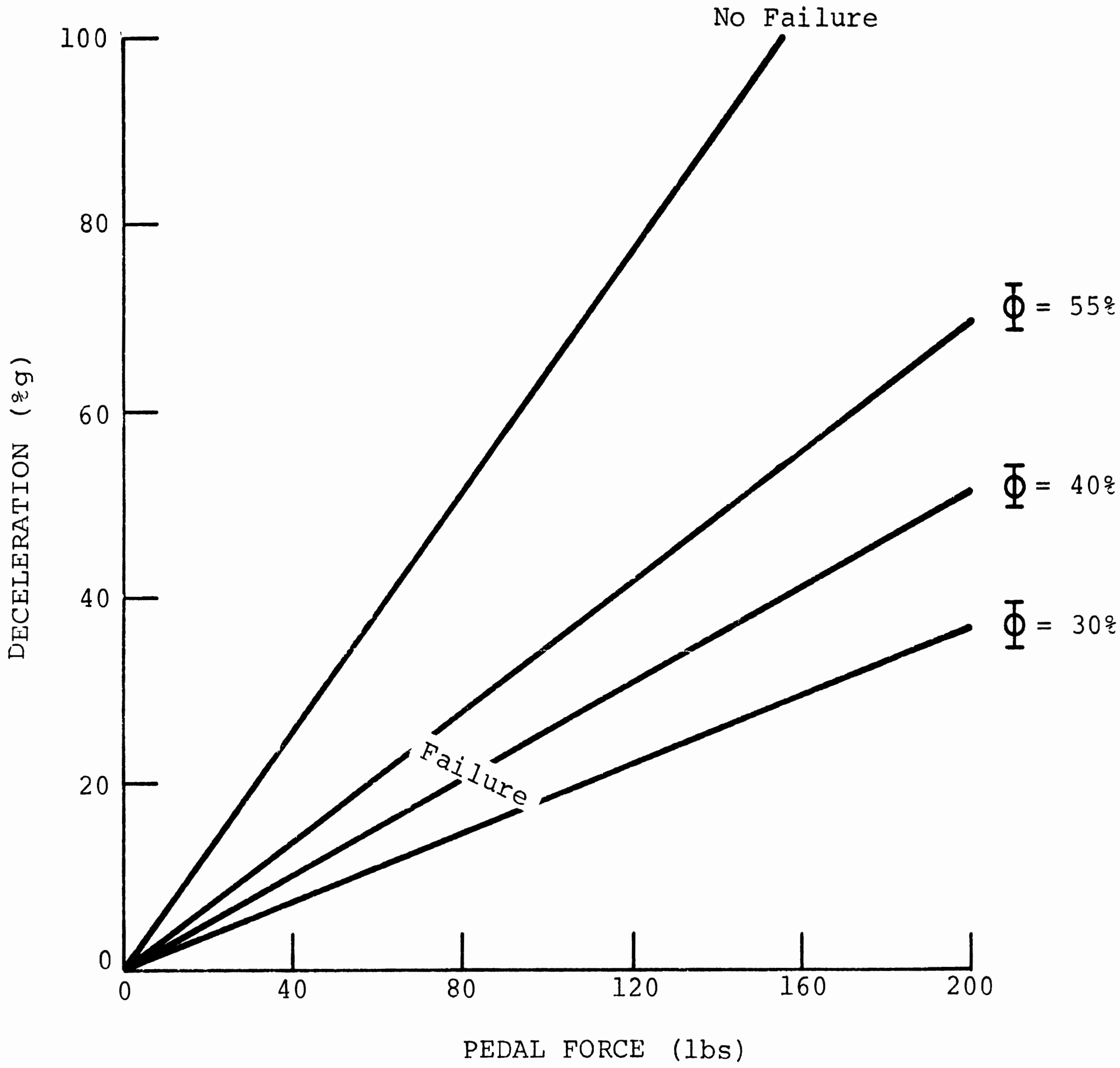



Figure 5.1. Deceleration/pedal force for a loaded passenger car without vacuum assist: front brakes operative and inoperative.

TABLE 5.1 TYPICAL DECELERATION/PEDAL FORCE RATIOS HYDRAULIC LINE FAILURES FOR VEHICLES WITHOUT POWER BOOST

| Loading Condition | No Failure $PF/a$ lbs/g | Front Line Failure $PF_R/a$ lbs/g | | | Rear Line Failure $PF_F/a$ lbs/g | | |
|---|---|---|---|---|---|---|---|
| | AVERAGE | MINIMUM | AVERAGE | MAXIMUM | MINIMUM | AVERAGE | MAXIMUM |
| Unloaded (Curb Weight and Driver) | 134 | 244 | 336 | 448 | 192 | 224 | 298 |
| Loaded | 154 | 282 | 387 | 518 | 221 | 268 | 345 |

$PF$ = Pedal force, lbs, front and rear system operational

$a$ = Deceleration, g-units

$PF_R$= Pedal force, lbs, rear system only operational

$PF_F$= Pedal force, lbs, front system only operational

For brake systems containing a power bo st element, the relationship between pedal force and deceleration can be approximated by two straight lines, one for the pedal forces developed below the saturation point of the booster and a second for the pedal forces that are developed above the saturation point. As before, the braking process yields that

$$PF_R = a \frac{PF_o}{a_o} \text{ assist } \frac{W}{W_o} \frac{1}{\Phi} \qquad a \leq a$$

$$PF_R = a_s \frac{PF_o}{a_o} \text{ assist } + \Delta a_m \frac{PF_o}{a_o} \text{ manual } \frac{W}{W_o} \frac{1}{\Phi} \qquad a >$$

$$PF_F = a\left(\frac{PF_o}{a_o}\right)_{assist} \frac{W}{W_o} \frac{1}{1-\Phi} \qquad a \leq$$

$$PF_F = \left[a_s \left(\frac{PF_o}{a_o}\right)_{assist} + \Delta a_m \left(\frac{PF_o}{a_o}\right)_{manual}\right] \frac{W}{W_o} \frac{1}{1-\Phi} \qquad a >$$

where

$a_s$ = deceleration at the saturation point

$\Delta a_m$ = increase in deceleration above the saturation point

For cars with vacuum assist, $PF_o \;/\; a_o \simeq 64$ lbs/g (Strien, 1968). When the booster is completely inoperative, $PF_o \;/\; a_o \simeq 460$ lbs/g, a gain that is much lower than that exhibited by cars that are not equipped with power assist. This result, in large measure, stems from the lower pedal lever ratio that is used in vehicles equipped with vacuum assist. Table 5.2 summarizes the results given by the above expressions, on assuming that $(PF_o\;/a_o)$ assist $\simeq 64$, $(PF_o\;/a_o)$ manual $\simeq 460$, and that the booster saturates at PF = 50 lb (a value computed on the basis of available technical data). As was true for the non-powered system, the largest values of pedal force are required when the front brake lines fail in loaded vehicle. A typical result is diagrammed in Figure 5.2.

In addition to the influence of line-pressure failures on the deceleration/pedal force relationship, there are additional implications with respect to pedal travel required and pedal travel available. It should be noted that the brake pedal acts through a linkage to move the pistons in the master cylinder, which in turn force hydraulic fluid through the lines to the individual wheel cylinders to actuate the brakes. Individual piston travel has to be designed such as to meet the fluid volume requirements at the front- and rear-axle wheel cylinders.

TABLE 5.2. TYPICAL DECELERATION/PEDAL FORCE CHARACTERISTICS. HYDRAULIC LINE FAILURE FOR VEHICLE WITH POWER BOOST (ASSUME POWER BOOST SATURATED AT PF = 50 LBS)

| | Loading Condition | No Failure | Front Line Failure | | | Rear Line Failure | | |
|---|---|---|---|---|---|---|---|---|
| | | PF lbs | $PF_R$ lbs | | | $PF_F$ lbs | | |
| | | AVERAGE | MINIMUM | AVERAGE | MAXIMUM | MINIMUM | AVERAGE | MAXIMUM |
| OPERATION BELOW BOOSTER SATURATION POINT | Unloaded (Curb Weight and Driver) | 64a | 116a | 160a | 214a | 92a | 107a | 142a |
| | Loaded $\frac{W}{W_O} = 1.15$ | 74a | 134a | 184a | 245a | 105a | 123a | 163a |
| OPERATION ABOVE BOOSTER SATURATION POINT | Unloaded (Curb Weight and Driver) | 460a-309 | 770a-286 | 1155a-311 | 1540a-310 | 656a-306 | 770a-310 | 1025a-311 |
| | Loaded $\frac{W}{W_O} = 1.15$ | 530a-308 | 970a-312 | 1330a-312 | 1760a-309 | 760a-312 | 920a-324 | 1184a-313 |

PF = Pedal force, lbs, front and rear operational

a = Deceleration, g-units

$PF_R$ = Pedal force, lbs, rear system only operational

$PF_F$ = Pedal force, lbs, front system only operational

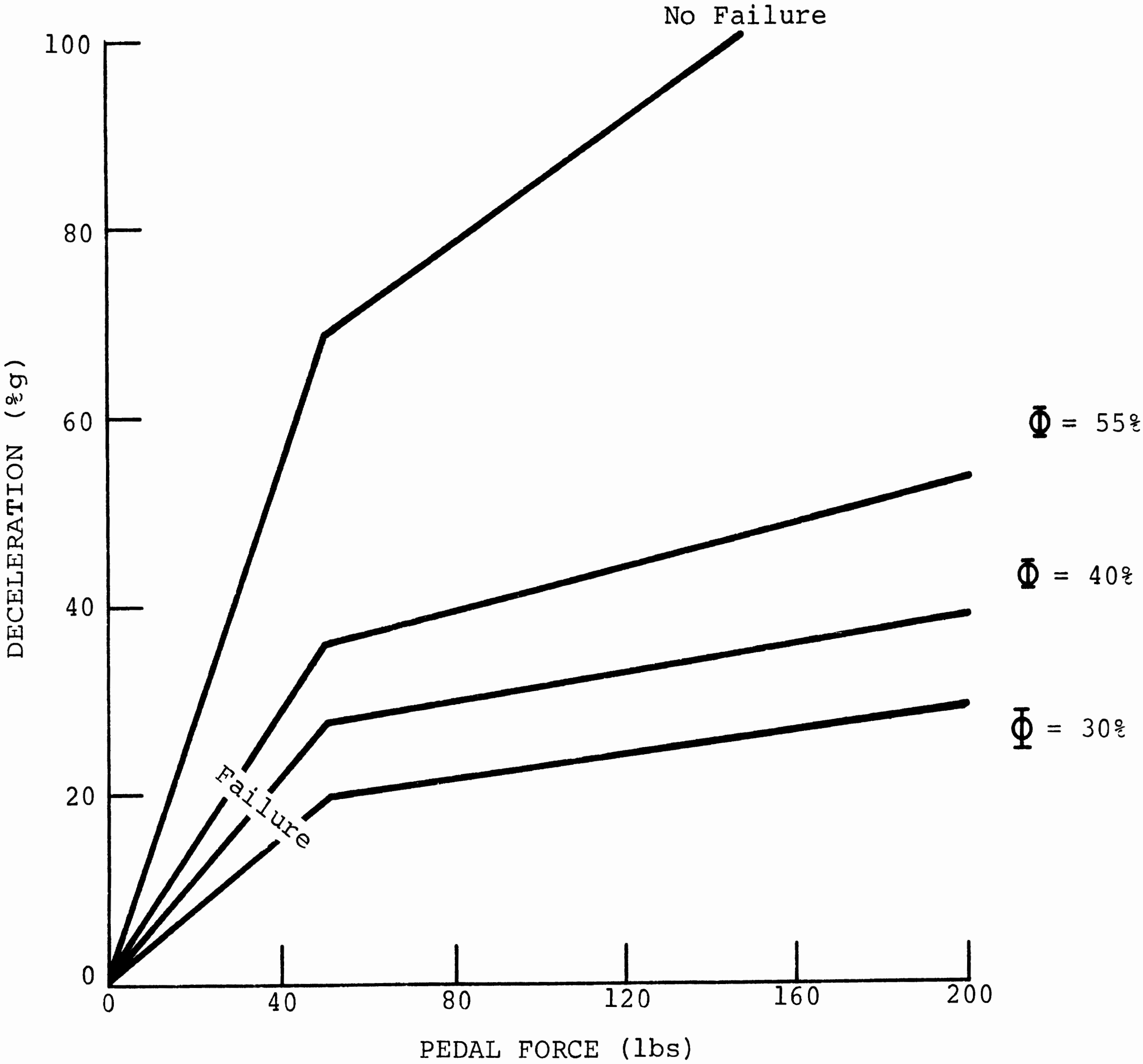



Figure 5.2. Deceleration/pedal force for a loaded passenger car with vacuum assist: front brakes operative and inoperative.

With the diameter of the wheel cylinders determined by the torque distribution, the fluid volume requirement becomes a function of the brake shoe or pad displacement necessary to account for hysteresis, lining compression, lining wear, and drum expansion.

For example, consider a car with a master-cylinder diameter of 3/4 inch. It has disc brakes on the front wheels with a wheel cylinder diameter of 1 3/4 inches. The rear wheels have 10 inch drum brakes with a wheel cylinder of 5/8 inch diameter. A piston displacement of 0.026 inches is considered adequate for the front disc brakes (Teves, 1960). The master-cylinder piston travel corresponding to the piston travel at the (four) front axle wheel cylinders is:

$$d_F = \frac{V_F}{A_{MC}} = \frac{4\ A_{WC}\ 0.026}{A_{MC}} = \frac{4 \times 2.4 \times 0.026}{0.441}$$

$$= 0.565 \text{ in.}$$

where

$V_F$ = volume of fluid

$A_{MC}$ = area of master cylinder

An additional piston travel of 0.08 inches is required to cover the port connecting the master cylinder and the reservoir (Teves, 1960). Thus the master-cylinder piston travel required to actuate the front brakes totals 0.645 inches.

The wheel-cylinder piston travel required for drum brakes can be approximated by the following relationship (Teves, 1960):

$$d_{wc} = 0.1 + 0.003 \times \text{drum dia. (in.)}$$

Therefore,

$$\left.d_{wc}\right)_R = 0.1 + 0.003 \times 10 = 0.13 \text{ in.}$$

and the master-cylinder piston travel necessary to actuate the rear brakes is given by:

$$d_R = \frac{V_R}{A_{MC}} = \frac{4 \times A_{WC} \times 0.13}{A_{MC}}$$

$$= \frac{4 \times 0.307 \times 0.13}{0.441} = 0.36 \text{ in.}$$

The total travel at the master cylinder is:

$$d = d_F + d_R = 0.925 \text{ in.}$$

With a pedal lever ratio of 3.2, this piston travel corresponds to a pedal travel of approximately 3.0 inches. In the case of a front axle failure the theoretical pedal travel required prior to building up pressure in the rear circuit becomes:

$$\frac{0.025 - 0.36}{0.925} \times 3.0 = 1.83 \text{ in.}$$

which distance is approximately 61 percent of the maximum travel.

The data employed in the illustrative calculation correspond to conditions of excellent brake adjustment (Teves, 1960). Since the majority of U.S. vehicles incorporate automatic brake adjustment, this is a realistic assumption. For poorer brake shoe adjustments, however, the pedal travel required to pressurize the rear brakes might well approach intolerable dimensions or may even take up the entire pedal travel available. These results indicate that the wisdom of the standard split (i.e., separate lines to the front to rear axle) is questionable in the case of a failure in the front brake lines. More effective splits (diagonal, horizontal, etc.) have been suggested by Vallin (1968).

POWER BOOST FAILURE. An analysis shall be made of systems that employ mechanical control of the vacuum assist since mechanically controlled boosters are more widely used.

Constraints influencing the design of hydraulic brakes yield the following approximate expression for the work into a master cylinder when a deceleration of 0.85 is required (Teves, 1960):

$$p_h V = (0.24 \text{ to } 0.29)W$$

where

$p_h$ = hydraulic pressure (psi)

$V$ = maximum fluid volume displaced by the master cylinder piston ($in^3$)

$W$ = vehicle weight (lb)

The hydraulic work into the master cylinder is the sum of the work done by the booster and the pedal. Thus:

$$p_h V = F_B \chi + PFy$$

where

$F_B$ = Effective booster force (lb)

$PF$ = Pedal force (lb)

$\chi$ = Effective Master cylinder travel (in)

$y$ = Pedal Travel (in)

In the case of a booster failure, the first term in the equation, $F_B\chi$, is equal to zero and only the work input of the driver, $P_F y$, produces a vehicle deceleration. On analyzing an 8-inch, single-diaphram vacuum booster, the results presented in Figure 5.3 are obtained.

In this figure, booster-exit force into the master cylinder is plotted versus pedal force times pedal-lever ratio, $F_p$ x ip.

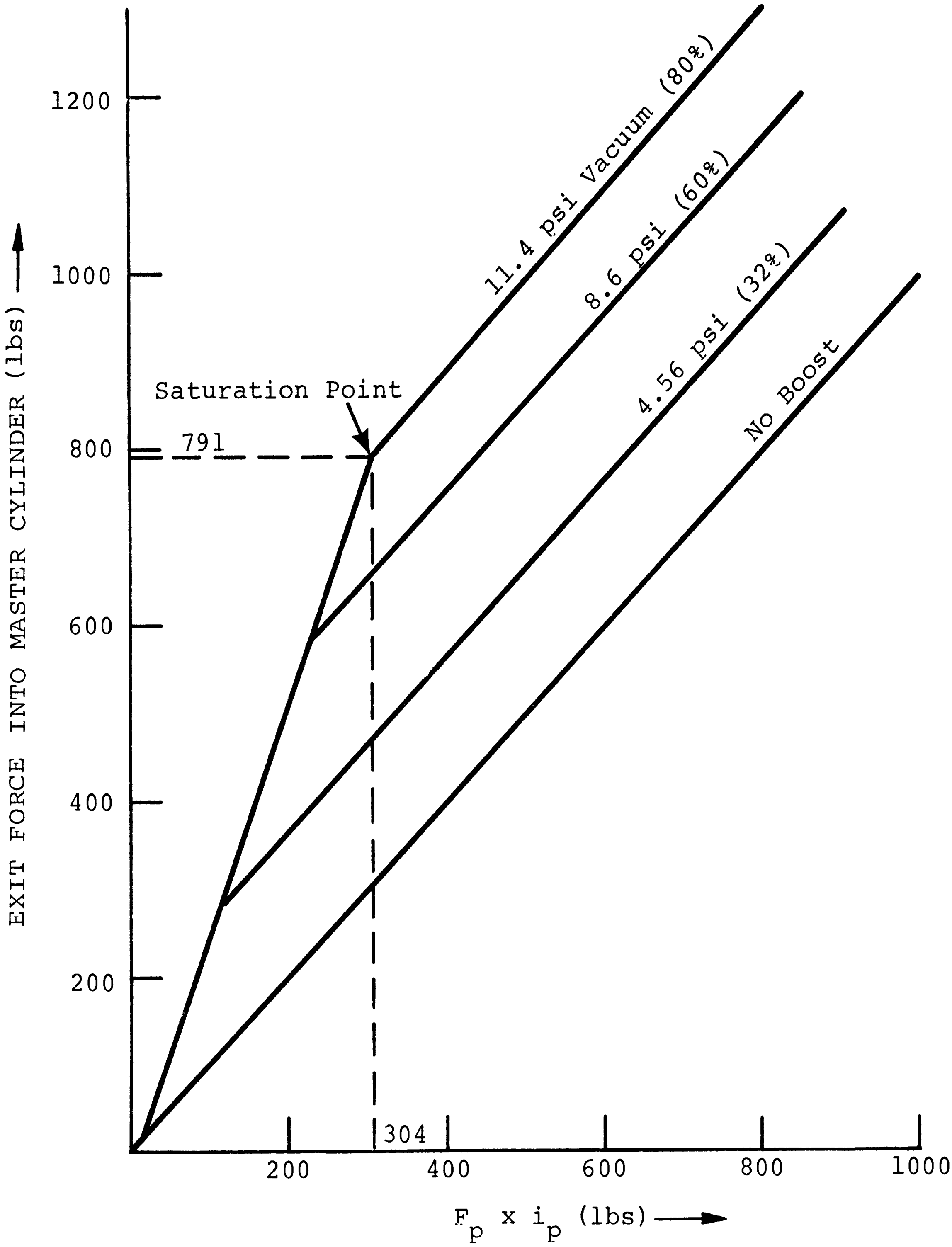



Figure 5.3. Power boost characteristics.

As can be seen from the diagram, maximum boost assistance is obtained at 791 lb. For decelerations requiring higher brake efforts, the additional work input has to come from the driver. The diagram also demonstrates the influence of partial vacuum and zero boost.

In order to show the influence of total- or partial-boost failure on the deceleration/pedal force relationship, Figure 5.4 was prepared. Typical dimensions were assumed for the elements in a brake system. The following observations can be made with respect to various levels of power boost failure:

1. No boost--to produce a deceleration of 90 percent g, a pedal force of approximately 270 lb is required. A deceleration of only 0.32 g is produced by a pedal force of 100 lb.
2. Thirty-two percent boost--the deceleration produced by a pedal force of 100 lb is 0.52 g. A deceleration of 0.90 g requires a pedal force of about 215 lb.
3. Sixty percent boost--the deceleration produced by a 100 lb pedal force is 0.76 g. A deceleration of 0.90 g requires a pedal force of about 150 lb.

BRAKE FADE. If a vehicle is subjected to a series of severe stops in rapid succession, it will be observed that for each successive stop a higher pedal force is necessary to maintain a specified deceleration level (SAE, 1967). This phenomenon is called fade. The phenomenon can be analyzed and predictions of the increase in pedal force can be made (Strien, 1949) provided that the variation of the brake factor (BF) as a function of the friction coefficient of the lining is known and that the variation of friction coefficient with velocity, pressure, temperature is known (Kruegel & Weber, 1964; Newcomb, 1960; Dorner, 1963).

The relationship between pedal force (PF) and brake factor is given by the following relationship on assuming a brake system without vacuum assist:

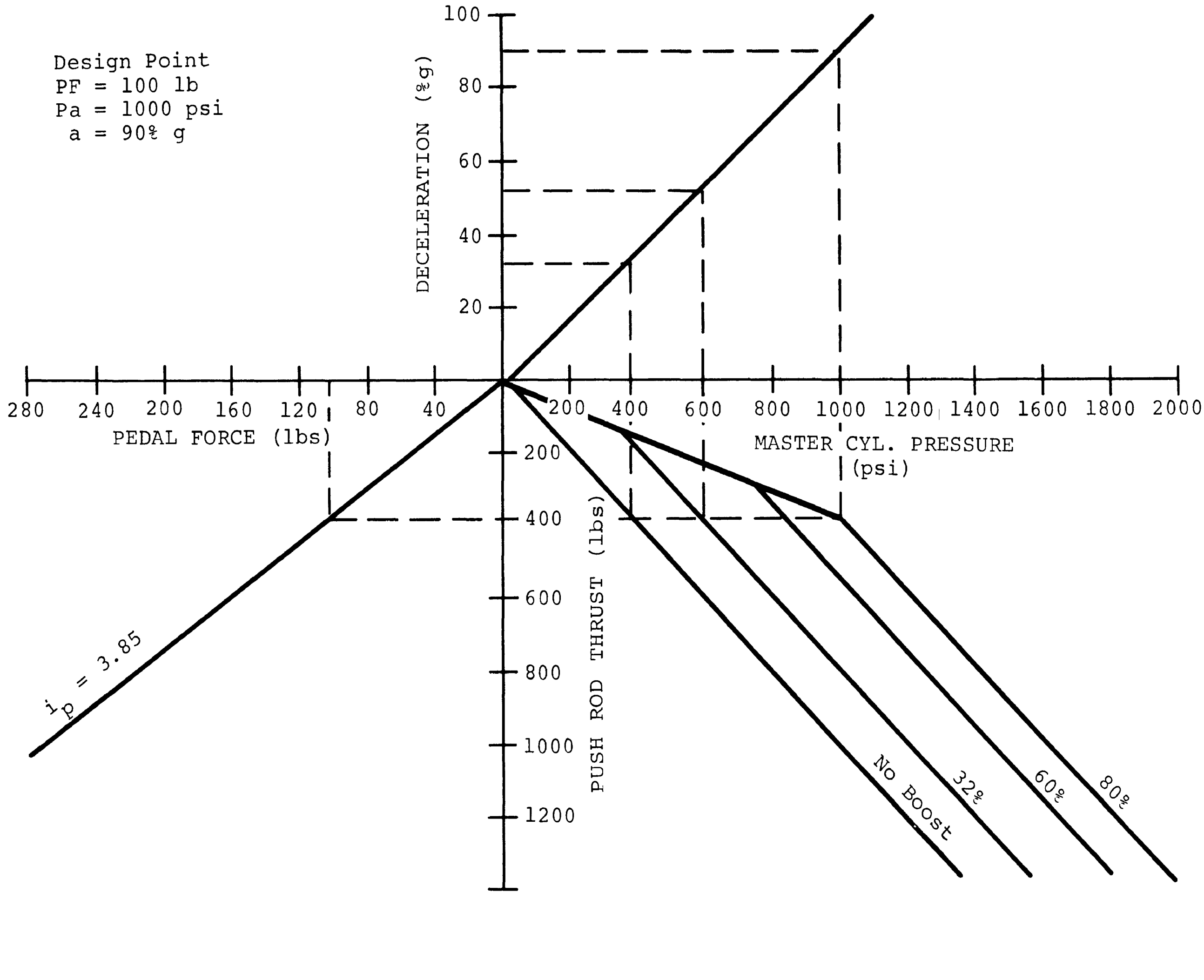



Figure 5.4. Braking performance diagram.

$$PF = \frac{p_h \; A_{MC}}{i_p \; \eta}$$

where

$PF$ = pedal force

$A_{MC}$ = master cylinder area

$i_p$ = pedal lever ratio

$h$ = hydraulic efficiency

$p_h$ = hydraulic line pressure

$$= \frac{W \; a \; R}{[(A_{WC} \; BF \; r)_F + (A_{WC} \; BF \; r)_R] \; 2 \; \eta}$$

and

$A_{WC}$ = wheel cylinder area

$a$ = deceleration in g units

$BF$ = brake factor, defined as the ratio of the summation of the circumferencial forces on the friction surface divided by the actuating force in the wheel cylinder.

$R$ = effective tire radius

$r$ = effective drum or disc radius

To illustrate the change in brake effectiveness due to fade, Limpert and Planck (1964) made three successive high speed 0.8 g stops with an instrumented vehicle. The vehicle was equipped with disc brakes on the front and rear axle ($W$ = 2000 lb, $A_{wc\ F}$ = 2.8 in$^2$, $A_{wc\ R}$ = 1.76 in$^2$, $r/R$ = 0.40, $A_{MC}$ = 0.64 in$^2$, $i_p$ = 5). Figure 5.5 shows the hydraulic pressures and the pedal forces measured in the non-faded condition and in each of the three high speed stops. The variations in brake factor as exhibited by the change in slope of line pressure versus deceleration is due to

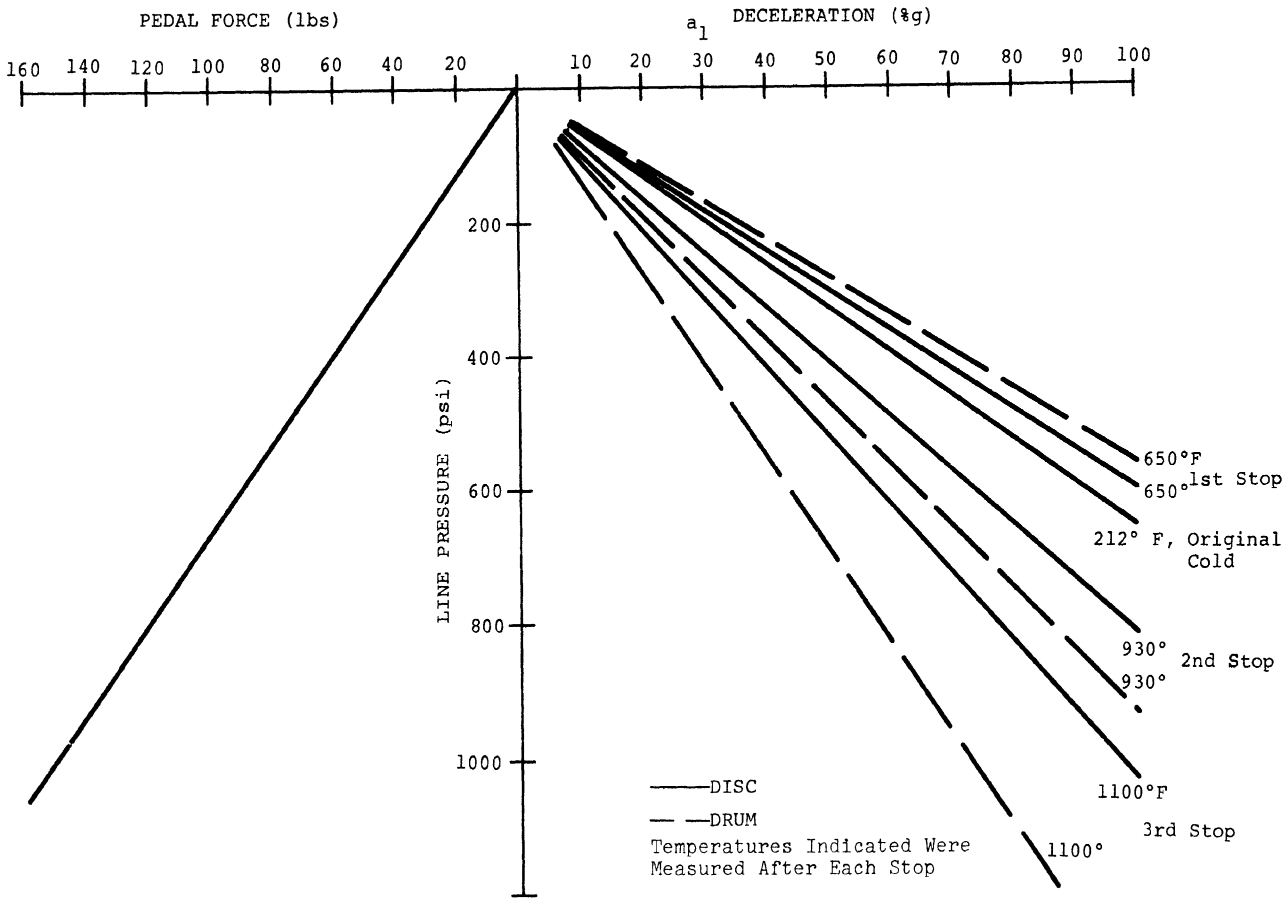



Figure 5.5. Fade-effectiveness diagram.

temperature increase which ranged from 212 degrees to 1100 degrees F as measured on the surface of the front discs. Examination of Figure 5.5 shows that the pedal force decreases after making the cold stop as a result of an increase in the rear disc brake factor. In the successive high speed stops, the pedal force required for an 0.8 g stop increases to 125 lb as compared to the 80 lb that are required when the brakes are cold.

When the same vehicle was equipped with drum brakes

$$(A_{wc})_F = 0.615 \text{ in}^2, \; A_{wc})_R = 0.39 \text{ in}^2)$$

the fade effects were greater, as is shown in Figure 5.5. The pedal force required for an 0.8 g stop increased from 80 lb to 165 lb after completing three high speed stops.

Deceleration/pedal force gain data obtained in compliance tests (1968 models) are summarized in Figures 5.6, 5.7, 5.8 and Tables 5.3 and 5.4. The plotted distributions of deceleration/pedal-force gain are derived from tests on 43 vehicles, 24 of which had power assisted brakes. As would be expected, the deceleration/pedal force gain for cars with power brakes is considerably higher than that for those without. Further, the power braked vehicles have gains that are highly variable compared to the gains built into vehicles with manual brakes. Note that in the faded condition, the gain on the power braked cars was higher than that exhibited by manually braked vehicles in an unfaded condition. It should also be noted that the fade induced in a single stop from 80 mph produces a decrease in pedal force gain nearly as great as that resulting from the standard SAE fade test procedure.

FACTORS INFLUENCING THE PARTIAL FAILURE OF BRAKE SYSTEMS.

Data showing the frequency of brake system component failure is nonexistent. In view of this information gap, it appears

TABLE 5.3. COMPLIANCE TESTS, MVSS-105 FOR VEHICLES WITH NON-POWER BRAKES

| VEHICLE NO. | BRAKE TYPE* | Average Values From Effectiveness Tests 30 MPH P.F. | 30 MPH P.F. GAIN | 60 MPH P.F. | 60 MPH P.F. GAIN | 80 MPH P.F. | 80 MPH P.F. GAIN | FIRST FADE TEST P.F. | FIRST FADE TEST P.F. GAIN |
|---|---|---|---|---|---|---|---|---|---|
| 1968 Plymouth Valiant | 1 | 68 | .29 | 73 | .27 | 79.5 | .25 | 80.5 | .19 |
| 1968 Ford Mustang | 1 | 46.5 | .43 | 48.5 | .41 | 48 | .42 | 65 | .23 |
| 1968 Plymouth Belvedere | 1 | 54 | .37 | 61 | .33 | 75.5 | .27 | 67 | .22 |
| 1968 Pontiac Tempest Safari | 1 | 53 | .38 | 71.5 | .28 | 99.5 | .20 | 70.5 | .21 |
| 1968 Checker Marathon | 1 | 77.5 | .26 | 77 | .26 | 112.5 | .18 | 99 | .15 |
| 1968 American Rebel 550 | 1 | 63.5 | .32 | 63.5 | .32 | 98.5 | .20 | 96 | .16 |
| 1968 Plymouth Suburban | 1 | 80.5 | .25 | 98.5 | .20 | 133.5 | .15 | 126.5 | .12 |
| 1968 Plymouth Satellite | 1 | 62 | .32 | 71.5 | .28 | 84.5 | .24 | 76 | .20 |
| 1968 Datson SRL 311 | 2 | 70 | .29 | 69.5 | .29 | 74 | .27 | 60.5 | .25 |
| 1968 Plymouth Sport Fury | 1 | 52 | .39 | 72.5 | .28 | 88.5 | .23 | 67 | .22 |
| 1968 MGB Mark II Rd. | 2 | 70 | .29 | 68.5 | .29 | 64.5 | .31 | 57 | .26 |
| 1968 Chevy II Nova | 1 | 59.5 | .34 | 73.5 | .27 | 100 | .20 | 86 | .17 |
| 1968 Ford Fairlane 500 | 1 | 69.5 | .29 | 82 | .24 | 99 | .20 | 86.5 | .17 |
| 1968 Ford Galaxie 500 | 1 | 63 | .32 | 70 | .29 | 83 | .24 | 79 | .19 |
| 1968 Chevelle Malibu | 1 | 57.5 | .35 | 75.5 | .27 | 101 | .20 | 84.5 | .18 |
| 1968 Ford Falcon Station Wagon | 1 | 58.5 | .34 | 68.5 | .29 | 75 | .27 | 65 | .23 |
| 1968 Chevrolet Impala | 1 | 50.5 | .40 | 58.5 | .34 | 75 | .27 | 72.5 | .21 |
| 1968 Buick Skylark | 1 | 28.5 | .70 | 35.5 | .56 | 61 | .33 | 45 | .33 |
| 1968 Buick Special | 1 | 38.5 | .52 | 48.5 | .41 | 76 | .26 | 47.5 | .32 |

*1 - Front Drum
Rear Drum

2 - Front Disc
Rear Drum

3 - Front Disc
Rear Disc

## TABLE 5.4. COMPLIANCE TESTS, MVSS-105 FOR VEHICLES WITH POWER BRAKES

| VEHICLE NO. | BRAKE TYPE* | Average Values From Effectiveness Tests: 30 MPH P.F. | 30 MPH P.F. GAIN | 60 MPH P.F. | 60 MPH P.F. GAIN | 80 MPH P.F. | 80 MPH P.F. GAIN | FIRST FADE TEST P.F. | FIRST FADE TEST P.F. GAIN |
|---|---|---|---|---|---|---|---|---|---|
| 1968 Lincoln Continental | 2 | 20.5 | .98 | 23.5 | .85 | 28.5 | .70 | .33 | .46 |
| 1968 Mercury Colony Park | 2 | 22 | .91 | 24.5 | .82 | 28.5 | .70 | 29.5 | .51 |
| 1968 Plymouth Road Runner | 1 | 61 | .33 | 77.5 | .26 | 87 | .23 | 76.5 | .20 |
| 1968 Rover 2000 TC | 3 | 36 | .56 | 35 | .57 | 35 | .57 | 31.5 | .48 |
| 1968 Mercury Cyclone | 2 | 29.5 | .68 | 33 | .61 | 35 | .57 | 34 | .44 |
| 1968 Buick Riviera | 1 | 30.5 | .66 | 29.5 | .68 | 43 | .47 | 30 | .50 |
| 1968 Pontiac Grand Prix | 1 | 21.5 | .93 | 30.5 | .66 | 50 | .40 | 29.5 | .51 |
| 1968 Dodge Polara | 1 | 61 | .33 | 84 | .24 | 119.5 | .17 | 87.5 | .17 |
| 1968 Chrysler Imperial | 2 | 27.5 | .73 | 31.5 | .64 | 41 | .49 | 29.5 | .51 |
| 1968 Oldsmobile Delta | 1 | 28.5 | .70 | 36.5 | .55 | 44 | .45 | 35.5 | .42 |
| 1968 Buick Le Sabra | 1 | 23.5 | .85 | 28.5 | .70 | 38 | .53 | 27.5 | .55 |
| 1968 Oldsmobile Delmont 88 | 1 | 24.5 | .82 | 29.5 | .68 | 41.5 | .48 | 31.5 | .48 |
| 1968 AMC Rebel 770 | 1 | 39.5 | .51 | 59.5 | .34 | 100 | .20 | 70 | .21 |
| 1968 Volvo 1445 | 3 | 48.5 | .41 | 46.5 | .43 | 43 | .47 | 31.5 | .48 |
| 1968 Plymouth Fury II | 1 | 33.5 | .60 | 46.5 | .43 | 83 | .24 | 63 | .24 |
| 1968 Dodge Charger | 1 | 20 | 1.00 | 33 | .61 | 63 | .32 | 46 | .33 |
| 1968 Ambassador SST | 1 | 35.5 | .56 | 43.5 | .46 | 60 | .33 | 53 | .28 |
| 1968 Chrysler 300 | 1 | 23 | .87 | 31 | .65 | 106 | .19 | 50.5 | .30 |
| 1968 Pontiac LeMans | 1 | 29 | .69 | 36.5 | .55 | 48 | .42 | 37.5 | .40 |
| 1968 Ford Galaxie | 2 | 31 | .65 | 36 | .56 | 34.5 | .58 | 35 | .43 |
| 1968 Ford Thunderbird | 2 | 26.5 | .75 | 31.5 | .64 | 32.5 | .62 | 32.5 | .46 |
| 1968 Oldsmobile Cutlass | 1 | 20 | 1.00 | 24.5 | .82 | 46 | .43 | 27.5 | .55 |
| 1968 Ford XL | 2 | 30 | .67 | 30 | .67 | 32.5 | .62 | 35 | .43 |
| 1968 Mercury Montclair | 2 | 35 | .57 | 34 | .59 | 45 | .44 | 37.5 | .40 |

*1 - Front Drum Rear Drum    2 - Front Disc Rear Drum    3 - Front Disc Rear Disc

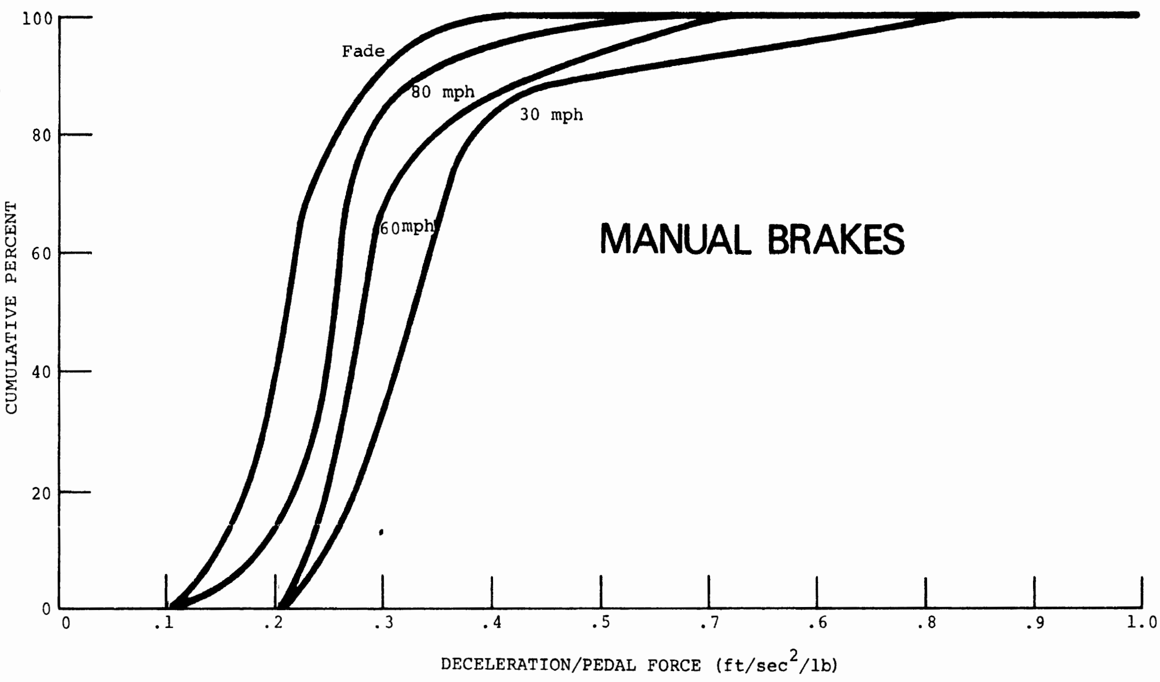



Figure 5.6. Cumulative percent of vehicles with lower gain: Manual brakes.

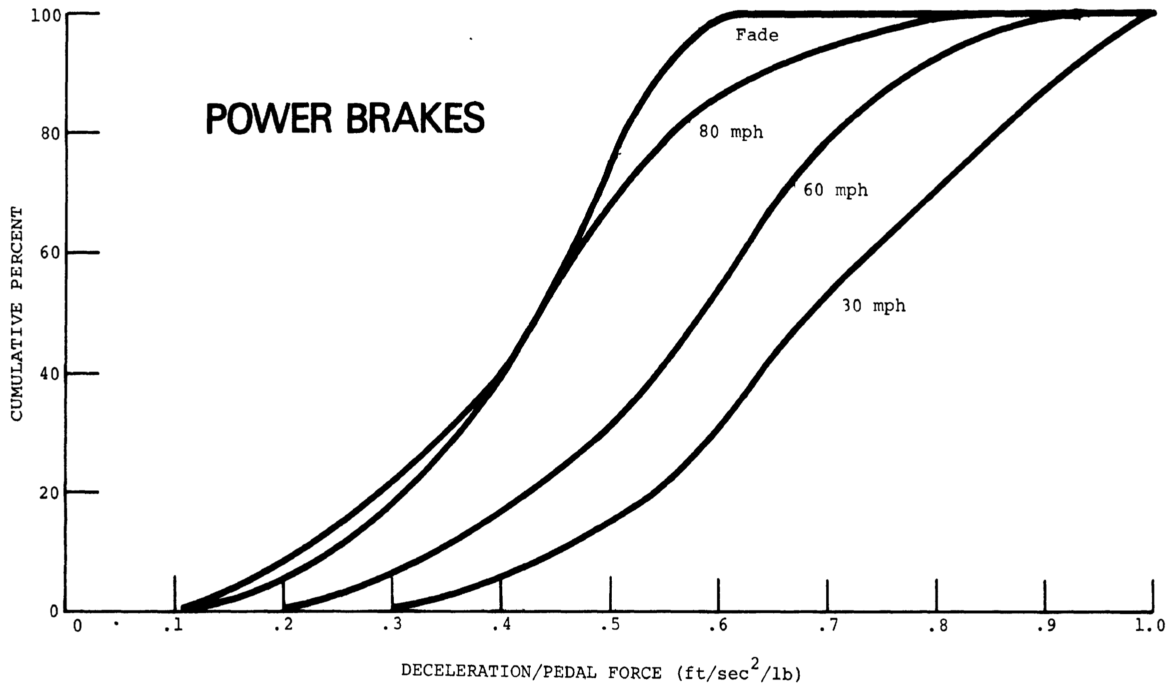



Figure 5.7. Cumulative percent of vehicles with lower gain : Power brakes.

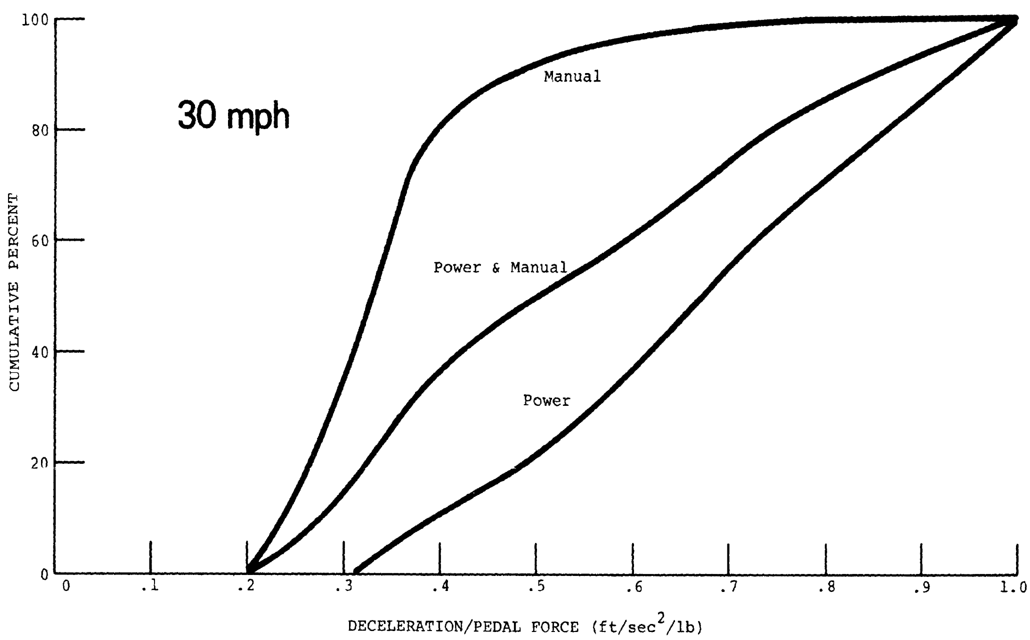



Figure 5.8. Cumulative percent of vehicles with lower gain: 30 mph.

appropriate to consider and discuss the various factors which presumably influence the occurrence of a partial failure in a brake system. Note that degradation of brake performance due to thermal effects is an operational problem, namely this partial failure is very much dependent on brake system usage. Under non-faded conditions, brake system failure is likely to occur only if:

1. Parts are defective (on either new or used vehicles).
2. Parts become degraded due to wear or corrosion.

Under normal driving conditions, it is not likely that new components will fail. It is clear that linings and drums that have been in use over a long period of time are more likely to fail than new ones. The same conclusion can be drawn with respect to master and wheel-cylinder housings, pistons and cups. Since wear in these components may produce a decrease in brake effectiveness or a complete loss of braking capability, manufacturers generally will specify tolerances on the wear dimensions of cylinders and drums.

It can be speculated that increases in traffic density will, over time, cause increased wear in master cylinders. It appears logical to conclude that the frequency of light brake applications will increase with an increase in traffic. These light brake applications require very little movement of the brake pedal. Consequently the displacement of the master cylinder piston is small, either causing the master cylinder cup to slide frequently over the port connecting the brake-fluid reservoir with the master cylinder, or causing the cup to operate right at the port during light brake applications. This situation can cause excessive wear and grooving of the cup, resulting in internal leakage within the brake system.

It is clear that degradation of brake components resulting from corrosion or aging can be a factor in causing brake failures. When these factors are involved, one would expect brake system failures to occur during severe brake application, i.e., during driving maneuvers requiring large pedal forces that severely stress the entire brake system. In general, it seems reasonable to conclude that failure due to excessive wear is most likely to occur at points of sliding motion such as exists at the brakes, master cylinder, wheel cylinders, and vacuum booster reaction unit. Failure due to corrosion and aging is likely to involve brake lines, brakeline hoses, and hoses connecting the vacuum booster with the intake manifold.

## CONSEQUENCES OF FAILURE

EFFECTS ON VEHICLE PERFORMANCE. The major effect of the three failure modes (line failure, booster failure, brake fade) on braking performance is the resulting departure of the deceleration/pedal force ratio from the design point. Accordingly, longer stopping distances may result if the driver is not able to produce the increased pedal forces.

In addition to this primary effect there are other influences at work that have consequences for safety. For example, if a line failure occurs in a vehicle with the standard front to rear split, the brakes which are still operational have to convert the kinetic energy of the vehicle into thermal energy resulting in an excessive temperature rise in the operating brake. The decrease in brake effectiveness due to heating will further compound the change in the deceleration/pedal force ratio. If should also be noted that the axle with the brake operational is likely to be overbraked, especially on road surfaces with a decreased coefficient of friction. In this instance, one may lose steering or stability, depending on whether the front or rear brakes are failed.

It should be noted that fading may also influence the directional stability of the vehicle in addition to causing a decrease in the deceleration/pedal force gain. For example, differential changes in brake effectiveness may occur on the left and right brakes of a vehicle. If this situation should occur, a yawing moment will be produced as a result of the difference in brake force produced on the right and left side of the vehicle (Mitschke, 1967). A difference in braking forces at the left and right front wheels, can also cause a steering displacement of the front wheels. This steering displacement will, of course, be a function of the compliance of the steering linkage and a function of the kingpin offset existing in the front suspension.

INFLUENCE OF PARTIAL FAILURES ON DRIVER-VEHICLE BRAKING PERFORMANCE. The occurrence of a brake system failure, plus a driver's limited pedal force capability, can obviously give rise to a situation in which a driver is not capable of the pedal force necessary to achieve the deceleration he desires. It is relevant to examine whether it is possible to compute the probability that a driver may not be able to decelerate at a desired level given that a partial failure exists. Obviously, it would be even more pertinent to predict the overall probability for such a situation to arise, but this cannot be done without data on the probability for failure.

The deceleration levels encountered during normal braking and the maximum pedal force capabilities of male and female drivers, as measured in this project, are approximately normally distributed. When the measured data are plotted on probability graph paper, they produce the approximate straight lines shown in Figure 5.9 and Figure 5.0. The peak deceleration data in Figure 5.9 may be mapped onto any one of the failure analysis curves, Figures 5.1 to 5.4, to obtain a distribution of required pedal forces. This is accomplished by selecting a series of deceleration values, i.e., .05 g to .4 g, and tabulating the cumulative distribution values from Figure 5.9, along with the pedal force required to achieve that deceleration. The latter can represent "normal" or "failed" brake system performance.

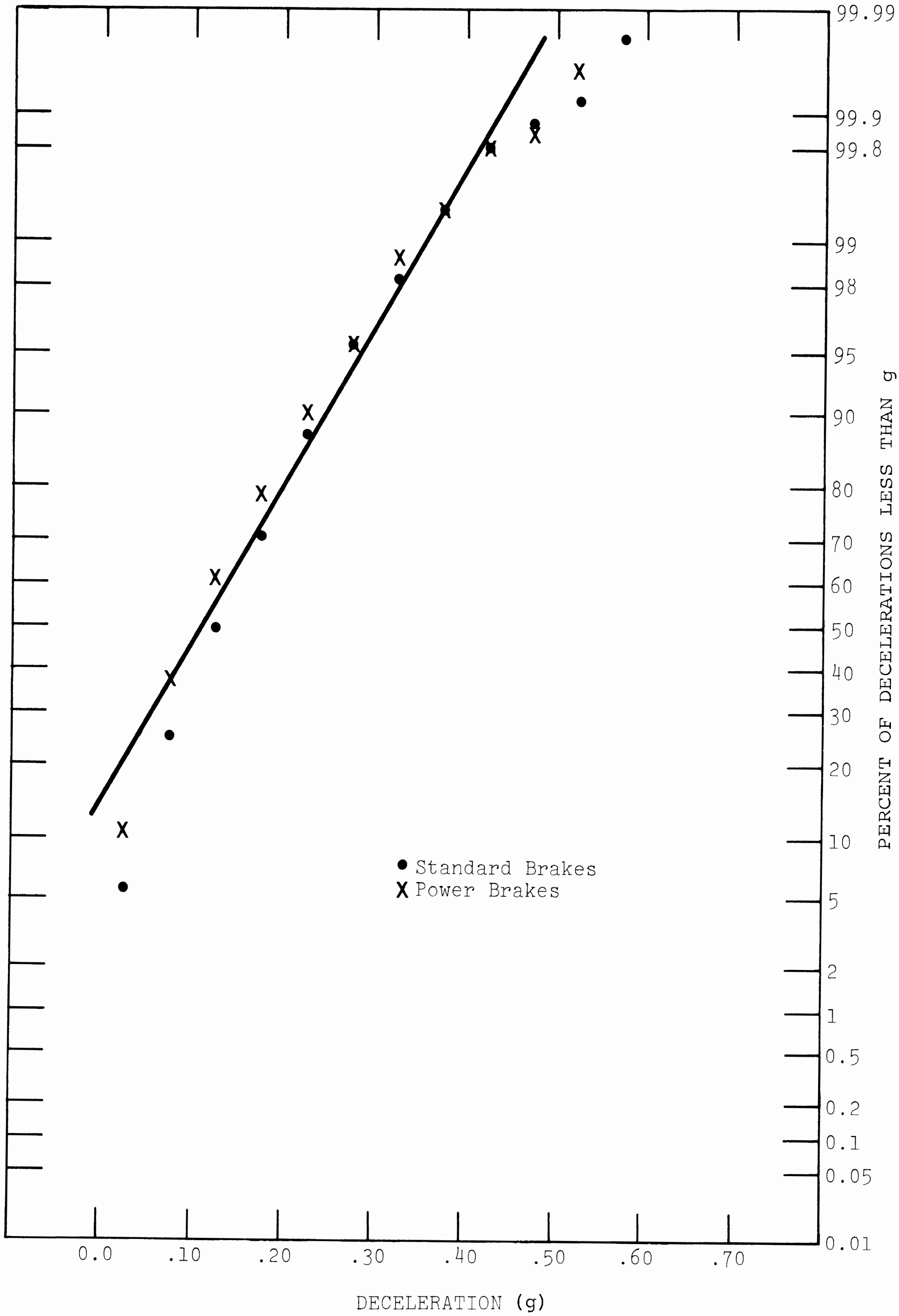



Figure 5.9. Peak deceleration-cumulative distribution of all data.

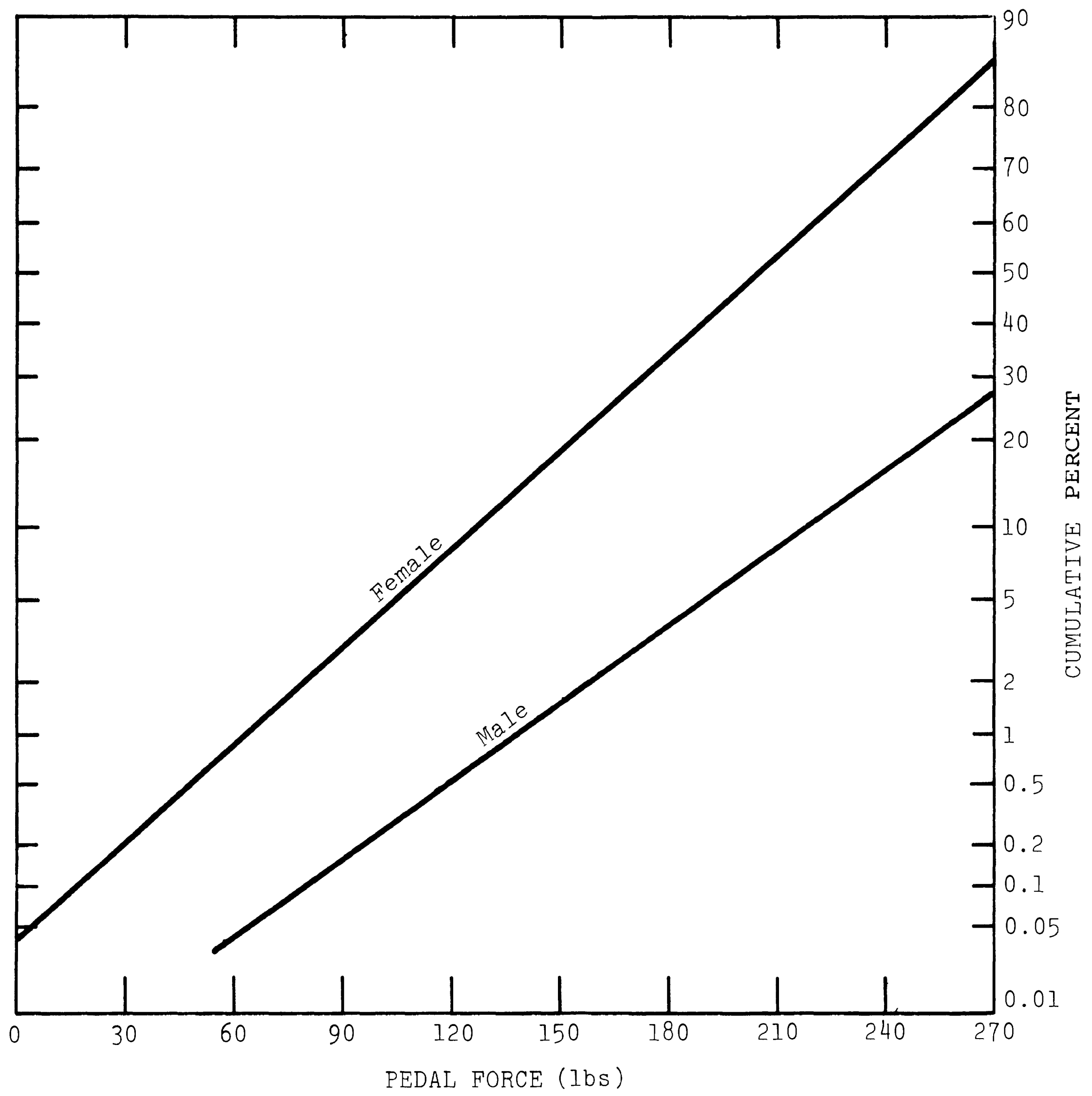



**Figure 5.10.** Pedal force capabilities of male and female drivers using right foot with induced motivation.

Table 5.5 presents the results of mapping the deceleration distribution onto Figure 5.1. Thus the peak deceleration data has been transformed into a distribution of required pedal forces. When the deceleration distribution data maps onto a linear deceleration/pedal force line, the required-pedal-force distribution also appears as a straight line on probability paper. When this distribution is plotted along with the straight line representing the driver's pedal force capability, it becomes convenient to determine the probability that the driver will be unable to achieve the decelerations that he normally carries out during his driving task.

Figures 5.11 and 5.12 illustrate the case of front brake circuit failures for manual- and power-assisted brakes, respectively, while Figure 5.13 represents a power boost failure. The use of these graphs is best shown by an example. Let us assume that it is desired to determine the probability that a 5th percentile female, driving a manually braked vehicle (whose torque distribution is given by $\Phi$ = .40) fails to achieve her desired deceleration during the stop following a front-brake circuit failure. Using the right hand scale of Figure 5.11, the 5th percentile line intersects the female capability line at "A". Proceeding vertically to the $\Phi$ = 40% line, point "B", and then again horizontally back to the right hand scale, we find that the probability of our 5th percentile driver failing to achieve her desired deceleration level is 8 percent, i.e., this is the probability of a given stop requiring higher pedal forces than she is capable of applying. Additional results, such as those shown in Table 5.6, are readily obtained in a similar manner.

Theoretically, it is possible to consider the effects of various brake system failures on the braking performance yielded by many combinations of vehicles and drivers. Prior to perform-

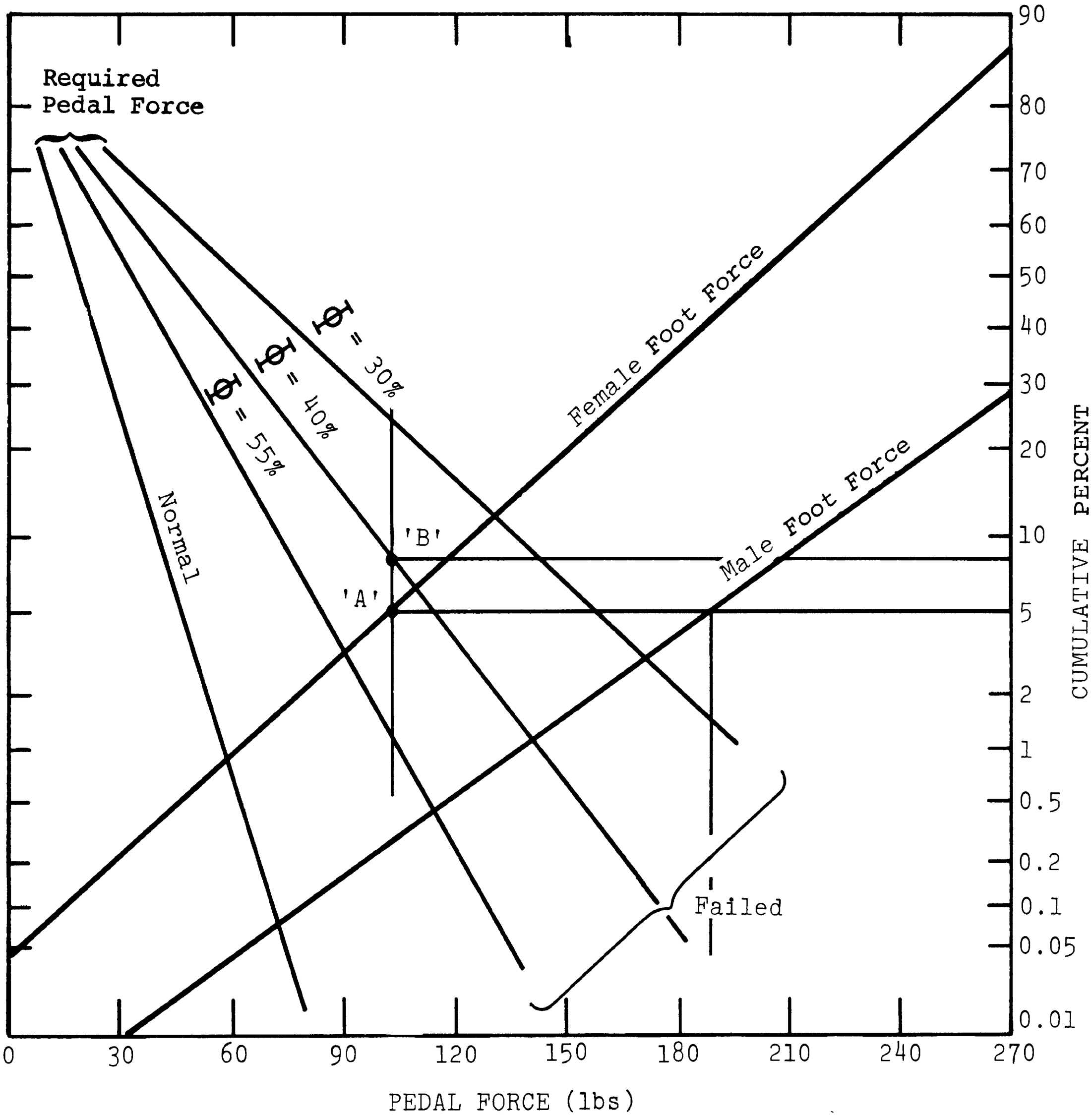



Figure 5.11. Cumulative pedal force distributions for front axle brake circuit failure in a loaded sedan with manual brakes.

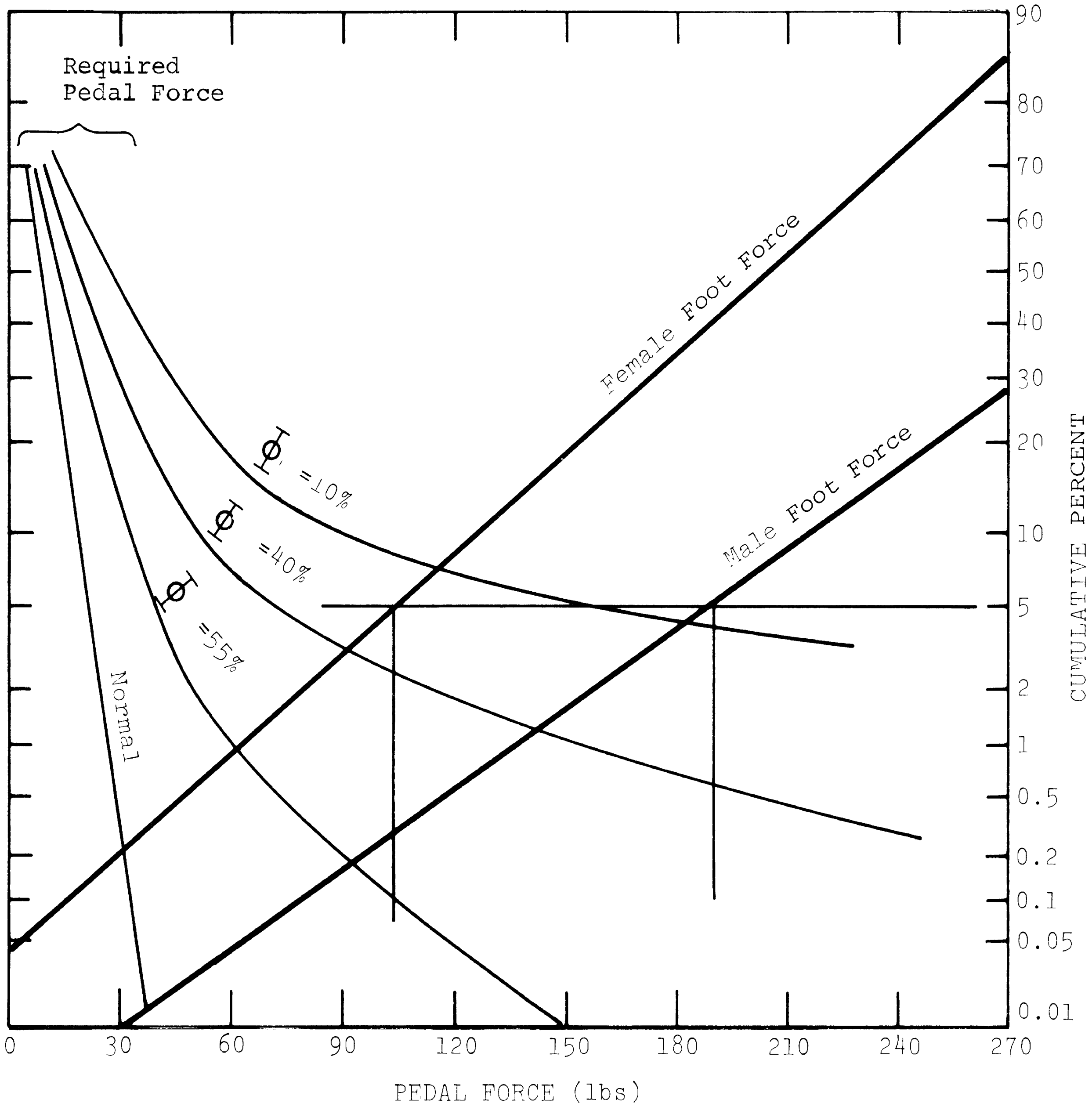



Figure 5.12. Cumulative pedal force distributions for front axle brake circuit failure in a loaded sedan with power brakes.

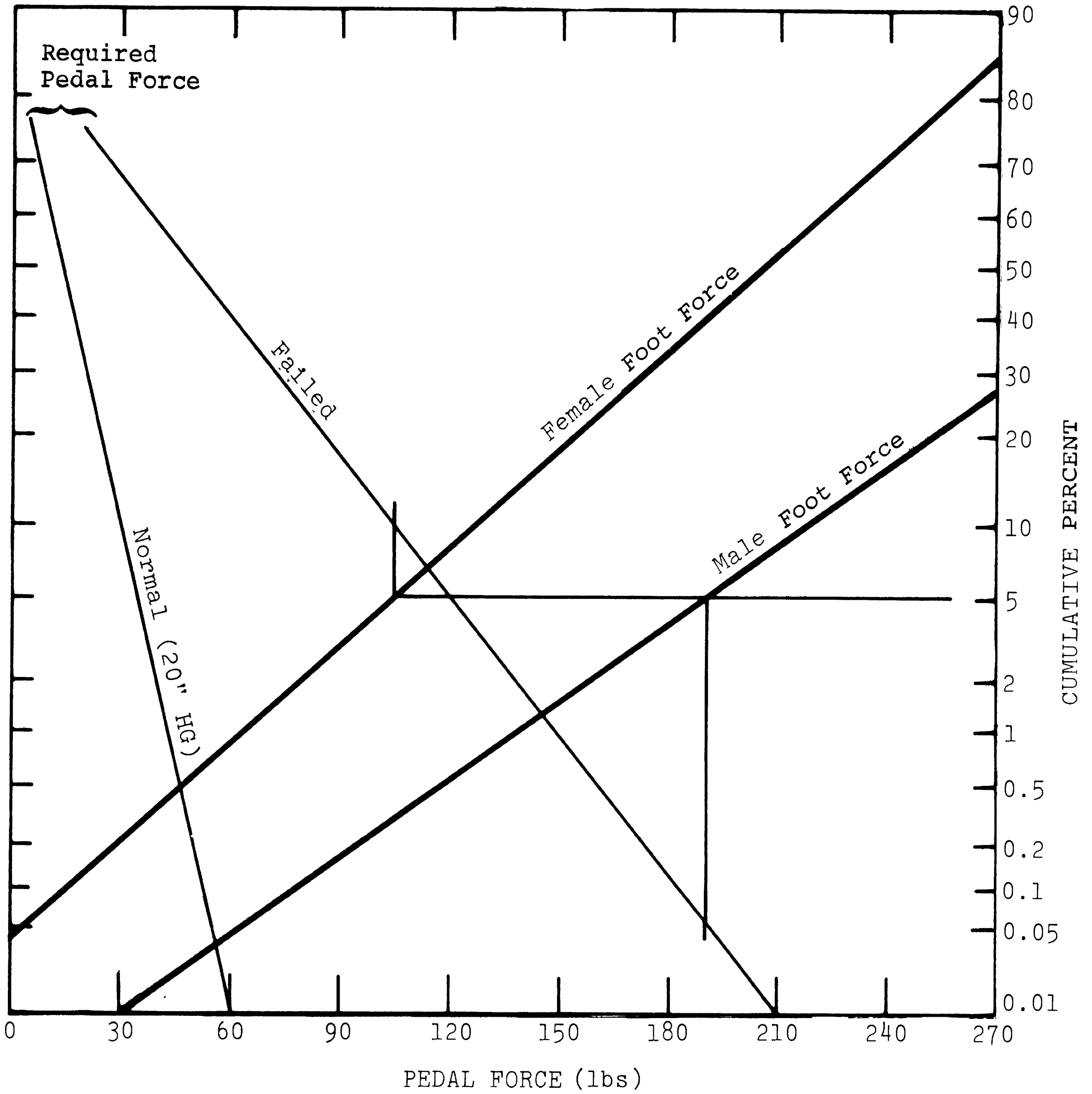



Figure 5.13. Cumulative pedal force distributions for power assist failure in a loaded sedan.

TABLE 5.5 TABLUATION OF REQUIRED PEDAL FORCES FOR FRONT BRAKE CIRCUIT FAILURES IN LOADED, MANUALLY BRAKED SEDAN

| | Required Pedal Force (lb) | | | | | |
|---|---|---|---|---|---|---|
| Decel (g) | CDF* | 1-CDF | No Failure | $\Phi$ = 55% | $\Phi$ = 40% | $\Phi$ = 30% |
| .05 | 28 | 72 | 8 | 15 | 20 | 28 |
| .1 | 46 | 54 | 16 | 29 | 39 | 53 |
| .2 | 80 | 20 | 32 | 58 | 78 | 109 |
| .3 | 96.2 | 3.8 | 48 | 67 | 117 | 164 |
| .4 | 99.7 | .3 | 64. | 115 | 156 | 217 |

*Cumulative Distribution Function

TABLE 5.6 PROBABILITY OF FRONT BRAKE CIRCUIT FAILURE RESULTING IN VEHICLE DECELERATION LOWER THAN DESIRED

| | | Probability of Failing to Achieve Desired Decel (%) | | |
|---|---|---|---|---|
| Brakes | 5% tile Driver | $\Phi$ = 55% | $\Phi$ = 40% | $\Phi$ = 30% |
| Manual | Female | 1.1 | 8.0 | 24.0 |
| | Male | .002 | .03 | 1.3 |
| Power | Female | .10 | 2.5 | 8.5 |
| | Male | .006 | .52 | 4.2 |

ing this task, however, a decision should be made as to what is an acceptable, safe, cost-effective probability that a driver is incapable of applying a required pedal force. Such a decision requires much more understanding and knowledge than is available at present. When this knowledge and understanding is obtained such as to permit the establishment of a criterion, it appears that it will be possible to develop performance guidelines for failed brake systems with the aid of the approach outlined herein.

# GENERAL DISCUSSION

The focus of this study was upon the dynamic driver-vehicle braking test. However, it was also important to obtain information of the static interface relationships between the driver and the brake control. These data were needed in order to set a limit upon the maximum force which drivers should have to exert on the brake pedal to obtain high deceleration from the vehicle. The measurements of the maximum pedal force of drivers revealed that pedal forces that can be achieved by the weaker segments of the population are clearly below 100 pounds. It was also evident that subtle factors, which were either practice or motivation effects resulting from the instructions given the subjects, considerably influenced these values. The 5th percentile female achieved about 70 pounds and 100 pounds, respectively, in the two trials or motivational sets that were used. The information obtained from the test showed that left and right foot maximum force is highly correlated. Maximum force was not found to be related to overall body weight, the weight of the leg itself, or to the driver's age. Therefore, the findings could not be attributed to sampling bias in these variables.

It was expected that males would produce higher foot forces than females and this was borne out by the results. Virtually none of the male drivers were incapable of producing a foot force equal to that of the 5th percentile female. The number of female drivers is increasing steadily and now constitutes about 42 percent of the driving population. For this reason, requirements of female drivers should be given close consideration. Therefore, it seems reasonable to take the female 5th percentile maximum foot force value, or a lower value, as an upper boundary of brake pedal force to obtain close to peak braking deceleration from a vehicle on a high coefficient of friction surface. On the basis of this work and that of Stoudt et. al. (1969) it was concluded that a maximum pedal force of 85 pounds would be a reasonable cut-off value. Since the highest level of deceleration which may be required and is reasonably attainable on a dry pavement is 0.75 g it is suggested that not more than a force of 85 pounds applied to the brake pedal should be needed to obtain this level of deceleration.

Having determined a maximum foot force level it was of interest to consider the requirements for brake force levels for vehicles equipped with manual and power brakes in both a normal operating mode and in a failed condition. For these reasons an analysis of failure conditions was carried out using typical vehicle data. The effects of failures in the brake booster, and front and rear brake line circuits have been described. These analyses were carried out to show the effects of each of the failures upon the pedal force levels that would be required to attain a given level of deceleration.

In order to assess the consequences of a failure as well as to learn of the operational conditions for which brakes are used, a vehicle was instrumented by which the peak deceleration level reached on each brake application was measured. Decelerations as high as 0.3 g were used less than 4 percent of the time. The results are shown in terms of cumulative percentage deceleration values.

These data are relevant to the failure analysis. This is because it is important to assess the consequences of a braking failure in terms of the likelihood that a particular deceleration may be required, when the failure occurs, by a driver capable of a particular pedal force. A deceleration level of 0.3 g, if used as a criterion for performance of failed brake systems, when required by a 5th percentile foot force female, would have to be provided by 85 pounds of force applied on the brake pedal. The use of a cut-off value of 0.3 g would ensure that in about 96 percent of such occurences, assuming that brake failures occur randomly in brake applications, the driver would be able to achieve the deceleration that he perceives to be needed. The cost of accomplishing a 0.3 g deceleration, and providing for estimated protection in 96 percent of brake application, can be computed. Any protection level required can be selected and the respective deceleration level derived from Figure 4.5. This deceleration value can then form the criterion to which the brake must perform in the failed condition. The pedal force data can be used in a similar fashion to select percentage levels of pedal force capability in the driving population as another criterion value for brake performance. Since brake performance can be stipulated in terms of the requirement to achieve a particular level of deceleration for a given pedal force these two distributions can be used together to define suitable requirements. The failure analysis shows that the various failure conditions require different pedal forces to achieve the same deceleration and, therefore, if the probabilities of different types of failures were known they could be used to define the pedal force requirements in terms of collisions (or desired deceleration) likelihood. In this study we have shown a procedure by which a brake performance standard could be developed for brake system failure modes.

The major thrust of this research effort was concerned with the operation of the brake system when it is in a normal operating, non-failed condition. The results of the dynamic braking test quite clearly showed that driver performance was affected by the gain of the brake control. The general nature of the results were as predicted, in that high deceleration/pedal force gain provided better stopping performance on the dry and intermediate friction surfaces compared to lower gain controls, and that this was reversed on the wet-painted surface. However, it could not have been predicted which specific gain levels that were used in the test would have provided significantly different performance on each of the surfaces used. The data showed that the highest gain (0.065 g/lb) produced lower mean deceleration and longer stopping distances compared to some lower gain levels. On the wet-painted surface the most effective performance was obtained with gain level 4 (0.012 g/lb). Therefore, both the highest and lowest gains were found to be undesirable in terms of maximizing deceleration in the braking task. These data alone would be adequate to set boundary conditions on pedal force requirements and deceleration/pedal force gain for a braking standard. However, because of the interaction between pedal force gain and the surface coefficient of friction these limits can be legitimately narrowed.

The suboptimal braking performance that was achieved with the highest gain condition was also shown by measures of wheel lockup, wheel lockup duration, loss of control runs and the subjective data. The importance of reducing the pedal force gain at low pedal force levels was clearly demonstrated in this study. A combination of high deceleration/ pedal force gain with a low absolute force level leads to a difficult brake modulation task for the driver, since he is controlling a highly responsive brake at a low pedal force level, at which his own sensitivity is low.

The cut-off that has been selected for maximum gain (0.021 g/lb) (Figure 3.29) will ensure that about 20 pounds of pedal force is the minimum for deceleration of 0.4 g. This boundary in the deceleration/pedal force envelope is of great importance, in view of the high frequency with which deceleration levels below 0.4 g are used by drivers, to provide comfortable and good brake modulation in normal, non-panic braking as well as to minimize stopping distance when the friction coefficient is low. It will also ensure that drivers can better attain a maximum longitudinal deceleration while retaining control over the path of the vehicle.

The boundary upon minimum gain will provide good brake modulation on low and high coefficient of friction conditions and ensures that the pedal force levels needed at high deceleration levels can be attained by most drivers.

The SAE brake effectiveness test which is incorporated into MVSS-105 calls for a minimum pedal force of 15 pounds and a maximum of 100 pounds, at a deceleration of 20 ft/sec/sec from 30 mph. These limits lie outside the boundaries that are recommended on the basis of this study (Figure 3.29), which requires a minimum brake pedal force of about 30 pounds and a maximum of about 75 pounds at this deceleration. Thirteen of the 24 power brake cars, for which brake compliance test results are shown in Table 5.4, and one of the 19 manual brake cars (Table 5.3) have gains that exceed the maximum gain boundary. Two of the 19 manual brake cars have less than the minimum gain. Thus, most U.S. passenger cars with either manual or power brakes appear to have pedal force requirements, in the 30 mph test, that fall within the defined space in the recommended deceleration/pedal force envelope.

# RECOMMENDATIONS

Based upon the analytical and experimental research conducted in this study some recommendations for a brake force standard and objective test and compliance procedures can be made:

(1) Standard should be written such as to insure that the pedal force required at some specified deceleration condition can be achieved by a specific percentile of the female driving population.

Recommendation: The pedal force required to decelerate a fully loaded vehicle at 0.75 g shall not exceed 85 pounds for brakes operating under nondegraded conditions in a stop initiated at 30 mph.

(2) Standard should be written such as to insure that deceleration/pedal force gain and pedal force level facilitate good braking modulation on surfaces of reduced friction coefficient.

Recommendation: The deceleration/pedal force relationship as measured on a high friction surface (skid number 0.75) with a lightly loaded vehicle should fall to the right of the maximum gain - minimum force boundary indicated on Figure 3.29, for brakes in a nondegraded condition and for stops initiated at 30 mph.

(3) Standard should be written such as to insure that low deceleration/pedal force gain and/or high pedal force do not unduly degrade driver-vehicle braking performance on moderate and high friction surfaces.

Recommendation: The deceleration/pedal force relationship as measured on a high friction surface (skid number 0.75) with a fully loaded vehicle should not be less than the gain associated with the minimum gain boundary on the right side of the recommended deceleration/pedal force space, nor should the pedal forces fall to the right of the boundary indicated in Figure 3.29 when the brakes are in a nondegraded condition and the initial velocity is 30 mph.

(4) Standard should be written such as to insure that brakes have sufficient energy absorption capacity such that stops initiated at the top speed capability of the vehicle shall not unduly increase the required pedal forces.

Recommendation: The deceleration/pedal force limits imposed for nondegraded brakes in making a stop from 30 mph shall be increased proportional to the increment in kinetic energy (above 30 mph ) that prevails when making a stop at initial speeds higher than 30 mph. The limit should be increased by 20 percent for a four-fold increase in kinetic energy.

(5) Compliance with the recommended standard on brake pedal force and deceleration/pedal force gain shall be measured by obtaining values for each vehicle of deceleration and pedal force at a number of deceleration levels and comparing the findings with the recommended standard. The test procedure shall be the same as that described in SAE Recommended Practice J-843.

# APPENDIX I

## DERIVATION OF CONSTANT PEDAL DISPLACEMENT/DECELERATION CHARACTERISTIC

The original specification for the displacement versus brake line pressure relationship called for brake line pressure to increase linearly with displacement up to 400 psi at 1 1/2 inches displacement, then to increase linearly at an augmented rate until it reached 1200 psi at 2 1/2 inches displacement.

Using this specification and the desired values for deceleration/pedal force gain, the spring canisters were constructed. Because of the limited number of different spring constants available, and the rather large deviations from catalog specifications which were found in individual springs, the degree of mismatch between the brake line pressure versus pedal displacement functions of the various canisters was found unacceptable.

A computer program was therefore written to determine the appropriate gain level to match each canister as closely as possible to the desired pressure-displacement function, using the empirically determined pedal force/displacement function as input. The error function to be minimized was the sum of the squared percentage errors in pressure for successive .25 inch increments in displacement over the range of 0.25 to 2.5 inches. While this procedure produced much improved uniformity of pressure-displacement functions, it was apparent from a graphic presentation of the results that still more improvement could be achieved by changing several of the springs. At the same time, these changes of springs could be employed to make the ratios between deceleration/pedal force gains, for successive canisters in the series, substantially equal.

It was also discovered that part of the difficulty in matching the pressure-displacement functions was due to the fact that the force-displacement functions which were used as input were nonlinear at the low displacements. This was attributable to friction in the pedal linkage and master and slave cylinders, and to the small degree of pedal travel required to take up slack in the system and close the port in the master cylinder. Because of the erratic nature of these residual effects, data for 0.25 and 0.5 inch displacements were excluded from further analyses. Extrapolation of the linear portions of the force-displacement curves yielded an origin at 0.375 inch displacement and three pounds pedal force. It was therefore decided to offset the zero setting on the force transducer so that zero output corresponded to three pounds of pedal force.

After the above modifications were made, the computer program was rerun using the new force-displacement functions and the new origin. An excellent fit was obtained between the pressure-displacement functions for the various canisters. The force gain levels found were approximately equally spaced logarithmically (each one was approximately a constant multiple of the next lower one) but the range covered was not satisfactory in that the highest gain was somewhat higher than necessary and the lowest was not as low as was desired.

Thefore, all gains were multiplied by a constant to obtain the desired range. The obtained values were then further adjusted to obtain precisely equal logarithmic steps.

The resulting pressure-displacement curves differed somewhat from those originally desired, in that 2.5 inches displacement yielded approximately 1000 psi rather than the 1200 originally envisioned. However, the highest and lowest pressures obtained with the canisters at a given displacement differed by less than 10 percent, which was considered to be acceptable.

# APPENDIX II

## INSTRUCTION TO TEST SUBJECTS

### INSTRUCTIONS-PRACTICE RUN

In this experiment I am interested in learning of your ability to bring the car to a safe stop in as short a distance as possible after initiating braking. In order for you to become familiar with the test lanes and the automobile I want you to make several practice runs today. Before making the runs, however, let me tell you more about the test lanes and automobile.

Three test lanes will be used; these are outlined by orange traffic cones. (Show subject the lanes outlined by cones). The left lane is the normal, dry asphalt lane, the center lane has asphalt that has been watered to simulate a rainy day, and the right lane has a yellow, painted asphalt surface which has been watered to simulate a slippery surface such as ice. You will notice that near the far end of each lane there are three lamps (point them out and make sure the subject sees them). Soon after you enter a test lane one of the lamps in that test lane will come on. When you see the lamp come on you are to begin braking with your right foot. Bring the car to a gradual, safe stop. By a safe stop I mean that you are to avoid knocking down any traffic cones.

During these trial runs I am not interested in how rapidly you can safely stop the car, but rather in giving you confidence that you can stop the car safely. You should realize that if you lock the brakes the car may skid. If you feel the car beginning to skid, let up on the brakes to permit the wheels to turn again and then apply the brakes so that they just avoid locking. Do you have any questions regarding the test lanes?

This car has a wheel attached to the rear bumper. To prevent damaging the wheel, the engine stops whenever the car is put in reverse. Therefore, when you shift from park to drive, move through reverse rapidly.

The braking system is powered by a pump which must be operated between runs to maintain proper pressure. Although it may be noisy, do not let it bother you.

This experiment is designed to study several factors related to emergency stopping distances. The factors considered are (1) the distance the brake pedal travels from resting point to the totally depressed position, and (2) brake pedal force, that is, the amount of pressure required to depress the pedal and bring the car to a complete stop. By varying the displacement and pedal force, we can simulate a variety of braking systems presently in use in most production cars. You will be given an opportunity to practice with each brake system before using it in the test runs.

You will notice a floor pedal to the left of the brake pedal. This is an emergency braking pedal. In an emergency you may use this pedal, but otherwise it should not be used.

The speed of the car will be automatically controlled at 35 mph or 50 mph. At the beginning of each run, I will inform you of the speed. It will be necessary for you to accelerate until the car is going three or four miles per hour above the desired speed, until the light on the dash comes on. You should then release the accelerator, but keep your foot resting lightly on it until you receive the signal to brake. Do you have any questions regarding the automobile or the procedure?

I want you to get in the driver's seat now and attach the seat belt and shoulder harness.

Please drive out toward the parked plane. Brake the car several times so you will be familiar with the brake system. I will tell you when we are out far enough. You should then turn around and line up with the dry test lane.

I am going to pump up the system as you approach the test lane. The speed for this run is 35 mph. Accelerate until you are going 35 mph and maintain this speed until you see one of the lamps come on. Are you ready?

The next run will be made at 50 mph. Accelerate until you are going just above the desired speed, then let your foot rest lightly on the accelerator as the speed control takes over. Brake when you see one of the lamps come on.

We are now going to do the same thing on the wet asphalt lane. The water sprinklers will be turned off whenever you enter a wet lane. Remember that if you feel the car beginning to skid, let up on the brakes to permit the wheels to turn again and then apply the brakes so that they just avoid locking.

Now we are going to do the same thing on the wet painted asphalt surface. Remember that if you feel the car beginning to skid, let up on the brakes to permit the wheels to turn again and then apply the brakes so that they just avoid locking.

INSTRUCTIONS-OFFICIAL RUN

We will now be making official runs. In this part of the experiment we are interested in your emergency braking ability. Let the onset of the light represent the presence of a child in your path. Try to bring the car to a stop as quickly as possible and in as short a distance as possible. If you can stop before reaching the light, you should do so as far in front of it as possible. In stopping the car, however, try to avoid knocking down any traffic cones or losing control of the car. Other than the emergency braking aspect of these runs, the test procedure will be the same as before. To summarize the important points of the procedure, remember that the speed of the car will be automatically controlled, and you will have to bring the car up to just above the desired speed and then let your foot rest lightly on the accelerator. When you see the lamp come on, apply the brake with your right foot.

Remember that in the following runs I am interested in your emergency braking ability; that is, your very best safe braking performance. Your best braking performance will occur just before lockup of your wheels, so if you can just keep the wheels from locking, stopping distance and time to stop will be at a minimum. If you lock the wheels, stopping distance and time to stop will be much greater and you may lose control of the car. Do you have any questions?

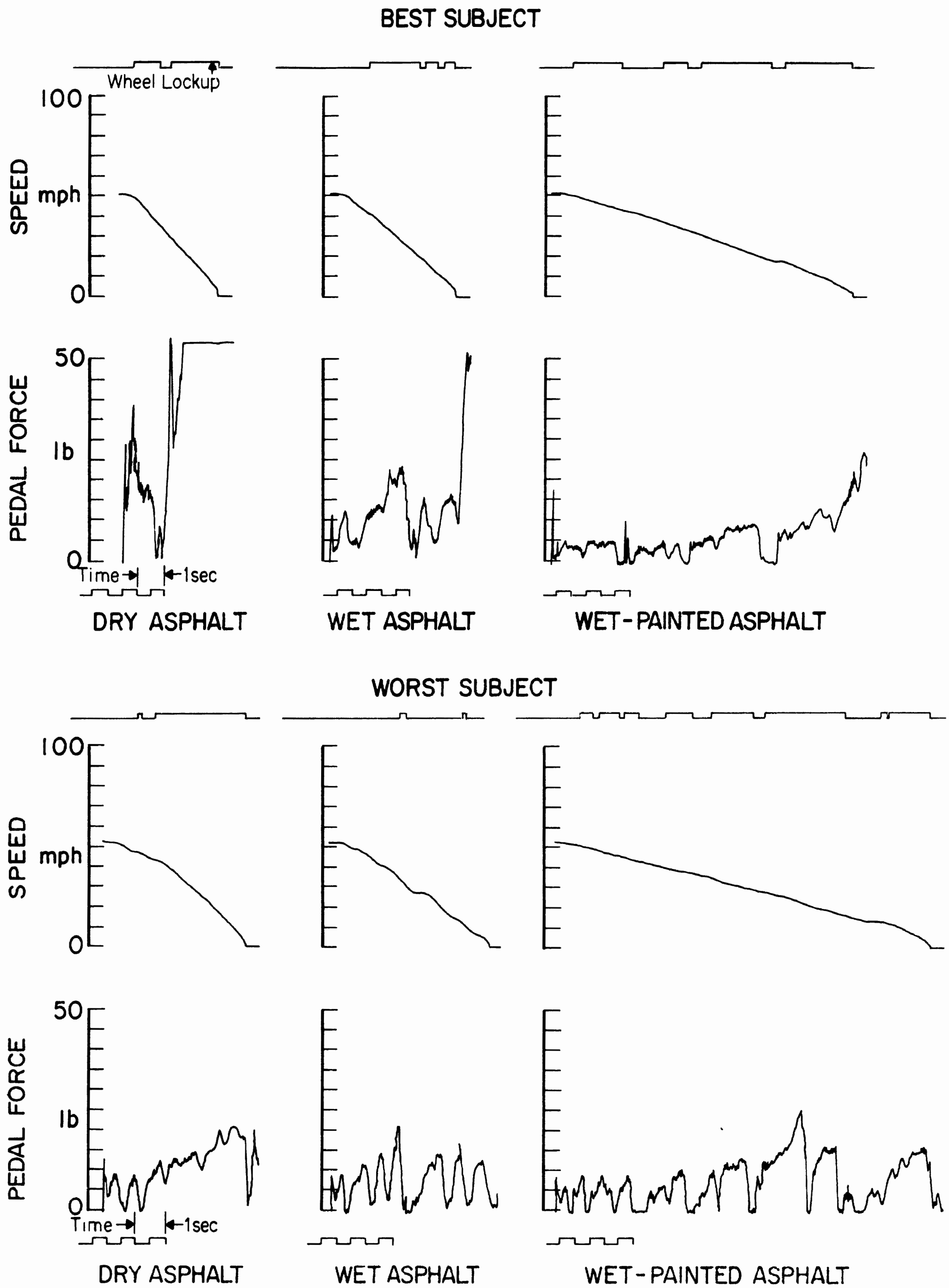



Figure A.II.1. Sample of speed, pedal force and wheel lockup time histories for the best and worst subject: deceleration/pedal force ratio = 0.065 g/lb.

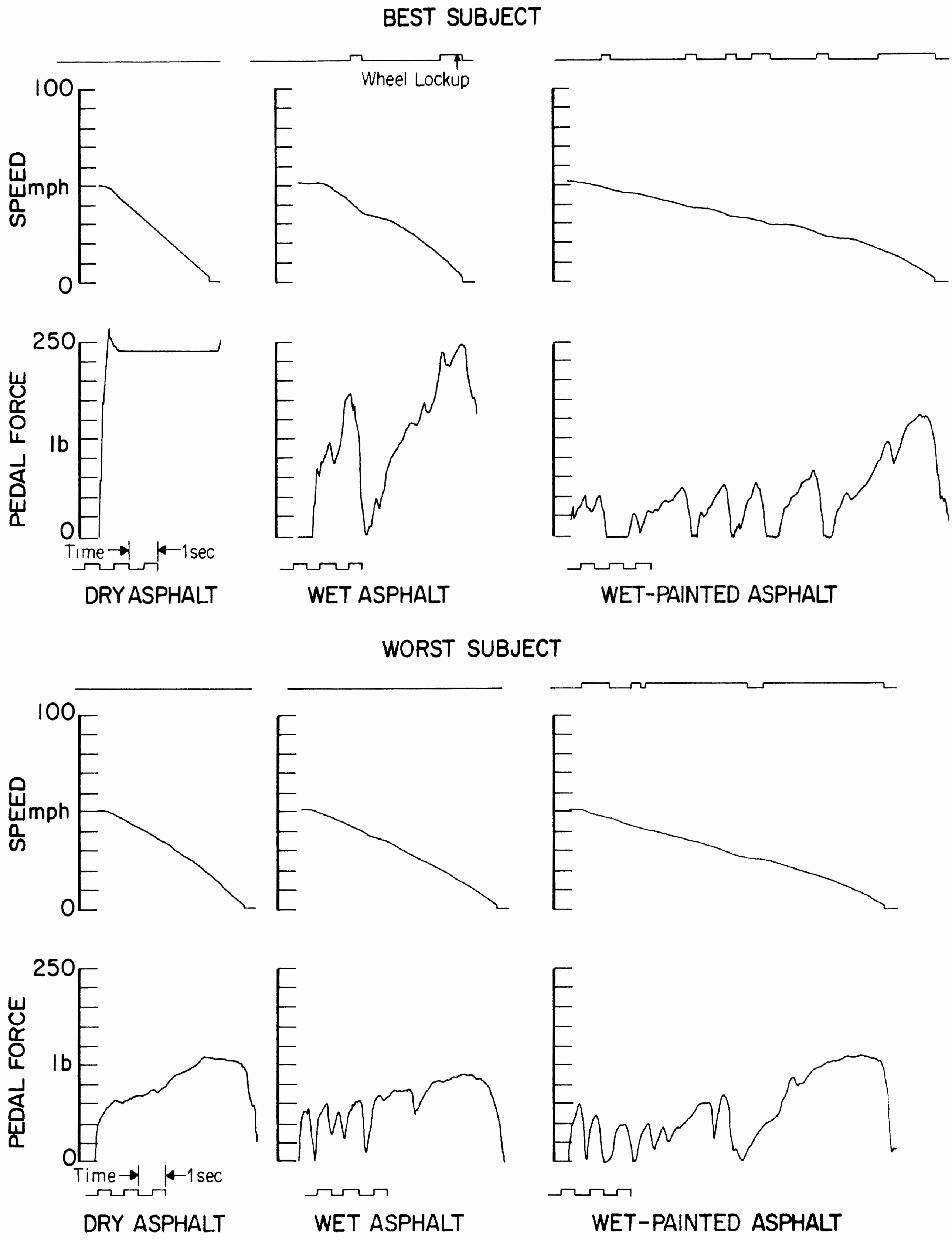



Figure A.II.2. Sample of speed, pedal force and wheel lockup time histories for the best and worst subject: deceleration/pedal force ratio = 0.004 g/lb.

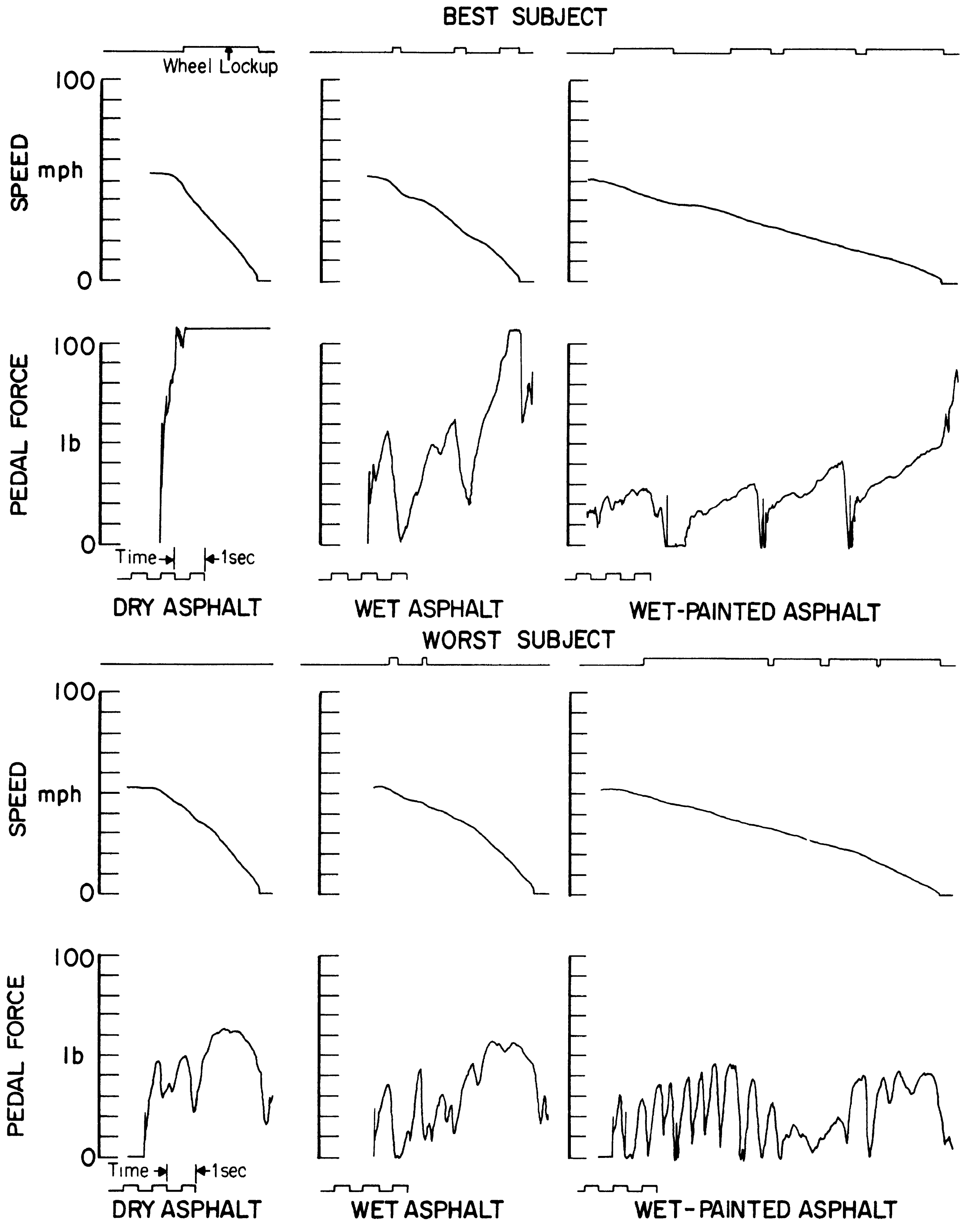



Figure A.II.3. Sample of speed, pedal force and wheel lockup time histories for the best and worst subject: deceleration/pedal force ratio = 0.012 g/lb.

## TABLE A.II. BRAKE FORCE MODULATION STUDY DATA COLLECTION SHEET - OFFICIAL RUNS

Subject: PG
Date: 11/7/69

Sex: F

Displacement: 0
Experimenter: DD

| | | 35 MILES PER HOUR | | | | | | 50 MILES PER HOUR | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Surface | Canister Order | Braking Time | Braking Distance | Wheel Lock Counter | Wheel Lock Time | Bad Runs | Canister Order | Braking Time | Braking Distance | Wheel Lock Counter | Wheel Lock Time | Bad Runs |
| 1 | | 3 | 2.98 | 72 | 0 | 0 | | 3 | 3.78 | 151 | 1 | 1.32 | |
| 2 | | 2 | 2.53 | 68 | 1 | .46 | | 2 | 3.44 | 146 | 1 | 1.17 | |
| 3 | | 4 | 2.58 | 78 | 1 | .89 | | 4 | 3.47 | 143 | 1 | 1.07 | |
| 4 | Dry | 1 | 3.24 | 81 | 3 | 1.05 | | 1 | 3.67 | 140 | 2 | .63 | |
| 5 | | 5 | 2.91 | 99 | 0 | 0 | | 5 | 4.71 | 214 | 0 | 0 | |
| 6 | | 6 | 4.06 | 131 | 0 | 0 | | 6 | 5.10 | 230 | 0 | 0 | |
| 7 | | | | | | | | | | | | | |
| 8 | | 3 | 3.06 | 98 | 0 | 0 | | 3 | 3.92 | 149 | 1 | .63 | |
| 9 | | 2 | 3.22 | 98 | 0 | 0 | | 3 | 4.83 | 206 | 4 | 2.62 | |
| 10 | | 4 | 3.56 | 110 | 0 | 0 | | 4 | 5.16 | 224 | 0 | 0 | |
| 11 | Wet | 1 | 3.39 | 104 | 2 | 1.18 | | 1 | 5.53 | 237 | 3 | 2.25 | |
| 12 | | 5 | 4.06 | 132 | 0 | 0 | | 5 | 5.86 | 266 | 0 | 0 | |
| 13 | | 6 | 4.87 | 147 | 0 | 0 | | 6 | 6.12 | 281 | 1 | .05 | |
| 14 | | | | | | | | | | | | | |
| 15 | | 3 | 7.83 | 239 | 5 | 4.15 | | 3 | 12.33 | 534 | 8 | 5.82 | |
| 16 | | 2 | 7.29 | 215 | 3 | 4.34 | | 2 | 11.34 | 482 | 8 | 6.69 | |
| 17 | Painted | 4 | 6.21 | 181 | 3 | 3.65 | | 4 | 12.34 | 547 | 2 | 3.64 | 1 |
| 18 | | 1 | 8.22 | 238 | 5 | 5.03 | | 1 | 13.43 | 661 | 5 | 5.11 | 6 |
| 19 | | 5 | 6.79 | 208 | 0 | 0 | | 5 | 13.76 | 574 | 1 | 2.11 | |
| 20 | | 6 | 7.94 | 249 | 1 | .11 | | 6 | 12.36 | 534 | 2 | .48 | |
| 21 | | | | | | | | | | | | | |

TABLE A.II. BRAKE FORCE MODULATION STUDY
DATA COLLECTION SHEET - OFFICIAL RUNS

Subject: PG
Date: 11/7/69

Sex: F

Displacement: 2.5
Experimenter: DD

35 MILES PER HOUR

| | Surface | Canister Order | Braking Time | Braking Distance | Wheel Lock Counter | Wheel Lock Time | Bad Runs | Canister Order | Braking Time | Braking Distance | Wheel Lock Counter | Wheel Lock Time | Bad Runs |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | | 4 | 3.14 | 98 | 0 | 0 | | 4 | 4.39 | 184 | 0 | 0 | |
| 2 | | 2 | 2.55 | 79 | 0 | 0 | | 2 | 3.46 | 146 | 1 | .98 | |
| 3 | | 3 | 2.89 | 98 | 1 | .43 | | 3 | 3.40 | 145 | 0 | 0 | |
| 4 | Dry | 1 | 2.55 | 71 | 1 | 1.67 | | 1 | 4.30 | 167 | 3 | 1.98 | |
| 5 | | 5 | 3.17 | 103 | 0 | 0 | | 5 | 3.99 | 182 | 0 | 0 | |
| 6 | | 6 | 4.16 | 125 | 0 | 0 | | 6 | 5.31 | 236 | 0 | 0 | |
| 7 | | | | | | | | | | | | | |
| 8 | | 4 | 3.83 | 119 | 0 | 0 | | 4 | 5.44 | 237 | 1 | 3.55 | |
| 9 | | 2 | 3.24 | 101 | 0 | 0 | | 2 | 4.91 | 225 | 2 | 3.34 | |
| 10 | | 3 | 3.27 | 106 | 0 | 0 | | 3 | 4.79 | 207 | 1 | 1.21 | |
| 11 | Wet | 1 | 3.80 | 127 | 1 | .21 | | 1 | 4.91 | 227 | 6 | 2.64 | |
| 12 | | 5 | 3.57 | 119 | 0 | 0 | | 5 | 5.55 | 237 | 0 | 0 | |
| 13 | | 6 | 5.14 | 148 | 0 | 0 | | 6 | 6.01 | 254 | 0 | 0 | |
| 14 | | | | | | | | | | | | | |
| 15 | | 4 | 6.33 | 194 | 2 | 4.35 | | 4 | 11.41 | 505 | 1 | .46 | 1 |
| 16 | | 2 | 7.15 | 216 | 5 | 4.72 | | 2 | 12.28 | 514 | 6 | 6.60 | |
| 17 | | 3 | 7.18 | 221 | 2 | 3.51 | | 3 | 11.99 | 533 | 4 | 6.89 | |
| 18 | Painted | 1 | 7.79 | 247 | 4 | 5.90 | | 1 | 13.19 | 564 | 8 | 6.75 | 2 |
| 19 | | 5 | 7.80 | 235 | 3 | 3.92 | | 5 | 12.54 | 545 | 4 | 1.28 | |
| 20 | | 6 | 7.62 | 236 | 2 | 2.60 | | 6 | 12.36 | 539 | 2 | 4.71 | |
| 21 | | | | | | | | | | | | | |

# APPENDIX III

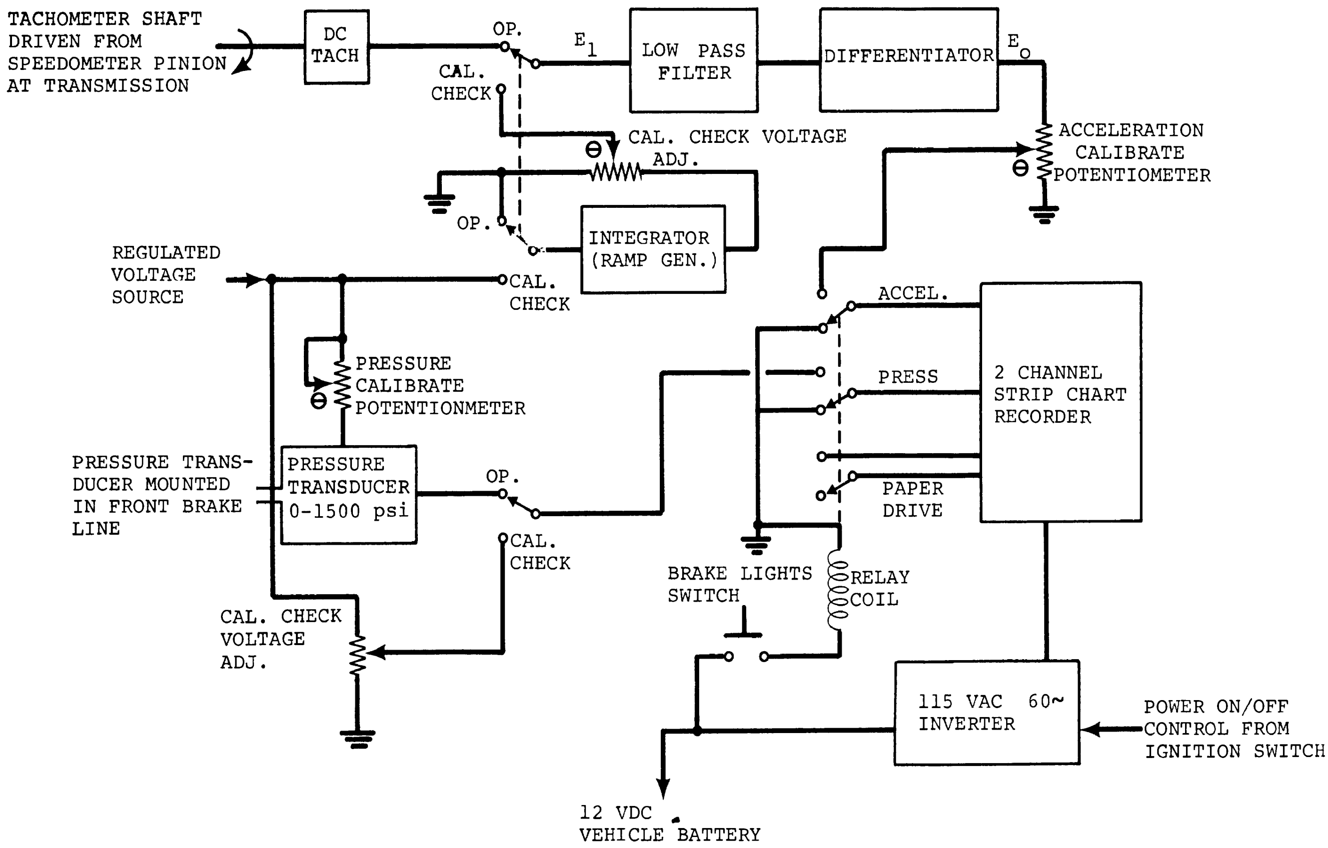
## VEHICLE INSTRUMENTATION FOR DECELERATION RECORDING

The deceleration magnitude instrumentation package, shown in block diagram form in Figure A-III.1, provides a strip chart pen recording of vehicle decleration and front brake line pressure during the time the brakes are applied. With the exception of the tachometer and pressure transducer, the instrumentation package was mounted in the trunk of a University of Michigan car pool vehicle. Power is supplied to the circuits when the vehicle ignition is on, thus permitting unattended operation of the system. In order to record only data of interest (while brakes are applied) and thus minimize the length of the data charts, the paper drive is turned on through a relay actuated from the brake light switch. The deceleration and pressure signals are switched through independent contacts on the same relay so that no pen deflection occurs except when the paper drive is on thus preventing ink smears.

### Deceleration Measurement

The deceleration signal was obtained by differentiating a velocity signal from a DC tachometer. This method was used rather than a standard accelerometer since the latter could have significant errors due to road slope and vehicle pitch. The velocity derivative is in error only if the wheels lock or slip excessively as in very hard stops.

The tachometer (Servo Tek Model SA-757A-2) was mounted at the transmission on one arm of a mechanical drive-tee driven by the speedometer pinion. The other arm of the tee carried the speedometer cable. In order to damp out tachometer commutator ripple and drive train vibrations which, unfiltered, would have



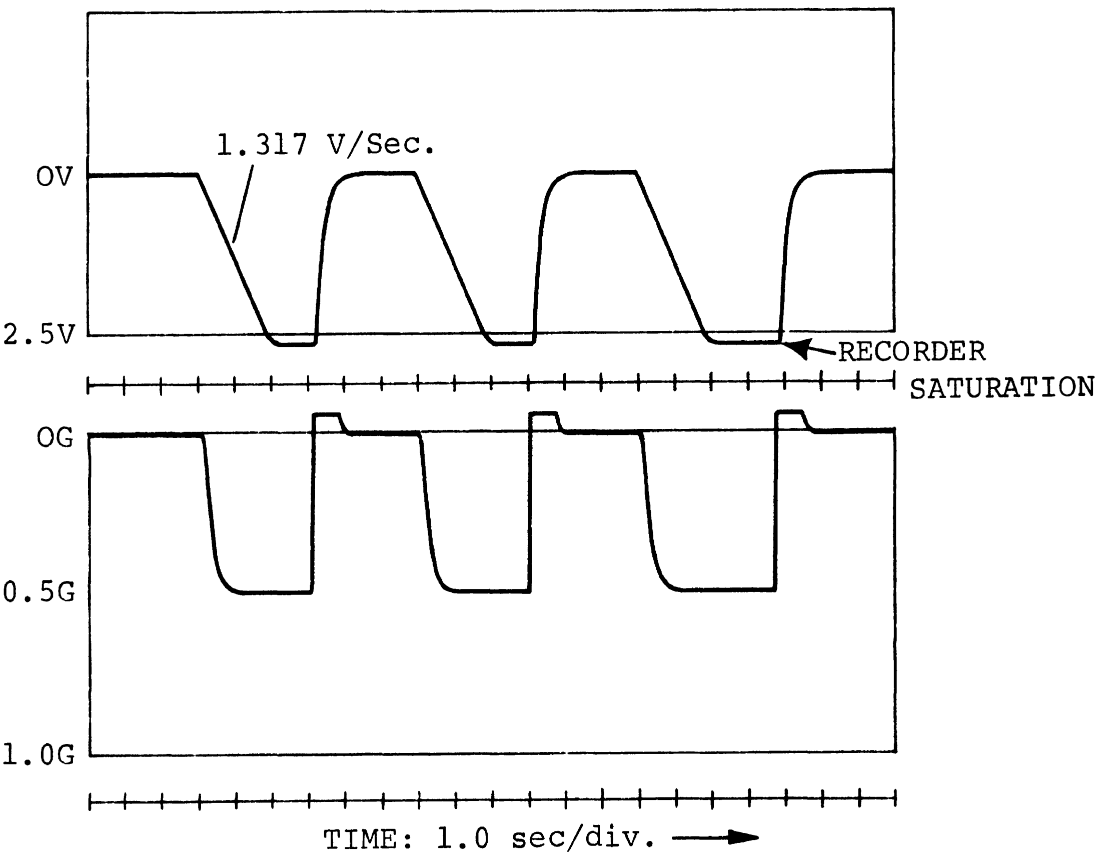

A.III.1. Deceleration magnitude instrumentation.

swamped the desired derivative signal, it was necessary to heavily filter the signal. This required a compromise between ripple remaining on the filtered signal and system response to sudden changes in acceleration. The response of the final circuit to a 0.5 G step in acceleration (16.1 ft/sec ramp in velocity) was about 0.75 seconds as shown in Figure A-III.2. Figure A-III.3 shows the measured frequency response curve of the low pass filter-differentiator circuit.

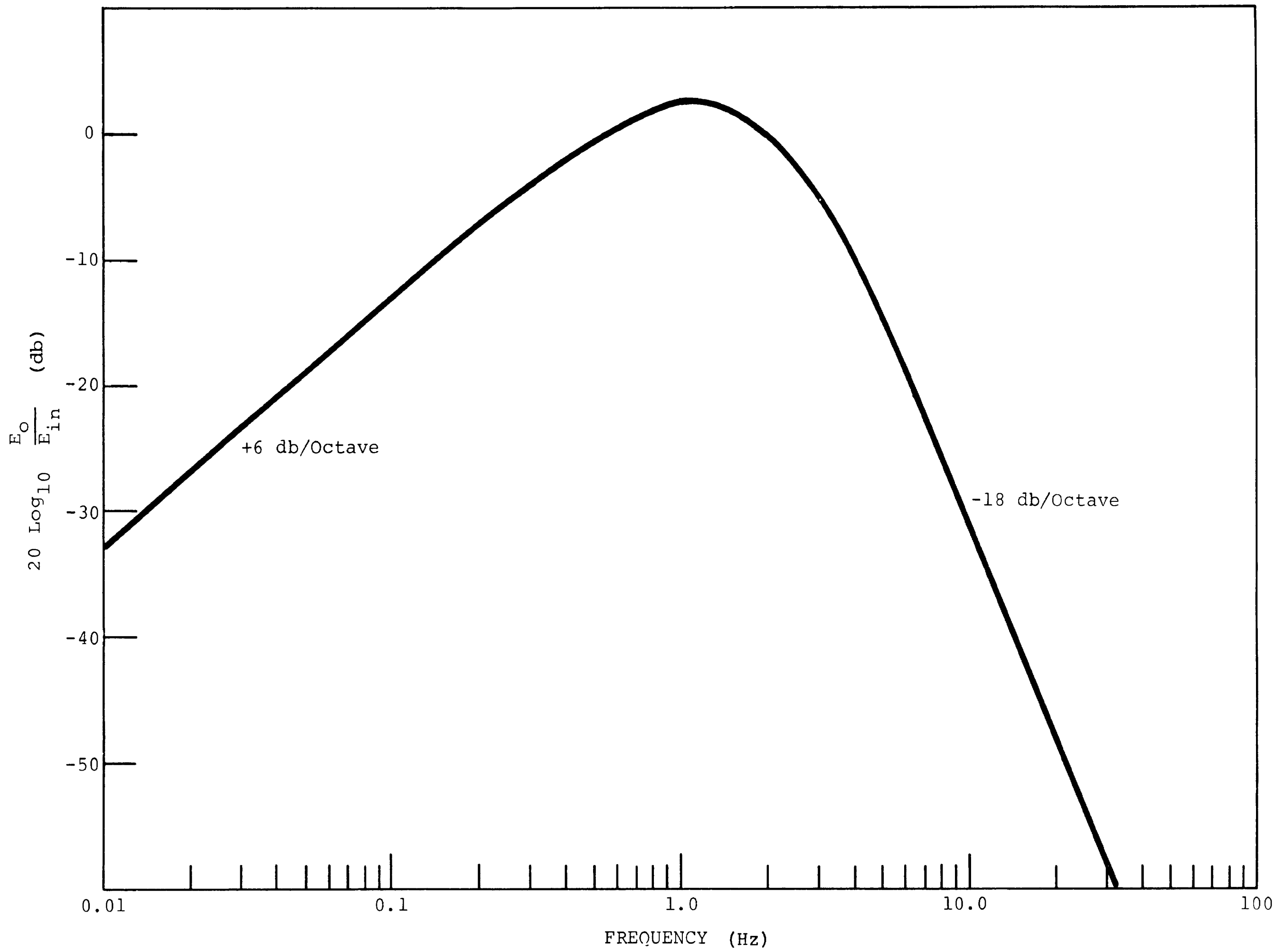
Deceleration Calibration

Calibration of the velocity-derivative accelerometer was carried out as follows. Using a fifth wheel to measure vehicle velocity and the strip chart recorder as a voltmeter, the tachometer output was calibrated in volts/feet/second ($E_1$ = 0.0818 V/fps). From this the rate of change or slope of the tachometer output voltage at a deceleration of 0.5 G, 16.1 ft/sec/sec, was calculated (0.5 G = 1.317 V/sec). The CAL CHECK VOLTAGE ADJ. potentiometer (Figure A-III.1) was then adjusted to give a ramp of voltage with this slope at the input to the low pass filter through the Operate/CAL. CHECK switch. Finally, the ACCELERATION CALIBRATE potentiometer at the output of the differentiator was set to obtain 1/2 of full scale deflection, i.e., 0.5 G on the recorder. The calibration ramp voltage and corresponding acceleration signal are shown in Figure A-III.4. The flat portion at the top of the ramp is due to recorder saturation and not to change in slope of the ramp.

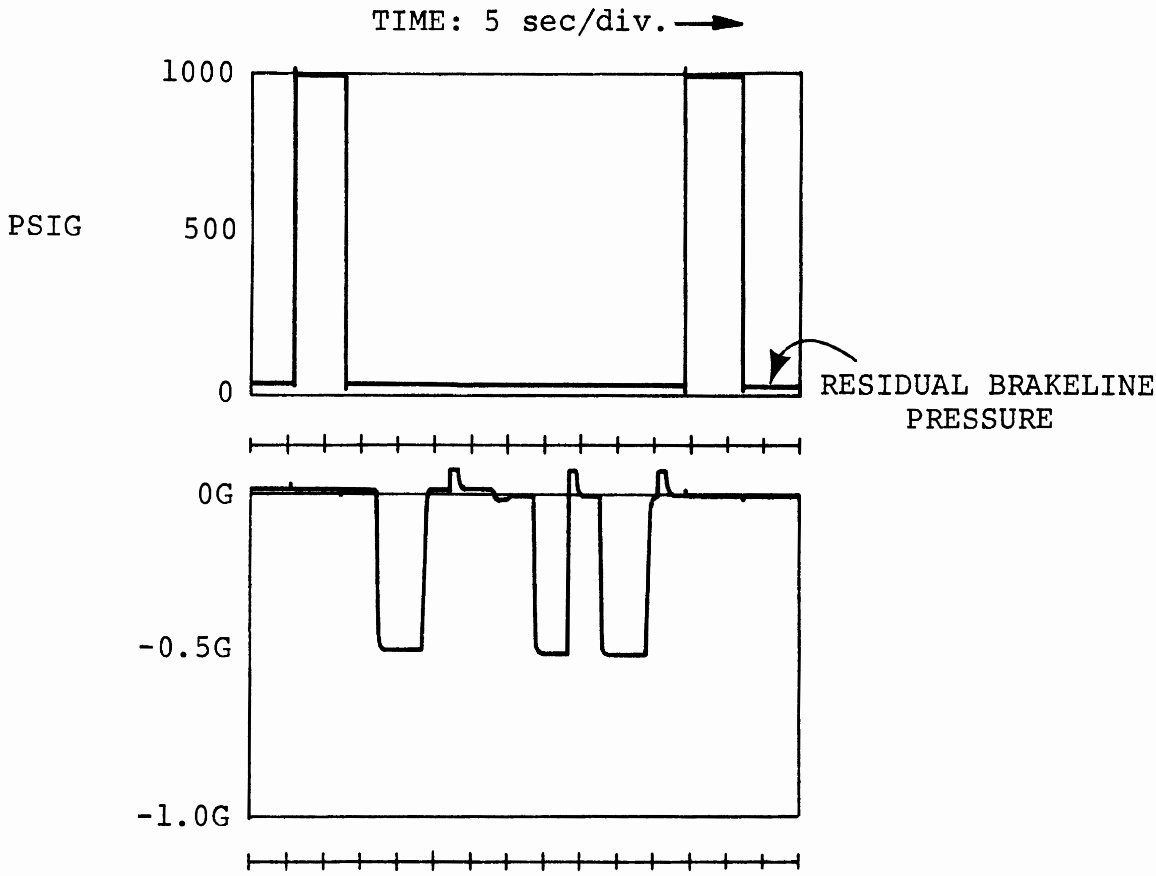
The ramp generator (integrator) was made an integral part of the equipment and was used to make a daily calibration check. During the data collection period the largest deviation from 0.5 G reading on calibration check was 0.02 G (one small division - 2 percent of full scale) and this was principally due to DC drift only requiring adjustment of the zero deflection (0.0 G) pen position. Accuracy of the deceleration data is estimated to be 3 percent of full scale (0.03 G) or better.




A.III.2. Top: Accelerometer calibration check voltage. Bottom: Acceleration check and accelerometer step response.




A.III.3. Velocity - derivative accelerometer low pass filter differentiator frequency response.




A.III.4. Top: Brake line pressure calibration check.
Botton: Deceleration calibration check.

## Brake Line Pressure

Brake line pressure signal was obtained from a Bourns Type 304 (0-1500 psig) pressure transducer mounted in the front brake line.

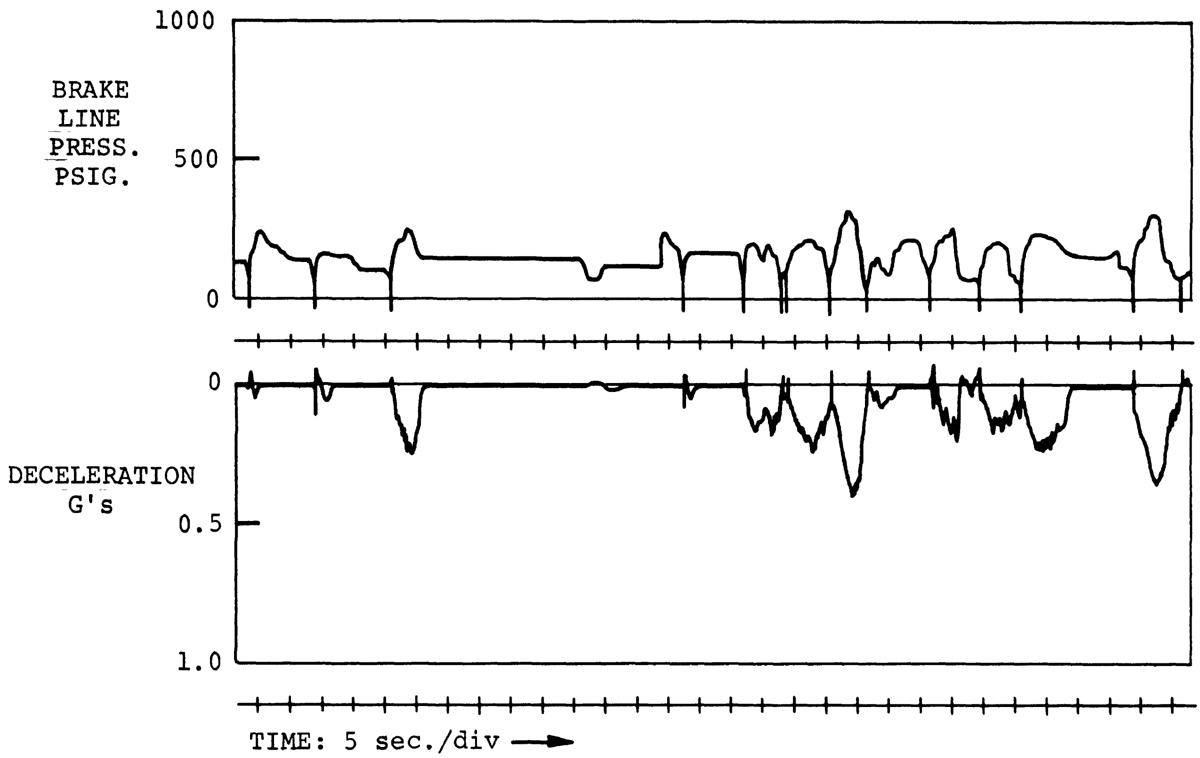
## Brake Line Pressure Calibration

Initial calibration of the brake line pressure channel was made with a 0-1500 psig test gauge mounted in the brake line along with the pressure transducer. A pressure of 1000 psig was applied through the brake pedal and the PRESSURE CALIBRATE potentiometer was adjusted to obtain full scale deflection on the recorder. Then the CAL CHECK VOLTAGE ADJ. potentiometer output was switched in through the Operate/CAL CHECK switch and this potentiometer was set to give full scale deflection on the pressure channel. The test gauge was removed. A daily check of the pressure channel calibration was made using the calibration check voltage corresponding to 1000 psig. A sample of this calibration check is shown in Figure A-III.4. The 20 to 30 psig pressure recorded with the brakes released is the residual pressure in the brake line due to the check valve in the master cylinder.

During the data collection period the largest deviation from 100 psig reading on calibration check was about 20 psi (1 small division - 2 percent of full scale) and this was principally due to DR drift of the zero deflection pen position on the recorder. Accuracy of the brake line pressure data obtained is estimated to be 3 percent of full scale (30 psig) or better.

## Sample Data

Figure A-III.5 gives a sample of the brake line pressure and deceleration data records obtained with the deceleration magnitude instrumentation package.




A.III.5. Brake line pressure and deceleration data sample.

## INSTRUCTIONS TO DRIVER

1. This car is on a special test which requires the use of some instrumentation that has been placed in the trunk.
   Please do not leave this vehicle without first locking it, and you should not surrender the keys to any other individual, i.e., parking lot attendants.
2. When driving this car please be sure not to ride the brake pedal.
3. Please FILL OUT THE ATTACHED TRIP SHEET BEFORE AND AFTER EACH TRIP.

Thank you

TRIP SHEET

| Trip No. | Name of Driver | Odometer at Start of Trip | Time at Start of Trip | Approximate Miles Driven in: | | | Odometer at End of Trip | Time at End of Trip | Do You Normally Brake With Right (R) or Left (L) Foot | Use this Space for Comments |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | City-Speed Limit 40 or Less | X-Way or Freeway | Country-Speed Limit 40 or Above | | | | |
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