BRAKING CAPABILITIES DURING DOWNHILL SPEED CONTROL AND EMERGENCY STOPPING

Paul S. Fancher

Research Scientist

University of Michigan Transportation Research Institute

SYNOPSIS:

The downhill braking performance of heavy duty vehicles is presented considering first the fundamentals of steady state, downhill, travel. The various parasitic drag functions are defined and the horsepower equivalents for engine drag, tire rolling resistance and aerodynamic drag are presented as a function of vehicle speed. For example U.S. vehicles, the results show that fuel economy improvements to vehicles in the late seventies caused as much as a 100 hp reduction in natural parasitic drag available at 55 mph. Accordingly, the demand for foundation braking and engine retarders has increased substantially. The performance of foundation brakes and retarders for absorbing energy during mountain descents is illustrated. Recommendations for safe descent speed as a function of the slope and length of the grade.

Emergency braking is discussed in terms of the static considerations of vehicle geometry and mass center location, static weight distribution, brake proportioning, and tire/road friction condition. Normalized illustrations of performance depict maximum deceleration capability over the range of tire/road friction. Examples of tractor semitrailer combinations show the difficulties faced in attempting to achieve good braking efficiency in the empty and fully loaded states.

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The Downhill Descent Process [2]

The control of heavy trucks during steep mountain descents is a safety problem that highway departments, truck drivers, fleet owners, brake and retarder manufacturers, and agencies of the federal government have addressed in various ways. For example, highway departments have built run-off ramps or provided "sand piles" for stopping heavy runaway trucks at selected sites [3]. To prevent brake fade and subsequent loss of speed control, drivers of heavy vehicles have learned to proceed down steep grades at moderate speed and in an appropriately low gear. Safety-conscious fleet owners have established maintenance and inspection programs to ensure proper brake adjustment. Equipment manufacturers have developed (1) economical, fade-resistant brakes and (2) auxiliary braking devices (retarders) for supplementing the foundation brakes. The federal government has supported work aimed at developing a "Grade Severity Rating System" [2] that would employ road signs to inform drivers of (1) the severity of an approaching hill and (2) safe operating speeds, depending upon the weight of the vehicle. The evidence from accident studies (see Section 3 of [2]) and records of run-off ramp usage indicates the existence of a significant truck runaway problem and thereby provides a safety-oriented justification for all efforts aimed at reducing the truck runaway problem.

Given the premise that a downgrade descent problem exists, the purpose of this section is to use physical principles and engineering methods to identify the retarding capability necessary for preventing a specific vehicle from accelerating on a particular highway grade.

The total retarding capability of a vehicle comes from a number of sources in addition to the foundation brakes and the retarder (if a retarder is installed). This situation is illustrated in Figure 1 which is a free-body diagram of a tractor-trailer combination making a constant-speed descent on a road whose grade, in percent, is given by 100 tan θ. For constant speed, the gravitational propelling force, W sin θ, is balanced by all of the forces resisting forward motion. With the drive wheels coupled to the engine, the forces resisting forward motion are

(1) aerodynamic drag,

(2) tire rolling resistance,

(3) retarding force at the drive wheels deriving from the torque created by the engine with throttle closed, F_{xeng}, and

(4) braking forces, F_{xbi}, created at each braked wheel by means of a mechanical friction brake and/or a retarder.

If we assume that a retarder is not provided and that, at a given line pressure, all brakes are generating an equal amount of brake torque,* the laws of physics yield the following expression for the horsepower that must be continuously absorbed by a single brake, viz.:

$$HP_{\substack{single\\brake}} = \frac{1}{n}\left[\left(W \sin\theta - F_{x_{RR}} - F_{x_{aero}}\right)\frac{V}{375} - HP_E\right] \qquad (1)$$

*In practice, this does not occur because the push-out pressures may vary from brake to brake, and the torque per unit line pressure may be set differently on each axle, and brake adjustment may vary from brake to brake.

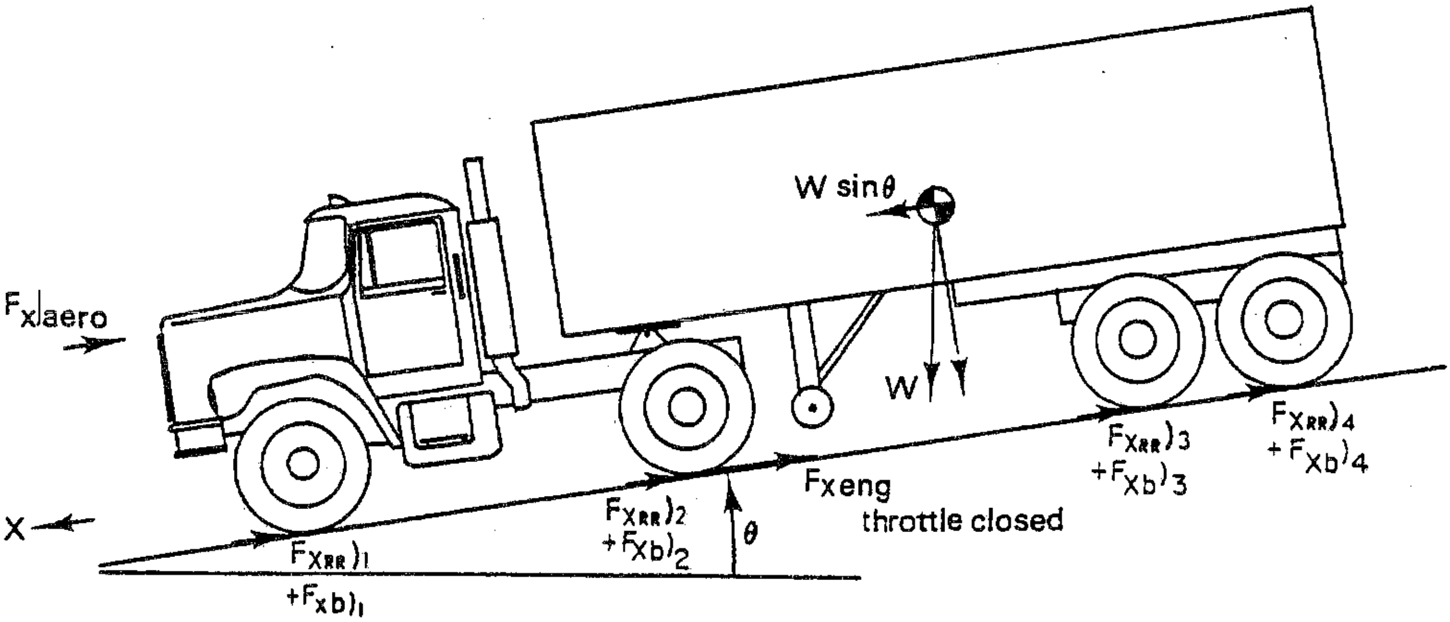


Figure 1. Free-body diagram of a four-axle tractor-trailer descending a grade at constant speed.

where

n = number of braked wheels

$F_{x_{RR}}$ = tire rolling resistance summed over all wheels, lb

$F_{x_{aero}}$ = aerodynamic drag force, lb

HP_E = horsepower absorbed by the engine with the throttle closed

V = speed of descent, mph

W = total weight of the combination vehicle, lb

θ = angle of the road plane with respect to the horizontal

Clearly, the horsepower to be absorbed by a single brake will increase:

(1) as the number of operational or installed brakes decreases

(2) with increased speed of descent

(3) with increased total weight

(4) with increased grade angle

(5) with decreased rolling resistance of the tires

(6) with decreased aerodynamic resistance

(7) as the horsepower that can be absorbed by the installed engine decreases

Accordingly, Equation (1) shows that existing plans to make trucking more fuel efficient by:

(1) increasing the total weight,

(2) reducing (a) the rolling resistance of tires and (b) aerodynamic drag, and

(3) reducing the internal losses in the engine

will require that each brake absorb more horsepower on a given grade at a given speed. If additional sources of retardation are not utilized, it can be anticipated that the trends to make trucking more fuel efficient will require that trucks descend grades at lower speeds to keep the horsepower absorbed by the mechanical friction brake within acceptable limits. It follows that trucking productivity will decrease in mountainous areas and that the potential for brake overheating and fade in long descents will likewise increase.

In order to reduce the above discussion to a quantitative basis, Equation (1) can be expanded to reflect the properties of both past and present-day (or future) trucks. Typical expressions for the retarding power provided by aerodynamic drag and rolling resistance are as follows:

$$HP_A = \frac{C_W \, AV^3 \, C_A}{375} \qquad (2)$$

where

HP_A is the horsepower absorbed through aerodynamic drag

A is the frontal area of the vehicle in ft^2

V is the velocity in mph

C_W is a drag coefficient (approximately 0.002)

and C_A is a coefficient representing the influence of drag reduction devices (C_A = 0.9 to 0.75 for various drag reduction improvements)

$$HP_{RR} = \frac{C_{RR} \, WV \, C_T}{375} \qquad (3)$$

where

HP_{RR} is the horsepower absorbed by rolling resistance

C_{RR} describes the tire/road interface (C_{RR} = 0.012 is a representative value for good roads)

C_T describes the tire construction (C_T = 1.0 for bias tires, C_T = 0.7 for radial tires)

W is the vehicle weight (GVW) in lbs.

With respect to engine friction, a standard 290 hp engine produced in 1974 absorbed approximately 113 hp including the effects of driveline efficiency and accessory power, while a 300 hp engine produced in 1980 will absorb approximately 75 hp under the same conditions [2]. Figure 2 has been constructed to illustrate representative magnitudes for these sources of "natural" retardation for an 80,000-lb vehicle operated at velocities from 10 to 60 mph. (The values plotted in Figure 2 are tabulated in Table 1.) Examination of these typical results indicates that fuel economy measures may reduce a vehicle's natural retardation by approximately 100 hp at 55 mph.

In addition, these results (Figure 2 and Table 1) show that the contributions of engine friction, aerodynamic drag, and rolling resistance are approximately equal at 55 mph, although the importance of aerodynamic drag reduces dramatically at lower speeds.

The influence of natural retardation on the power balance needed to maintain constant velocity on a downgrade is summarized by the following equation:

$$HP_{B/R} = HP_H - HP_N \tag{4}$$

where

$HP_{B/R}$ is the required braking/retarder horsepower

HP_H is the power supplied by the downgrade

and

$$HP_N = HP_E + HP_{RR} + HP_A \tag{5}$$

The horsepower of a particular downgrade, θ, is linearly related to the vehicle speed by the following equation:

$$HP_H = \frac{W \theta V}{375} \tag{6}$$

(where $\theta \approx \sin \theta$ for highway grades).

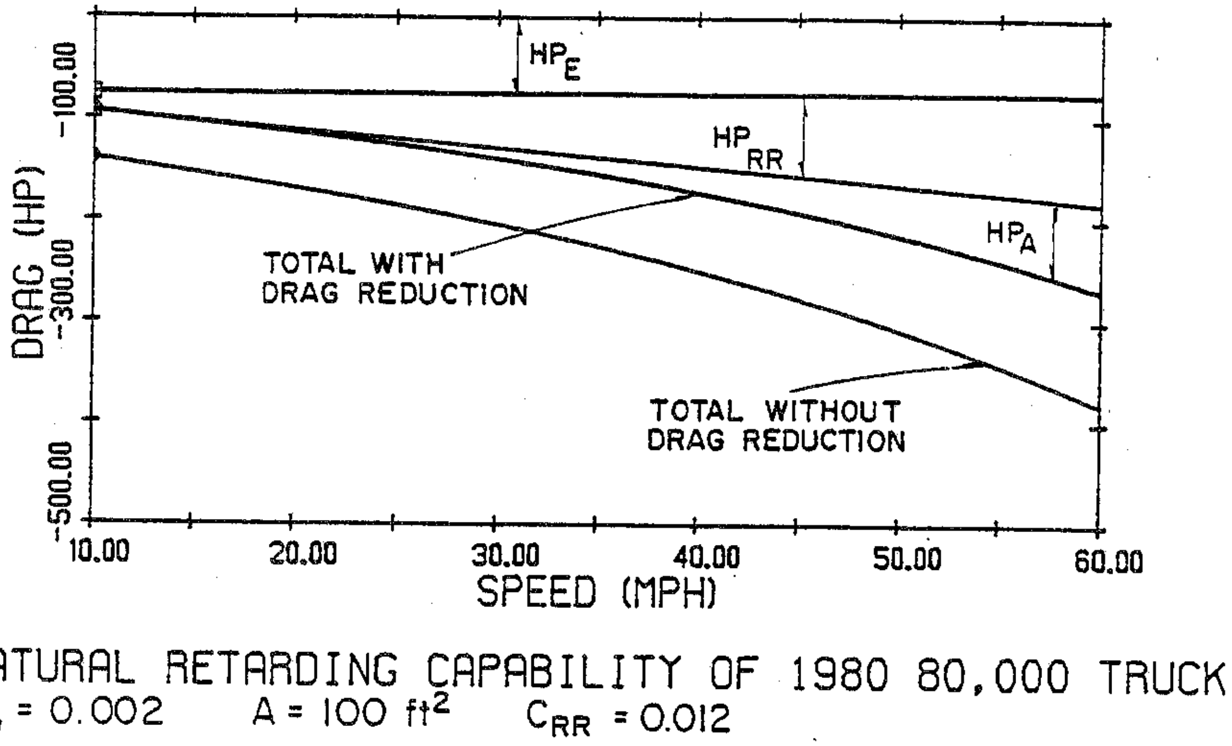


Figure 2. Magnitude of the sources of natural retardation.

Table 1. Sources of Natural Retardation

V_{mph}	Aerodynamic HP_A (C_A = 0.75)	Rolling Resistance HP_{RR} (C_T = 0.7)	Engine HP_E	Total With Fuel Saving Devices	Total Without Fuel Saving Devices	ΔHP
10	0	18	75	93	140	47
20	3	36	75	114	168	54
30	11	54	75	140	204	64
40	26	71	75	172	249	77
50	50	90	75	215	307	88
60	86	108	75	269	381	112

Example results from applying Equations (4), (5), and (6) to grades ranging from 2 percent to 10 percent are presented in Figure 3. This figure graphically illustrates the influence of natural retardation on the required braking and/or retarder horsepower for the example vehicle used in constructing Figure 2. In this case, for velocities above 30 mph the required braking/retarder horsepower, $HP_{B/R}$, happens to be approximately equivalent to the horsepower on a grade that is 2 percent less than the actual grade. As indicated in Figure 3, the natural retardation of this example vehicle is sufficiently large for preventing runaway on all grades less than or equal to 2 percent.

If the example vehicle were not equipped with radial tires, aerodynamic aids, and a low-friction engine, the natural retardation would have been enough for holding velocity on a grade of approximately 3 percent rather than on the 2-percent grade shown in Figure Hence, the reduction in natural retardation due to fuel economy measures (roughly 100 hp) has approximately the same influence as operating on grades that are effectively 1 percent steeper than they are for a comparable vehicle without fuel economy improvements.

Now consider the use of a retarder to absorb the required braking/retarder horsepower.

For the purposes of this discussion, retarders will be divided into two major categories, either "driveline" or "engine speed" retarders. A driveline retarder applies torque to a rotating element connected to the wheels without an intervening transmission. As the name implies, an engine speed retarder operates on the engine side of the transmission. The engine speed retarder produces a braking force at the wheels only when the transmission is in gear.

Since the horsepower capability of a driveline retarder is independent of engine speed, the determination of the downgrade performance of a vehicle equipped with this type of retarder is easy to

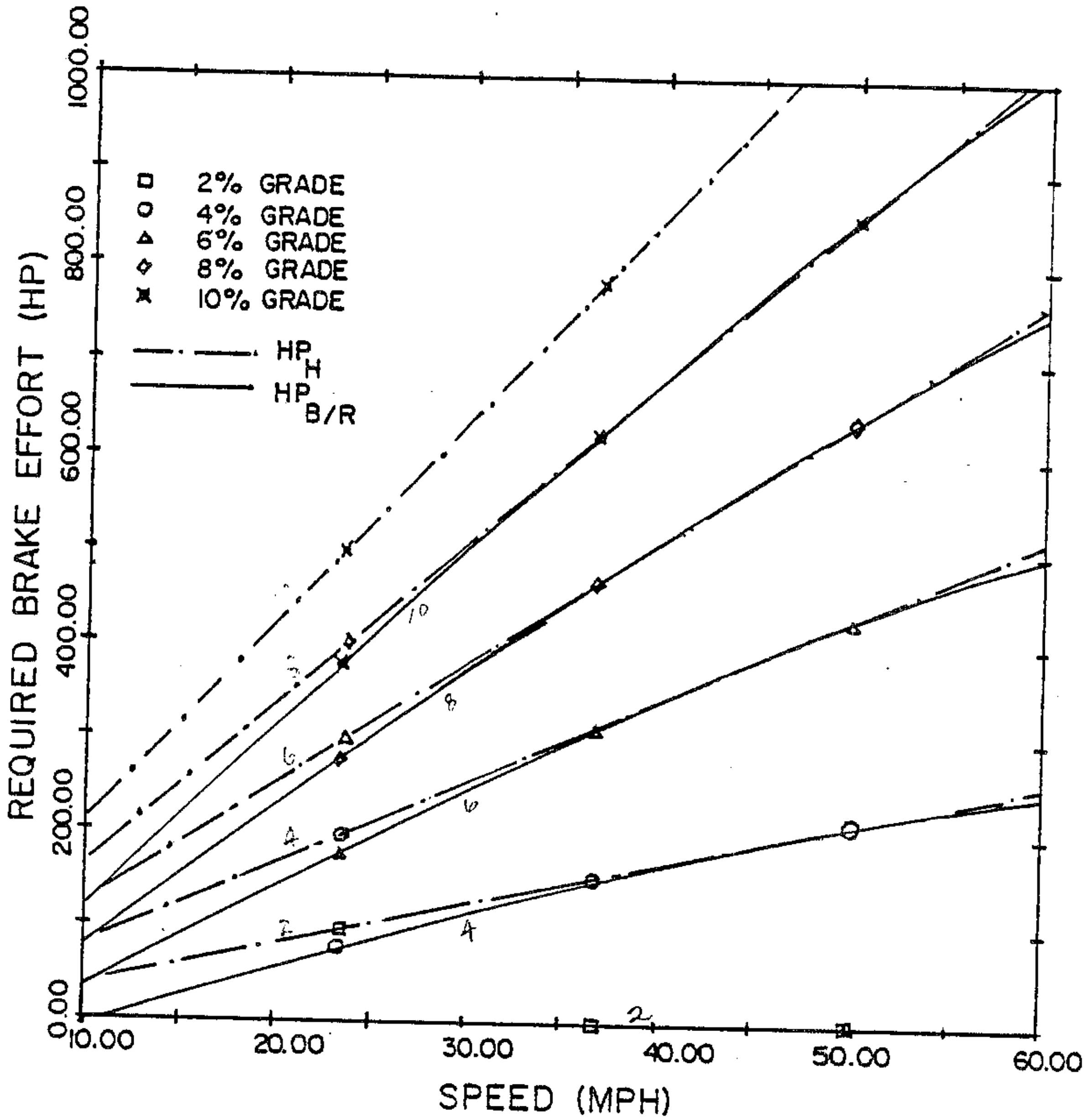


Figure 3. Required braking power.

explain. As indicated by Equation (6), the weight and forward velocity of the vehicle plus the slope of the grade are the important factors contributing to the horsepower of a given hill, i.e., HP_H. Given information on the natural retardation, the maximum vehicle weight, the slope of the steepest downgrade along the route, and an acceptable velocity on the steepest downgrade, the required retarding horsepower can be readily determined. A driveline retarder that can absorb this amount of required horsepower will maintain the desired velocity on the steepest hill to be encountered.

The characteristics of retarder horsepower as a function of speed can have an influence on the type of equilibrium that exists at a selected maximum speed. Figure 4 contains two examples illustrating a stable and an unstable equilibrium. In both examples, the required braking horsepower curve for a 6 percent grade (from Figure 3) represents the steepest hill to be considered. Also, in both examples, 40 mph is selected as the acceptable speed. In the first example, operation above 40 mph will result in surplus braking power tending to slow the vehicle to 40 mph, while operation at less than 40 mph will result in a deficiency of braking power causing the vehicle speed to increase towards 40 mph. Thus a stable equilibrium is achieved at 40 mph. In example 2, 40 mph is an unstable operating condition—above 40 mph the grade is sufficient to cause the vehicle to speed up, below 40 mph the retarder will reduce vehicle speed. If vehicle speed is less than 40 mph, the driver could cycle the retarder on and off to increase speed, but if the speed ever got above 40 mph, the retarder could not control speed and the foundation brakes would have to be used to reduce speed to 40 mph. Clearly, the retarder with an unstable equilibrium requires driver control actions that are not necessary in the stable equilibrium situation.

For an engine speed retarder, the selection of adequate retarder horsepower is easily demonstrated graphically. Figure 5 shows the power versus velocity characteristics of a hypothetical engine speed retarder superimposed on the 6 percent grade curve from Figure 3.

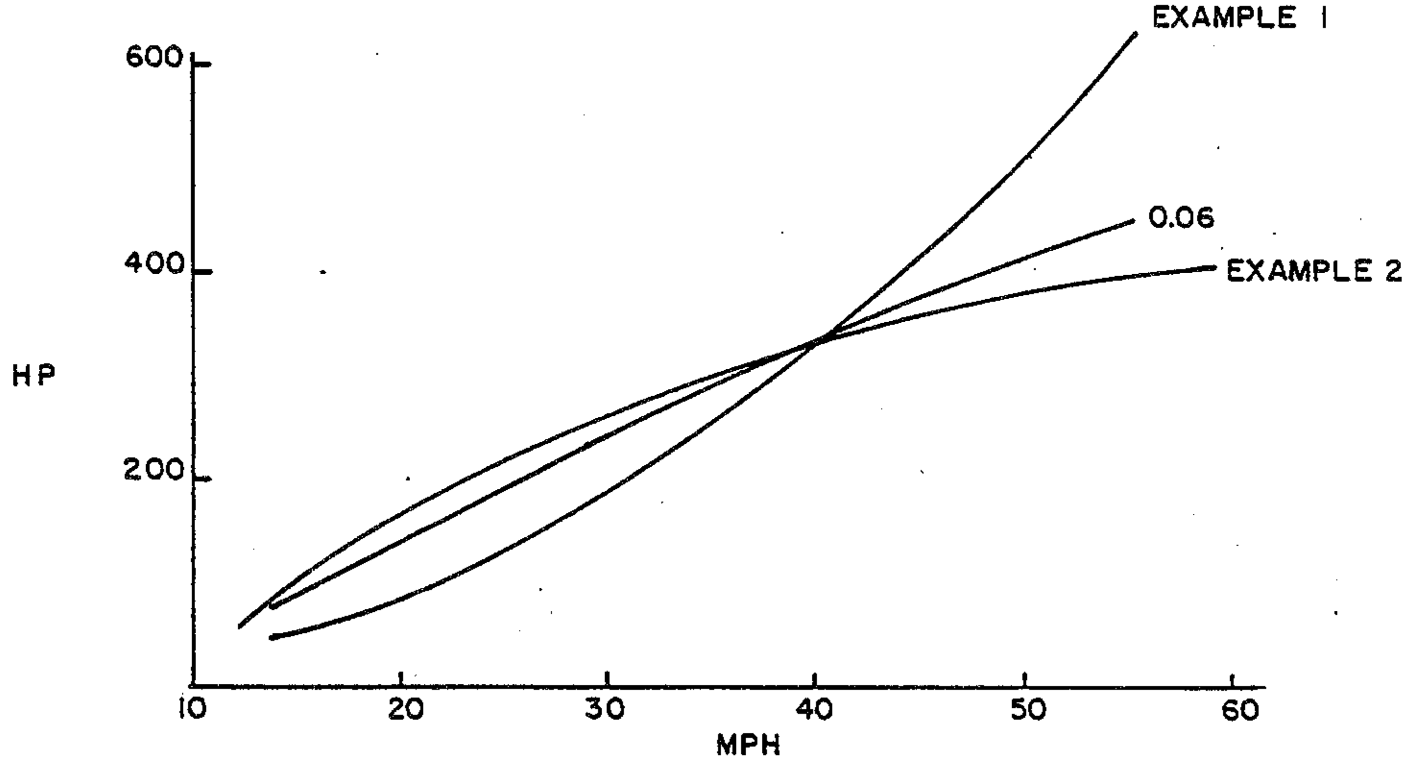


Figure 4. Examples of stable and unstable equilibrium.

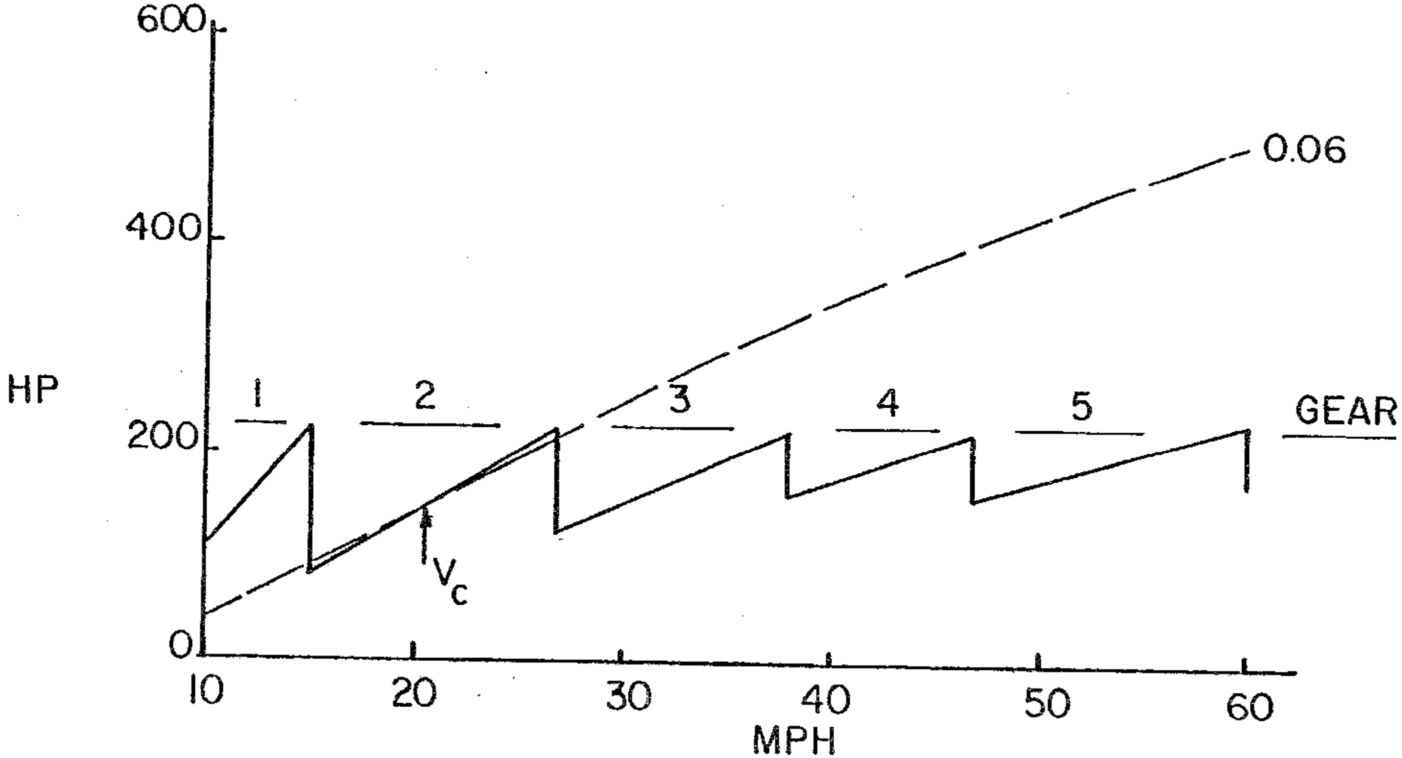


Figure 5. Control speed, V_c, for an engine speed retarder.

The retarding horsepower is shown to fall off with engine speed in each gear range. For the example shown in Figure 5, the equilibrium speed is approximately 20 mph in second gear. This speed, V_C, occurs at a stable equilibrium condition.

If 20 mph were not fast enough to be acceptable, one could consider (1) a different transmission with a more favorable set of speed ranges for its gears, (2) a higher horsepower retarder, or (3) the use of the foundation brakes in addition to the retarder.

Although discussions with personnel from retarder manufacturing companies have indicated that they specify retarders capable of maintaining speed control without utilizing the foundation brakes, it is of interest to consider the use of foundation brakes for maintaining speed control with and without the aid of a retarder. For a number of years highway engineers have been interested in this problem and in devising schemes of rating downhill sections of road to aid truck drivers. This interest has led to the development of a proposed grade severity rating system based on brake temperature [4]. The proposed rating system represents a trade-off between the desire to travel rapidly and the need to prevent overheating the brakes to the point where they can no longer supply the torque required to control vehicle speed. The following discussion examines the implications of restricting brake temperature to be at, or below, a specified maximum value.

Chapter 10 contains an analysis of the brake temperature changes taking place during a constant velocity descent on a fixed grade of given length. The basic result obtained in Chapter 10 for the maximum temperature (which occurs at the bottom of the hill) is expressed by the following equation:

$$Q_f = Q_o \, e^{-L/V\tau} + \left(\frac{HP_B}{h(V)} + Q_a\right)\left(1 - e^{-L/V\tau}\right) \quad (7)$$

where

Q_f is the final brake temperature

Q_o is the initial brake temperature at the top of the grade

L is the length of the grade

V is the velocity

τ is the thermal time constant of the brakes

HP_B is the horsepower input to the brakes (i.e., the absorbed horsepower)

h(V) is a cooling coefficient that is a function of velocity

Q_a is the ambient temperature

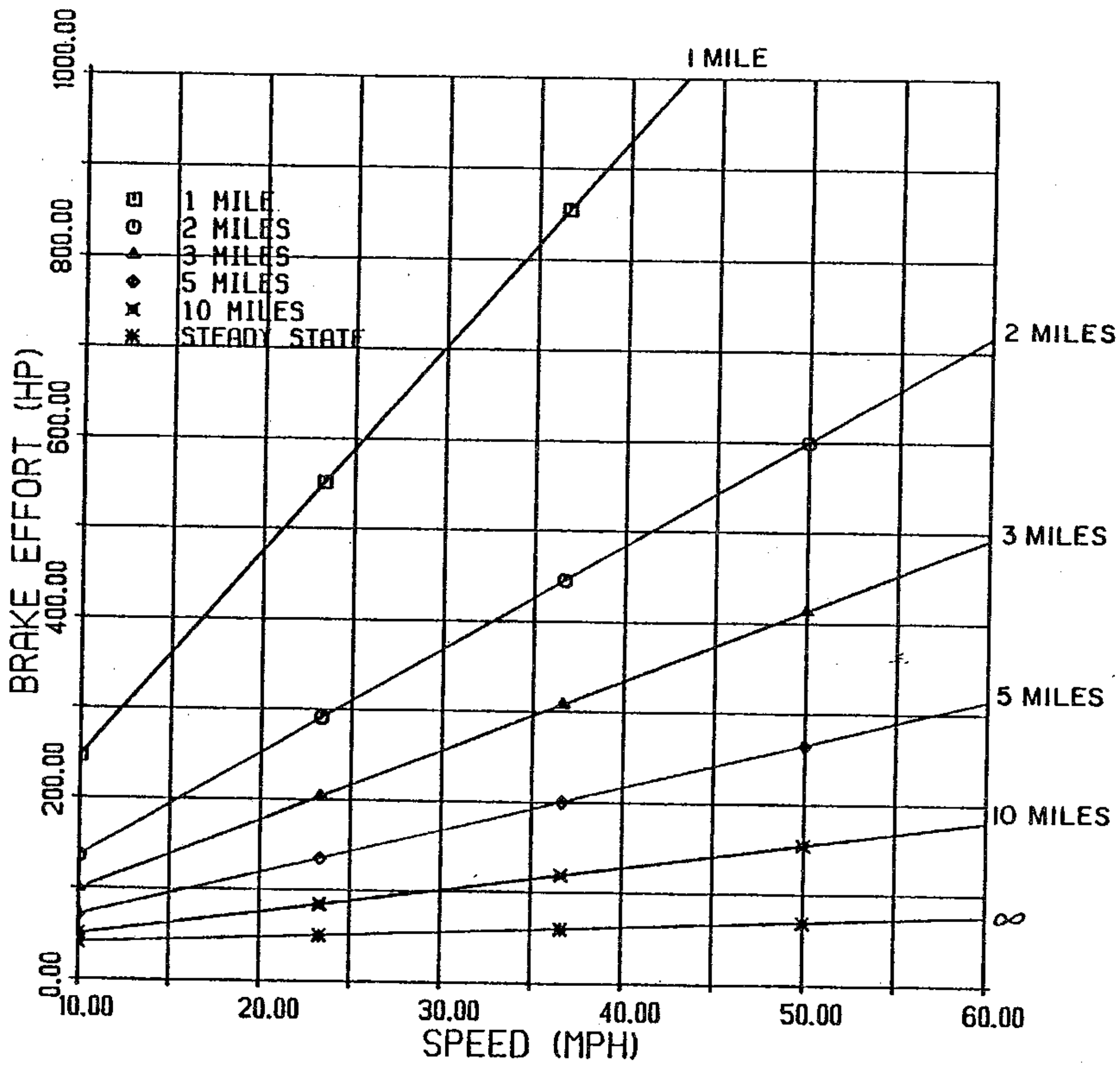
Note that $L/V = t_f$, the length of time required to descend the grade.

In order to emphasize the influence of the length of grade, and control velocity, V_c, on the horsepower that the brakes can absorb without exceeding the temperature boundary, Q_f, Equation (7) can be restated (rearranged) as follows:

$$HP_B = \left[\left(\frac{Q_f - Q_o e^{-L/V_c\tau}}{1 - e^{-L/V_c\tau}}\right) - Q_a\right] h(V_c) \qquad (8)$$

Figure 6 presents the results of applying Equation (8) to various length grades over the range of velocities from 10 to 60 mph. This figure shows the horsepower that the brakes can absorb without violating the temperature constraint for the five-axle, tractor-semitrailer vehicle studied in [4].

Equations (4) and (8) form a set of simultaneous equations for HP_B and V_c with the independent variables being L and θ. (An example graphical solution of these equations can be obtained by (1) superimposing Figures 3 and 6 and (2) reading off the velocity and horsepower at points corresponding to known (selected) grades, θ, and lengths of grade, L.) The solution of these equations for a hill specified by a grade, θ, and a length of grade, L, consists of the



$Q_o = 150°$ F $Q_g = 90°$ F $Q_f = 425°$F

Figure 6. Brake capabilities based on a maximum temperature.

safe descent speed, V_C, and the amount of brake horsepower, HP_B, required for descending that hill at the safe speed. ("Safe speed" means that speed for which Q_f, the final temperature, will not exceed the selected maximum temperature (e.g., 425°F for drum-type brakes). In fact, using the approach described, Q_f will equal the selected maximum temperature, thereby providing the minimum time (maximum velocity) solution.)

A typical solution for the safe descent speed as a function of grade and grade length can be used to develop an understanding of the implications of setting a temperature limit. Figure 7 has been generated using the vehicle and brake parameters employed in constructing Figures 3 and 6. As illustrated in Figure 7, the allowable speed on a steep grade has a rather abrupt transition between being almost independent of length for long grades to being a very sensitive function of length in the region near the minimum length at speeds approaching 55 mph. For example, on a 6-percent grade ($\theta = 0.06$ radians) the vehicle can be operated at 55 mph if the grade is 2.9 miles long. However, if the grade is 3.0 miles long, the safe speed is 36 mph, and, if the grade is 5.0 miles long, the safe speed is 16 mph. For steeper grades this trend is even more accentuated. On an 8-percent grade, a change in length from 1.9 to 2.0 miles reduces the safe speed from 55 mph to 28 mph. These results indicate that for steep grades there is a sharply defined critical length above which the allowable speed of descent falls rapidly from 55 mph to below 20 mph.

Further insight into the meaning of setting a brake temperature limit can be derived from looking at graphically obtained solutions for horsepower and control speed on grades of 6 and 8 percent and at grade lengths of 2 and 3 miles, as portrayed in Figures 3 and 6, respectively. The appropriate curves from Figures 3 and 6 are displayed in Figure 8. The lower pair of curves (one for a 6 percent grade and the other for a length of 3 miles) are seen to merge at 40 mph and remain very nearly equal up to 60 mph. In this speed range, the increase in required braking horsepower due to increased speed on the grade is nearly matched by (1) the higher convective heat

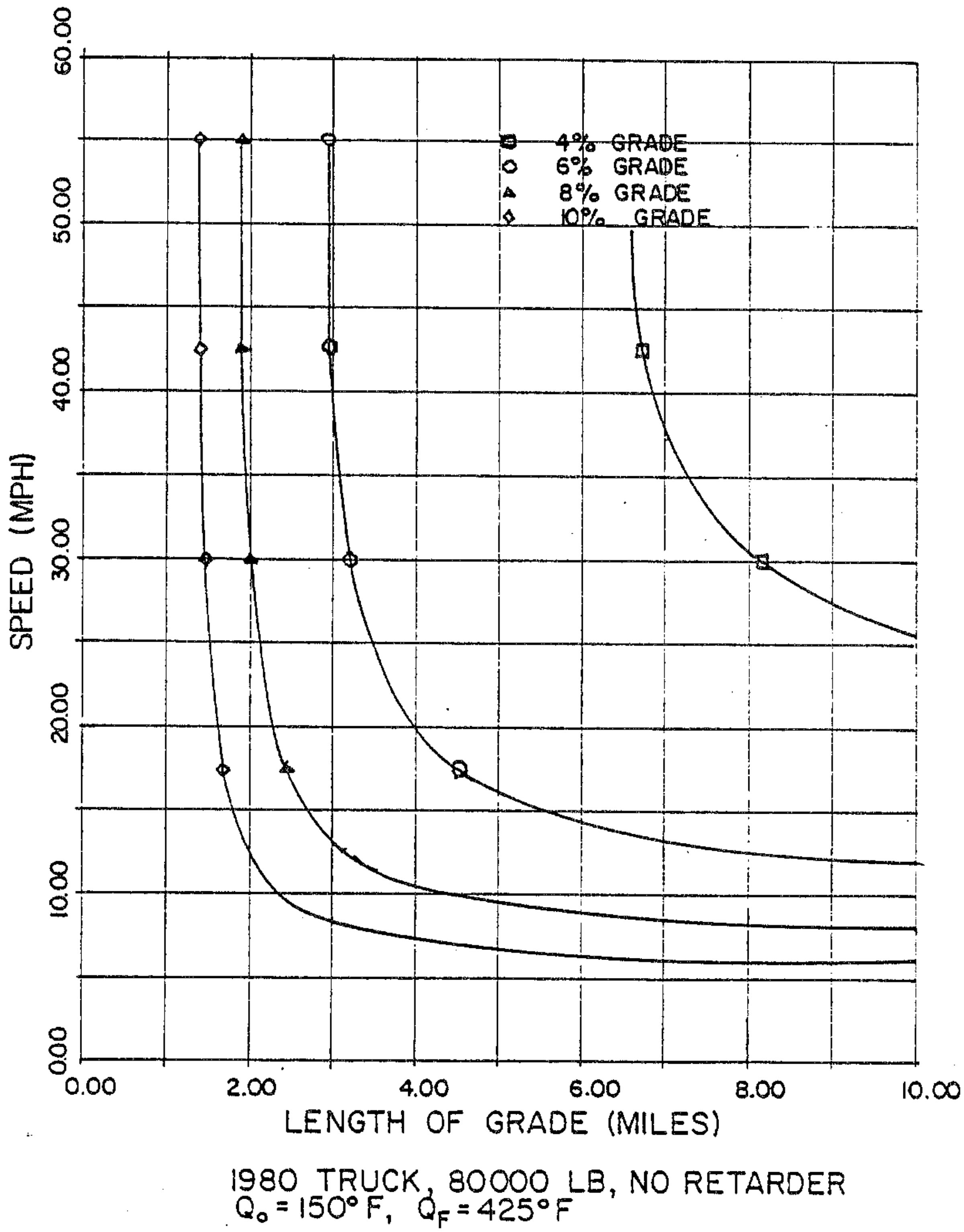


Figure 7. Allowable speed versus length of grade for various grades.

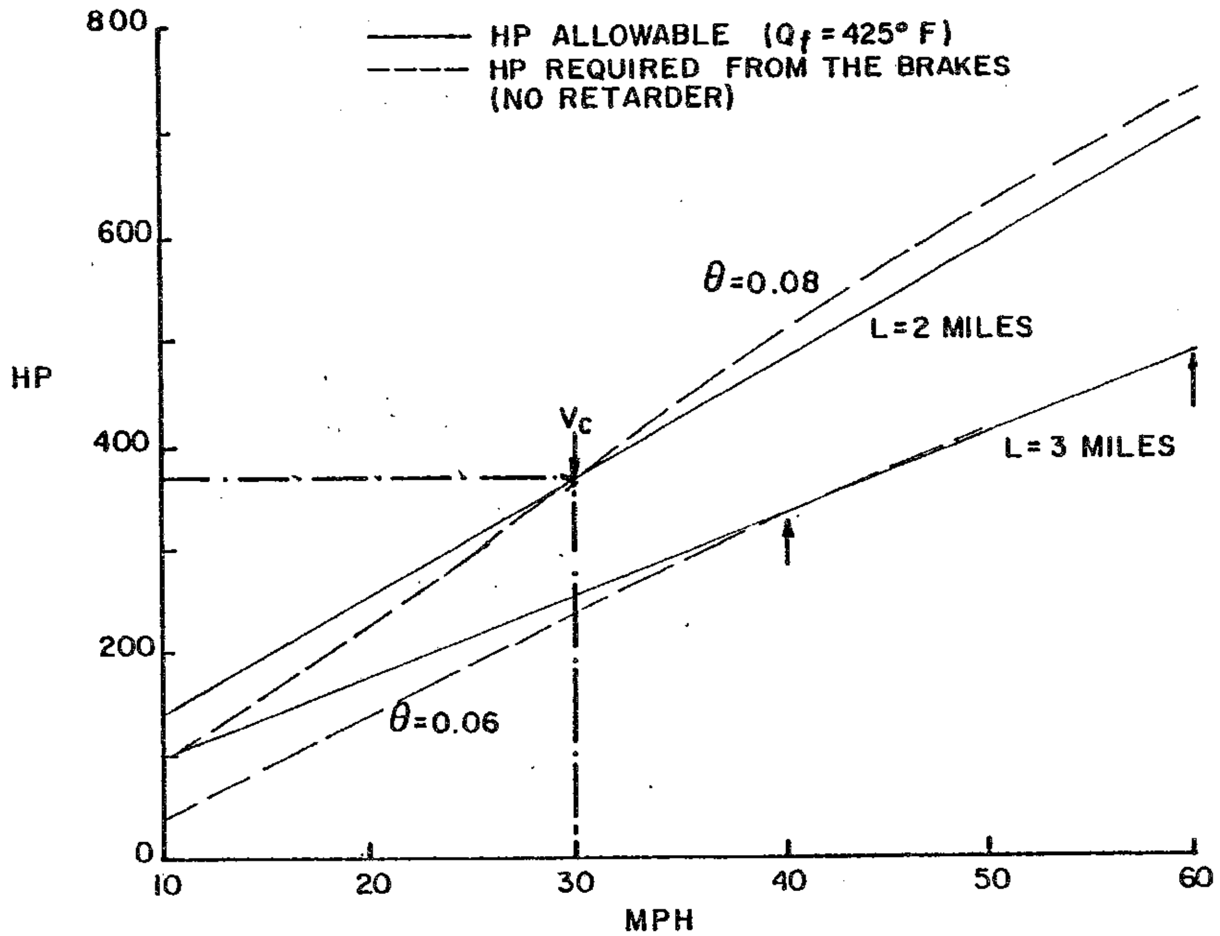


Figure 8. Graphical solutions for control speed and absorbed horsepower.

transfer and (2) the shorter time on the grade as speed increases. Clearly, small changes in length or grade can make large changes in the control speed in this case.

The upper pair of curves (for θ = 8 percent and L = 2 miles) intersect at a control speed of approximately 30 mph with an accompanying requirement for the brakes to absorb approximately 370 horsepower. The solution at 30 mph and 370 horsepower is a point of unstable "equilibrium" in the sense that if the speed exceeds 30 mph, there is no inherent mechanism to force the vehicle's speed back to 30 mph without exceeding the temperature boundary. However, for speeds up to 35 mph, an additional 10 horsepower of braking effort would be enough to cause the velocity to fall off towards 30 mph (the equilibrium point for a final temperature of 425°F). Hence, even if the vehicle speed did approach 35 mph and some slight additional braking were required to reduce speed, the final temperature would not necessarily exceed 425°F by a significant amount. Thus, it appears that small errors (on the order of 2 or 3 mph) in controlling speed will not lead to excessive temperatures.

However, a major difficulty associated with setting a temperature limit is the slowness of the process of cooling the foundation brakes. The length of time for cooling a brake from 425°F to 150°F (e.g., as might be considered in a grade severity rating system [4]). is on the order of 40 minutes, depending upon vehicle speed. For mountainous regions with closely spaced downgrades, the distance between applications of the brakes may not be far enough to allow the brakes to cool to 150°F. This point is illustrated in Figure 9, which was constructed using Equation (7) with $HP_B \equiv 0$. As shown in the figure, the example vehicle would have to travel 39 miles at 60 mph or 26 miles at 30 mph (without applying the brakes) to cool the brakes from 425°F to 150°F.* For a mountainous region with downgrades spaced approximately 7 to 10 miles apart, Figure 9 indicates that once the

*Two competing factors influence these results: (1) slower speed means longer cooling time and (2) higher speed provides a higher cooling rate. In this case, the slower speed yields the shorter distance.

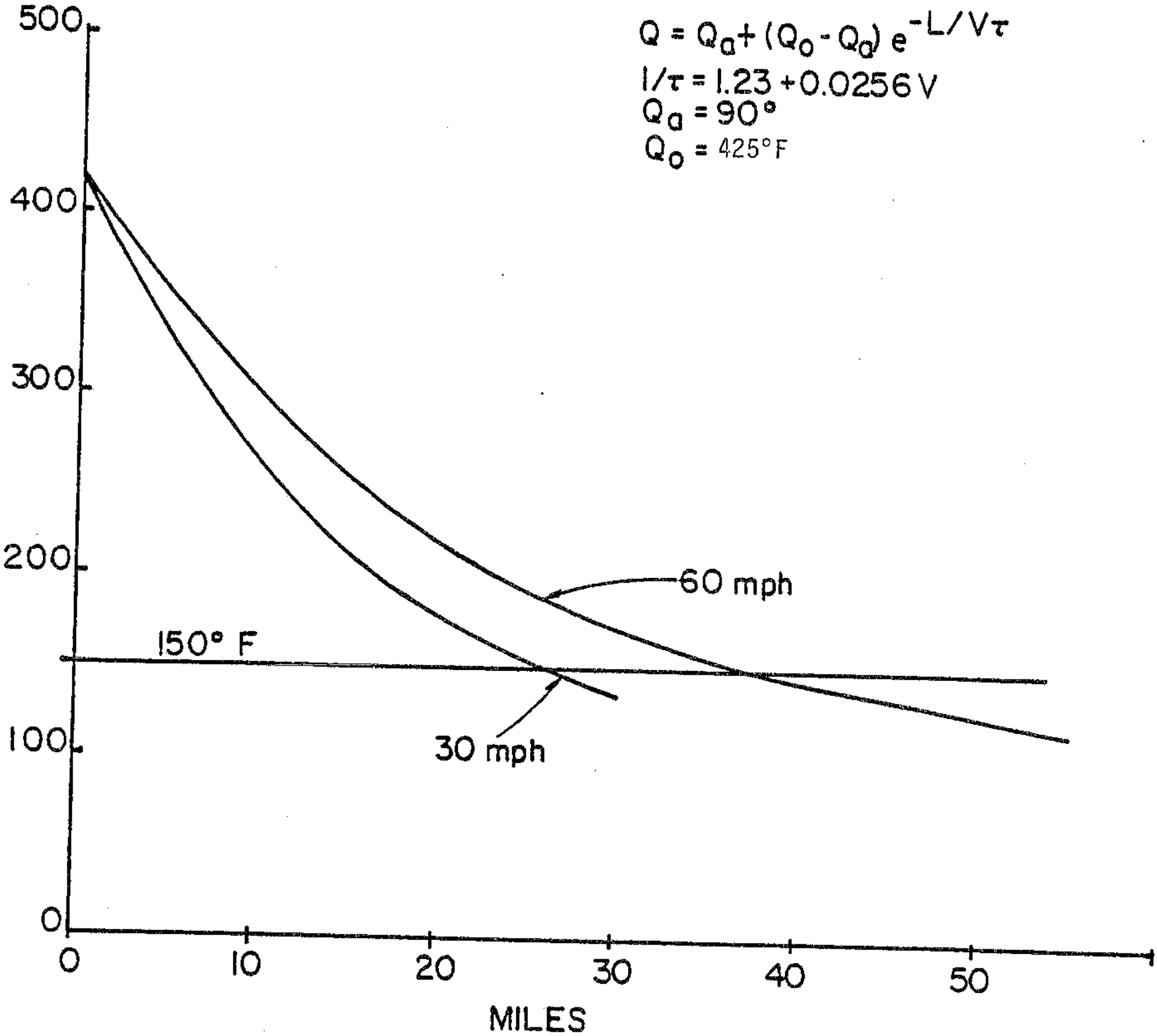


Figure 9. Miles to reach 150°F.

brakes are heated to 425°F, they will cool only to approximately 300°F for speeds in the range from 30 to 60 mph.

The initial brake temperature at the top of a downgrade is an important parameter in determining the control speed for descending the grade without exceeding the temperature limit. Specifically, a change from Q_0 = 150°F to Q_0 = 300°F has a large influence on the selected control speed, as shown in Figure 10. At 55 mph, for example, the allowable lengths of grade for various grades are shown in Table 2. Given that the driver may be unaware of brake temperature, the potential for an erroneous choice of speed for various length grades appears to be a hazard in mountainous areas unless the driver is able to follow carefully determined control speed information.

Table 2. Length of Grade, L, in Miles for V_C = 55 mph for Two Initial Brake Temperatures and Four Grades.

Q_0 °F \ θ Percent	4%	6%	8%	10%
150	6.6	2.9	1.9	1.4
300	3.3	1.4	0.9	0.7

Clearly, if both the foundation brakes and a retarder are used for downhill speed control, then speeds of descent faster than those applicable to operation with the brakes alone can be allowed without absorbing too much power in the foundation brakes. For example, Figure 11 indicates that a vehicle equipped with a retarder producing 200 hp over the normal influence of engine drag can operate at 55 mph on 4 percent grades up to at least 10 miles long without exceeding a brake temperature limit of 425°F even if the initial brake temperature at the top of the hill were 300°F. On a 6-percent grade that is 10 miles long, the control speed is shown to be 34 mph in Figure 11. In comparison, the results for a comparable vehicle using the foundation brakes alone (see Figure 10) are (1) 3.3 miles at 55 mph on a 4-percent grade and (2) a control speed of 11 mph on a 6-percent grade that

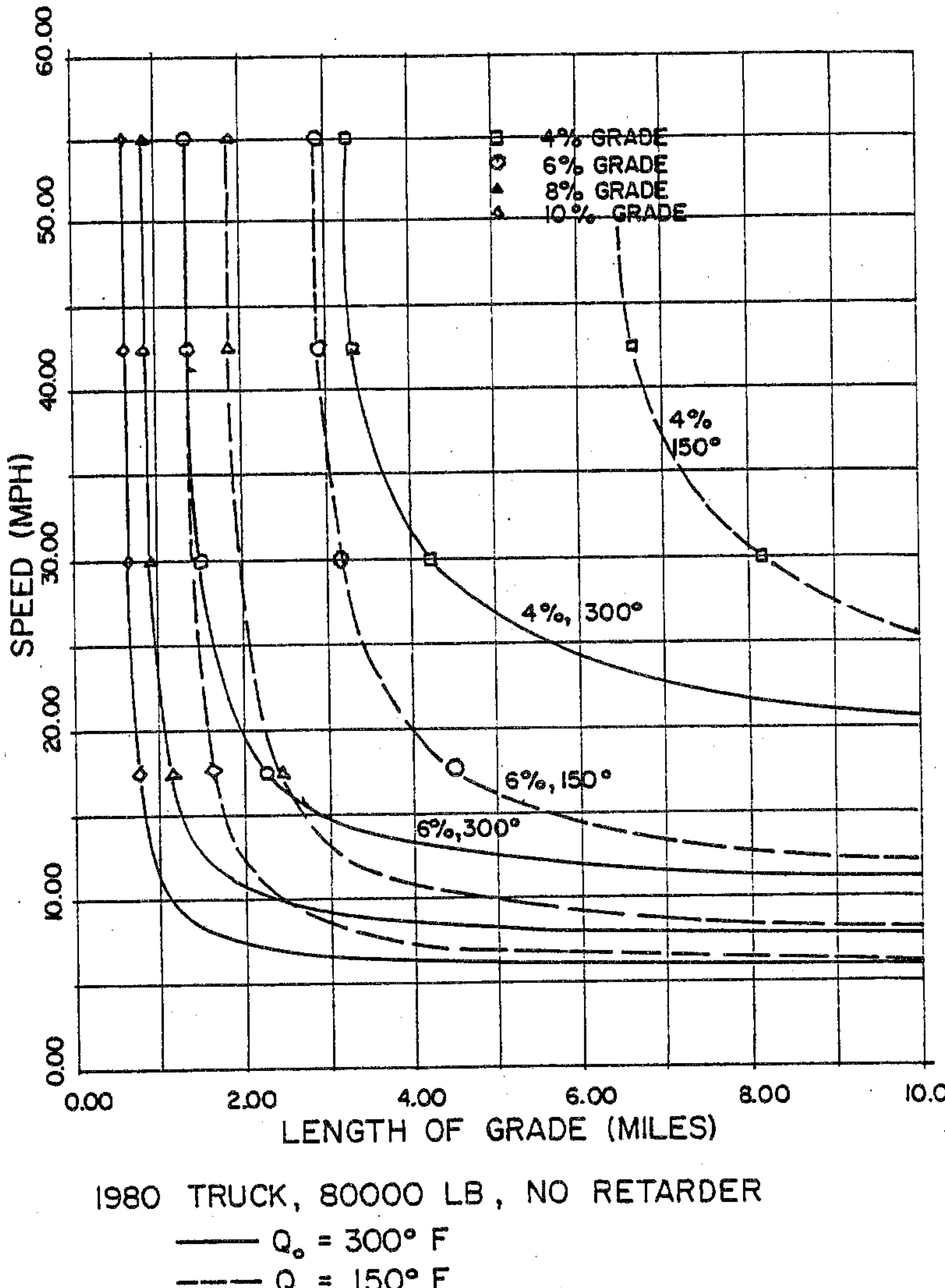
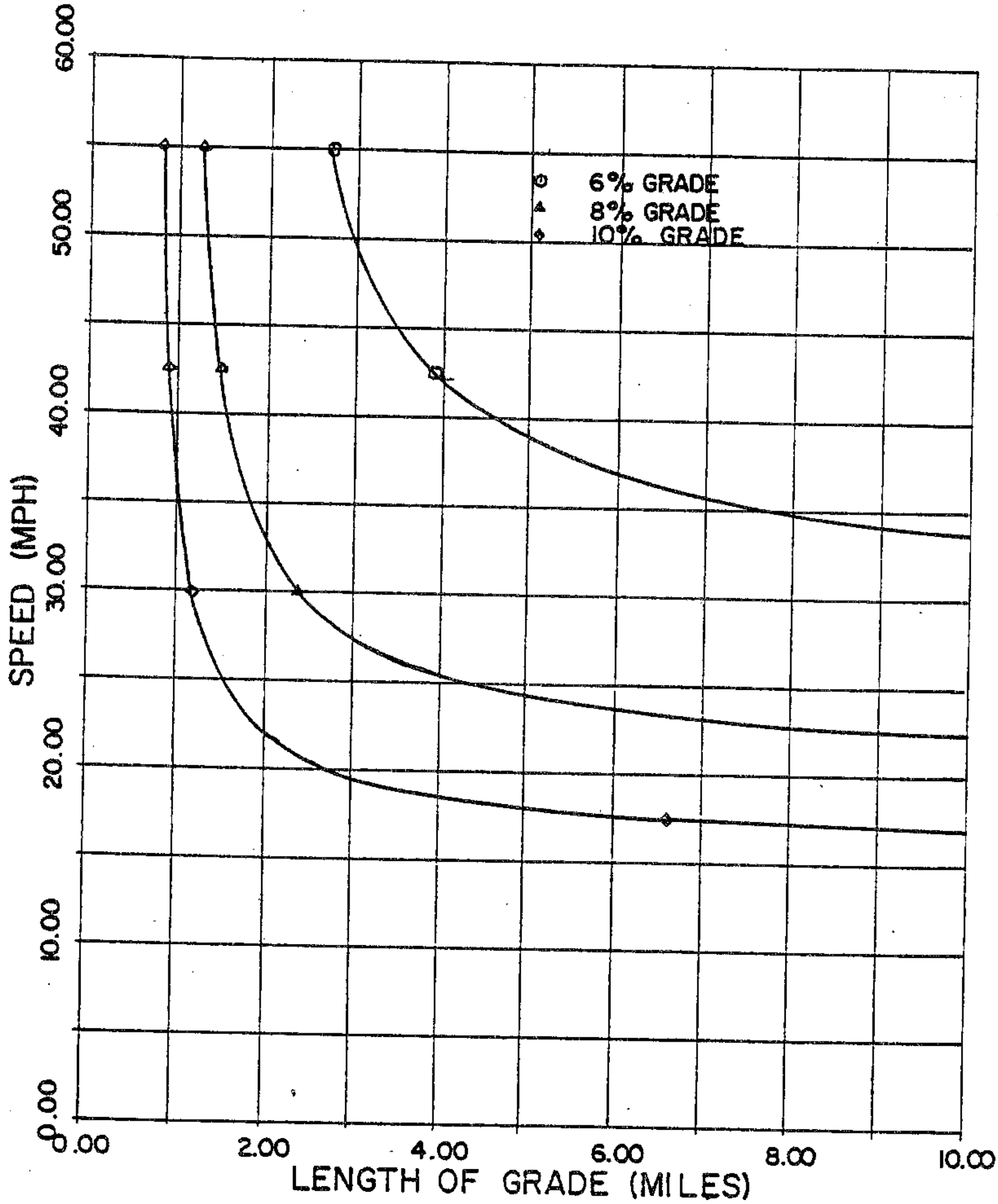


Figure 10. Influence of initial brake temperature on control speed.



1980 TRUCK, 80000 LB W, Q_o = 300, 200 HP RETARDER

Figure 11. Results for combined use of a retarder and the brakes.

is 10 miles long. Although the combined use of both the retarder and the foundation brakes has the disadvantage of not reserving the foundation brakes for emergency situations, the combined use is very effective in increasing speed and may well represent the actions of drivers that are pressed for time.*

*In practice, drivers would need to be very familiar (or well informed) with regard to the route, the weight of their vehicles plus load, and thermal properties of the vehicle's brake system in order to operate safely while using both the foundation brakes and the retarder to minimize time.

Analysis of Tractor-Semitrailer Braking as a Quasi-Static Process [1]

The preliminary ideas presented in this section are the result of a short-term effort to develop a simplified, analytical approach for assessing the merits of various methods of meeting braking regulations on high and low friction surfaces for loaded and unloaded vehicles. The techniques used herein to study the braking of tractor-semitrailer vehicles were developed by M. Sayers and these techniques are extensions of unique methods developed in an MVMA passenger car study.*

Even though stopping distance is the measure of braking performance selected by the federal government, it is much more convenient to work with deceleration in simplified braking analyses. Accordingly, stopping distance requirements can be converted to approximately equivalent average deceleration requirements using the following basic formula:

$$D = t_L V_o + \frac{V_o^2}{2A} \qquad (1)$$

where

D is the stopping distance

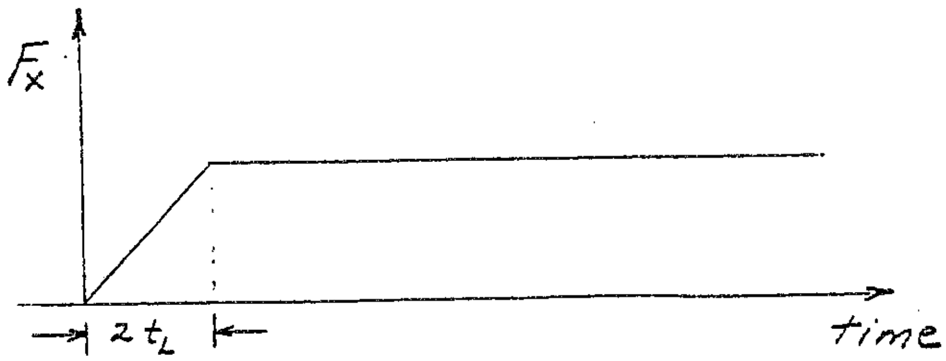
t_L is the "equivalent time lag" in the braking system

V_o is the initial velocity

and A is the average deceleration

For a situation in which the total braking force, F_x, can be approximated by a ramp-step type of time history, t_L is equal to 1/2 of the ramp time as illustrated in the following figure. The material which follows assumes that the braking requirements to be satisfied have been expressed, or if necessary re-expressed, in terms of the average deceleration capability desired on surfaces with given levels of tire-road friction potential.

*Sayers, M. and Segel, L. "Investigation of the Influence of Various Braking Regulations on Accident-Avoidance Performance." Final Technical Report, MVMA Project #4.31, November 1978.



Having established a deceleration condition to study, three fundamental questions need to be addressed. First, how do the laws of dynamics influence braking capabilities? Second, how will various arrangements of brake system hardware influence stopping performance? And, third, what levels of tire-road friction are necessary to achieve a selected deceleration level? The analytical approach which will be described in the following material uses the simplest concepts possible to answer these three questions simultaneously. The results of this simplified analysis would be almost trivial if it were not for the fact that the results describe a variety of solutions for various braking configurations. The challenge is to be able to interpret the results and to make meaningful comparisons between possible arrangements of braking capabilities.

The equations of "steady" motion for a tractor-semitrailer vehicle making a nearly constant deceleration stop are given in Equations (2) through (5) using the symbols illustrated in Figure 1 and defined in Table 1.

Tractor Plus Trailer Accelerations

$$A(W_1+W_2) = F_{x_1} + F_{x_2} + F_{x_3} \quad (2)$$

$$W_1 + W_2 = F_{z_1} + F_{z_2} + F_{z_3} \quad (3)$$

Trailer: Pitch Moments about the Fifth Wheel

$$0 = F_{x_3}h_f + F_{z_3}(a_2+b_2) - W_2a_2 + AW_2(h_2-h_f) \quad (4)$$

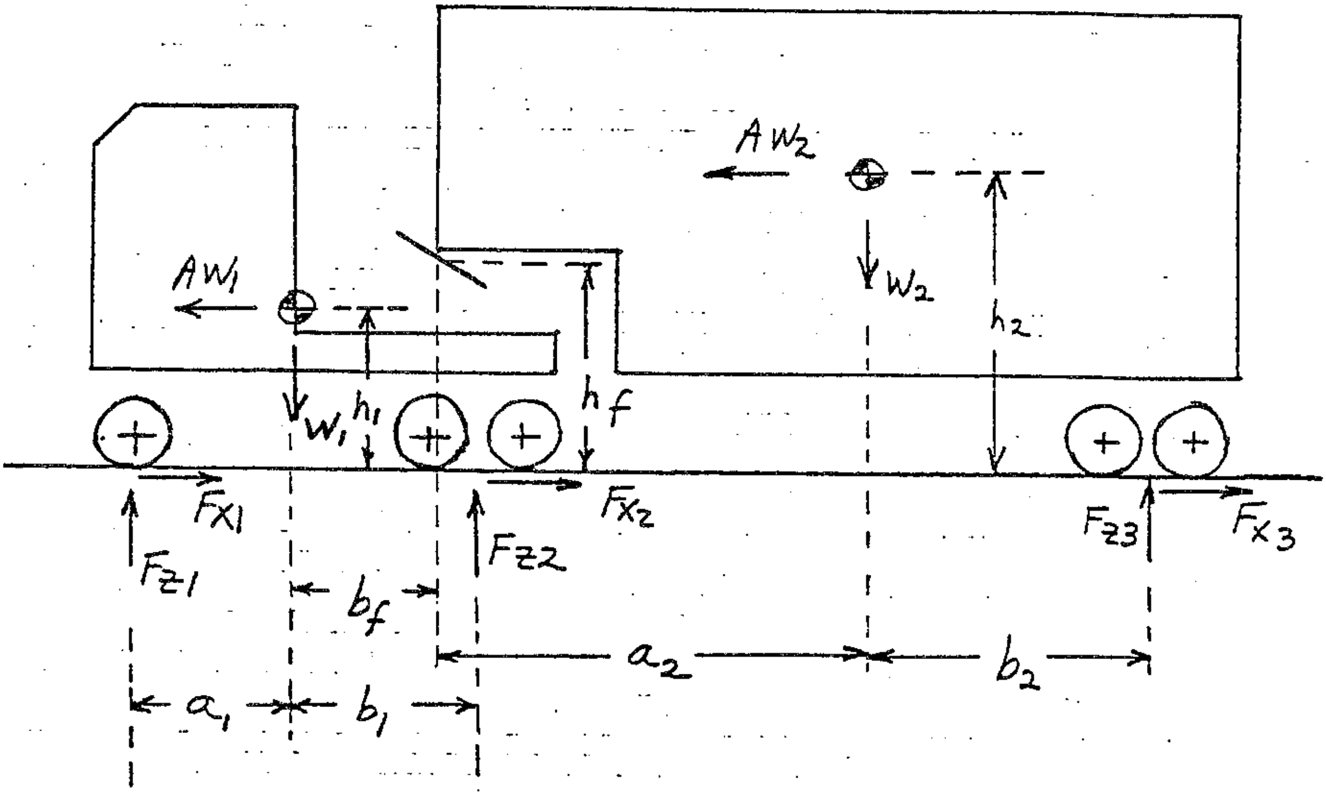


FIGURE 1 Illustration of Symbols

Table 1. Definition of Symbols

A	Deceleration of the vehicle in g's
a_1	Front axle to c.g. of the tractor
a_2	Fifth wheel to c.g. of the trailer
b_1	c.g. of tractor to center of tractor rear suspension
b_2	c.g. of trailer to center of trailer rear suspension
b_f	c.g. of tractor to fifth wheel
F_{x_1}	Brake force produced by the front tires
F_{x_2}	Brake force produced by the tires on the tractor rear suspension
F_{x_3}	Brake force produced by the tires on the trailer suspension
F_{z_1}	Vertical load on the front tires
F_{z_2}	Vertical load on the tires on the tractor rear suspension
F_{z_3}	Vertical load on the tires on the trailer suspension
h_1	Height of the tractor c.g.
h_2	Height of the trailer c.g.
h_f	Height of the fifth wheel
W_1	Weight of the tractor
W_2	Weight of the trailer

Tractor: Pitch Moments about the Fifth Wheel

$$0 = (F_{x_1}+F_{x_2})h_f + F_{z_2}(b_1-b_f) + W_1b_f - F_{z_1}(a_1+b_f) - AW_1(h_f-h_1) \quad (5)$$

We could consider interaxle load transfer in tandem suspensions. Interaxle load transfer could be included using a "percent concept" in which a fraction of the total brake torque generated at a suspension is reacted through changes in vertical load on each of the axles in that suspension. However, the present analysis uses the total braking force and the total vertical load for each suspension.

With regard to the braking system, special parameters (P_1, P_2, and P_3) are introduced to describe the proportioning of braking effort between the various suspensions. These parameters are rigorously defined by Equations (6) through (9).

$$F_{x_1} = P_1A(W_1+W_2) \quad (6)$$

$$F_{x_2} = P_2A(W_1+W_2) \quad (7)$$

$$F_{x_3} = P_3A(W_1+W_2) \quad (8)$$

$$P_1 + P_2 + P_3 = 1 \qquad P_1 = 1 - P_2 - P_3 \quad (9)$$

Equations (6) through (9) describe the manner in which the braking system distributes braking effort within the longitudinal deceleration condition of the total vehicle as expressed by Equation (2).

A concept referred to as "adhesion utilization" or "friction utilization" is employed to relate longitudinal braking force to vertical load at each suspension, viz.,

$$F_{x_1} = K_1 F_{z_1} \quad (10)$$

$$F_{x_2} = K_2 F_{z_2} \quad (11)$$

$$F_{x_3} = K_3 F_{z_3} \quad (12)$$

The constraint upon braking performance caused by the level of available tire-road friction is incorporated into the analysis by observing that permissible values of K_i (i=1,2,3) cannot exceed the available friction, μ. Or, alternatively, results for various values of K_i can be used to find the minimum level of friction needed to perform a given deceleration stop with a particular vehicle and a selected brake proportioning (where "proportioning" means a set of values for P_1, P_2, and P_3).

Hopefully, the example "solutions" which follow will clarify the interpretation of the brake proportioning and the friction utilization parameters.

Figure 2 contains a graphical presentation of a family of solutions for the braking of a tractor-semitrailer vehicle in the loaded condition during an 0.6 g stop. (The parameters describing this particular vehicle when fully loaded or empty are given in Table 2.) Each member of the

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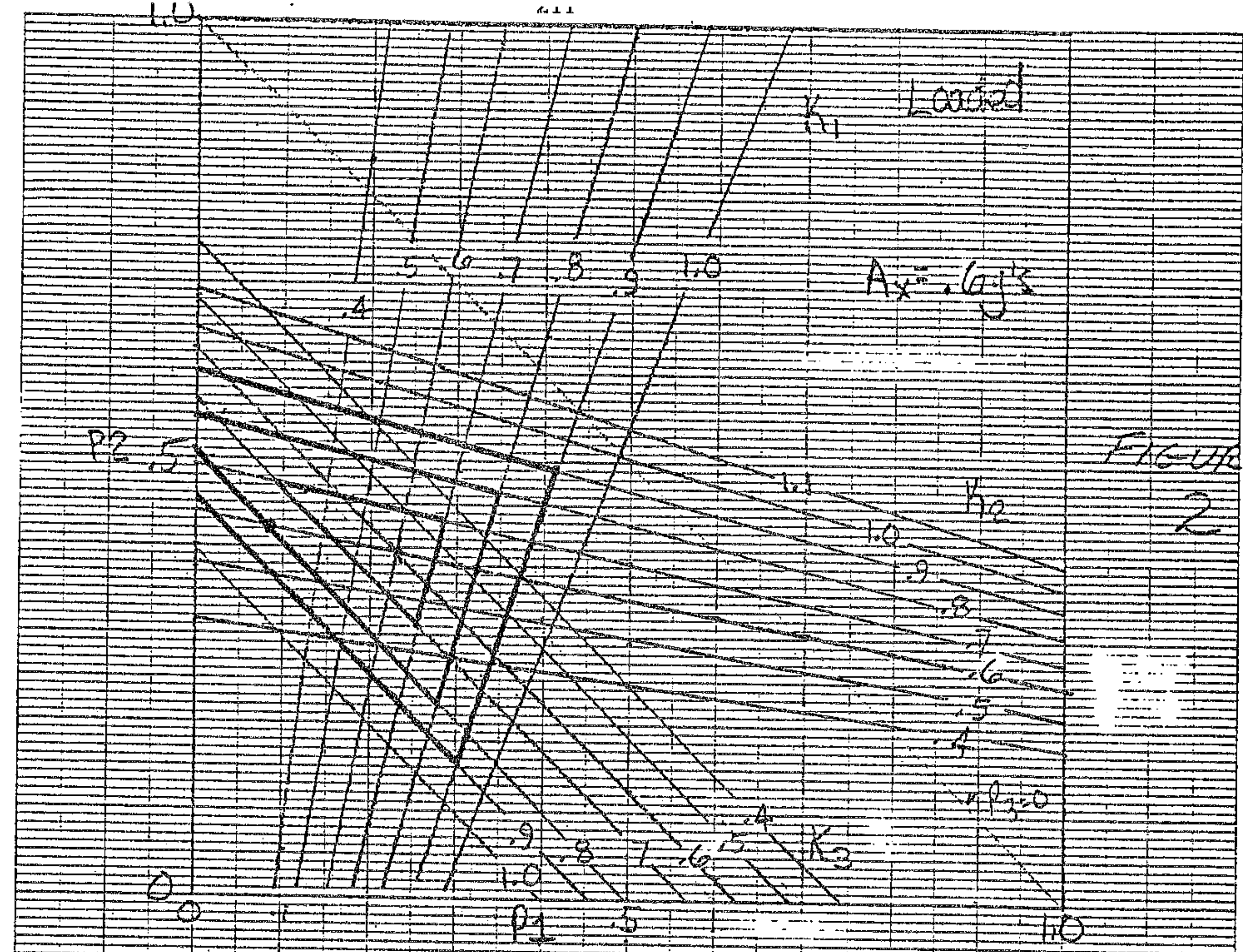
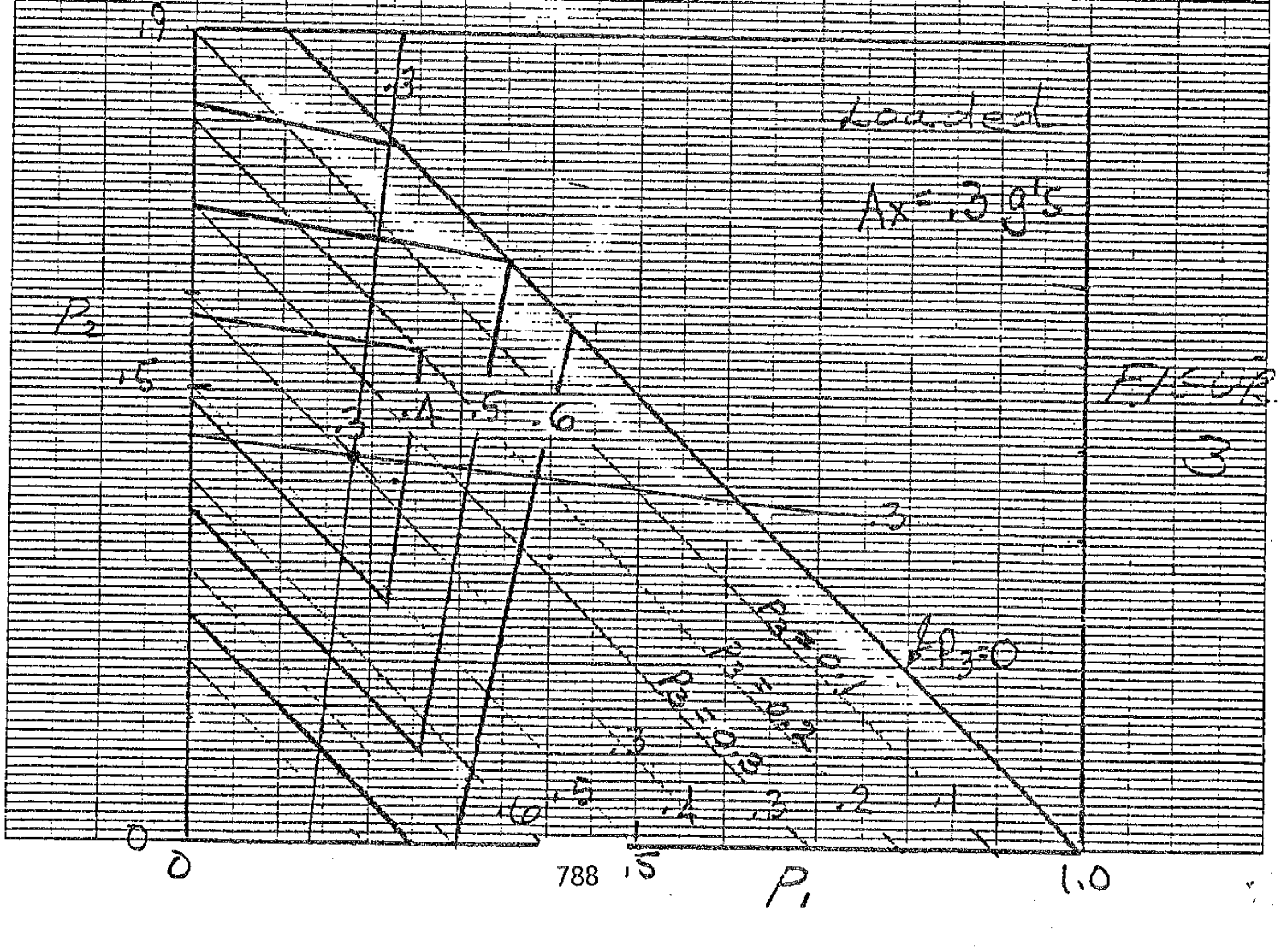
Table 2

(These parameters describe a two-axle tractor pulling a semitrailer with a tandem suspension. They are taken from SAE Paper No. 710044 by R. Limpert.)

Tractor		Trailer	Loaded	Empty
a_1	84"	a_2	200"	289"
b_1	78"	b_2	192"	103"
b_f	67"	h_2	102"	47"
h_1	35.6"	h_f	47"	47"
h_f	47"	W_2	30,000 lb	5,333 lbs
W_1	8,000 lb			


K1 Loaded
Ax = .6g's
K2
P2 .5
P1 .5
1.0
K3
FIGURE 2


Loaded
Ax = .3g's
P2
.5
P3 = 0
P3 = 0.1
P3 = 0.2
P3 = 0.3
P1
1.0
FIGURE 3

family of solutions (i.e., lines of constant K_1, K_2, or K_3) presented in Figure 2 was constructed using the indicated value of K_1, K_2, or K_3 at various levels of P_1. For example, the line K_1=1 is determined from the simultaneous solution of (a) Equations (2) through (5) using the vehicle parameters from Table 2, (b) Equation (10) with K_1=1, and (c) Equation (6) using various values of P_1. At each value of K_1 and P_1, the longitudinal forces (F_{x_1}, F_{x_2}, and F_{x_3}) and the vertical forces (F_{z_1}, F_{z_2}, and F_{z_3}) are determined. These forces are, in turn, used in the appropriate equations to express the relationship of P_2 to P_1 along the line K_1=1.0 for an 0.6 g stop for the loaded tractor-semitrailer described by the parameters given in Table 2.

Different vehicles will have different braking diagrams, but all braking diagrams will be similar to Figure 2 in that the lines of equal K_1, K_2, and K_3 will make "concentric triangles." An important point is located at the intersection of the $K_1=K_2=K_3=0.6$ lines in Figure 2. This point represents "perfect" proportioning for an 0.6 g stop for this vehicle. At no other proportioning can an 0.6 g stop be achieved with a tire-road friction (μ) equal to 0.6. However, if μ = 0.7, then any proportioning which fits within the triangular region defined by $K_1 \leqq 0.7$, $K_2 \leqq 0.7$, and $K_3 \leqq 0.7$ will produce an 0.6 g stop. If μ is interpreted as the peak friction (maximum of the μ-slip curve), then any proportioning which fits within the $K_i \leq 0.7$ region can produce a wheels-unlocked 0.6 g stop with a tire-road friction of 0.7.

Figure 3 is the braking diagram for the same vehicle making an 0.3 g stop. The constant K_i boundaries are emphasized in this diagram. As is well known, the proportioning which was "perfect" for an 0.6 g stop is not perfect for an 0.3 g stop.

Braking diagrams for the empty vehicle at 0.6 and 0.3 g are presented in Figures 4 and 5. The set of braking diagrams consisting of Figures 2 through 5 correspond, roughly, to a range of conditions which might be considered in a braking regulation which examines performance on wet and dry surfaces for vehicles in loaded and empty conditions. By

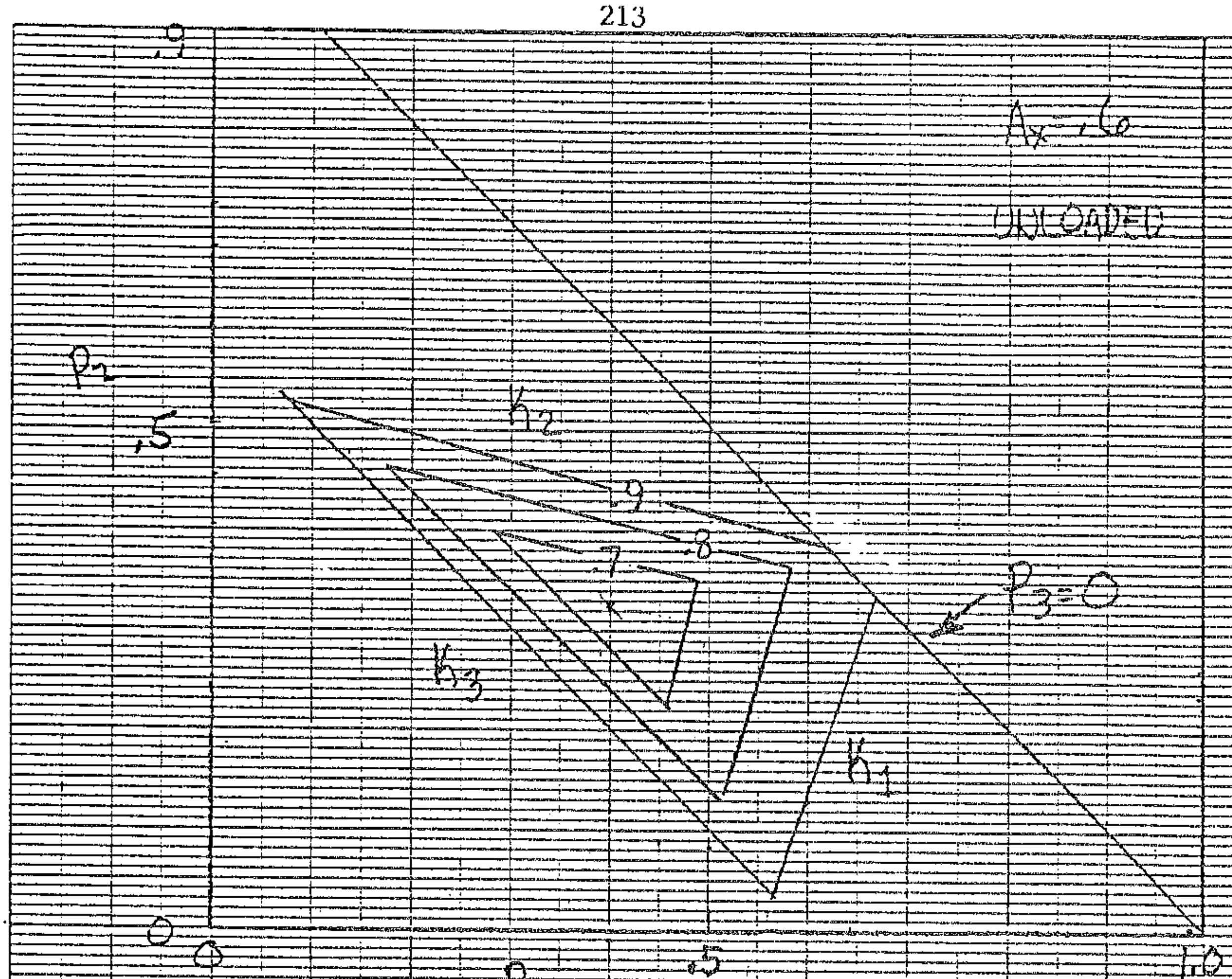


FIGURE 4

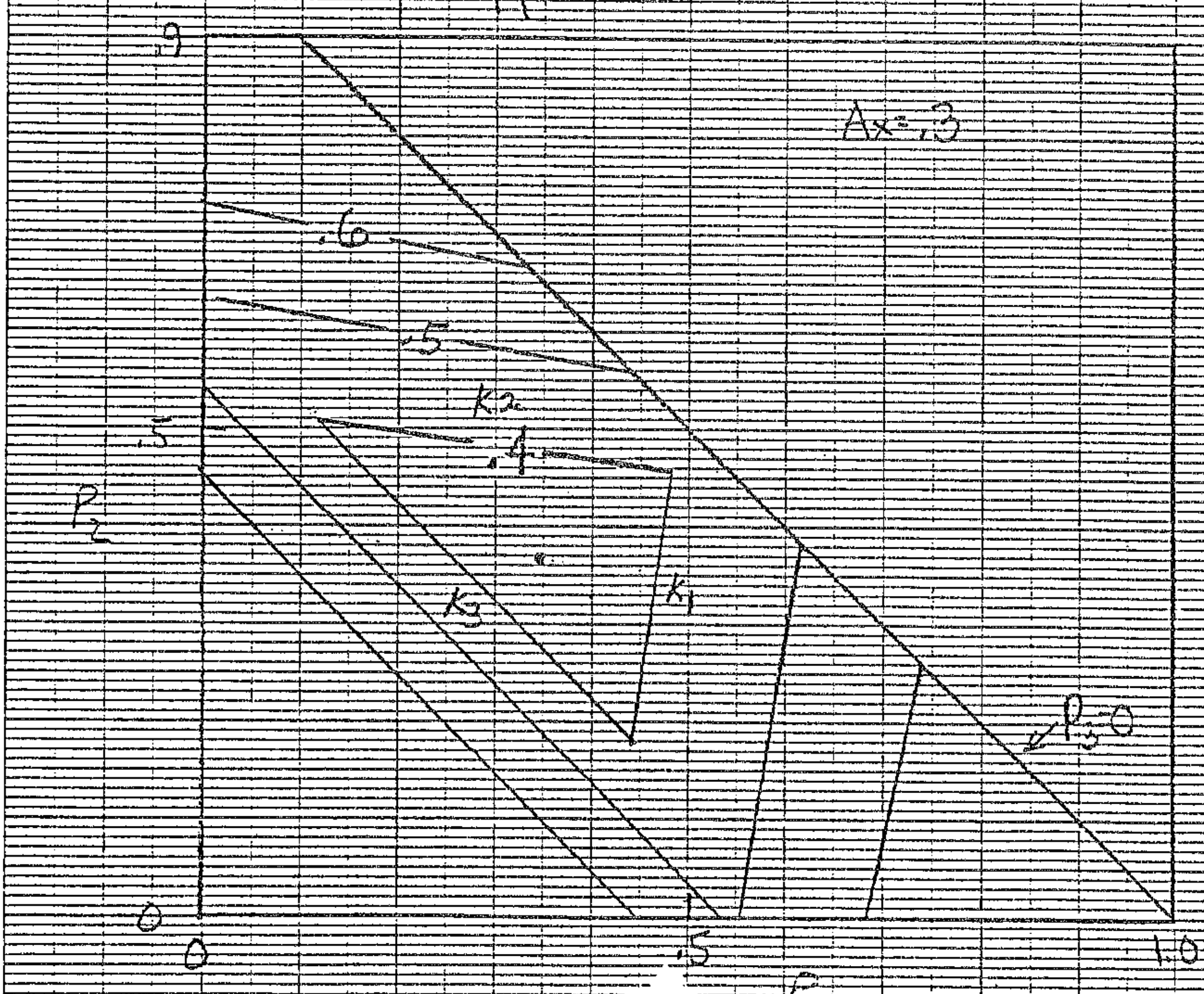


FIGURE 5

comparing Figures 2 through 5 it can be seen that the changes in proportioning needed to compensate for differences in loading state are greater than the changes needed to compensate for g level at a given load.

To illustrate how results from Figures 2 through 5 may be combined, consider an efficiency, E_i, defined by the following ratio:

$$E_i = (A/K_i)\ 100\ (i=1,2,\text{ or }3) \tag{13}$$

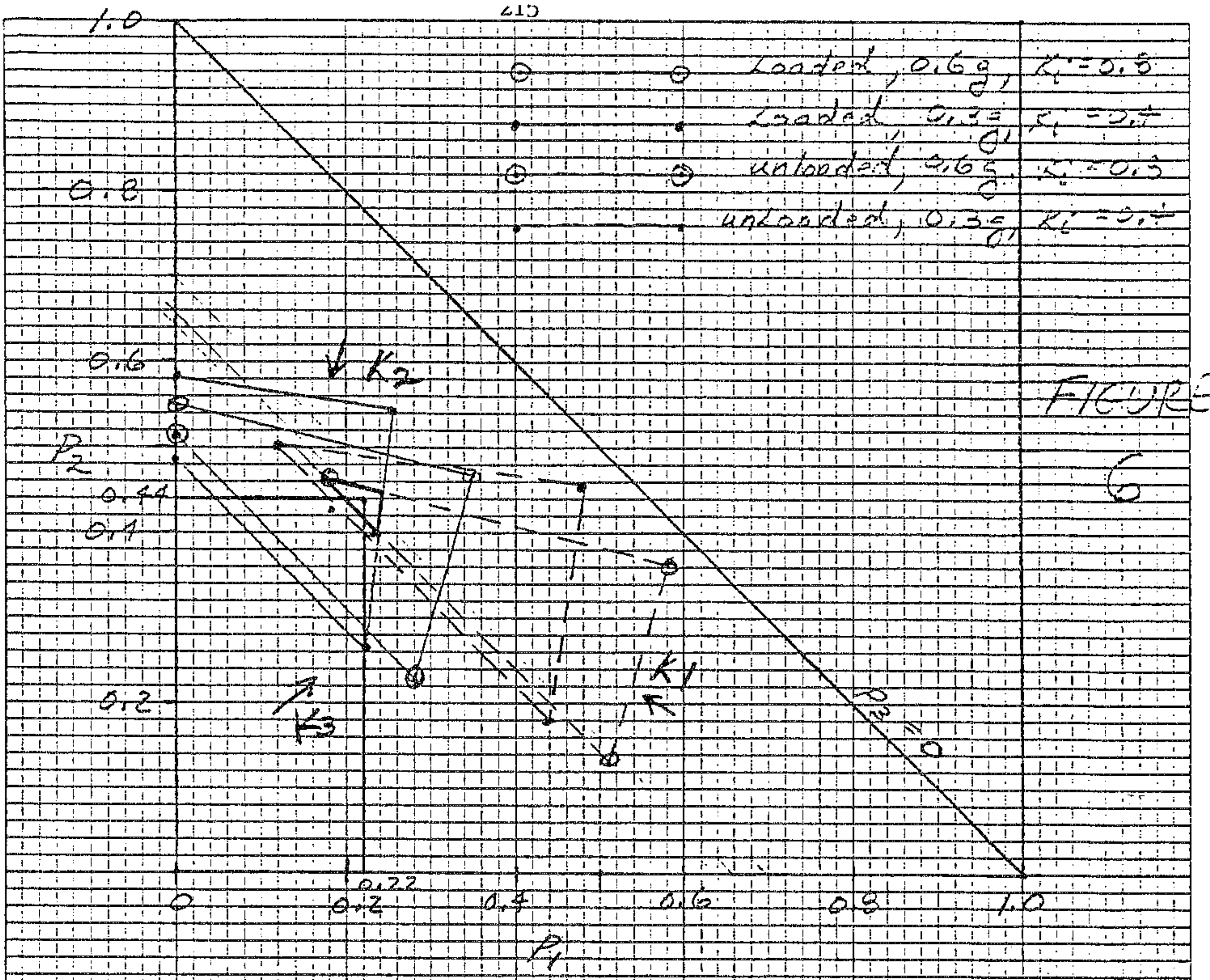
For example, if the overall braking effort were to be at least 75% efficient, then it is desired that the following inequality be satisfied, $K_i \leqq (A/0.75)$, that is, at 0.6 g, K_i must be less than 0.8 and at 0.3 g, K_i must be less than 0.4.

Figure 6 has been constructed by tracing the appropriate boundaries for the 75% efficiency regions from Figures 2, 3, 4, and 5. It should be noted that there is a small region of overlap of the four regions of 75% efficiency. As indicated on Figure 6, a proportioning of $P_1 = 0.22$, $P_2 = 0.44$, and $P_3 = 0.34$ would allow an efficiency of at least 75% for stops on "high and low μ surfaces."

The results obtained from Figure 6 are intended to illustrate a method of using the braking diagrams. Clearly, practical and pragmatic reasons may preclude the use of the particular proportioning arrangement arrived at in Figure 6. Possibly, $P_1 = 0.22$ may imply either front brakes which are hard to package within the available space or more effective front brakes than the customer desires. In addition, the tractor manufacturer may have no means for assuring that the trailer brakes will have a proportioning of $P_3 = 0.34$.

Nevertheless, this example does show that it is difficult to meet both a loaded 0.3 g and an empty 0.6 g stopping requirement with a fixed proportioning arrangement.

The previous example was for a vehicle with a relatively small tractor. In the next example, a typical heavy vehicle with a three-axle tractor and a tandem suspension trailer will be considered. The parameters describing this vehicle are given in Table 3.



permissible solution: $P_1 = 0.22$

$P_2 = 0.44$

$1 - P_1 - P_2 = P_3 = 0.34$

For $\frac{A_x}{K_{i\,max}} > 0.75$, i.e., $K_i = 0.8$ for $A_x = 0.6g$

$K_i = 0.4$ for $A_x = 0.3g$

Table 3

(A Prototypical Heavy Vehicle)

Tractor	
a_1	64"
b_1	78"
b_f	78"
h_1	40"
h_f	48"
W_1	15,000 lbs

Trailer	Loaded	Empty
a_2	198"	198"
b_2	168"	168"
h_2	96"	56"
h_f	48"	48"
W_2	58,000 lbs	11,000 lbs

In the loaded condition the static loads are as follows:

$$F_{z1L} = 8,200 \text{ lbs}$$
$$F_{z2L} = 33,400 \text{ lbs}$$
$$F_{z3L} = 31,400 \text{ lbs}$$

In this case, the analysis will start by selecting brake characteristics based on the static loads for the loaded vehicle.. The brakes (including the influence of tire radius, drum radius, air chamber size, lever ratio, etc.) will be "sized" as illustrated in Figure 7 so that at 100 psi treadle pressure the total braking force at a suspension could be equal to the static loads given in Table 3 if enough tire-road friction were available.

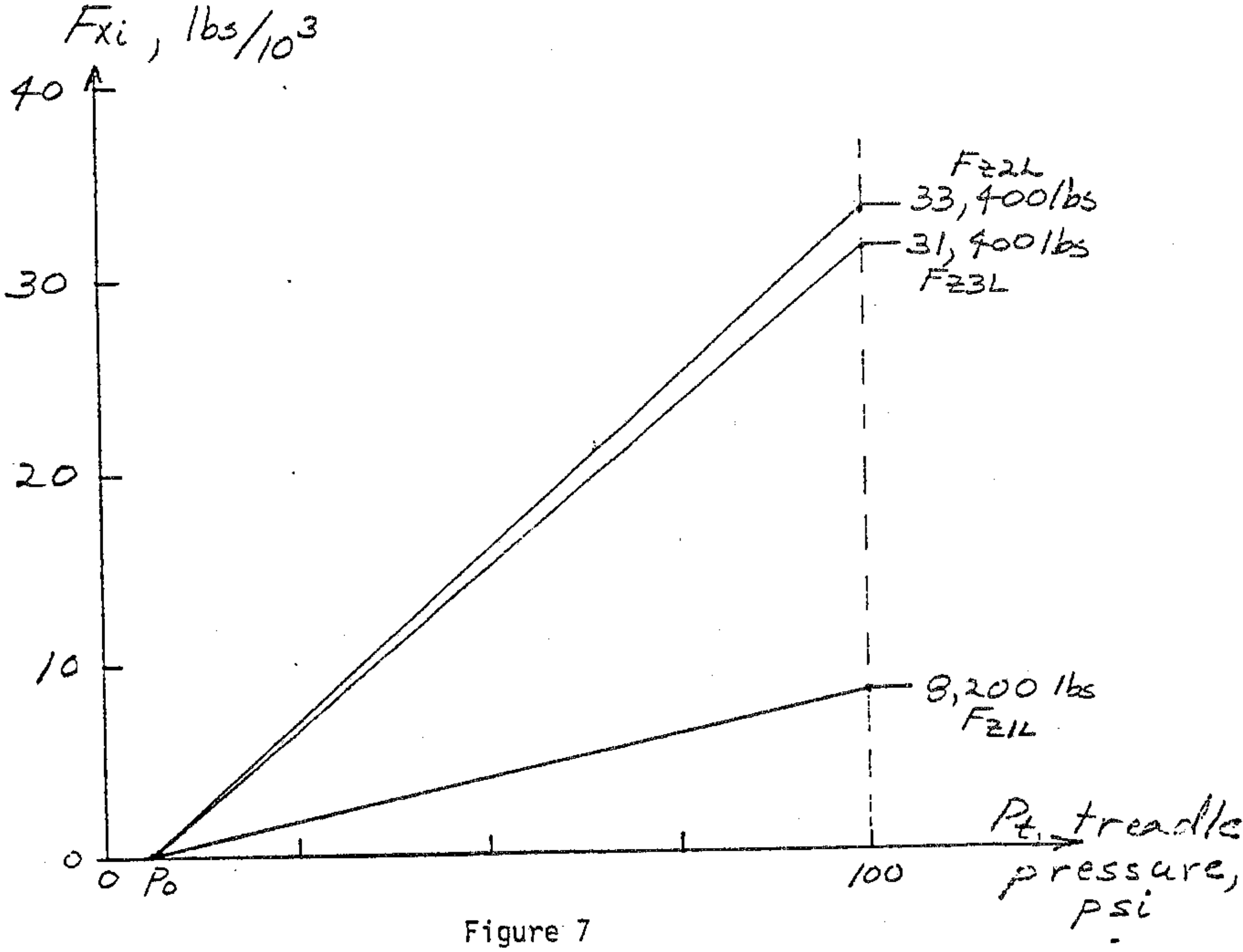


Figure 7

The following equations pertain to the arrangement shown in Figure 7:

$$F_{x_1} = F_{z1L}\,\frac{(P_t-P_o)}{(100-P_o)}$$

$$F_{x_2} = F_{z2L}\,\frac{(P_t-P_o)}{(100-P_o)}$$

$$F_{x_3} = F_{z3L}\left(\frac{P_t-P_o}{100-P_o}\right)$$

$$A(W_1+W_2) = F_{x_1} + F_{x_2} + F_{x_3}$$

$$= \left((P_t-P_o)/(100-P_o)\right)(W_1+W_2)$$

Therefore

$$P_1 = (F_{x_1}/A(W_1+W_2)) = F_{z1L}/(W_1+W_2) = 0.11$$

and similarly

$$P_2 = F_{z2L}/(W_1+W_2) = 0.46$$

and $$P_3 = F_{z3L}/(W_1+W_2) = 0.43$$

Or, in "short-hand" form, the proportioning is (0.11/0.46/0.43).

Braking diagrams for this heavy vehicle are given in Figures 8, 9, 10, and 11. Superimposed on these diagrams are points corresponding to the proportioning (0.11/0.46/0.43). The results presented in Figure 8 show that the selected proportioning is very efficient for the loaded vehicle making an 0.3 g stop. (Of course, the selected proportioning is expected to achieve good friction utilization on slippery surfaces using the loaded vehicle because it is "optimum" for friction levels approaching zero.) By examining Figure 9, it can be seen that the so-called "static loaded proportioning" is at least 75% efficient ($K_i < 0.8$) for an 0.6 g stop of the loaded vehicle. However, examining Figure 10 for the empty vehicle indicates that $K_i \doteq 0.6$ is needed for an 0.3 g stop, that is, a μ of 0.6 is needed to make an 0.3 g stop using static loaded proportioning. Furthermore, Figure 11 reveals that it is not possible to make an 0.6 g stop with the empty vehicle using static loaded proportioning and operating on an excellent road surface (μ = 0.9).

Since the static loaded proportioning does not appear to be efficient enough for use with the empty vehicle, it seems appropriate to investigate

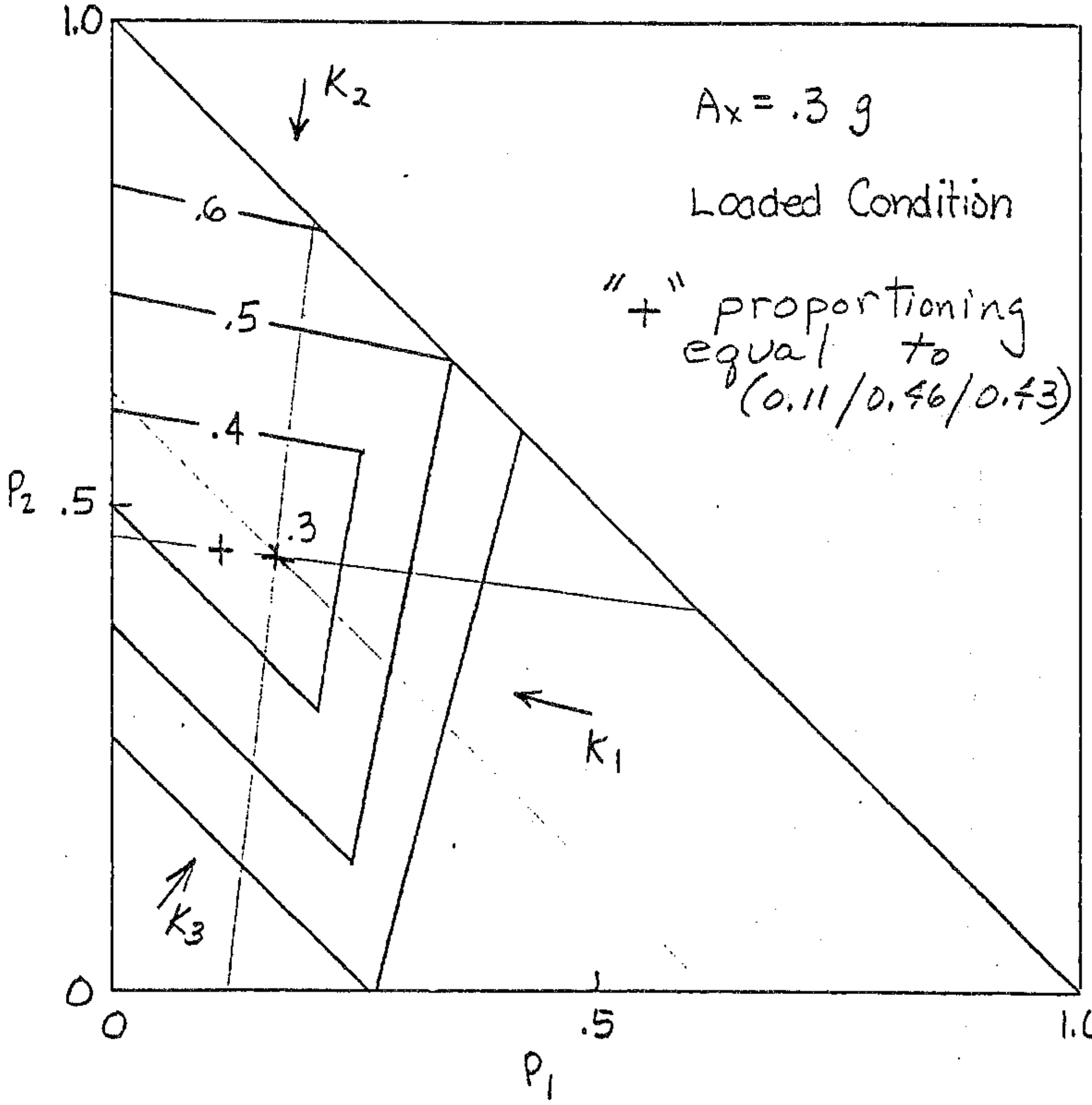

1.0
K_2
$A_x = .3$ g
Loaded Condition
"+" proportioning equal to (0.11/0.46/0.43)
.6
.5
.4
P_2 .5
.3
+
K_1
K_3
0
0
.5
1.0
P_1

FIGURE 8

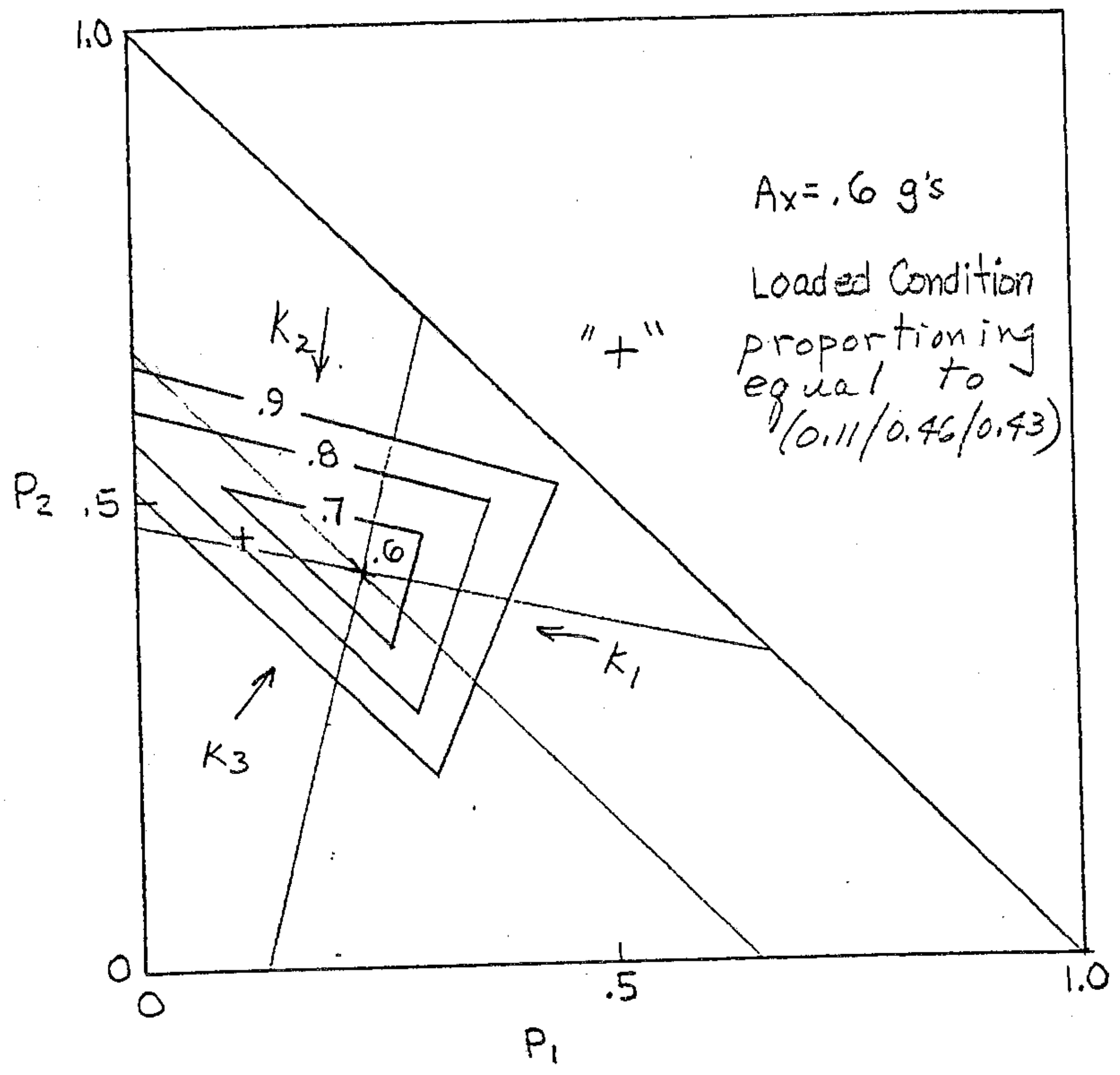

1.0
Ax= .6 g's
Loaded Condition
proportioning
equal to
(0.11/0.46/0.43)
"+"
K2
.9
.8
.7
.6
P2
.5
+
K1
K3
0
0
.5
1.0
P1

FIGURE 9

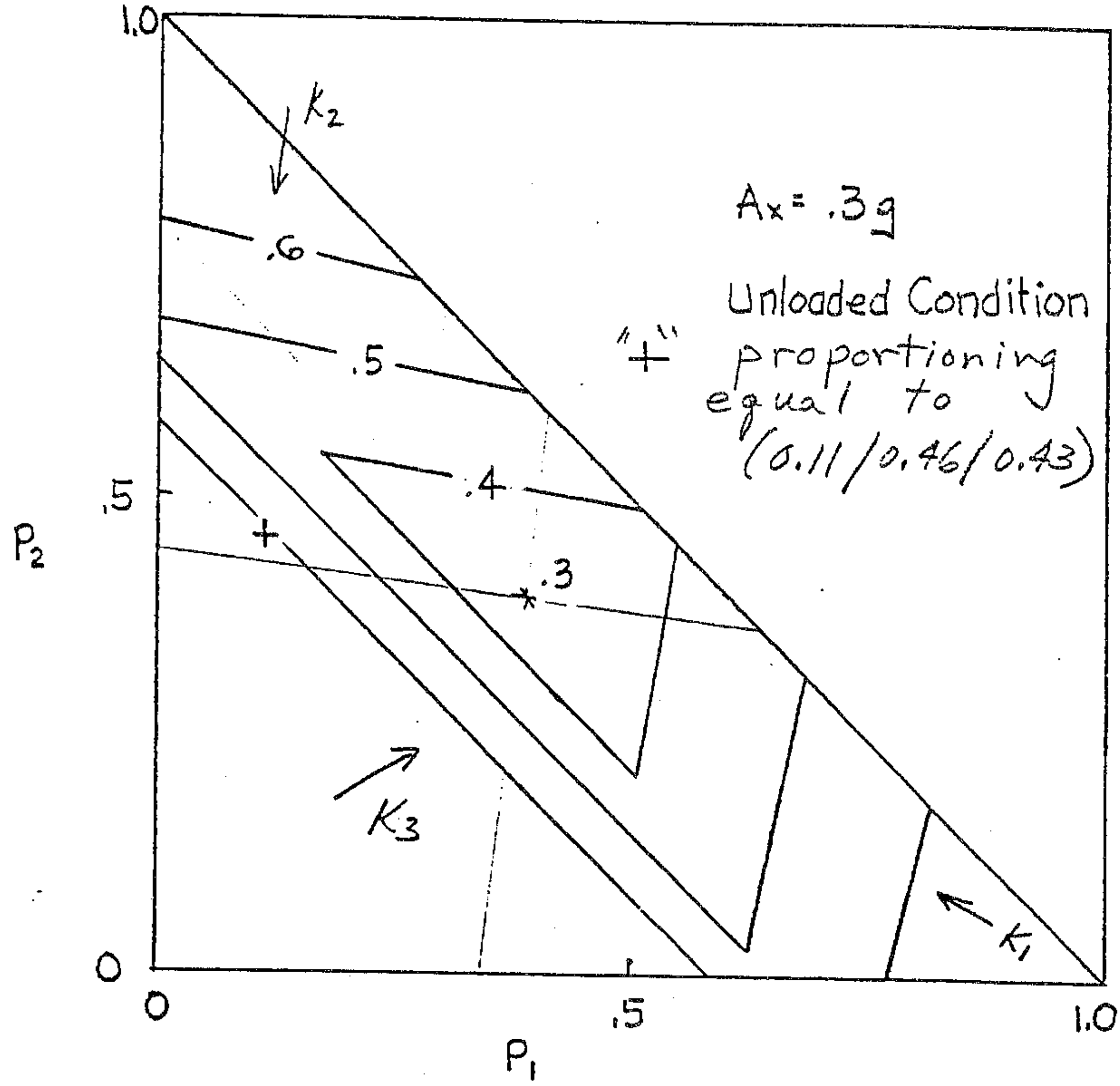

1.0
K_2
A_x = .3g
Unloaded Condition
"+" proportioning equal to (0.11/0.46/0.43)
.6
.5
.4
.3
.5
P_2
+
K_3
K_1
0
0
.5
1.0
P_1

FIGURE 10

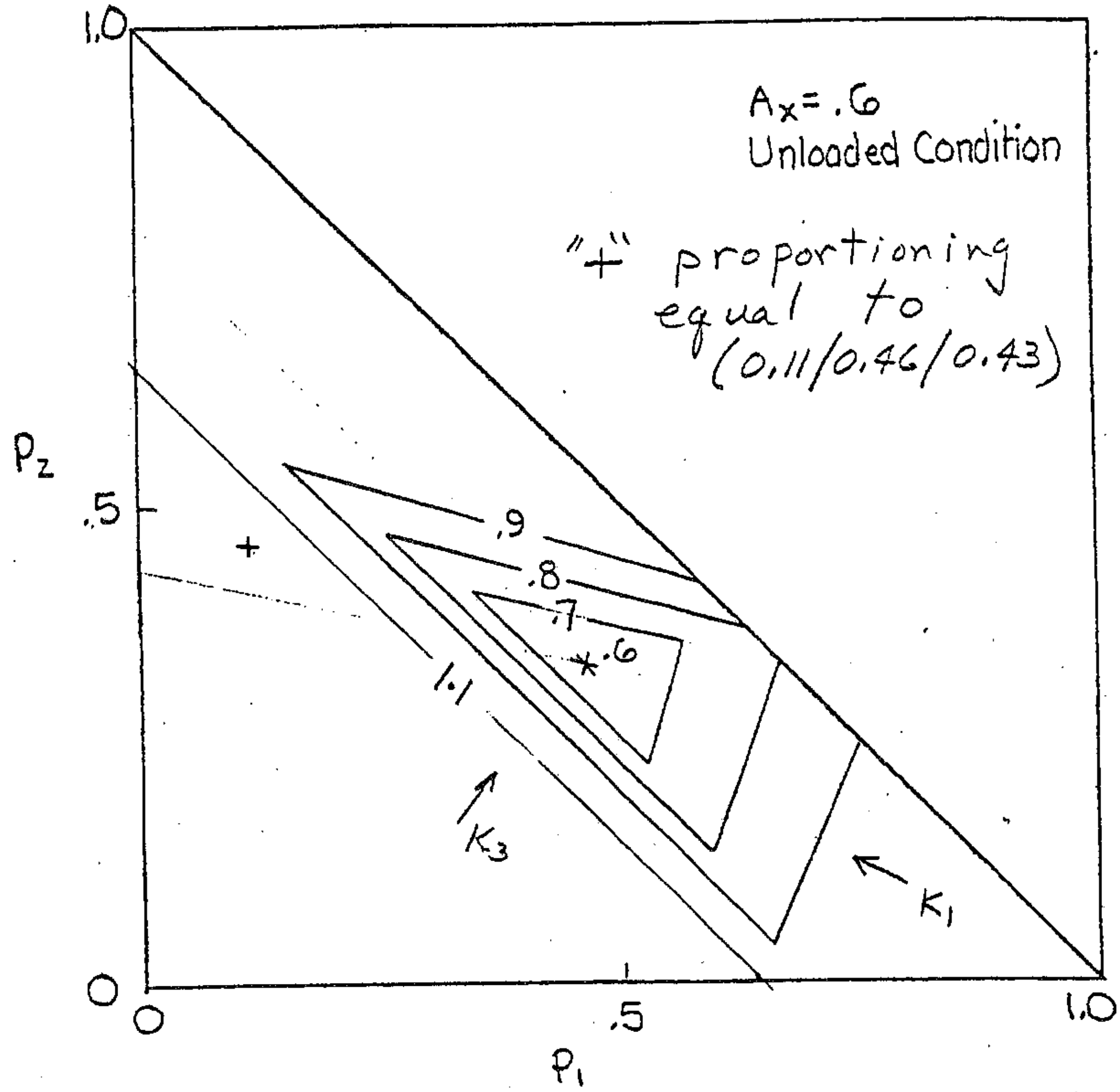

1.0
A_x= .6
Unloaded Condition
"+" proportioning equal to (0.11/0.46/0.43)
P_2
.5
+
.9
.8
.7
.6
1.1
K_3
K_1
0
0
.5
1.0
P_1

FIGURE 11

what can be done with regard to finding a "best compromise," fixed proportioning as was done for the first vehicle. By superimposing regions of the braking diagrams (as was done before), it can be seen (see Figure 12) that it is not quite possible to achieve at least 75% efficiency at 0.3 g and 0.6 g for loaded and empty conditions.
Rather, something like 72 or 73% efficiency could be obtained if a proportioning of (0.25/0.47/0.28) were used.

Again, it should be emphasized that the results presented here are to illustrate a method. Nevertheless, the figures obtained emphasize the difficulty of meeting various braking requirements with fixed proportioning.

Before closing this section, it is of interest to (1) indicate how antilock braking on a suspension can be included in the braking diagram, and (2) mention some ideas concerning load-sensitive proportioning.

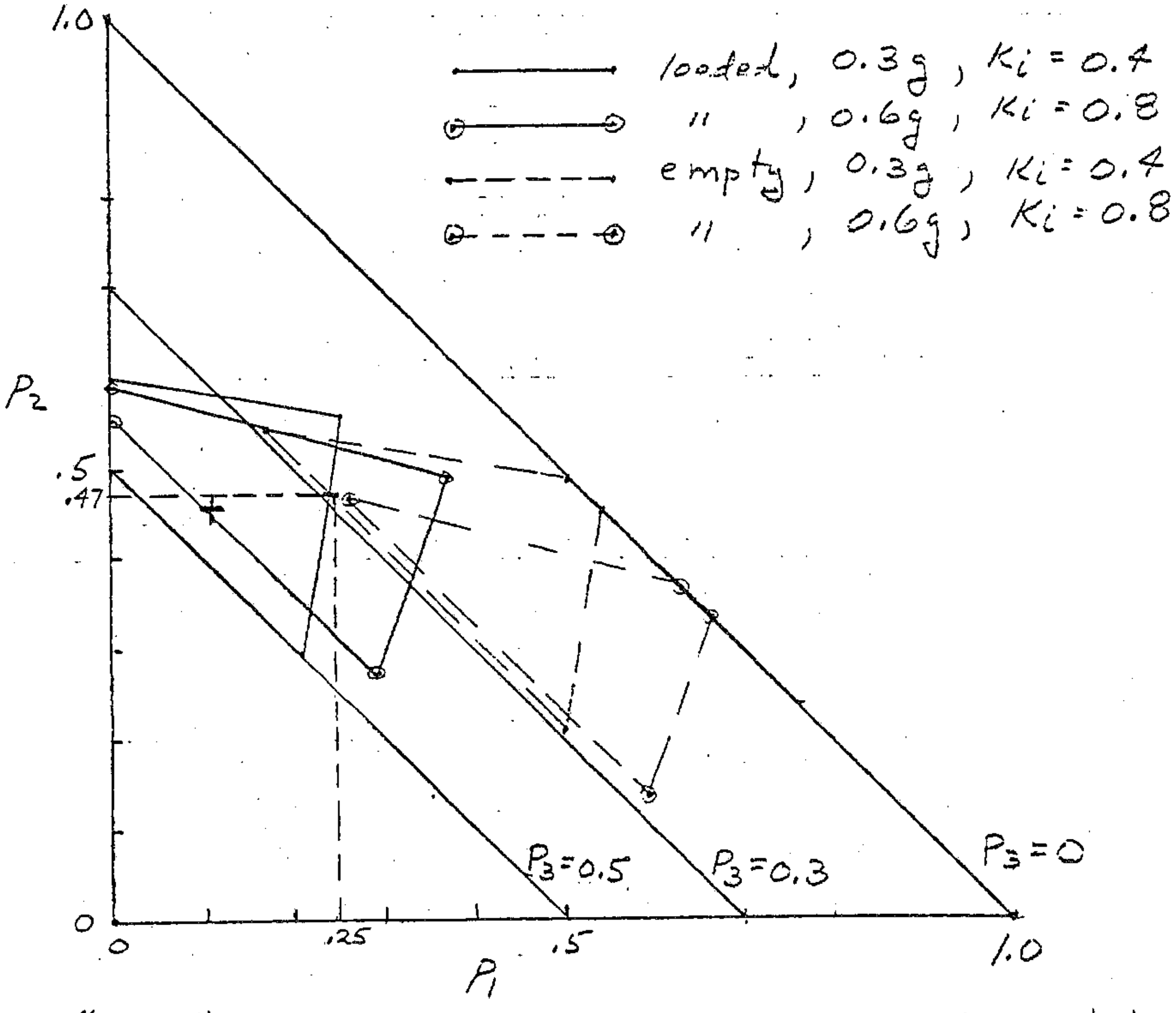
An antilock system, if operating reasonably well, establishes (on the average) a relationship between vertical load and braking force for the tires controlled by the antilock system. In the simplest interpretation, $F_{x_i} = K_i F_{z_i}$ where K_i equals the deceleration, A, or $F_{x_i} = AF_{z_i}$. Thus, the K_i-line for $K_i = A$ represents antilock braking for the i^{th} suspension in the braking diagram.

It should be noted that available antilock systems are not perfect and, accordingly, a tire-road friction level greater than K_i may be needed for the antilock system to achieve a given level of deceleration (i.e., A). A measure of antilock system efficiency can be used to relate the desired or specified deceleration level, A, to the tire-road friction, μ, required, viz.,

$$A = (E_{AL}/100)\mu$$

where

E_{AL} is the efficiency of the antilock system (where E_{AL} may be a function of μ).



"Best" compromise:

P_1 = .25

P_2 = .47

P_3 = .28

+ Based on static loaded conditions

P_1 = .11

P_2 = .46

P_3 = .43

FIGURE 12

With regard to load-sensitive proportioning, braking efficiency can be improved by allowing brake proportioning parameters to vary with operating conditions. One approach to variable proportioning is to develop valves which control the gain of a brake as a function of suspension deflection. (Suspension deflection is taken to be a useful measure of vertical load in this case.)

Although the following analysis is highly idealized, it indicates how "perfect" load-sensitive proportioning would fit into the present analysis scheme. Under ideal braking conditions

$$F_{x_i} = \mu F_{z_i} \tag{14}$$

and

$$\mu = A \tag{15}$$

Combining Equations (14) and (15) with the proportioning Equations (6), (7), and (8), yields the following equations for "optimum" braking under ideal conditions:

$$P_1 = F_{z_1}/(W_1+W_2) \tag{16}$$

$$P_2 = F_{z_2}/(W_1+W_2) \tag{17}$$

$$P_3 = F_{z_3}/(W_1+W_2) \tag{18}$$

Clearly, Equations (16), (17), and (18) state the well-known fact that optimum proportioning is achieved if the proportioning of the brakes at each suspension is proportional to the instantaneous vertical load carried by that suspension.

In practice, various types of load sensing proportioning have been used in Great Britain and in other parts of Europe. Some load-sensitive proportioning devices operate slowly in a static manner and others respond rapidly to dynamic changes in load.

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