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BUS, TRUCK, TRACTOR-TRAILER BRAKING SYSTEM PERFORMANCE
Volume 1 of 2: Research Findings

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16 Abstract

The objectives of this study were to determine the range of braking performance currently exhibited by buses, trucks, and tractor-trailers and to establish the maximum braking performance capabilities of these vehicles based upon full utilization of the technology related to brake system design. Both vehicle testing and analytical techniques, including dynamic modeling and simulation, were used to accomplish these objectives. Performance measures were defined which serve to quantify the degree to which a given vehicle-braking system possesses those qualities necessary for adequate braking performance. Using these measures, a braking performance standard is recommended based upon a comparative analysis of (1) current braking performance, (2) the maximum performance achievable by full exploitation of existing technology, and (3) performance as constrained by a host of associated factors.

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SYMBOLS

a	=	thermal diffusity
a_x	=	deceleration
$a_{x(av)}$	=	mean deceleration; see Fig. 13
$a_{x(max)}$	=	maximum deceleration (ft/sec^2)
A	=	brake force distribution constant; see Eq. 2-9
A_c	=	brake chamber area
A_{MC}	=	master cylinder area
A_{wc}	=	wheel cylinder area
b	=	a constant depending on characteristics of friction surfaces
B	=	a brake force distribution constant; see Eq. 2-9
BF	=	brake factor, defined as the ratio of drag force on the drum friction surface to the actuating force of one shoe
$BRHP$	=	horsepower rating of the braking system
$B*$	=	vacuum assist characteristics, defined by the ratio of pushrod force upon the master cylinder piston to the pedal force multiplied by the pedal lever ratio
c	=	specific heat
E	=	braking efficiency
E_t	=	total energy (ft/lb)
E_{min}	=	braking efficiency, specified minimum value
E_F	=	braking efficiency as determined by the front axle
E_R	=	braking efficiency as determined by the rear axle
f	=	fade coefficient
F_p	=	pedal force
$F_{x,a}$	=	brake force per axle (air brake system)
$F_{x,h}$	=	brake force per axle (hydraulic brake system)
$F_{X,F}$	=	brake force on front axle
$F_{Z,F/dyn}$	=	vertical force on front axle
F_2	=	spring force at extended length
g	=	gravitational constant
h	=	convective heat transfer coefficient; variable dependent upon vehicle speed
i	=	number of braked axles
J	=	energy conversion factor, 778 ft-lb/Btu
k	=	thermal conductivity
k_1, k_2	=	thermal conductivities of the moving and stationary friction surfaces, respectively
K	=	spring rate $(lb/in.)$
K_f	=	K-factor
ℓ	=	length parameter; see Fig. 11
ℓ_c	=	effective cam radius
ℓ_h	=	hydraulic ratio between master cylinder and wheel cylinder
ℓ_r	=	pedal lever ratio
ℓ_s	=	effective slack adjuster length

n_B = number of brakes

n_S = number of brake shoes

N = number of snubs in 720 sec

p_h = line pressure

p_ℓ = brakeline pressure

p_m = mean pressure between brake shoe and drum

p_o = pushout pressure

P = mean lining pressure

P_n = nominal pressure on the friction surface

P_y = yield strength in compression

q = heat received by one hot spot

q_D = heat flux into drum $(\text{Btu/ft}^2\text{-sec})$

q_L = horsepower per unit area of lining (hp/ft^2)

r = drum radius (effective drum or disk radius)

R = effective tire radius

S = brake sensitivity

S_t = distance travelled during the test

S_2 = actual stopping distance (ft)

S_{60} = stopping distance (ft) corrected to 60 mph initial speed

t_1 = calculated time (sec) that brakes are applied in a single snub at the velocity and deceleration employed in the test

t_2 = average drift time (sec) when decelerating from 50 to 15 mph

t_a = application time (sec)

t_b = buildup time (sec)

T = brake torque

V = velocity (ft/sec)

V_i = initial velocity (mph)

V_{rel} = relative velocity

V_1 = actual test velocity (mph)

W = vehicle weight (lb)

X_1 = caged length of spring

X_2 = length of spring when spring brakes are set

β = a factor between 2 and 6 relating hot-spot size and sliding distance

ΔD = incremental stopping distance (wheels unlocked)

$\Delta \theta$ = change in temperature

ε = brake sensitivity

η = mechanical efficiency

η_c = cylinder efficiency

η_m = mechanical efficiency between brake chamber and shoe actuation

η_l = lever efficiency

θ = temperature

θ_i = initial brake temperature

θ_s = friction surface temperature

λ = relative load ratio; see Fig. 7

μ = coefficient of friction

μ_{oo} = value of lining friction coefficient at high values of sliding velocity

μ_{road} = coefficient of friction at tire-road interface

μ_x = longitudinal coefficient of friction at tire-road interface

μ_y = lateral friction coefficient at tire-road interface

μ_L = coefficient of friction of lining

$\mu_{L\ell}$ = lining coefficient of friction at arbitrarily specified "faded" condition

μ_{Lh} = lining coefficient of friction, nonfaded condition

μP_m = mean pressure times lining friction coefficient (lb/in.^{2})

μ_S = value of lining friction coefficient at high relative velocity

ρ = lever ratio between brake chamber and brake shoe

ρ_d = density

σ = heat absorbed by the drum divided by total thermal braking energy

ϕ_{1F} = brake force on front axle/total brake force

ϕ_{1R} = brake force on rear axle/total brake force

ϕ_{2R} = brake force on trailer axle/total brake force

ϕ = brake force distribution, defined as brake force between tire and roadway generated at the rear axle divided by total brake force

χ = height of center of gravity divided by the wheel base

ψ = static rear axle load divided by the total vehicle weight

1. INTRODUCTION

This report presents findings, conclusions, and recommendations derived by the Highway Safety Research Institute (HSRI) of The University of Michigan in a research program for the National Highway Traffic Safety Administration (NHTSA) entitled, "Bus, Truck, Tractor-Trailer Braking System Performance." The broad objectives of this program are the formulation of techniques and the production of data designed to aid NHTSA in fulfilling its mandate to issue reasonable and desirable safety standards.

1.1 STATEMENT OF THE PROBLEM

The complex relationship between current braking capabilities of commercial vehicles and the frequency of accidents with these vehicles due to deficiencies in braking performance is neither completely understood nor statistically documented. There is, nevertheless, ample intuitive basis to hypothesize that such a relationship exists, and further, that there are certain specific vehicle braking performance characteristics which, during either the normal driving process or emergency situations, causes the potential for loss of control to rise above a threshold beyond which even the skill and experience of the professional driver are of little avail. It is known that the braking performance of buses, trucks, and tractor-trailers varies significantly over a wide range, and is, on the average, less than that for passenger cars, and certainly less than the maximum performance achievable. Thus, it can be argued that this performance differential, if great enough, can constitute a significant safety hazard since all vehicle types are subjected to the same traffic and physical environments. The performance demands of these environments and the integration of vehicle types within the environments indicate a need for uniform and higher braking performance levels to be achieved by commercial vehicles. The study described herein addresses this need by focusing on the establishment of braking system performance requirements for buses, trucks, and tractor-trailers.

1.2 OBJECTIVES

The specific objectives of this study are threefold:

(1) To determine, by means of vehicle testing, the range of braking performance currently exhibited by buses, trucks, and tractor-trailers.

(2) To establish the maximum braking performance capabilities of these vehicles based on full utilization of the technology related to brake system design.

(3) To recommend a rational braking performance standard based upon a comparative analysis of (a) current braking performance, (b) the maximum performance achievable by full exploitation of existing technology, and (c) performance as constrained by a host of associated factors.

1.3 SUMMARY OF TASKS

In order to meet the objectives of this program, four major experimental

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and analytical tasks were carried out.

1.3.1 LITERATURE REVIEW. The foreign and domestic literature was surveyed with the objective of finding information pertinent to: accurate analyses of braking systems, experimental test procedures, and means of measuring and evaluating the braking performance of buses, trucks, and tractor-trailers. Factors considered important in the review were brake system design, braking performance, brake usage, brake testing, brake failure, and performance standards.

1.3.2 VEHICLE-BRAKE SYSTEM PERFORMANCE TESTS. In order to determine the braking performance capabilities of vehicles equipped with standard braking systems, three integral trucks, three buses, and four tractor-trailer combinations were subjected to a series of effectiveness, fade and recovery, and brake rating tests. So that the improvement in performance through use of more effective brakes and advanced braking systems could be determined, three additional vehicles were tested, namely:

(1) An integral truck equipped with disk brakes and a full power hydraulic brake actuation system.

(2) A tractor-trailer equipped with proportioning valves, adaptive braking system,* and trailer brake synchronization device.

(3) A tractor-trailer equipped with a wheel antilock system.**

The disk brake truck was subjected to effectiveness, fade and recovery, and brake rating tests, while the two tractor-trailer vehicles were tested primarily for effectiveness and minimum stopping capability.

1.3.3 ANALYTICAL PROGRAM. The analytical study was directed toward establishing mathematical and computation procedures for predicting braking performance based upon vehicle and braking system design factors. Analysis of the performance of several of the test vehicles demonstrated the feasibility of making accurate predictions of brake effectiveness, braking efficiency, pedal force gain, and thermal response. Dynamic modeling and simulation were employed to determine the extent to which vehicle braking performance can be improved with increased effectiveness, refinements in brake torque distribution, and load- and deceleration-sensitive proportioning systems.

1.3.4 RECOMMENDATIONS FOR A SAFETY STANDARD. Based upon the results of the three aforementioned tasks, performance requirements are suggested which would effectively upgrade the braking performance of these vehicles to a level approaching that of passenger cars. Procedures and tests for ensuring compliance with the requirements are also suggested.

The interrelation of the tasks and important subtasks is shown in block diagram form in Fig. 1.

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*Designation used throughout the report for the system developed by Bendix-Westinghouse.

**Designation used throughout the report for the system developed by Eaton, Yale, and Towne.

Test Planning and Selection of Vehicles (Task 2)

Vehicle Testing—Baseline Vehicles (Task 2)

Literature Survey (Task 1)

Vehicle Testing—Advanced Systems (Task 2)

Development of Analytical Techniques (Task 3)

Analysis of Test Vehicles (Task 3)

Evaluation of Analytical and Test Results (Task 4)

Recommended Braking Performance Standard (Task 4)

Development of Dynamic Simulation Models (Task 3)

Simulation Analysis of Advanced Systems (Task 3)

FIGURE 1. INTERRELATIONSHIP OF PROJECT TASKS AND SUBTASKS

1.4 CONCLUSIONS AND RECOMMENDATIONS

The results of the analytical and testing work accomplished for this study indicate that the maximum braking performance that can be achieved by a vehicle on a given test surface is limited by five factors:

(1) The frictional forces available at the tire-road interface

(2) The effectiveness of the vehicle's brakes, that is, the maximum torque capacity of the brakes

(3) The braking efficiency of the vehicle, that is, how well the brake torque is balanced axle to axle such that the tire-road frictional forces are best utilized

(4) The ability of the driver to modulate the pedal force such that maximum deceleration and minimum stopping distance are achieved without loss of directional control and stability

(5) The time response of the brake system to an applied pedal force

The results of this study also incidate that three major steps will have to be taken to significantly upgrade the maximum braking performance of commercial vehicles.

First. The basic braking systems of the majority of these vehicles will have to be improved by use of more effective brakes, better balance, and faster system response on air-braked vehicles.

Second. The traction characteristics of tires used on the majority of medium and heavy commercial vehicles will have to be improved so that the

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advantage of improved brake effectiveness can be fully utilized at the tire-road interface.

Third. Advanced brake control systems will have to be employed to allow rapid brake applications without instigating vehicle instability, whether the vehicle be loaded or empty, and whether operating on a dry or slippery surface. A number of design alternatives exist for achieving these objectives:

(1) The effectiveness and fade resistance of the braking systems on medium and heavy trucks can be improved significantly by use of disk brakes, powered either by a vacuum assist unit or a full power hydraulic system.

(2) The effectiveness of the braking systems of tractors can be improved by use of large brakes on the front axle of tractors with tandem rear axles (a design configuration in which front brakes are generally absent) and by use of larger brakes on the front axle of two-axle tractors.

(3) The braking efficiency of many trucks and tractor-trailers can be improved by careful distribution of braking effort among the axles of the vehicle.

(4) The brake response time of air-braked systems can be improved significantly through use of larger hoses, improved connectors and fittings, quick release valves, relay valves on tractors, and trailer brake synchronization.

(5) Braking performance can be improved significantly on trucks, buses, and tractor-trailers through use of the advanced brake control systems, which were evaluated by test and/or simulation in this program. These systems, ranked in order of potential for improving braking performance, are:

(a) adaptive braking or antilock system
(b) dynamic load-sensitive proportioning system
(c) static load-sensitive proportioning system

In making specific recommendations for a standard, careful consideration has been given to the necessity to upgrade commercial vehicle braking performance as quickly as possible to acceptable levels. Taking into account the system design problems which will result from increased performance requirements, and the state of development of advanced systems, it is recommended that rules be promulgated which require upgrading the performance of trucks, buses, and tractor-trailers in three discrete steps, separated by appropriate periods of time. As a first step, it is recommended that the rules require immediate action to upgrade braking performance to a level achievable by current design practice; that is, to the best performance already demonstrated by those tested vehicles which were equipped with standard braking systems. For the second step, it is recommended that the rules require performance be improved to the limit of the tire-road interface tractive capabilities of truck tires now available, with due regard to realistic braking efficiencies. The second step may require use of load-sensitive proportioning systems on certain vehicles, and therefore sufficient lead time should be allowed for further development and testing of these devices. After an appropriate period

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to allow for development and testing of a reliable antilock system, the development of truck tires with better tractive characteristics, and the necessary design modifications of vehicle brake, suspension, and structural systems, it is recommended as a third step that performance equal to or approaching that of passenger cars be required along with use of an antilock system to insure vehicle stability over a wide range of vehicle loadings and road surface conditions. Summaries of the suggested performance requirements for each step are given as follows:

Step 1:
- Maximum deceleration capability: 16 ft/sec^2*
- Minimum braking efficiency: 65% for surfaces having peak truck tire-road friction coefficients between 0.2 and 0.8
- Thermal capacity: same as requirements of SAE J786 fade and recovery test except that 15 ft/sec^2 deceleration is required for fade snubs
- Deceleration/pedal force gain: within limits such that the driver can modulate pedal force to prevent wheel lockup on a wide range of surfaces.
- Air brake response time: application, 0.25 sec tractor, 0.35 sec trailer; release, 0.50 tractor, 0.70 trailer
- Special systems required: none

Step 2:
- Maximum deceleration capability: 20 ft/sec^2*
- Minimum braking efficiency: 75% for surfaces (as in Step 1)
- Thermal capacity: test upgraded to correspond with heaviest duty cycles experienced in class of service
- Air brake response time: application, 0.25 sec, tractor, 0.30 sec trailer; release, 0.30 tractor, 0.40 trailer
- Special systems required: static load proportioning (if necessary)

Step 3:
- Maximum deceleration capability: 24 ft/sec^2 with upgraded tires such that peak tire-road surface coefficient is at least 0.85 on a typical dry asphalt or concrete surface
- Minimum braking efficiency: 35% for surfaces having peak truck tire-road friction coefficient between 0.2 and 0.9
- Special systems required: antilock system
- Improved tires also required

Test procedures similar to those used in the test program are recommended for determining brake effectiveness, fade resistance, deceleration/pedal force gain characteristics, and static response time of air brake systems. If stopping distance tests are required, they should be made in conjunction with the effectiveness tests to ensure maximum unlocked wheel decelerations are achieved. Compliance with braking efficiency requirements can be affected by requiring calculations based upon vehicle and brake system design data and validating the calculations by effectiveness tests on high-coefficient and low-coefficient surfaces.

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*It is presumed that this deceleration would be measured on a surface having a peak truck tire-road friction coefficient of at least 0.75.

Testing of tractor-trailer combinations presents a special challenge. Becuase of brake balance problems, a tractor which may perform very well with one trailer may perform poorly with another. Conversely, a trailer whose brakes were tested on a brake dynamometer and deemed adequate may perform well with one tractor but perform poorly with another. Also a tractor could perform well as a loaded straight truck, but poorly in combination with a trailer. For this reason it is recommended that tractors be certified to pull only those trailers for which design calculation and test have demonstrated adequate combined performance.

1.5 REPORT ORGANIZATION

The work performed in each of the four tasks mentioned in Section 1.3 is described in Sections 2 through 6 of this report. Section 2 constitutes the Literature Review. This review is not only a catalogue of the technical literature surveyed during the course of this study, but a critical discussion of each aspect of the analysis, testing, design, and performance of braking systems found to be pertinent to the work at hand. The results and conclusions of this review were used extensively in formulating the test program and in planning and executing the analytical phases of the study.

Section 3 is devoted to the Test Program. In this section the criteria for (and the method of) selection of the test vehicles are set forth, test requirements are cited, and test results given.

Section 4 describes the Analytical Program. Analysis of the braking performance of several of the test vehicles has demonstrated the feasibility of making accurate predictions of braking performance based upon vehicle and brake-system design data. Thermal analyses are also described, as are the results of the dynamic modeling and simulation activities undertaken in this study.

In Section 5, the Analytical Results and Experimental Findings are evaluated, leading to Section 6 which presents recommendations for a braking performance standard for trucks, buses, and tractor-trailers.

There are seven appendices to this report. Appendix A lists Data and Specifications for the test vehicles. Appendix B documents test procedures employed for the vehicles equipped with standard brakes and for those with improved brakes and/or advanced brake control systems. The tire-road interface tests, made on the surface upon which the braking performance tests were conducted, are described in Appendix C, where the findings are presented. Appendix D describes the hybrid computer simulation model used in the analytical phase of this study, and Appendix E lists and describes the digital computer program used for making calculations of braking performance. Appendix F details the formulation of the three measures used to rate potential vehicle performance in the survey conducted for purposes of selecting the test vehicles. Appendix G describes the trailer brake synchronization system.

2. LITERATURE REVIEW

2.1 INTRODUCTION

   2.1.1 SCOPE OF SURVEY. This review contains a summary of domestic and foreign literature dealing with braking of commercial vehicles. The literature was searched with the objective of finding information pertinent to accurate analyses of braking systems, experimental test procedures, and improved means of measuring the performance of braking systems. Important factors in the review are brake system design, braking performance, brake usage, brake testing, brake failure, and performance standards.

   The review is divided into several sections, corresponding to these factors. In Section 2.2 (The Decelerating Vehicle) the braking performance of integral trucks and articulated vehicles with fixed brake force distributions, proportional braking systems, and wheel antilock systems are discussed. In Section 2.3 (Brake System Elements) the actual hardware and operation of different braking systems are described. Literature found on the mechanical and thermal analysis of brakes is discussed in Section 2.4 (Analysis of the Friction Brake). In Section 2.5 (System Design Considerations), equations and procedures for the design of braking systems and braking system components are presented. Important factors to be considered in brake testing are discussed in Section 2.6 (Braking Test Procedures). Motor truck braking performance standards existing in the U.S. and in foreign countries are discussed in Section 2.7 (Performance Standards).

   2.1.2 METHODOLOGY. In conducting this survey, we first reviewed the technical literature available in the HSRI library. (Included in the HSRI material were three previously published literature reviews on brakes.) Engineering and technical indices and abstracts were surveyed. Finally, references listed in pertinent publications were cross-checked and examined. A total of 800 papers were processed—including German, Russian, French, Japanese, and English publications. More than 300 documents were examined in detail and are included in this review.

   2.1.3 TERMINOLOGY. The terminology established by the Society of Automotive Engineers (SAE) for the automotive braking system and brake operation (1,2)* is used throughout this report. To minimize the confusion resulting from a multiplicity of symbols, all equations were rewritten using symbols commonly used in U.S. publications at the present time. The symbols are defined in the text when first introduced and are summarized at the beginning of this report.

2.2 THE DECELERATING VEHICLE

   2.2.1 BRAKE FORCE MODULATION. During braking, the kinetic energy of the vehicle is converted into thermal energy at the friction surfaces of the brake and at the tire-roadway interface. In the braking process, the brakes

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*References referred to in text by numbers in parentheses are listed numerically in <u>References</u>, following Appendix G, Volume II.

generate a retarding torque as a function of the applied pedal force. The pedal force/braking torque characteristics are determined by the mechanical/ hydraulic or mechanical/pneumatic parameters of the braking system whereas the actual deceleration of the vehicle is determined by the braking torques, the tire-roadway friction coefficient, and the normal force between tire and road-way. The normal forces change with the dynamic load transfer from the rear axle(s) to the front axle. For vehicles equipped with tandem axles, load transfer occurs on individual axles of the tandem suspension (3-5).

Control of the braking process through the pedal force depends on the static and dynamic characteristics of the entire braking system. For a hydraulic braking system without power assist, the necessary work input (in.-lb) at the master cylinder (in terms of the hydraulic pressure multiplied by the displaced fluid volume) is directly related to the required work output at the individual brakes. The work output is determined by the product of hydraulic pressure and the displacement of the wheel-cylinder piston. The minimum piston displacement is dependent upon brake type, brake size, compression of the lining material, and lining wear as well as drum expansion due to thermal or mechanical deformation (6). The work output of a compressed-air brake system is limited by reservoir pressure and volume of the reservoir (7).

The fundamental expression relating deceleration and pedal force for a hydraulically actuated drum braking system with identical brakes at each wheel is given by (6,8):

$$a_x = \frac{g}{W} \cdot F_p \cdot \frac{n_B}{n_S} \cdot \frac{r}{R} \cdot BF \cdot \rho \cdot \eta \tag{2-1}$$

where

a_x = deceleration

BF = brake factor defined as the ratio of drag force on the drum friction surface over the actuating force of one shoe

F_p = pedal force

g = gravitational constant

ℓ_h = hydraulic ratio between master cylinder and wheel cylinder

ℓ_r = pedal lever ratio

n_B = number of brakes

n_S = number of brake shoes

r = drum radius

R = effective tire radius

η = total efficiency

ρ = total gain ℓ_r x ℓ_h = pedal travel divided by actuating travel of one brake shoe

W = vehicle weight

For example, Equation 2-1 reduces to

$$a_x = 0.5 \cdot \frac{g}{W} \cdot F_p \cdot \frac{r}{R} \cdot BF \cdot \rho \cdot \eta \tag{2-2}$$

for a two-axle vehicle equipped with four brakes.

It is clear that the gain ρ—and hence both the pedal travel and the

shoe displacement—is as important as force F_p, in determining decelerations (9). Relationships similar to Equation 2-1 can be derived for other types of brake systems.

For hydraulic braking systems with vacuum assist, the work input at the master cylinder is equal to the work done at the pedal plus the work performed by the booster (10). Depending upon the booster characteristics, the pedal force serves to a greater or lesser extent as an element for controlling the braking process rather than as a work-producing element. Consequently, in the case of a booster failure, due to the increase in required pedal force levels the decelerations obtainable are considerably less than those achievable under normal operating conditions, as is illustrated in Fig. 2 (11-13).

The fundamental principle utilized in pneumatic braking systems is pressure equalization between two volumes: the volume of the reservoir and the sum of all brake chamber and connecting pipe volumes (7,12,14). (See Fig. 3.) The energy stored in the compressed air of the reservoir is transmitted to the brakes through the brake lines. The brake valve enables the driver to apply and release the brakes gradually. In the case of a reservoir failure, no braking action is possible unless an emergency system is provided. A detailed discussion on the design and evaluation of hydraulic and air brake systems is given in Section 2.5 (System Design Considerations).

2.2.2 TIRE-ROADWAY FRICTION. The friction between tire and roadway determines to a large extent the braking capability of a vehicle. It can be defined in terms of the coefficient of friction μ_x, which is equal to the ratio of the tangential force transmitted by the tire to the normal load carried by the tire (15). The coefficient of friction depends upon the tread material and the geometry of the tread in addition to the surface geometry of the roadway, the nature and thickness of any film (water, mud, oil, etc.) present in the contact area, the sliding velocity, and operating temperature (10,16-26).

A typical curve indicating the manner in which friction coefficient varies with longitudinal slip of the tire is given in Fig. 4. (Longitudinal slip is defined as the ratio of the difference between the actual speed of the vehicle and the equivalent ground speed of the tire to the speed of the vehicle.) Available data indicate that the peak of the friction curve generally lies between 5 and 25% slip for most tires and test conditions (24,27). Antiskid devices try to make use of this behavior of the tire. Note that the friction of a given tire-roadway combination is frequently specified by a peak and locked-wheel coefficient of friction (10,18).

As can be seen from Fig. 4, adaptive brake force modulation, i.e., a wheel antilock system, will not bring about a substantial improvement in friction utilization or stopping distance on dry surfaces because the ratio of the friction coefficient peak/slide is nearly unity. A considerable decrease in stopping distance can be expected, however, with brake force modulation on wet surfaces (18,27,28). Figure 4 also illustrates that the effective lateral friction coefficient of a free-rolling wheel is significantly
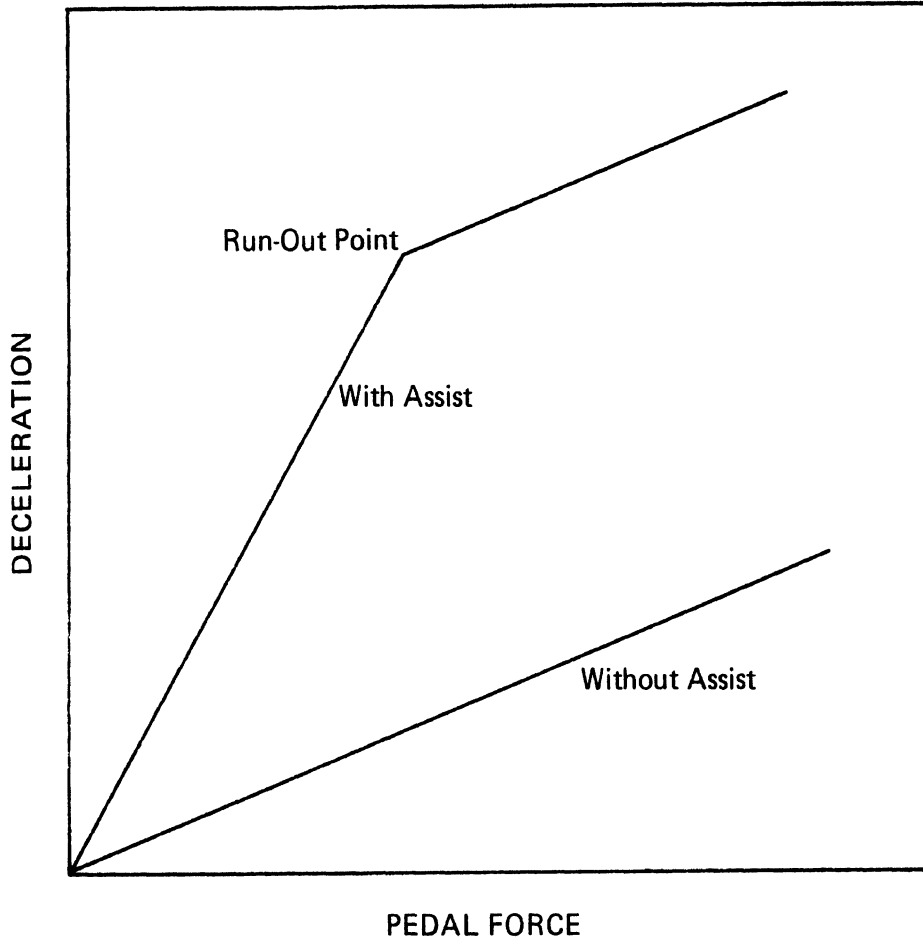
FIGURE 2. VACUUM-ASSIST CHARACTERISTICS
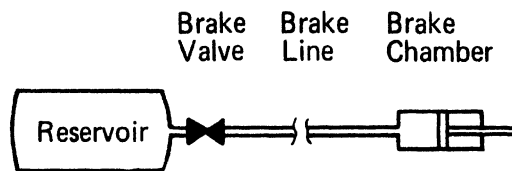
FIGURE 3. AIR-BRAKE SCHEMATIC
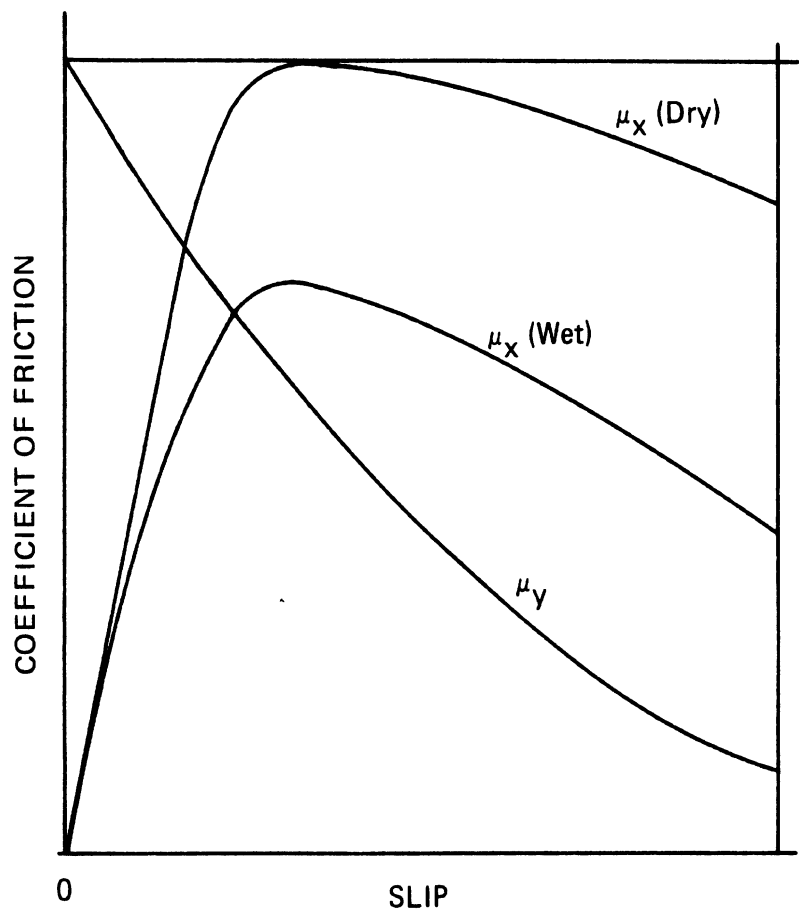
FIGURE 4. COEFFICIENT OF FRICTION AS A FUNCTION OF SLIP

reduced at large values of longitudinal slip (18,29,30).

The measured friction coefficient of a public road varies both season-
ally (31-33) and from lane to lane (34) on the highway. These measurements
of road friction can be accomplished in several ways (35-37). Three methods
have been widely used: skid trailer (36), vehicle stopping distance measure-
ments (38), and portable testers (39). Skid trailers can be used to measure
peak and sliding friction coefficients (40,41). Comparisons between the
British Portable Tester and automobile stopping distance measurements (42)
show good correlation (43) of the results when patterned tires are used. In
all skid resistance measurements, conditions must be carefully controlled in
order to obtain consistent results (44). On treating the braking process as
a static phenomenon, the braking efficiency is derived in the following man-
ner. The coefficient of friction between the tires and the road necessary to
prevent the lockup of wheels on the front axle is given by:

$$\mu_x = \frac{F_{X,F}}{F_{Z,F}|dyn} = \frac{(1 - \phi)\cdot(a_x/g)W}{[1 - \psi + \chi(a_x/g)]W}$$

where

$F_{X,F}$ = brake force on front axle

$F_{Z,F}|dyn$ = vertical force on front axle

ψ = static rear axle load divided by total vehicle weight

ϕ = brake force between tire and roadway generated at the
     rear axle, divided by total brake force

χ = height of center of gravity divided by wheel base

The braking efficiency, as determined from the forces acting on the
front wheels, can by computed from

$$E_F = \left(\frac{a_x/g}{\mu_x}\right)_F = \frac{1 - \psi}{1 - \phi - \mu_x\chi} \tag{2-4}$$

Similarly, the braking efficiency, as determined from the forces acting on
the rear wheels, can be computed from

$$E_R = \left(\frac{a_x/g}{\mu_x}\right)_R = \frac{\psi}{\phi + \mu_x\chi} \tag{2-5}$$

2.2.3 BRAKING EFFICIENCY. Friction utilization or braking efficiency
E is defined as the ratio of the maximum, wheels-unlocked deceleration a_x/g
to the coefficient of friction μ_x existing between tire and roadway. Thus,
braking efficiency can be expressed as

$$E = \frac{a_x/g}{\mu_x} \tag{2-3}$$

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In general, E will be different for the individual axles over a wide range of loading and driving conditions due to dynamic weight transfer (45-51) and, in the case of articulated vehicles, due to forces transmitted through the king-pin of the trailer (3,52).

2.2.3.1 Integral Trucks with Fixed Brake Force Distribution. Braking efficiency is directly influenced by the distribution of brake force braking effort among the axles. Accordingly, commercial vehicles with a large weight difference, loaded and empty, tend to have reduced braking efficiencies at driving conditions departing from the design point. A fixed brake force distribution is usually selected for integral trucks such that the braking process gives reasonably satisfactory results for the load condition and deceleration level that occur most frequently.

Braking efficiency E of the integral motor truck is determined by the center of gravity location, the brake force distribution, and the tire-road-way friction coefficient (45,46,50). The brake force distribution ϕ is typically not constant during the braking process. At low decelerations, ϕ will depend upon the difference in pushout pressures on front and rear axle, whereas at higher decelerations, ϕ may be affected by fade (3). For a truck with typical values for ψ, χ, and ϕ, Equations 2-4 and 2-5 produce the result shown in Fig. 5. Note that for the loaded condition, the limiting value of braking efficiency is determined by the front axle, while for the empty condition, the front axle limits up to a μ of 0.68, at which point the rear axle becomes the limiting factor.

The stopping distance that can be achieved prior to wheel lockup is directly related to braking efficiency (46,50,51). For a braking efficiency of one, that is, a deceleration in g units equal to the existing friction coefficient, the stopping distance of the vehicle is the minimum achievable. If it is assumed that the deceleration a_x/g and the friction coefficient μ_x remain constant during the braking process, the incremental stopping distance ΔD (wheels unlocked), resulting from a braking efficiency less than unity, is given by the expression

$$\Delta D = \frac{V^2}{2g} \left(\frac{\mu_x}{a_x/g} - 1 \right) \qquad (2-6)$$

where V = initial speed of the vehicle. On using Equation 2-3, the expression given in Equation 2-6 becomes

$$\frac{\Delta D}{D_{min}} = \frac{1 - E}{E} \, 100 \; [\%] \qquad (2-7)$$

where

$$D_{min} = \frac{V^2}{2g\mu_x} = \text{minimum stopping distance.}$$

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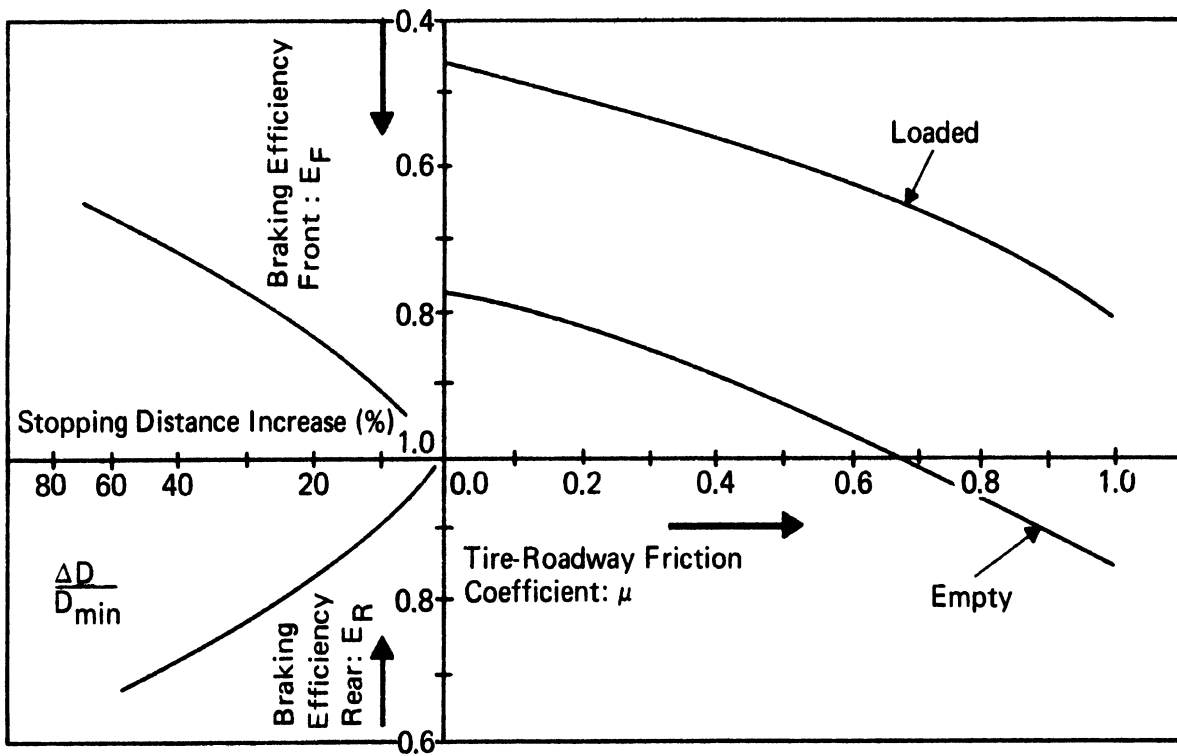
FIGURE 5. BRAKING EFFICIENCY FOR TWO-AXLE VEHICLE EQUIPPED WITH FIXED BRAKE FORCE

The increase in stopping distance as calculated from Equation 2-7 for a typical truck is also shown in Fig. 5.

If it were required that the efficiency E never be less than a specified value E_{min} irrespective of roadway and loading conditions, a very tight bound is placed on the distributions of brake force ϕ that will satisfy this requirement. If the requirement for E_{min} is set too high, it will not be possible to achieve it with a fixed brake force distribution. The existence of a fixed value of ϕ that provides $E > E_{min}$ can be checked through the following inequality (50)

$$\left[1 - \mu\chi - \frac{1 - \psi}{E_{min}}\right] \leq \phi \leq \left[\frac{\psi}{E_{min}} - \mu\chi\right] \qquad (2-8)$$

Application of this inequality to the wet and dry roadway ($0.2 \leq \mu \leq 0.8$) and over the full range of ψ and χ defines an envelope for values of ϕ that yield $E \geq E_{min}$ (50).

Experimental work has been reported by Alexander (53) comparing different fixed brake force distributions and load conditions for passenger cars. In tests of cars with three different brake force distributions, he showed that only one distribution resulted in minimum stopping distance.

The relationships discussed above (Equations 2-4, 2-5, and 2-8) are also applicable to integral trucks equipped with tandem-axle suspensions. Tandem-axle suspensions are designed to distribute the static load among both axles independent of irregularities in the road surface. They can be classified into three basic groups (according to their attachment to the truck frame) as is indicated in Fig. 6: (a) walking beam suspension, (b) two-elliptic leaf spring suspension, and (c) multiple leaf-multiple rod suspension. [The history of tandem suspensions has been reviewed by Hendrickson (54).] In general, the reaction moments during braking cause a change in load distribution among both axles of the tandem suspension. During braking, the forward axle of a walking beam or multiple leaf-multiple rod suspension will be subjected to an increase in load, while the forward axle of a two-elliptic leaf spring suspension will experience a decrease in axle load from their respective static load values (3-5,55). Since load transfer among axles of a tandem suspension can lead to premature wheel lockup, tandem-axle geometry and the brake force distribution among individual axles of a tandem suspension have a pronounced effect upon peak, wheels-unlocked deceleration performance.

2.2.3.2. Articulated Vehicles with Fixed Brake Force Distribution. The sum of the dynamic axle loads of each unit of a truck-trailer combination is equal to the weight of the particular unit, assuming no forces are transmitted by the hitch. In the case of a tractor-semitrailer, the axle loads of the tractor are influenced by the loading and brake forces of the trailer. The equations expressing weight transfer, brake forces and decelerations achievable without wheel lockup for a tractor-semitrailer are consequently more complicated than those for the integral truck and the truck-trailer combination (3,55-61). Experimental work in Europe has indicated that the locking of
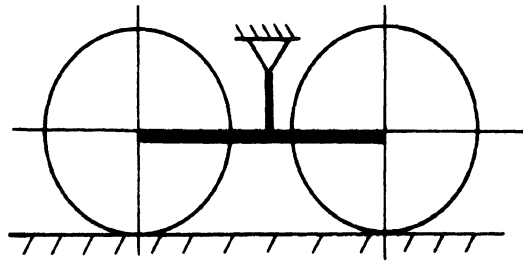
(A) Walking-beam suspension.
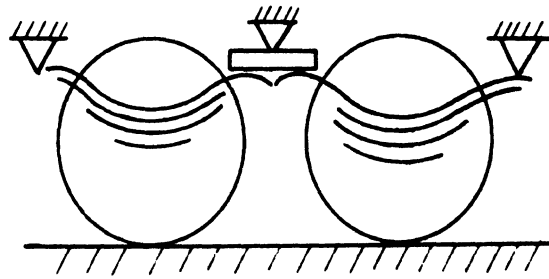
(B) Two-leaf spring suspension.
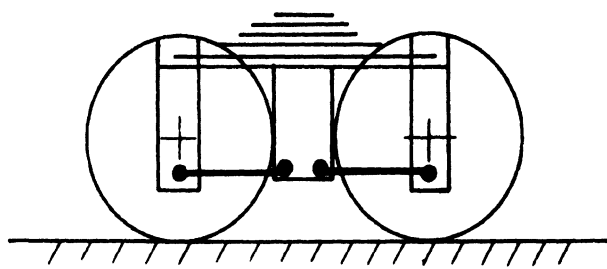
(C) Multiple spring-multiple rod suspension

FIGURE 6. TANDEM AXLE SUSPENSIONS

individual axles influences the stability of the combination in accordance with theoretical predictions (62-64). It has been observed that the tractor-trailer combination is stable with a locked front axle, unstable with the trailer axle locked (producing "trailer swing"), and violently unstable with the tractor axle locked (producing "jackknifing"). The potential for jack-knifing is greatest at the beginning of a severe braking process due to the large longitudinal forces that are created at the kingpin if the application and buildup time of the brake force on the trailer is greater than for the tractor (52,57,61,62).

Prevention of wheel lockup appears to be the greatest deterrent to artic-ulated vehicle instability. Assuming that wheel lockup does occur, undesired articulation and directional response can be minimized by having the front wheels lock up first, then the trailer wheels, and then the tractor rear wheels. The often observed practice in the U.S. of disconnecting or removing the front brakes of a tandem-axle tractor is, however, in conflict with the above observation (65). Eliminating the front brakes without changing the baseline distribution of the combination has, generally, an unfavorable in-fluence on the braking performance of the vehicle combination (3).

Several means for determining the optimum fixed brake force distributions on a tractor-semitrailer have been suggested. They consist of either comput-ing the decelerations achievable for a given tire-roadway friction coefficient for several assumed brake force distributions or computing the brake force distribution as a function of vehicle data and the desired range of braking efficiencies (3,58,59). Given the brake force produced by a trailer, brake force distribution on the tractor that will yield a minimum stopping distance can be computed from data defining the geometry and loading of the combination (3). A static analysis yields that

$$\phi_{1R} = \frac{A}{E_{1R}} - \mu_{1R} \cdot B \qquad\qquad (2-9)$$

where

$A = \lambda \psi_1 + (1 - \lambda)(1 - \psi_2)y + \rho(z_2 y + z_1)$

$B = \lambda \chi_1 + (1 - \lambda)z_1 - (1 - \lambda)(\chi_2 - z_2)y$

E_{1R} = braking efficiency produced by tractor rear axle

μ_{1R} = coefficient of friction between tractor rear axle and roadway

$\lambda = \dfrac{W_1}{W}$, $\rho = \dfrac{F_{zR}}{W}$, $W = W_1 + W_2$

(See Fig. 7 for other symbols)

Since in most cases the constants A and B in Equation 2-9 are different for the empty and loaded vehicle, Equation 2-9 will usually yield two dif-ferent values for ϕ_{1R}. For most cases, the tractor rear axle brake force distribution ϕ_{1R} should not exceed 50% (63). A relatively small value of ϕ_{1R}, and hence a moderate utilization by the tractor rear axle of road friction in the longitudinal direction, means that increased lateral forces are avail-able from the tires for maintenance of directional stability. This result is important since the danger of jackknifing is directly related to the lateral forces that can be produced at the tractor rear axle (61,62).

Brake forces should be distributed on a truck-trailer combination such

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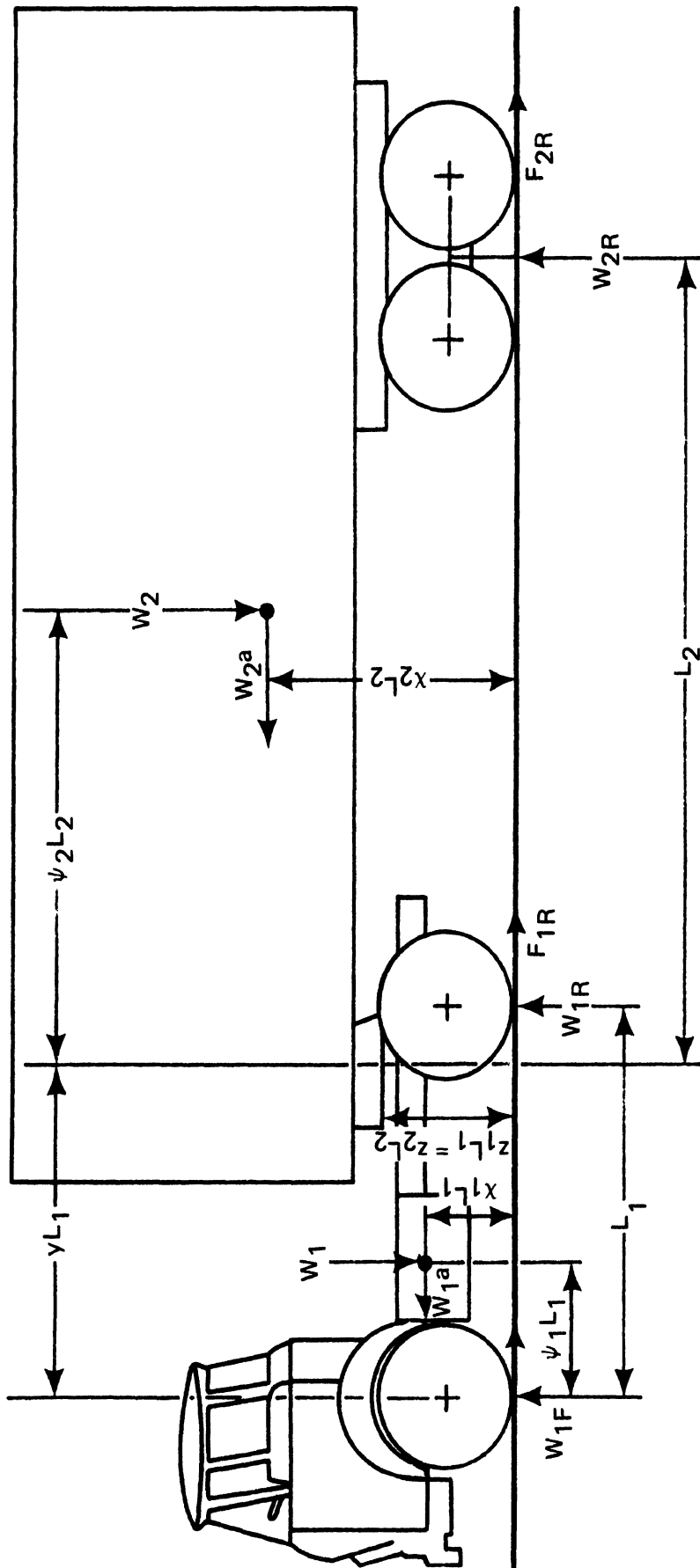
FIGURE 7. GEOMETRIC AND LOAD CONFIGURATION OF TRACTOR-SEMITRAILER

that the longitudinal forces at the hitch point are approximately equal to zero during braking. This practice is especially important on combinations in which large differences exist between the loaded and unloaded driving condition (66-68). Time differences in the application of brakes on the individual axles is also an important factor (56,69). On a semitrailer, for example, if brake torque builds up on the tractor rear axle much faster than on the trailer axle, the combination may jackknife, especially on slippery roadways. This phenomenon can occur very quickly, with the result that the driver loses steering control because he cannot correct for the jackknife in the short time available (70).

It should be noted that dynamic load transfer between the forward and rearward axle of a tandem axle suspension has been neglected in deriving Equation 2-9. This assumption is valid only for tandem axle designs with equalization levers resulting in an approximately constant load distribution. For axles without equalization, however, the reaction moments during braking cause a change in load distribution among the individual axles (3,4-6,76). In this instance, the braking efficiencies of each axle have to be computed individually.

2.2.3.3 Variable Brake Force Distribution. If the deceleration levels achieved prior to wheel lock, as limited by a fixed distribution of brake force, are considered to be insufficient, a variable distribution of brake force can be employed. In the latter instance, the brake forces are proportioned so that they more closely approximate the ideal brake force distribution over a wide range of driving conditions. The ideal brake forces are related to the vertical loads on the axles produced by the dynamic weight transfer during the braking process (3,8,45,46,50,53,71-75). A braking efficiency curve typically produced by a braking system with a variable distribution of brake force is shown in Fig. 8. It is clear that the system as designed produces better braking efficiencies on road surfaces with higher coefficients of friction. The braking efficiency curve in Fig. 8 also demonstrates that the designer wanted to avoid lockup of the rear wheels at the higher coefficients of road friction.

For design purposes, it proves to be more convenient to compare the "ideal" pressures (either hydraulic or pneumatic) desired at the wheel cylinders or brake chambers to the pressures actually delivered to the wheel cylinders by the braking system. "Ideal" pressures are those that correspond to the ideal braking forces, i.e., they are dependent upon the dynamic weight transfer (46,50,72,76). Ideal and actual pressures, corresponding to the example used for computing Fig. 8, are shown in Fig. 9.

Proportioning valves have been designed that modulate line pressure as a function of the static or dynamic axle loads (72,74-83). In the case of static modulation, the valve setting is not affected by suspension movement during the braking process. For dynamic modulation, the proportioning system must, however, be made sensitive to deceleration. For example, suspension deflection during braking has been used to modulate individual brake line pressures. With this type of proportioning, suspension and road noise must be filtered out while still retaining an adequate signal.
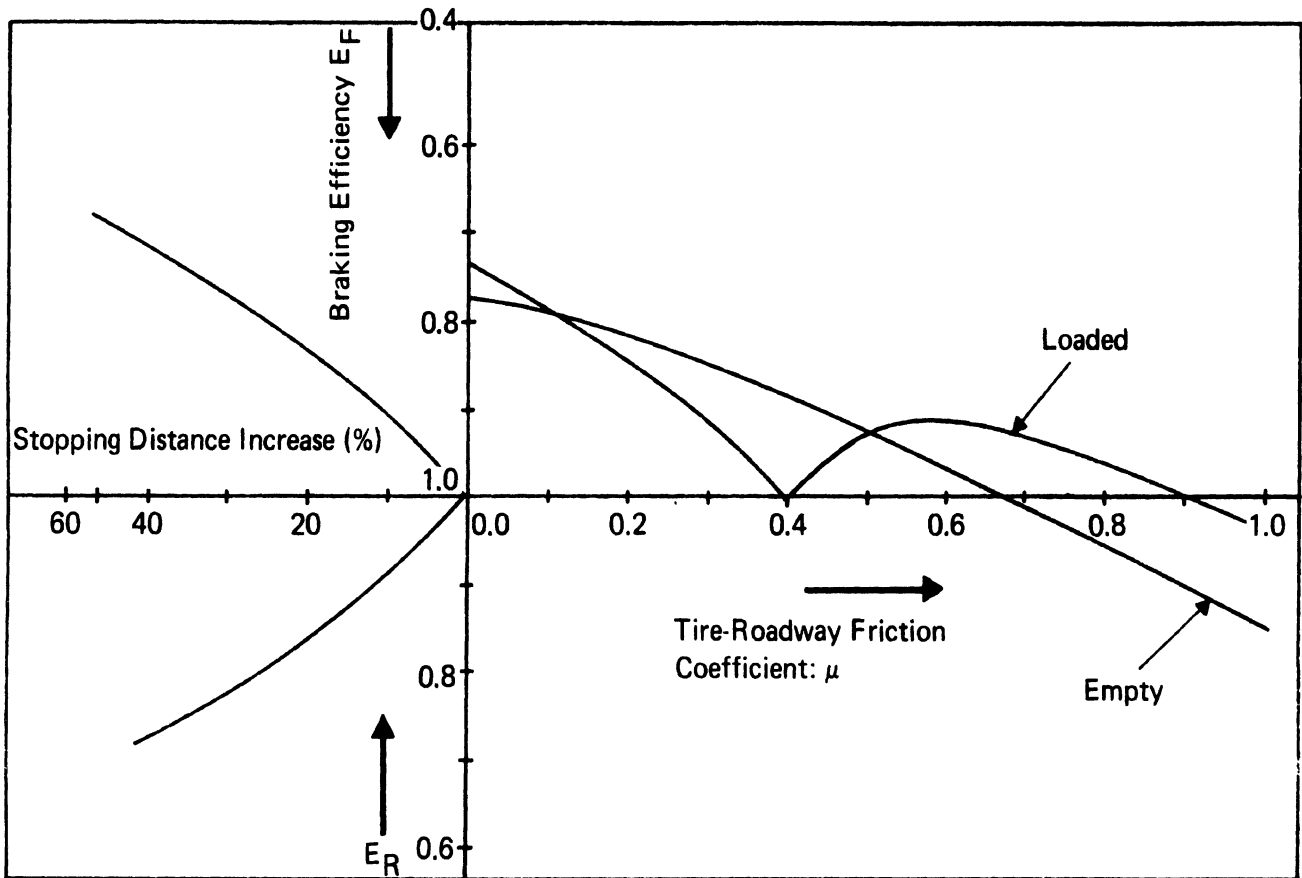
FIGURE 8. BRAKING EFFICIENCY FOR PROPORTIONAL BRAKING
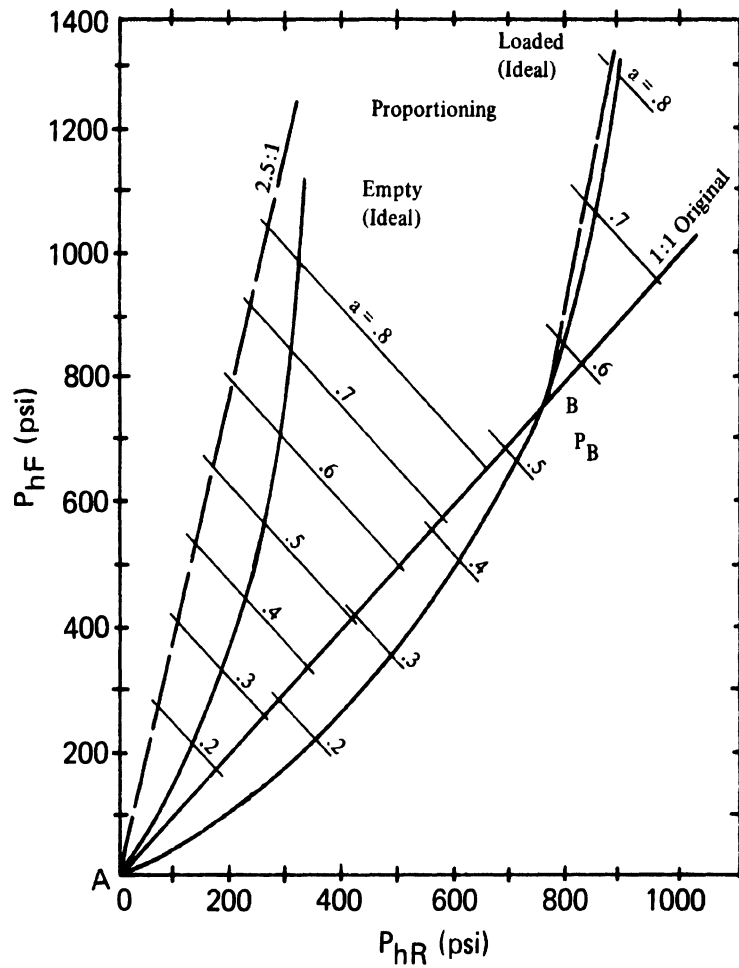
FIGURE 9. IDEAL AND ACTUAL HYDRAULIC PRESSURES

The ideal brake forces for a tractor-semitrailer in the loaded and empty driving condition are shown in Fig. 10. These curves show that the ideal brake force on the front axle varies little with change in vehicle loading, whereas the ideal brake force on the rear axle of the tractor and on the trailer axle is heavily influenced by the loading condition (3,4,56). Studies have shown that the following procedure is convenient for implementing variable brake torque distribution in an articulated vehicle (79):

(1) The front axle brake force of the tractor is designed to be proportional to brake line pressure.

(2) The brake force at the rear axle of the tractor is determined by a load sensitive proportioning valve. Depending on the design of the valve, the brake torque on the tractor rear axle may vary, for example, from 50 to 120% of the front axle brake torque of the tractor.

(3) It is convenient and sufficient to control the brake force of the trailer axle by a manually positioned limiting valve which has settings for the empty, half-loaded, and loaded conditions resulting in different limiting brake torques on the trailer axle (56,66,80-83).

When brake proportioning schemes are employed, it becomes necessary to examine the variability in performance that may result from using different trailers with the same tractor. Calculations of braking efficiency should be made by computing the coefficient of friction required by each axle to achieve a given deceleration without wheel locking. In order to calculate friction utilization, the effective axle loads have to be determined. For tractor-semitrailer and truck-trailer combinations, the equations relating deceleration achievable and tire-roadway friction have the same appearance as those derived for braking systems with fixed distribution of brake torque. For vehicles with proportioning systems, the brake force distribution is determined by the static loads on the tractor rear axle and trailer axle (3,66,82,83) or by the dynamic axle loads (80,81) during braking.

Since braking systems with proportioning are often implemented by making the line pressure a nonlinear function of the pedal force rather than a function of the actual deceleration of the vehicle or a function of the friction coefficient, it is still possible to lock the wheels, especially on low friction surfaces. Although proportioning can provide a brake force distribution to match a wide range of loading and dynamic driving conditions, a practical system is still subject to some of the basic limitations of systems with fixed torque distribtuion (75).

2.2.4 WHEEL ANTILOCK SYSTEMS. Wheel antilock systems prevent the wheels from locking up during braking by adjusting the braking effort to the traction force available at the tire-roadway interface. These systems were employed on aircraft as early as 1952 (10,84). Shortly thereafter several antilock devices were introduced for use on automobiles (85-91). However, by 1961 no satisfactory wheel antilock device was in production for road vehicles in Europe (48) or the United States. During the last ten years, extensive research and development has been performed on wheel antilock braking systems for both passenger cars and commercial vehicles (27,28,92-118).

(A) Loaded truck

Ideal Brake Forces

Tractor Rear

Trailer

Front

Deceleration

(B) Empty truck

Ideal Brake Forces

Front

Tractor Rear

Trailer

Deceleration
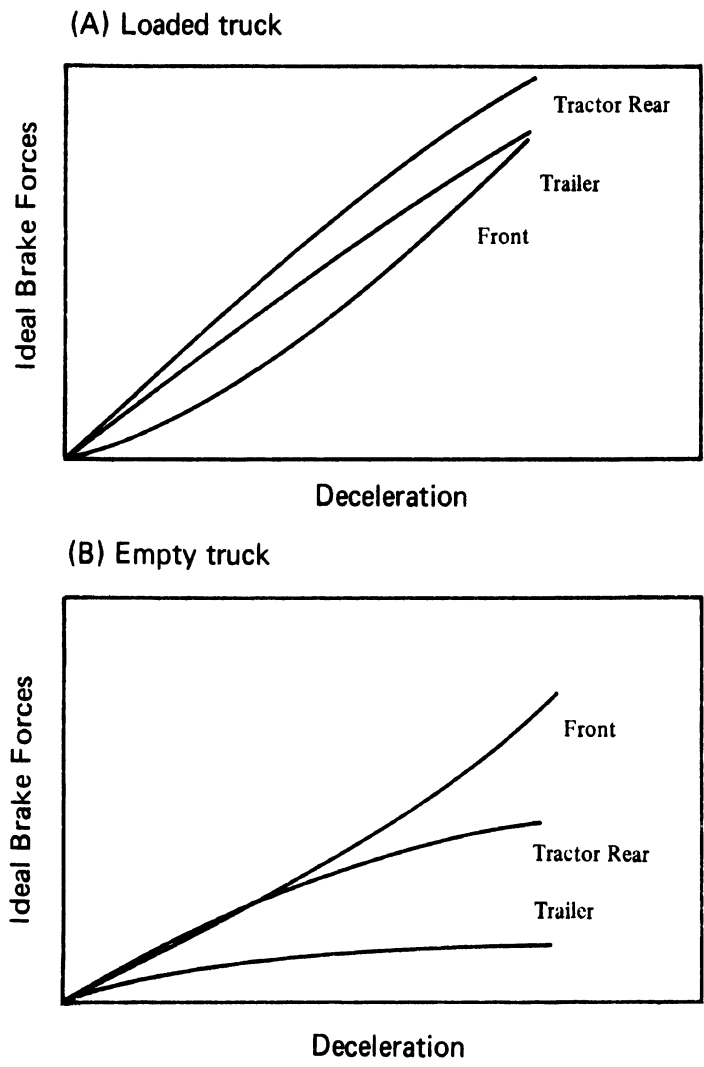
FIGURE 10. IDEAL BRAKE FORCES ON TRACTOR-SEMITRAILER

When wheel antilock systems are installed in a motor vehicle, the driver operates the brakes as usual under normal conditions. On slippery roadways or during severe braking, the device takes over and modulates the brake force whenever the driver causes the wheels to approach lockup (119). The angular deceleration of the wheel is frequently used as a control variable as it becomes very large when wheel lockup is impending and can be measured rather easily (27). In order to retain directional stability during braking and achieve minimum stopping distances on wet road surfaces, the wheel deceleration control limit is set such as to cause the wheel to operate at slip values below the peak of the wet μ-slip curve (see Fig. 3). Tests using wheel antilock devices on dry pavement have demonstrated that, in some instances, a slight increase in stopping distance results, while in other cases a slight decrease in stopping distance is noted (27,120).

2.2.5 DRIVER-VEHICLE INTERFACE. Modulation of the brake force to achieve minimum stopping distance and to retain directional stability during braking must be provided by the operator unless the vehicle is equipped with a wheel antilock system. Several investigators have examined the problem of the driver-vehicle interface as it impinges on this modulation task (13,119, 120). Although the ability of the driver to perform as a controller is affected by the design of the cab, a study of ten truck cabs indicated that only a few were designed with the comfort of the driver in mind (121). Many cabs were found to be below minimum standards essential to safe and efficient operations. In some cases, the leg movement was hindered during braking by obstructions.

Important factors in the design of truck cabs are: efficient operation, visual efficiency, comfort, and safety. For example, controls should be assigned to proper body segments. Important and frequently used controls should be assigned to preferred locations. Controls should be grouped according to use and function, instruments should be readable from normal position, seats should be adjustable, etc. (121). A study of the driver's position relative to the brake pedal indicates that the pedal force will be at a maximum for a particular angle between leg and thigh (122). A seating design study (123) has concluded that adequate seat adjustment is needed in addition to a good spring suspension.

2.3 BRAKE SYSTEM ELEMENTS

2.3.1 MECHANICAL SYSTEMS. In mechanical brake systems, the brake pedal is connected to the brake assembly by levers and rods or cables. Maintenance is an important factor in obtaining a maximum overall mechanical efficiency. Due to frictional losses, the total braking effect, even in well-maintained systems, is reduced considerably (10). Road tests indicate brake system efficiencies of approximately 69% for mechanical, 92% for hydraulic, and 73% for pneumatic systems (124). Pure mechanical systems are, therefore, no longer used for service brakes. Mechanical systems are used, however, for parking brakes since large displacements are more easily accommodated by hand (125). Mechanical systems with automatic adjustment devices generally have time lags small enough that vehicle braking response is not adversely affected (124).

2.3.2 HYDRAULIC SYSTEMS. Hydraulic brake systems without power assist are commonly used on light-weight trucks. Brake efficiencies are high, as mentioned above. The foot pedal connects to a single or tandem master cylinder. The pedal displacement is transmitted via the hydraulic brake fluid to the wheel cylinder (10,11,119). Hydraulic brake systems exhibit very little time lag between pedal displacement and buildup in brake torque (10,110,124). However, on adding a hydraulically braked trailer, the time lag between pedal actuation and brake torque buildup can become significant (124,125). This time lag can be reduced by decreasing the fluid volume at the wheels through smaller wheel cylinders and automatic adjusting devices (110).

Since the pedal forces that can be developed by the operator are limited, additional power sources are frequently used to generate higher brake forces. Among such devices are vacuum boosters which utilize the vacuum developed in the engine manifold. The vacuum assist is either hydraulically (10,11,126) or mechanically controlled (10,120). For motor vehicles equipped with diesel engines and for heavy vehicles, a vacuum pump and reservoir become necessary. Tests on articulated vehicles equipped with vacuum hydraulic brakes have shown that there may be a time lag of as much as two seconds between pedal operation and application of the trailer brakes. Improved vacuum systems have been developed that show little or no time lag (125-127).

Full hydraulic power brakes employ a pump reservoir system in which the hydraulic fluid is under pressure at all times. Since the system can be brought into operation by opening a valve, i.e., the hydraulic pressure does not have to be built up from zero, the time lag will be small (11,128,129) and the stopping distances are shorter compared to air or vacuum assist brake systems. Hydraulic power brakes seem particularly suited for vehicle combinations since the length of the brake lines has little effect upon the time lag (124). The system can be designed easily for dual circuit operation (10).

A large number of heavy commercial vehicles in England and Germany are equipped with air boost over hydraulic brake systems (130). In this case compressed air is used to assist in building up large hydraulic pressures (11,131,132). From a time lag point of view, a hydraulic pump assist is preferable to a compressed air over hydraulic system (124).

2.3.3 COMPRESSED AIR SYSTEMS. The functions of the major components of an air brake system (10,133,134) are described briefly below, in order to familiarize the reader with the terminology.

The function of the air compressor is to build up and maintain the pressure required for the operation of the braking system and additional air-powered devices. The governor automatically controls the air pressure between the maximum and minimum pressures desired. The brake valve is the control unit of the brake system and provides the operator with an easily operated device to apply or release the brakes. Quick release valves speed the release of air from the brake chambers. Brake chambers at the wheels utilize the air pressure to actuate the brakes. Slack adjusters provide a quick and simple method of automatically adjusting the brakes.

Compressed air brake systems provide large operating forces at the brakes, and are particularly suited to articulated vehicles because of the ease with

which brake line connections are made between tractors and trailers (10). However, air brakes have relatively long response times and high power losses. With an overall mechanical efficiency of about 73%, the total power losses in air systems are greater than those associated with hydraulic systems (124). The time lag can be kept small through adequate pneumatic piping design.

Air pressure systems have been designed for use on light-weight vehicles (135); however, they have not been used extensively on these vehicles. A comparison made of different air brake system designs used in the U.S. and several European countries (136) shows that the design of braking systems of heavy vehicles operating in the U.S. is almost identical to air brake systems used in Europe. Only design components such as reservoir sizes, types of piping, couplings, etc., were examined in this comparison.

Improved control systems for air brakes and dual air brake systems have been introduced in the last few years (137-140). It appears that an emergency braking system can be easily incorporated into air brake systems. One such example is the MGM brake (141). It provides a secondary spring-actuated braking system capable of stopping and holding the loaded vehicle when the service brakes fail. Air pressure is used to retract the spring to maintain the off position of the brake. The parking brake is applied by exhausting the cylinder and is released by pressurizing the cylinder. When the primary or service system fails, the MGM brakes automatically apply and generate approximately 70% of the maximum retardation. In the case of air leakage, the brake will begin to operate once the system pressure falls below a certain level. Alfieri (142) discusses the influence of different failure modes of braking systems of tractor-trailer combinations on the braking behavior of the combination. Results indicate that in the case of a brake failure on the trailer, not too much of the total braking action is lost.

2.3.4 PROPORTIONING AND ANTISKID BRAKING SYSTEMS. As mentioned earlier, brake force proportioning devices are either statically or dynamically load-sensitive. The static-load-dependent valves are in operation only when the brakes are not applied, i.e., they are locked in a fixed position at the moment of brake application (82). Several different devices have been designed for use on heavy commercial vehicles (74,79).

Although wheel antilock systems were introduced several years ago (143-145), and a number of companies are currently developing these systems, there is limited informative material on these devices (27,28,117,146,147).

2.3.5 SYSTEMS FOR SUSTAINED BRAKING. Tests have shown that brake lining wear can be reduced considerably during sustained braking if vehicle retarders (148,149) or exhaust brakes (150-152) are employed. For commercial vehicles, savings on brake linings of 20% and 35% are reported when operating on flat and mountainous roads, respectively (148,153). Another advantage offered by retarders is the increased downhill speed of the vehicle that can be tolerated without overheating the primary brakes. Retarders also provide a braking system that is unlikely to lock the wheels (149,154). Some disadvantages are their initial cost, their large weight (148), and the fact that they can be used economically only on the powered unit; if used on trailers, they require an additional cooling system.

The exhaust brake is simple and inexpensive. Valves close off the exhaust manifold and the fuel supply, and the engine operates as a compressor (150-152,155-159). The brake torque generated depends on the gearing and engine speed. In general, at moderate and high velocities the primary braking system must be applied since the generated brake torque is only about 70% of the motor drive torque (156,158).

In the case of the motor brake, the timing of the camshaft is altered (154,160,161) such that the compressor action of the engine is increased. The engine brake torque may be over 100% of the maximum drive torque of the engine. Large retarding torques, however, can only be achieved by using a low gear, which in turn results in undesirably low cruising speeds. No effects upon engine wear have been observed (160).

Use of hydraulic retarders is obligatory on heavy commercial vehicles and buses in Germany. The kinetic energy of the vehicle is converted into thermal energy via the mechanism of viscous friction in the retarder. The heat is given off to the atmosphere by a radiator (130,148,149,154,160,162-164). Hydraulic retarders operate independently of engine, clutch, transmission, or electrical power supply. They are connected to the drive axle and represent an almost indestructible braking element when properly designed (160). When used on a trailer, a separate cooler becomes necessary (163,165). Skidding at the wheels is impossible since the retarding torque becomes zero when the retarder drive shaft does not rotate. When a retarder is installed in the powered unit, it prevents cooling of the engine below normal operating temperatures on long mountain grades by transferring the thermal energy generated through viscous damping in the retarder to the engine cooling system.

The eddy current retarder consists of a metal disk rotating in a magnetic field (67,160,166-168). The magnetic field can be altered to obtain a continuous variation in brake torque. It appears that eddy current retarders are simpler to construct than hydraulic retarders. Disadvantages, however, are large weight, dependency upon electric power, cooling fan losses, and cooling of the engine below normal operating temperatures (160).

2.4 ANALYSIS OF THE FRICTION BRAKE

2.4.1 MECHANICAL ANALYSIS.
The mechanical performance of a friction brake can be described by (among other things) the "brake factor" and "brake sensitivity" (6,9-11,169).

2.4.1.1 Brake Factor. The "brake factor" of a brake is defined as the ratio of the sum of all tangential forces acting on the friction surface, i.e., the drag on the drum, divided by the actuating force on a single shoe. A large number of publications have been addressed to a graphical (10,170) or analytical (9,169,171-178) determination of the brake factor. In these analyses it is usually assumed that brake drum, shoe, and shoe pivot are rigid and that the motion of the lining is constrained to the cylindrical shape of the drum. With these assumptions, the pressure distribution and forces acting on the brake can be determined (10,169,170,173,179). For brake shoes sliding on an abutment, a pressure distribution is assumed and the frictional forces are computed (169,172). Current design practice yields the following brake fac-

tors (75)(on assuming the coefficient of friction of the lining equal to 0.35):

disk brake	0.7 to 0.9
simplex brake (leading/trailing shoe)	2.0 to 2.8
duplex brake (two-leading shoe)	2.5 to 3.5
duo-servo brake	3.0 to 7.0

It should be noted that these analyses determine the brake factor as a function only of brake geometry and lining friction. They do not take into account the influence of velocity, mean pressure, temperature, wear, time, and drum or shoe distortion. Although drum distortions have been both computed analytically (180) and measured (181), the influence of distortion has yet to be incorporated into the brake factor analysis. These simplifications can lead to a disparity between the results of analysis and test.

Some improvement in the correlation of test results and theoretical predictions has been achieved by using experimental results in obtaining a functional relationship $[\mu_L = \mu_L(P, \theta, V)]$ between the lining friction coefficient μ_L, mean lining pressure P, temperature θ, and velocity V (110). For constant temperature and pressure this relationship can be expressed as

$$\mu_L = \mu_{oo} + (\mu_S - \mu_{oo})e^{-bV_{rel}} \qquad (2\text{-}10)$$

where (182,183)

V_{rel} = relative velocity
μ_{oo} = value of friction coefficient at high values of sliding velocity
μ_S = value of static coefficient of friction
b = a constant depending on characteristics of friction surfaces

Given the brake factor BF, the torque generated by a brake is given by (3):

$$T = BF \times A_{wc} \times p_h \times r \times \eta \qquad (2\text{-}11)$$

where

A_{wc} = wheel cylinder area
BF = brake factor
p_h = line pressure
r = effective drum or disk radius
η = mechanical efficiency

A similar relationship can be used to compute the brake torque for air-actuated brakes.

Limited attention has been given to the analysis of the dynamic behavior of brakes, due to the complexity of the problem and to the fact that a knowledge of the time response of the brake was relatively unimportant until the advent of antilock braking systems. A few investigations have been carried out in which the brake is treated as a dynamic system element (109,184).

2.4.1.2 Brake Sensitivity. Two definitions are given in the literature for "brake sensitivity." One expresses the change in brake factor with change in the friction coefficient of the lining, the other expresses the change of

brake <u>torque</u> with the friction coefficient of the lining. As early as 1949 (169) the brake sensitivity ε was defined as a change in brake factor divided by a change in the friction coefficient of the lining (6,15,171,175,185,186):

$$\frac{d(BF)}{d(\mu_L)} = \varepsilon \qquad (2-12)$$

For a design friction coefficient of μ_L = 0.35, and a change in μ_L of ± 0.05, the brake factor (and hence the brake torque) and the brake sensitivities ε vary approximately as follows (75):

	ΔBF	ε
disk brake	+14% to -14%	2
simplex brake	+26% to -21%	8
duplex brake	+36% to -28%	12
duo-servo brake	+51% to -33%	24

In the second definition, the brake sensitivity S is related to the brake torque T and the lining coefficient of friction μ_L, in the following manner (187-189):

$$\frac{dT}{T} = S \frac{d\mu_L}{\mu_L} \qquad (2-13)$$

Note that Equation 2-12 is the slope of the brake factor curve, whereas S should be regarded as the percentage change in brake torque for 1% change in μ_L. According to Equation 2-13. S = 1.0 for non-self-energizing disk brakes, and S > 1 for leading/trailing shoe, two-leading shoe, or duo-servo brakes. For two-trailing shoe brakes, S < 1.

Duo-servo drum brakes with a high self-energizing effect will always tend to exhibit a large sensitivity. This property may result in an undesirable directional response of the vehicle during braking (78,190). The term "brake fade" is also used to describe a change in brake factor during the braking process. In most cases, this phenomenon is recognized as a decrease in brake torque/line pressure gain (2,191,192). However, since the brake factor is a function only of the geometry and the friction coefficient of the lining, brake fade is also only a function of these two parameters. Geometry changes are caused by drum and shoe distortion due to thermal and mechanical loading. Changes in the friction coefficient derive from the dependency upon temperature (heat fading) (193), velocity (speed fading) (9,15,182,183), pressure, and the presence of water or oil at the friction interface (194-198).

2.4.2 THERMAL CAPACITY. The thermal energy produced during braking results from plastic deformations, elastic hysteresis losses, viscous drag, and phase changes in the materials. About 90% of the plastic deformation energy is converted into thermal energy. The rest is converted into decaying vibrations at other locations where internal damping produces thermal energy. A small remainder is converted into kinetic energy of the ambient air, resulting in noise (15).

As a result of the conversion of energy, a friction force is produced

between the sliding surfaces. The generation of friction force and thermal energy is directly related to the area of true contact (15,194,195,198). The work done during sliding consists of plastic work occurring during the deformation of the contact area, shear work, viscous drag work, and elastic nonrecoverable work (hysteresis losses). Deformations occur not only in the sliding direction but also laterally. Due to the large number of micro asperities in contact, the lateral forces integrate to zero; however, a corresponding deformation work (and hence thermal energy production) exists.

A sizeable body of literature is concerned with the thermal analysis of brakes. Since the torque production of a brake is related to the temperature at the friction surface (199,200), most theoretical investigations have been addressed to the determination of the average temperature rise expected during a single stop (179,200-217) or during continuous braking (148,160,166, 167,218). These analyses indicate that in the case of a single stop the friction surface should be as large as possible to reduce the temperature. However, in continued braking, heat capacity and convective heat-transfer are essential. Thus, the design parameters important for a single stop differ from those important to continued braking (216). The theoretical investigations also indicate that approximately 95% of the heat generated during a single stop is absorbed by the drum or disk, and 5% by the organic linings or pads. Sintered-iron linings, on the other hand, transfer a greater portion of the generated heat to the brake shoes or pad support as a result of the increased thermal conductivity of this friction material (201,210,215,216).

During a single stop, the temperatures achieved in both drum and disk brakes are approximately the same. During continued braking, however, the increased convective cooling capacity of the disk brake results in lower average temperatures (205,214-217). Some difficulties arise in determining the convective heat transfer coefficient of the drum or disk (217). Although the heat transfer from a rotating disk has been studied (219,220), the influence of a caliper located on the disk has not yet been incorporated into an analysis.

If a ventilated and solid disk are of equal weight, only a small temperature difference can be expected during the first few stops. In continued braking, the ventilated disk will tend to reach approximately 60% of the temperature attained by a solid disk (221,222).

The effects of radiation are neglected in most applications, since radiation contributes only about 5 to 10% to the heat transfer from the drum or disk (200). However, for brakes that attain high temperatures, as can be expected in applications using sintered-iron plug linings (216), thermal radiation may contribute substantially to the heat transfer. Dorner made a thermal analysis of brakes which included the heat transfer to the wheel cylinder and the bearings (217).

Brake dynamometer tests have shown that for continued braking the drum temperatures at the center of the friction path are initially 100° to 150°F higher than at the edges (223). At about 500°F to 600°F the temperatures in the center and at the edges of the friction path reached approximately the same value. Similar conditions were observed for single brake applications.

A dynamometer evaluation of three different drum alloys indicates that a cast iron drum achieved higher temperatures than bimetallic or chromium copper drums (224). Phase changes of the drum material into martensite indicated that the surface temperature (hot spot) reached 1300°F or higher. A comparison of rotor alloys for automotive disk brakes has resulted in similar findings (225).

Experimental results also indicate that localized interface temperatures are much higher than are determined with conventional thermocouples (224). Conventional thermocouples cannot respond to rapid changes in temperature, since their relatively large mass provides a sink for conducting heat away from the point at which temperature is measured. With a specially designed thermocouple having fast response, peak temperatures of 1705°F were measured on a disk brake pad (226).

If the actual contact region is assumed to be a square, the expected hot spot temperature rise can be computed as (195):

$$\Delta\theta = \frac{\mu WgV}{4.24\ell J} \cdot \frac{1}{K_1 + K_2}$$

(2-14)

where

2ℓ = length of one side

K_1 and K_2 = thermal conductivities of the moving and stationary friction surfaces, respectively (Fig. 11)

W = load

μ = coefficient of friction

V = sliding velocity

J = 778 ft-lb/Btu

Fazekas, on the other hand, has derived the following approximate expression for determining hot spot temperatures (212):

$$\theta \simeq \frac{2}{\sqrt{\Pi}} \cdot \frac{q}{\sqrt{V}} \cdot \frac{1}{\sqrt{kc\rho_d}} \cdot \frac{\sqrt{\beta a}}{P_n/P_y}$$

(2-15)

where

q = heat received by one hot spot

V = sliding velocity ⎫

k = conductivity   ⎪

c = specific heat   ⎬ of the rotor material

ρ_d = density   ⎪

a = thermal diffusivity ⎭

β = a factor between 2 and 6 relating hot spot size and sliding distance

P_n = nominal pressure on the friction surface

P_y = yield strength in compression of the softer material

High rates of heat generation have resulted in thermal cracks oriented at right angles to the friction path whereas low rates of heat generation have resulted in cracks oriented randomly over the drum or disk friction
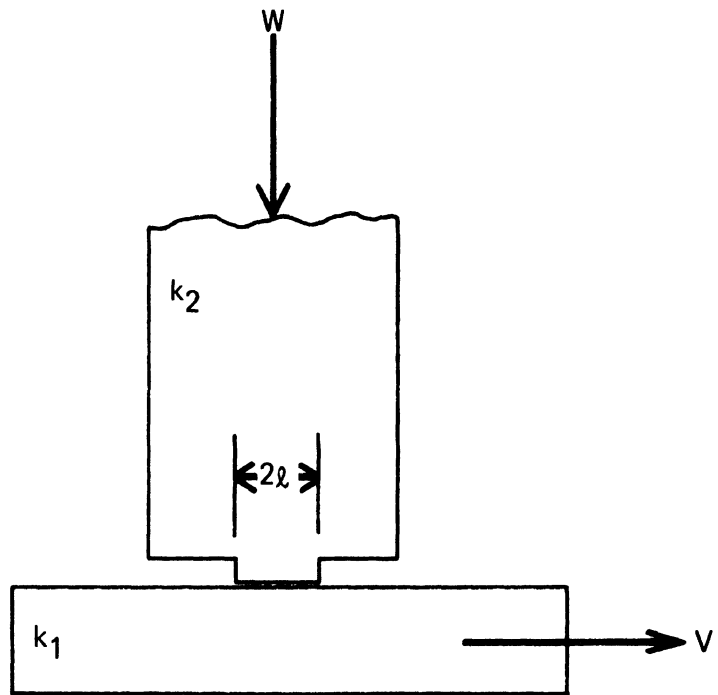
FIGURE 11. SLIDING CONTACT SCHEMATIC

surface (227-229). Thermal cracking occurs when the temperature gradient between the friction surface and the interior of the rotor is sufficiently large. This phenomenon is generally termed "heat checking" (10).

Surface cracking appears (227) to be closely related to:

(a) Initial hot spots on the braking surface
(b) Localized interference with heat transfer, caused, for example, by subsurface porosity
(c) Nonuniform temperature distribution
(d) Distortion of rotor during braking

Hot spot temperature ranges have been experimentally verified by examining the change in matrix structure of the drum or disk material (224,229,230). The heat flux entering the rotor during braking results in an unsteady, nonuniform distribution of temperature. For single short-duration stops, a large temperature gradient will initially exist over a relatively thin region of the material.

Thermal expansion in the friction interface is inhibited by the colder material surrounding the interface. Consequently, high compressive stresses result which, in the case of large temperature gradients, exceed the yield strength of the material, causing plastic deformation. After temperature equalization during the braking process or after cooling, the original length cannot be attained, resulting in residual tensile stresses within the generally highly loaded friction surface. Repeated brake application results in a continuous change between tensile and compressive stresses, or a "fatigue loading." Additional stresses can occur due to volume changes resulting from phase changes, internal oxidation, etc. The thermal fatigue eventually becomes visible in the form of heat cracks oriented at right angles to the brake lining path. These cracks are the result of a combination of material properties and loading conditions (227,228,230,231). Several investigators have computed the thermal stresses attained in cylinders or disks during heating or cooling (232-235), and more specifically in brake drums during braking (212, 236). In a low-heat-flux duration test (downhill braking or constant-velocity, low-energy-rate testing), randomly oriented fine hairline cracks are observed. These cracks, described as crazing, are similar to patterns formed on alternately heated and cooled bars (227). The differences observed in the thermal crack patterns suggest taking a cautious approach in replacing brake snub testing by continuous horsepower tests. In general, heat checking is less likely to occur in materials having:

(a) higher yield strength at elevated temperatures
(b) smaller E-modulus
(c) smaller thermal expansion coefficient
(d) greater thermal conductivity
(e) greater composition stability (i.e., no carbide decay)
(f) less tendency to internal oxidation
(g) increased phase change temperatures
(h) smaller wall thickness of the rotor (228,237)

The effects of drum or disk surface condition upon the deceleration capability have been investigated (238). Cracked and hot-spotted drum or disk

surfaces decreased the stopping ability of two vehicles whereas the stopping ability of another vehicle was increased. Scored rotor surfaces decreased the deceleration capability of all vehicles tested.

2.4.3 PERFORMANCE COMPARISONS. Drum brakes can be grouped according to (a) the configuration of the brake shoe—i.e., duo-servo, duplex (two leading shoes), and simplex (one leading-one trailing shoe); (b) the manner in which brake torque is transmitted to the back plate by pivot or sliding (parallel or inclined) abutments; and (c) the manner in which the brakes are actuated (Fig. 12).

Duo-servo brakes show the highest brake factor and hence the highest brake torque/line pressure gain (169,221). However, the high sensitivity is a disadvantage. A vehicle equipped with high gain duo-servo brakes may easily undergo a change in brake force distribution front-to-rear or side-to-side during the braking process (180). Also, the lining on the secondary shoe wears more rapidly than on the primary shoe. This latter problem is often resolved by using different friction materials on the primary and secondary shoes (10).

Duplex brakes exhibit a moderate brake factor as well as a moderate brake sensitivity. Their main disadvantage lies in the fact that if the direction of rotation is reversed the brake factor decreases by as much as 70% due to a change from a two-leading shoe to a two-trailing shoe configuration (11,221).

Simplex brakes exhibit the lowest brake factor and sensitivity. Since approximately 70% of the brake torque is generated on the leading shoe, the lining of the leading shoe will wear more rapidly than the trailing shoe.

The wear along a lining should ideally be uniform (169). A shoe held by an abutment will wear more uniformly than a pivoted shoe. An inclined abutment will result in a more uniform wear than a parallel abutment (10,15).

The wear life theoretically predicted for S-cam (239) and wedge-actuated drum brakes (179) agrees with experimental results. The wedge brake appears to cause lesser drum wear than occurs with the S-cam brake (240,241).

Experiments with disk brakes on commercial vehicles were carried out in Europe as early as 1957 (242,243). Measurements of braking performance achieved by trucks equipped with disk brakes were published in the United States in 1969 (129). The measured decelerations, 26 to 32 ft/sec, have been exceptionally high compared with decelerations (19 to 20 ft/sec) achieved with vehicles equipped with drum brakes. The constant sensitivity resulting from the linear relationship between the brake factor and the friction coefficient of the lining is one of the main advantages of disk brakes over self-energizing drum brakes. For a constant friction coefficient, the brake factor is little affected by thermal expansion of the caliper or disk. In the case of drum brakes, however, the effective drum radius will increase more than the radius of the brake shoe during severe braking due to the smaller thermal conductivity of the brake linings. This phenomenon results in a change in the pressure distribution between lining and drum, and can cause a reduction in the brake factor up to 20%. During the cooling period, the drum will attain lower temperatures than the brake shoe, causing the effective drum
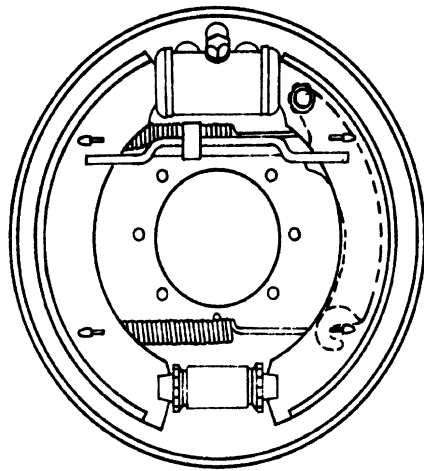
Simplex Brake
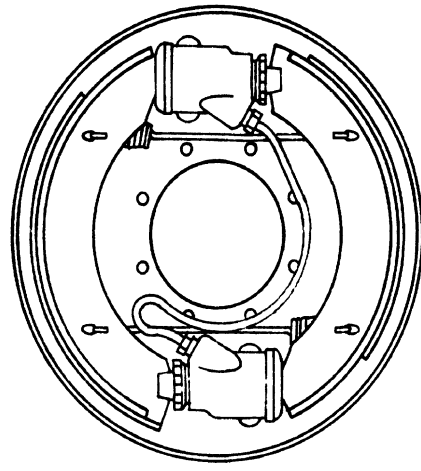
Duplex Brake
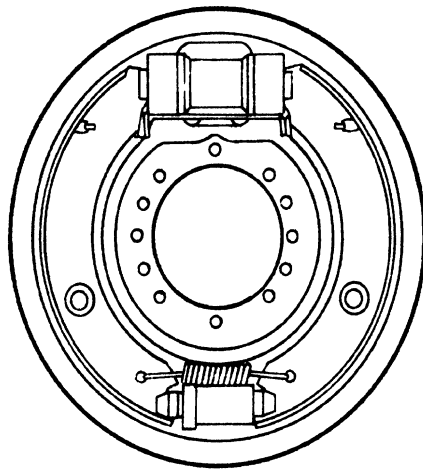
Duo-Servo Brake
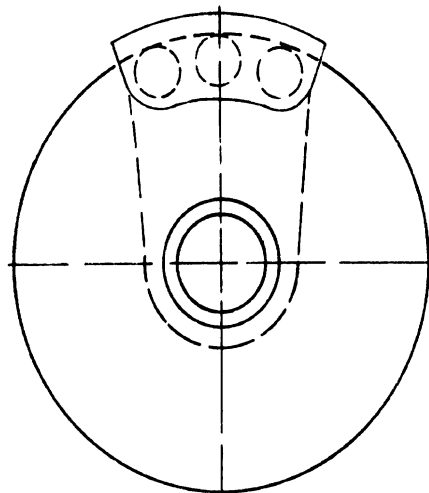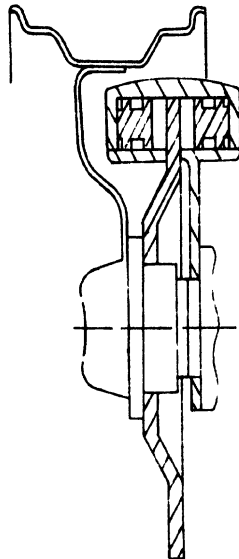
Disc Brake

FIGURE 12. AUTOMOTIVE BRAKES

radius to be smaller. A high pressure between lining and drum may occur at both ends of the lining and cause an increase (up to 40%) in the design brake factor for normal operating conditions (6,244).

The greatest disadvantage of disk brakes is their low brake factor. On the average, the brake factor of disk brakes is only about 25% of that for a two-leading shoe drum brake (6,8,185). This deficiency can be resolved, however, by use of power assist.

The pedal travel required to sustain a certain deceleration will generally be greater for drum brakes than disk brakes, due to greater thermal expansion of the drum (6). An expansion of the disk in the transverse direction resulting from high temperatures does not cause a loss in pedal travel (202). However, experiments indicate that under certain conditions pedal travel may increase less for drum brakes than for disk brakes, due to different lining stiffness characteristics. Mueller reports that a soft pad material may result in a considerable pedal travel increase (191).

Self-energizing disk brakes of the ball and ramp type were used in 1939 in heavy military vehicles in Germany (244). Depending on the degree of self-energizing, such brakes are more or less sensitive to changes in lining friction. Thus, one of the main advantages of disk brakes, namely low and constant sensitivity, is lost (169,201,245).

2.5 SYSTEM DESIGN CONSIDERATIONS

2.5.1 HYDRAULIC BRAKING SYSTEMS. The important parameters describing the braking performance of a two-axle vehicle have been given in Equation 2-2, repeated below for convenience:

$$a_x = 0.5 \cdot \frac{g}{W} \cdot F_p \cdot \frac{r}{R} \cdot BF \cdot \rho \cdot \eta \qquad (2\text{-}2)$$

It is clear that the parameters r/R, BF, and ρ must be made as large as possible (6,49) in order that a vehicle with a weight W achieve high decelerations with small or moderate pedal forces. However, the ratio r/R is limited by the size of the tire and rim. The trend to small travel of the foot pedal and the requirement for providing a minimum travel of the brake shoe tends to decrease ρ. The only alternative remaining is to increase the brake factor. This goal can be accomplished by either increasing the friction coefficient of the lining, or using two-leading shoe brakes, or duo-servo brakes, all of which steps increase the braking performance, but at the sacrifice of stability. With the introduction of disk brakes having a lower brake factor ($BF = 2_{\mu L}$) the total gain ρ had to be increased by a factor of about four in order to achieve a satisfactory relationship between deceleration and pedal force (6).

On considering the brake shoe displacement required to cover lining compression, lining wear, and drum or caliper expansion, a lower limit on shoe actuation can be established (6,11,185). Since the pedal travel is limited by ergonomic considerations, a weight restriction on vehicles with unassisted brake systems can be derived (246,247).

2.5.2 PNEUMATIC BRAKE SYSTEMS. It is common practice in the U.S. to

use a K-factor to characterize the retarding force of each brake instead of
the brake factor defined earlier. The K-factor is defined as the ratio of
retarding force to rated axle load, as achieved at a reservoir pressure of
60 psi (248-250). Use of a K-factor of 0.6 constitutes common design prac-
tice in the U.S. In some cases the K-factor has been raised to 0.7 in order
to obtain acceptable performance on city buses (249).

Studies have shown that it is desirable that the brakes on the rear ax-
le of the trailer be applied first in order to avoid large kingpin or hitch
forces (52,250,251). With an increase in the length of combination vehicles,
dynamic considerations arise which usually are neglected in the analysis of
shorter vehicle combinations (14,69). A detailed investigation of the dy-
namic behavior of air brake systems indicates (119) that the time required
to overcome clearance between the brake shoes and drum becomes greater with
increased line pressure and decreased brake chamber piston travel; the time
required to build up the brake torque also decreases with increasing line
pressure, increasing reservoir pressure, decreased piston travel, and de-
creased line length. For example, increasing the line length between the
brake valve and the brake chamber from 6.5 to 35 ft increases the application
time only a little, while the buildup time is nearly doubled (124). The ap-
plication time is defined as the time elapsed between the instant of first
foot-pedal movement and the instant the brake shoes are put against the drum.
The buildup time is defined as the time elapsed between the instant the brake
shoes are contacting the drum and the instant maximum deceleration is obtained.
The optimum result should therefore be obtained with minimum volume and max-
imum line pressure. Experiments have also shown that considerable time de-
lays are associated with the control and flow processes in the brake valve
(124).

Experiments with scaled physical models (69,70) indicate that braking
lags may be considered as composed of three parts, each influenced by dif-
ferent factors. In the first part, a time lag derives from the speed with
which the pressure wave travels through a brake line of given length. The
second part of the time lag derives from the motion of brake chamber pistons
required to overcome slack. This portion of the time lag is proportional to
the volume of the brake chambers The third part of the lag consists of the
time required for the line pressure to reach 90% of the reservoir pressure.
This lag is proportional to both the total volume and the flow resistance of
the brake system (69). In a related study (12), a dynamic analysis and simu-
lation was performed for pneumatic braking systems on railways. The results
indicate behavior similar to that experienced with the brake systems on motor
trucks.

2.5.3 COMPONENT DESIGN CONSIDERATIONS. Factors considered important in
the design of a friction brake are: force levels on specific parts of the
brake, operating temperatures, convective heat transfer characteristics, ef-
fect of thermal deformation on brake chamber piston travel, and shielding
against dirt and water contamination (175,227). The maximum temperature al-
lowable in a brake is limited by the structural integrity of the lining mater-
ial (171,206), while high temperature gradients result in large thermal

stresses in the drum or disk (212,236). Expressions for calculating the required friction area for a specified temperature rise on the interface have been derived. For a well ventilated drum and a surface temperature increase of $(T_s - T_i)$ the minimum rubbing path area of the rear drum (ft^2) is given by (210):

$$A_{min} = \frac{0.296 \cdot \sigma \cdot (1 - \phi) \cdot W}{(T_s - T_i)} \qquad (2-16)$$

where

T_s = temperature of friction surface, °F

T_i = initial brake temperature, °F

σ = heat absorbed by the drum divided by total thermal braking energy

W = vehicle weight, lb

ϕ = brake force distribution = brake force of the rear axle divided by total brake force

However, since the temperature is a dependent variable, depending upon kinetic energy, brake geometry, etc., a method has been developed by which it becomes possible to determine the required friction area from the independent variables such as vehicle speed and weight (206). In this method, the brake rating is calculated in terms of either the horsepower per unit area of brake lining or the swept surface area. According to Newcomb, typical power ratings for drum brakes based on the lining area are 0.5 hp/in.^{2} for normal operation and 0.25 hp/in.^{2} for heavy duty operation (206). In 1956, Girling found that if a brake is designed with an upper limit of 2.5 hp absorption per square inch of lining, fade will usually not be a problem (252). For disk brakes, power ratings range from 3.5 to 10 hp/in.^{2} (210,253,254).

The horsepower rating method is equivalent to allowing a limiting heat-flux at the interface of drum and lining. The interface temperature, which is related to the integrity of the lining material, and the temperature gradient, which is related to thermal stresses, are not determined by the heat-flux alone. They are determined by a complete solution of the thermal problem with its corresponding boundary conditions (255). The horsepower rating method can give useful results in terms of the required swept area. The choice of rating values, however, is not well defined, especially for repeated braking (206).

Design alterations, such as ventilation openings in the wheels, usually do not result in appreciable reduction in temperature levels (256). Experiments with dust shields have shown that at low speeds the cooling rates differ little for shielded and unshielded disks. At higher velocities, however, the cooling rates were increased by as much as 30% by elimination of the dust shield.

The wear of a brake lining can be related to the mean mechanical pressure between lining and drum. German regulations specify that the product of mean pressure and lining friction coefficient is not to exceed 64 lb/in.^{2} (257). Methods for predicting wear life in good agreement with experimental results have been developed (258). It is important to realize that the value

of the friction coefficient between brake lining and drum or disk is a function of $\underline{both}$ friction partners, and not of the lining alone (247). Consequently, an improvement in brake performance, stability, wear, life, etc., has to take the lining as well as the drum material into consideration. Several comparative studies of different alloys for automotive brake drums have been carried out (228,259), as well as investigations on the failure of cast iron drums (229).

Also of importance to the design of braking systems are the performance characteristics of hydraulic brake fluids. They are affected by the boiling point, water avidity, freezing point, viscosity, and corrosive action on rubber parts (260-265). For air brake systems it is necessary to consider the possible freezing of the water condensate in the system when operating at low ambient temperatures.

2.6 BRAKING TEST PROCEDURES

The deceleration capability of a vehicle may be determined indirectly by measuring initial velocity and stopping distance, or directly through measuring the actual deceleration during braking (18,185,266-268).

2.6.1 MEASUREMENT OF MEAN DECELERATION. The mean deceleration, as determined from velocity and stopping distance, yields a direct evaluation of the braking process with respect to the shortest stopping distance. Deceleration as a function of time can be plotted as shown in Fig. 13 (16,53,266,267, 269,270). Mean deceleration $a_{x(av)}$ can be computed from the initial velocity V_i (mph), the maximum deceleration $a_{x(max)}$ (ft/sec^2), the application time t_a (sec), and the buildup time t_b (sec) by the expression (185,269):

$$a_{x(av)}(ft/sec^2) = \frac{a_{x(max)}}{1 + \dfrac{1.365\, a_{x(max)}}{V_i}(t_a + t_b/2)} \cdot \qquad (2\text{-}17)$$

The stopping distance may be measured experimentally by (a) integration of the velocity signal from a fifth wheel, (b) a revolution counter attached to the fifth wheel, or (c) direct distance measurement employing an explosively-fired chalk pellet. The time lag associated with the latter method has to be considered in determining the actual stopping distance of the vehicle (271).

The stopping distance is greatly affected by the driver and is difficult to measure accurately at all speeds. It seems advisable to evaluate the performance of a vehicular brake system in terms of the retarding force actually produced as measured by the deceleration of the vehicle (51).

2.6.2 INSTANTANEOUS DECELERATION. An instantaneous value of deceleration can be obtained either by differentiating the velocity signal from a fifth wheel or by direct measurement with a decelerometer. Decelerometers mounted in the vehicle are sensitive to vibrations, pitch motion, and road inclination, unless a stabilized platform is provided. The application and buildup time as well as the maximum deceleration can be obtained from actual time histories (267). Consequently, this test method also allows the evalua-
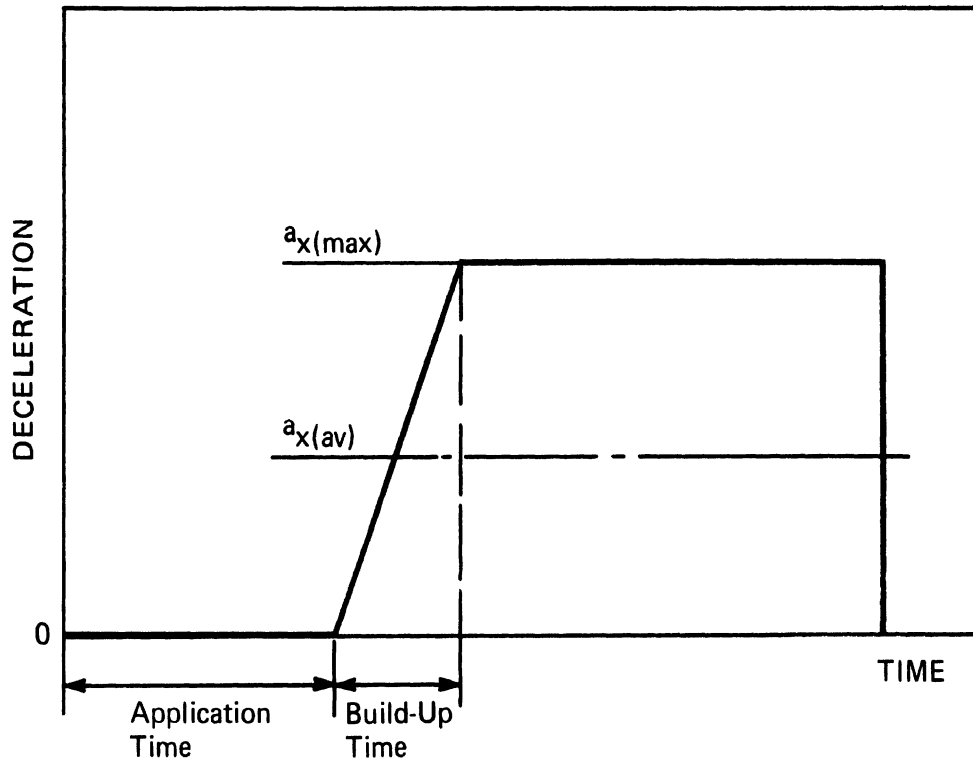
FIGURE 13. SIMPLIFIED DECELERATION DIAGRAM

tion of the braking process with respect to the shortest stopping distance.

2.6.3 TEST PROCEDURES AND VEHICLE USE. Test procedures should be directed at determining if the stopping capability of the vehicle as determined by brake torque levels, brake response, and brake force distribution designed into the vehicle is satisfactory when the vehicle is subjected to the most severe duty cycles encountered in normal service. Brake usage on commercial vehicles has been studied by several investigators (10,258,272,273). European studies on an Alpine route indicate that vehicles experience maximum decelerations as high as 0.14 g, but that heavy vehicles descend the grades at nearly uniform velocity (273). For this route the average work done by the brakes was 41% of the total energy dissipated with the remainder of the work done by engine retardation. The results of brake usage investigations on heavy trucks seem to correlate well with earlier results obtained for passenger cars (119,268). Other investigations show that on flat roads only a few decelerations exceed 0.35 g. A public service vehicle, for example, rarely brakes in excess of 0.25 g in city traffic, but experiences a large number of decelerations below 0.2 g (258). For passenger cars, it appears that the deceleration during routine driving depends on the driver and the top speed performance of the car (119). For commercial vehicles the three main controlling variables on the brake usage seem to be engine horsepower output, driver and route type, and energy transfer, a factor determined experimentally on the vehicle (258).

2.6.4 PREPARATION OF VEHICLES FOR TESTING. Before testing a vehicle, the brake system including tires should be brought to a mechanical level corresponding to the manufacturer's specifications. Even small factors such as balancing shoe return springs may have a pronounced effect when braking at relatively low line pressures on slippery road surfaces (274-279).

The following is a brief description of the more important maintenance tasks to be carried out before testing:

The brakes should be relined with the lining material specified by the vehicle or brake manufacturer, and complete sets of brake blocks should be installed even when one lining does not show any apparent wear (274). The brake drum surface should be smooth and concentric. If the drum is scarred or worn unevenly, it should be reconditioned by reboring the friction surface. Shoe return springs should be balanced to insure specific distribution of braking forces at low line pressures. Brake chambers and wheel cylinders should be in good mechanical condition in order to guarantee good brake balancing. On air-braked vehicles, worn or loose slack adjusters should be replaced since they may unfavorably affect time delay and force transmission (274). With all brake components in good mechanical condition, a carefully conducted brake balancing test should be carried out. It will reveal if the individual axles produce the brake force levels specified by the manufacturers (274, 280,281).

2.6.5 ROAD TESTING. Detailed road test schedules are usually arranged so as to test the brakes according to their expected usage. Depending on the experimental data desired, instrumentation required can be minimal or fairly extensive (10,272,282-286).

Test and rating procedures in current usage in the U. S. were established by SAE together with the automotive industry during tests conducted in 1960 and 1967 (287,288). For these tests vehicles are instrumented to determine deceleration, velocity, stopping distance, line pressure, stopping time, and brake temperature. After burnishing the lining according to outlined procedures, the fully loaded vehicles are subjected to a series of effectiveness, fade, and recovery tests. Performance requirements of SAE Recommended Practice J992a (289) for the effectiveness tests are as follows:

(1) Maximum stopping distance from 20 mph:
    (a) 25 ft, light trucks, 10,000 lb, GVW or less
    (b) 35 ft, truck and bus, over 10,000 GVW
    (c) 45 ft, vehicle combinations

(2) Deceleration requirements:
    (a) 18 fpsps from 60 mph, 6000 GVW or less
    (b) 15 fpsps from 60 mph, 6000-10,000 GVW
    (c) 12 fpsps from 50 mph, truck and bus over 10,000 lb GVW and vehicle combinations

(3) Directional stability must be such as to remain in a 12-ft-wide lane during braking applications throughout all phases of the test procedures.

(4) Pedal force must not exceed 200 lb and maximum air pressure must not exceed vehicle manufacturer's cutout pressure on any stop.

(5) A final inspection showing no visual evidence of permanent deficiency in functional or structural integrity of the part is required.

In 1968 Nelson and Fitch investigated the directional stability of longer truck combinations during braking (67). They studied stopping distance performance and vehicle stability on wet and dry road surfaces for a variety of loading conditions with double and triple combinations. The test results have shown that longer combinations can achieve braking and stability performance comparable to shorter combinations, provided the braking system is maintained properly.

In 1969 the Virginia Twin Trailer Study Commission sponsored an investigation in which the operational characteristics of a maximum size tractor-semitrailer combination allowed on Virginia highways at that time was compared with those of a tractor-semitrailer with full trailer combination (290). Stopping distance tests of the two combinations were part of the test requirements. The vehicles were tested in the fully loaded (70,000 lb) and empty driving conditions on both wet and dry pavements. The test results indicate that the vehicles performed equally well in stopping ability as well as in stability.

To determine the horsepower rating of brakes, SAE J880 specifies a test in which the vehicle is braked in a series of snubs from an initial velocity of 50 mph and a final velocity of 15 mph. The horsepower rating of the braking system (BRHP) is calculated from the experimental data by the following equation:

$$BRHP = 1.01 \times 10^{-4}W \times N(1 - \frac{t_1}{t_2})$$ (2-18)

where

      BRHP = brake rating horsepower

      W   = GVW of the vehicle (lb)

      N   = number of snubs in 720 sec

      t_1 = calculated time (sec) that brakes are applied in a single
              snub at the velocity and deceleration employed in the test

      t_2 = average drift time (sec) when decelerating from 50 to 15 mph

The test results of 1960 were plotted in terms of brake rating horsepower as
a function of vehicle weight. Curve fitting procedures resulted in a straight
line relationship given by

$$\text{"required"} \; BRHP = 12 + \frac{1.4 \; GVW}{1000}$$ (2-19)

Except for some very extreme cases, this equation permitted all vehicles
participating in the 1960 tests to pass the rating procedures (287).

The results obtained from brake testing on the vehicle clearly indicate
whether or not the braking system performance is satisfactory. In the case
of fair or poor performance, however, it is not always possible to point out
the reasons for poor performance from the experimental data alone (291).
These aspects of brake testing will be clearly revealed in single component
testing. Also, no specific information pertinent to lining and drum material
development can be obtained since the experimental results are influenced by
driver, tire, roadway, relative humidity, etc. The tire-roadway friction co-
efficient is, among other things, both speed and load dependent. Consequent-
ly, a braking system designed for adequate performance may be classified as
poor due only to uncontrolled external circumstances (174,247).

    2.6.6 LABORATORY TESTING. The testing of a brake on a stationary dy-
namometer has several advantages. Test conditions are much more easily con-
trolled and are repeatable, and velocity, pressure, temperature, torque, etc.,
can be accurately measured (10,247,292). Care must be taken in designing the
dynamometer such that the test conditions correspond to the intended brake
usage. Several different design concepts for dynamometers have been utilized
(9,10,169,202).

For the type of testing required for vehicle inspection or diagnosis of
brake system defects, two main types of stationary dynamometers are in use
(10,292,293): the chassis dynamometer employing rollers, and the platform
type. On the chassis dynamometer, the vehicle is anchored and the braked
wheels are driven by rollers. Published data indicate that test results
agree well with actual road test results (6). Using the platform type, the
vehicle is driven on to the platform, then braked sharply, with the reaction
force of each wheel on the platform measured. However, it has been reported
that test results from the platform type are inaccurate (6). For brake and
brake lining testing, the inertia type dynamometer is used (169,292,293).
Test machines and procedures for testing only a segment of the brake lining

have been developed (169,247,294-299). A test termed Friction Assessment Screening Test (FAST) has been in use since 1967 for all Ford passenger cars to specify quality controls on brake linings. According to one source (300), results obtained from the FAST machine correlate well with vehicle performance tests. In order to obtain consistent results, extreme care must be taken in controlling and measuring the important parameters. This is especially true for complete brake lining tests (169).

2.7 PERFORMANCE STANDARDS

2.7.1 BRAKE LEGISLATION. Several agencies are concerned with regulating brake system performance. Before 1955 the Interstate Commerce Commission (ICC) was the Federal agency with jurisdiction over commercial vehicles operating in interstate traffic. Since then this task has been given to the Bureau of Motor Carrier Safety (BMCS). The National Committee on Uniform Traffic Laws and Ordinances (Uniform Vehicle Code, UVC), has developed braking performance regulations which have been adopted by several states. In addition, every state has published its own specific brake regulations. The National Highway Users Conference, Inc., annually publishes the "Motor Vehicle Law Reporting Service." The American National Standards Institute (ANSI) publishes recommended standards for the inspections of motor vehicles. In 1965, 21 states required periodic motor vehicle inspection modeled after the ANSI recommendations. The National Education Association has developed brake performance requirements for school buses published in "Minimum Standards for School Buses" (301-302), which standards have been adopted by several states.

2.7.2 U.S. STANDARDS. It has been argued that brake regulations should be devised so as not to hinder technical development but to prevent unsafe brake systems from coming into use by specifying performance requirements rather than design specifications (301,303). Goepfrich gives a typical example of a design specification wherein lining area is related to gross vehicle weight, which specification would preclude the use of disk brakes (301).

The Society of Automotive Engineers in cooperation with the industry has developed and published recommended procedures for testing braking systems on the vehicle as well as components on stationary test machines (304). Experimental and theoretical work leading to recommended braking performance is widely discussed in the literature (287,305-315).

At present, performance requirements for the service brake are based on a speed of 20 mph for stopping distance and 50 mph for deceleration capability (316). Manufacturers usually test their vehicles at 60 mph, however (301). Performance measures commonly used are stopping distance, mean deceleration, and pedal effort under normal and fade conditions (267,301). The effect of current and proposed brake regulations on hydraulic and air brake systems have been discussed by Goepfrich (301) and Andrews (306). Design changes and innovations such as the wedge brake have been introduced by the industry in order to meet or surpass requirements imposed by brake legislation (306).

Minimum performance requirements for emergency brakes have been established but differ from state to state. For motor vehicles under 10,000 lb GVW,

a dual circuit brake system is obligatory. For medium and heavy commercial vehicles, a standard split of the braking system in front and rear circuit might not be a good practice due to the low thermal capacity of the front brakes (301,317).

Requirements for improved performance of the emergency brake brought about design changes of the actuators and control devices (306). The performance of the brake system under emergency conditions for school buses has been upgraded over those of commercial vehicles (307). Brake regulations published by the Bureau of Motor Carrier Safety as well as those of certain states require motor vehicles to be equipped with parking brakes adequate to hold the loaded vehicle on any public road free of snow and ice (306). Developments during the last years indicate that the parking brake will be upgraded in order to hold vehicle units on 16 to 20% grades as already required by California and the National Education Association minimum standards. Proposed Federal motor vehicle safety standards (318) require a stopping distance of 216 ft from a speed of 60 mph on a road surface with a skid number of 75. For wet road surfaces with a skid number of 30, the proposed stopping distance is 436 ft.

2.7.3 EUROPEAN STANDARDS. Several different agencies in Europe have developed brake standards (130,191,319). Since the German draft code established in 1964 (257) serves as a pattern to member countries of the Economic Commission of Europe (Common Market Nations) it will be discussed in detail. Some major requirements are:
1. Service brakes must have a deceleration capability of 0.45 g.
2. Emergency brakes must have a deceleration capability of 0.25 g.
3. Parking brakes must hold loaded tractors and buses on a 25% slope, trailers on a 20% slope.
4. Tractor/trailer combinations must have a third brake or retarder.
5. Tractor/trailer combinations must have a load-sensitive brake proportioning system.
6. Buses over 12,125 lb (GVW) are required to have a retarder.
7. Trailers must have a two-line brake system.
8. Split brake systems are required.

The fade tests differ from those conducted in the U.S. (287) in the method of heating the brakes. In Germany, a deceleration for the purpose of heating the brakes is computed from the vehicle weight (GVW) and the maximum engine horsepower. With a pedal force applied which would result in the computed deceleration the vehicle is driven over a distance of 1.06 miles at a speed of 25 mph. After heating, the vehicle is accelerated to 31 mph as fast as possible. The ensuing hot brake performance has to be within 60% of the performance under cold conditions (191,320).

In general, performance requirements established by other European countries are lower than those drafted by Germany (130). One exception is England, which requires a 0.5 g deceleration capability of the service brake, 0.25 g from a backup system, and a parking brake that will hold on a 16% grade. The English Code of Practice, established jointly by the Ministry of Transport and the motor industry recommends a performance of 0.6 g from the service brakes,

0.3 g from the backup system, and a parking brake that will hold the vehicle on a 19% grade. The brake regulations and operational conditions for Europe led to the design and implementation of heavy duty disk and drum brakes, hydraulic retarders, proportioning devices, and automatic slack adjusters (130, 319,321-324).

3. TEST PROGRAM

3.1 SCOPE OF THE PROGRAM

In order to determine the braking performance capabilities of vehicles equipped with standard braking systems, three integral trucks, three buses, and four tractor-trailer combinations were subjected to a series of effectiveness, fade and recovery, and braking rating tests. So that the improvement in performance through use of more effective brakes and advanced braking systems could be determined, three additional vehicles were tested:

(1) An integral truck equipped with disk brakes and a full power hydraulic brake actuation system.

(2) A tractor-trailer equipped with proportioning valves, adaptive braking system, and trailer brake synchronization device.

(3) A tractor-trailer equipped with a wheel antilock system.

The disk brake truck was subjected to effectiveness, fade and recovery, and brake rating tests, while the two tractor-trailer vehicles were tested primarily for effectiveness and minimum stopping distance capability.

Testing reported herein was performed at the Bendix Automotive Development Center (BADC) during the period September 1969 to October 1970.

3.2 SELECTION OF VEHICLES

The procedure used for selecting specific test vehicles was necessarily pragmatic because vehicles had to be leased or borrowed, and thus choice was somewhat restricted. It was decided, therefore, to analyze the braking systems of available vehicles, and compare their expected performance with performance of vehicles in the same class. This comparison resulted in surveying manufacturers' published data on 246 trucks and 218 tractors and the formulation of design measures by means of which expected performance could be estimated from the available data. The sample surveyed included vehicles from the following manufacturers: GMC, Dodge, Chevrolet, International Harvester, White, Diamond-Reo, Ford, and Mack. Vehicles surveyed had a variety of braking systems: hydraulic, with and without vacuum assist, and air brakes. Configurations included duo-servo, wedge, S-cam, twin-plex, and stopmaster.

Three measures were used for comparing the brakes of the different vehicles:

$$q_D = \text{heat flux into drum, } Btu/ft^2\text{-sec}$$

$$q_L = \text{horsepower per unit area of lining, } hp/ft^2$$

$$\mu P_m = \text{mean pressure x lining friction coefficient, } lb/in^2$$

These measures indicate the capability of the braking system to convert mechanical energy into thermal energy (q_L), act as a dissipation element for

thermal energy (q_D), and provide wear resistance (μP_m).* Distribution plots of these measures for the vehicles surveyed are given in Figs. 14 and 15. Note that low values of these arbitrarily selected measures reflect design practice which would be expected to yield higher levels of energy absorption-performance.

These criteria could not by themselves form the basis of selection. Other factors had to be considered. In selecting the integral trucks (see Table 1 and Fig. 16) other factors considered included truck size, type of brake, means of actuation, loaded/unloaded weight ratio, braking efficiency, and modes of brake system failure. As can be seem from Table 1, the C30, Loadstar 1700, and the J70 represent the small, medium, and heavy truck categories with loaded to unloaded ratios of 1.6, 4.1, and 2.8, respectively. Various brake types are represented, as are types of actuation and failure modes.

In the case of buses (see Table 2 and Fig. 17) factors of significance in selection were type of service, availability, type of brake system failure protection, and the degree to which a given bus was typical of vehicles in service. The school bus was selected because its size was representative and the vehicle was equipped with air brakes meeting National Education Association standards for braking performance and brake system failure protection. The following combinations were deemed to be most representative:

(1) Two-axle tractor (one front, one rear) with 35-ft, van-type, single-axle semitrailer

(2) Two-axle tractor with 40-ft, van-type, tandem axle semitrailer

(3) Three-axle tractor (one front, tandem in rear) with 40-ft, van-type, tandem-axle semitrailer

(4) Two-axle tractor with two 27-ft, van-type, single-axle semitrailers joined with a dolly (27-ft doubles)

Combination vehicles selected are shown in Figs. 18 and 19, with brake type and loading information given in Table 3.

Use of the above defined design measures, together with the cited additional considerations led to a selection of ten vehicles which are located in the distribution of design measures as shown in Figs. 14 and 15, and which indeed demonstrated a wide range of performance capabilities.

In Figs. 19 and 20 are shown the vehicles tested with advanced braking systems. Specifications, brake and loading data are given in Table 4. These vehicles and systems were selected directly by the National Highway Traffic Safety Administration for testing in this program. For complete specifications on each of the test vehicles, including loading diagrams and braking system schematics, see Appendix A.

3.3 TEST REQUIREMENTS

3.3.1 PREPARATION OF VEHICLES. Since most of the vehicles selected for testing had previously been used in various types of service, it was deemed necessary to take the following steps to insure that the braking system of

---

*Expressions used to calculate these measures are given in Appendix F.

(A)

Vehicles (%)

30

20

10

0

City Bus — Intercity Bus — Medium Truck — Light Truck — Heavy Truck — School Bus

0   100   200   300   400 q_D Btu/ft^2-sec)

(A) Energy Dissipation Measure, Heat Flux into Drum

(B)

Vehicles (%)

30

20

10

0

Medium Truck — City Bus — Intercity Bus — Light Truck — School Bus — Heavy Truck

0   200   400   600   800 $(q_L$ hp/ft$^2)$

(B) Energy Conversion Measure, hp into Lining

(C)

Vehicles (%)

30

20

10

0

Medium Truck — City Bus — Intercity Bus — Light Truck — School Bus — Heavy Truck

0   100   200   300   400 $(\mu p_M$ lb./in.$^2)$

(C) Wear Measure

FIGURE 14. BUS AND TRUCK SURVEY

49
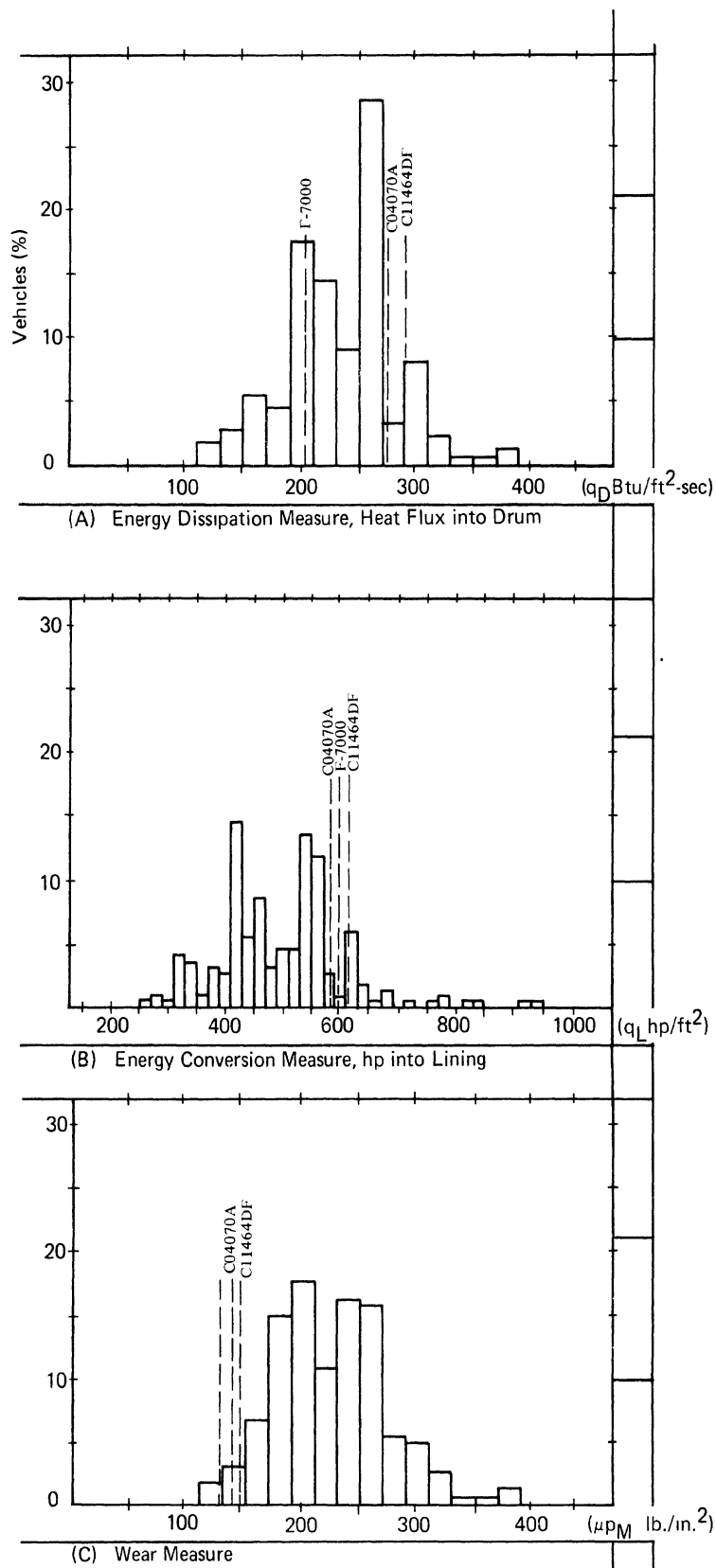
FIGURE 15. TRACTOR SURVEY

(A) Light truck.

(B) Medium truck.

(C) Heavy truck.

FIGURE 16. TEST VEHICLES, INTEGRAL TRUCKS

(A) School bus.

(B) Intercity bus.

(C) City bus.

FIGURE 17. TEST VEHICLES, BUSES

(A) 2-S1.
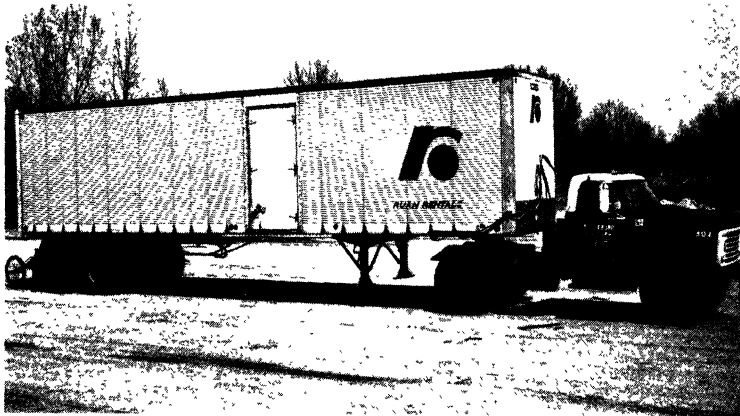
(B) 2-S2.

(C) 3-S2.

FIGURE 18. TEST VEHICLES, TRACTOR-TRAILERS

(A) Doubles combination.
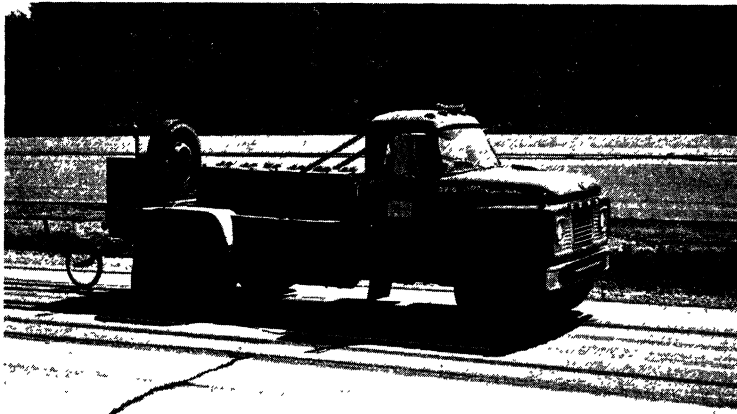
(B) Disc brake truck.

(C) Vehicle 14.

FIGURE 19. TEST VEHICLES

(A) Loaded tractor.
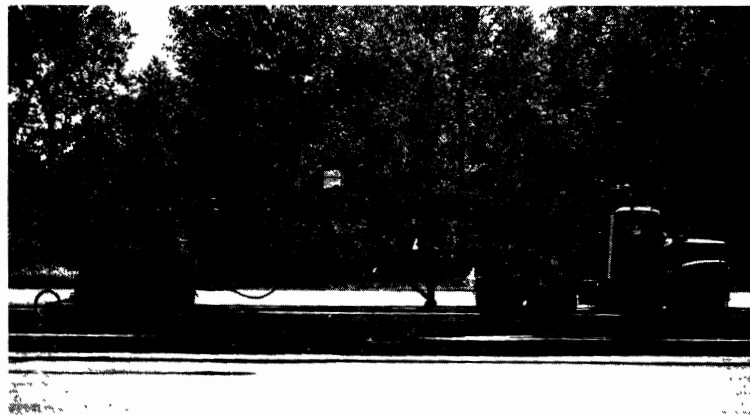
(B) Tractor-trailer with high load center of gravity.

(C) Tractor-trailer with low load center of gravity.

FIGURE 20. TEST VEHICLE 12

TABLE 1

INTEGRAL TRUCKS

Vehicle Manufacturer	Model	Test Weights, lb		Axle Configuration	Brake Actuation	Brake Type		Brake Size	
		Unloaded	Loaded			Front	Rear	Front	Rear
1. Chevrolet (Light Truck)	C30 (1969)	6,764	10,732	2	Hydraulic, split master cylinder with booster	DSSA*	DSSA	11-1/8 x 2-3/4	13 x 2-1/2
2. International Harvester (Medium Truck)	Loadstar 1700 (1969)	10,920	25,500	2	Vacuum hydraulic single master cylinder	Two leading shoe	Twin-plex	15 x 3	16 x 6
3. Chevrolet (GMC) (Heavy Truck)	J-70 (1968)	15,600	39,000	3	Air	S-cam	Wedge Wedge	15 x 3-1/2	15 x 5, 15 x 5

*Duo-servo self-actuating

TABLE 2

BUSES

	Vehicle Manufacturer	Model	Test Weights, lb Unloaded	Loaded	Axle Configuration	Brake Actuation	Brake Type Front	Rear	Brake Size Front	Rear
1.	Ford-McFadden 66 passenger school bus	B-753 (1969)	14,050	24,500	2	Air (meets NEA standards)	Wedge	Wedge	15 x 3	15 x 5
2.	Motor Coach Industries 43-passenger intercity bus	MC-7 (1969)	27,890	35,940	3	Air, with limit valve on tag axle	S-cam	S-cam	14-1/2 x 5	14-1/2 x 8 14-1/2 x 5
3.	GMC 53-passenger city bus	T6H-5305 (1969)	21,200	32,145	2	Air	S-cam	S-cam	14-1/2 x 5	14-1/2 x 10

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TABLE 3

TRACTOR-TRAILER COMBINATIONS

	Vehicle Manufacturer	Model	Test Weights, lb		Axle Configuration	Brake Actuation	Brake Type		Brake Size	
			Unloaded	Loaded			Front	Rear	Front	Rear
1.	Tractor-Ford	F-7000 (1970)	18,930	41,480	2-S1	Air	S-cam	S-cam	16 x 2-1/2	16-1/2 x 6
	Trailer-Trailmobile 35 ft van	A32D2AAE (1970)						S-cam		16-1/2 x 7
2.	Tractor-Ford	F-7000 (1970)	20,850	56,380	2-S2	Air	S-cam	S-cam	16 x 2-1/2	16-1/2 x 6
	Trailer-Fruehauf 40 ft van	VB6F2 (1969)						Wedge		16-1/2 x 7
3.	Tractor-Diamond Reo	C11464DF (1970)	27,770	75,650	3-S2	Air		Wedge		15-1/2 x 7
	Trailer-Fruehauf 40 ft van	VB6F2 (1969)						Wedge		16-1/2 x 7
4.	Tractor-International Harvester	CO4070A (1969)	28,870	83,265	2-S1-2	Air	Wedge	Wedge	15 x 3-1/2	15 x 7
	Trailers-Brown 27 ft van	266CXD3H (1969) with DO2 dolly						S-cam	16-1/2 x 7	16-1/2 x 7

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TABLE 4

VEHICLES EQUIPPED WITH ADVANCED SYSTEMS

Vehicle Manufacturer	Model	Test Weights, lb Unloaded	Loaded	Axle Configuration	Brake Actuation	Brake Type Front	Brake Type Rear	Brake Size Front	Brake Size Rear
1. Ford	F-1000 (1966)	13,590	27,570	2	Full power hydraulic, separate system front/rear	Bendix disk	Bendix disk	15.63" diameter series B-1	15.63" diameter series B-2
2. Tractor-White	4654TD (1967)	29,360	77,780	3-S2		Double wedge	S-cam	15 x 4	16-1/2 x 7
Trailer-Fruehauf 40 ft platform	PBF-240-SP				Air		S-cam		16-1/2 x 7
3. Tractor-White	4654TD (1967)	16,770	44,320	3	Air	Double wedge	S-cam	15 x 4	16-1/2 x 7
4. Tractor-Brockway	N457T-4	29,960	61,500	2-S2		S-cam	S-cam	16-1/2 x 4-1/4	16-1/2 x 7
Trailer-Arrow 30 ft lowboy	SCR-22-35				Air		S-cam		16-1/2 x 7
5. Tractor-Brockway	N457T-4	14,200	--	2	Air	S-cam	S-cam	16-1/2 x 4-1/4	16-1/2 x 7

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each vehicle be as near to its "as new" condition as possible:

(1) Install new original equipment brake linings, and replace any heat checked drums or drums that have been rebored in excess of 0.060 inches.

(2) Burnish new brake linings of the trucks and buses in accordance with procedures specified in SAE Recommended Practice J786. For the combination vehicles, burnish according to procedures specified in SAE J880a.

(3) Replace any tires worn beyond 50% of original tread depth or having flat spots or other defects.

(4) Adjust all brakes to manufacturers specifications.

(5) Inspect all other brake-system components to insure that the total system is in good mechanical condition.

Instrumentation specified for each test is given in Table 5. A typical installation in a tractor cab is shown in Fig. 21. Light beam oscillographs were used for continuous records. Brake temperatures were measured by thermocouples placed in the brake linings according to the method specified in SAE Recommended Practice J843a. Brake line pressure was calibrated to pedal force for each vehicle tested.

3.3.2 GENERAL TEST REQUIREMENTS. All vehicle tests were conducted at the Bendix Automotive Development Center at New Carlisle, Indiana (see Fig. 22). During testing, HSRI provided a test engineer to monitor the tests, act as liaison between HSRI and Bendix, and handle data obtained from the tests. Ambient temperatures during the tests were required to be between 32°F and 100°F. The skid number at various locations on the test track and skid pad was measured by the Michigan State Highway Department Skid Trailer, first in September 1969 and again in July 1970, using ASTM Procedure E274, omitting water delivery. The skid number was checked each week tests were conducted by making locked-wheel partial stops from 40 mph with a passenger car equipped with a decelerometer. So that results from the skid trailer and instrumented car would be comparable, ASTM Standard Pavement Test Tires (E249) were mounted on the car. All dry pavement braking tests were conducted on the oval track or the skid pad approach road, each of which had an average skid number of 0.85. Low-coefficient tests were conducted on a portion of the skid pad surface which had been treated with an asphalt sealant. The skid number of this surface when wetted by a sprinkler truck was in the range of 0.21 to 0.24. Tire-road interface tests were also conducted on these surfaces using a medium truck equipped with typical 10 x 20 nylon truck tires. Results from these tests indicate an average peak/locked wheel coefficient of 0.75/0.60 for the oval track, 0.72/0.57 for the skid pad approach road, and 0.35/0.26 for the wet, sealed-asphalt surface of the skid pad. Results of all the tire-road interface tests are given in Appendix C.

For testing under fully loaded conditions, vehicles were loaded to their rated gross weight in such a way as to insure that the height of the center of gravity corresponded to that location generally found under normal service conditions. In tests made on the two articulated vehicles equipped with advanced brake control systems, this practice was omitted because of stability problems being encountered at high deceleration levels.

In testing combination vehicles the anti-jackknife bumper pictured in
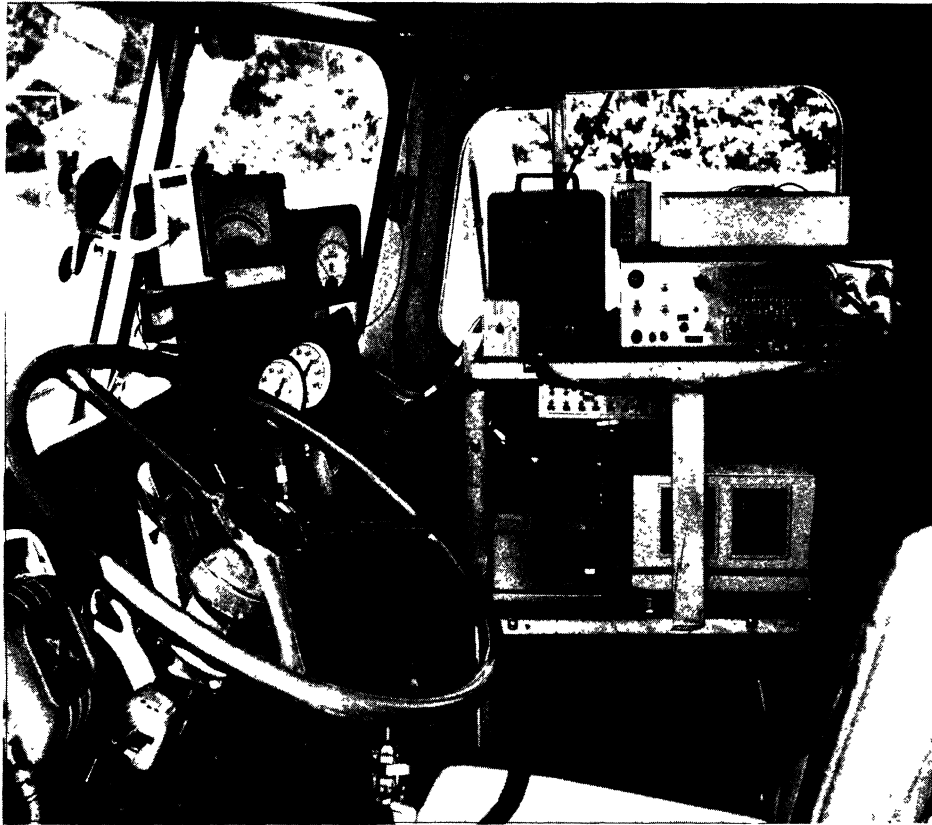
FIGURE 21. INSTRUMENTS INSTALLED IN TRACTOR CAB, VEHICLE 12

FIGURE 22. TEST TRACK AND SKID PAD, BENDIX AUTOMOTIVE DEVELOPMENT CENTER

TABLE 5

INSTRUMENTATION FOR TEST VEHICLES

Test	Variable Measured	Instrument Required	Record Continuously	Monitor
1. Effectiveness	deceleration	longitudinal accelerometer	X	
	velocity	5th wheel	X	
	stopping distance	detonator and tape measure, and 5th wheel counter		
	brake line pressures	pressure transducers	X	
	brake temperatures	thermocouples		
	wheel rotation	wheel lockup indicators	X	X
2. Emergency	Same as effectiveness test			
3. Fade	deceleration	longitudinal accelerometer	X	
	velocity	5th wheel	X	
	brake line pressure	pressure transducer	X	
	brake temperatures	thermocouples	X	
4. Brake Rating	velocity	5th wheel	X	
	brake line pressure	pressure transducer	X	
	tow bar force	instrumented tow bar	X	
	brake temperatures	thermocouples	X	
5. Minimum Stopping Distance	Same as effectiveness test			
6. Brake Balance Test	Same as effectiveness test			

TABLE 5 (Concluded)

INSTRUMENTATION FOR TEST VEHICLES

Test	Variable Measured	Instrument Required	Record Continuously	Monitor
7. Parking Brake Test	velocity	5th wheel	X	
	tow bar force	instrumented tow bar	X	
	wheel rotation	wheel lockup indicator	X	
8. Static Timing Test	pedal force	force transducer	X	
	brake line pressures	pressure transducers	X	

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Fig. 23 was used, which restricted the articulation angle between the tractor and trailer to 15 degrees.

3.3.3 TEST SPECIFICATIONS. Procedures employed in the effectiveness, brake failure, fade and recovery, brake rating, brake balance, minimum stopping distance, static timing tests, and parking brake tests are outlined in Appendix B. Table 6 summarizes the test program which was conducted on vehicles equipped with standard braking systems and on the truck equipped with disk brakes.

For the two tractor-trailer combinations equipped with advanced systems, changes in some of the test procedures were made because the test objectives were different from those of the other eleven vehicles. For example, in testing the White 6x4 tractor and the Fruehauf tandem-axle platform trailer (hereinafter referred to as Vehicle 12), five different vehicle and loading configurations were used, the effectiveness of the tractor front and rear brakes was varied, and the aforementioned brake control systems (i.e., proportioning valves, adaptive braking, and trailer brake synchronization) were employed in various combinations. The purpose in testing Vehicle 12 was to determine the best braking performance which could be achieved using a standard test vehicle equipped with commercially available brakes, and employing state-of-the-art brake control systems. The effectiveness of the tractor brakes was varied by changing the wedge angles and brake chambers on the front brakes, and the slack adjuster lengths on the rear brakes (see Table 7). Five different vehicle and load configurations were tested using Vehicle 12, namely:

(1) tractor-trailer, empty

(2) tractor-trailer, loaded, high c.g.

(3) tractor-trailer, loaded, low c.g.

(4) tractor only, bobtail configuration

(5) tractor only, loaded to gross axle weight rating

Prior to commencement of tests with Vehicle 12, the following advanced brake control systems were installed:

·Proportioning valves for the tractor rear brakes and trailer brakes; supplied by Borg-Warner.

·Adaptive braking system, with a sensor and controller mounted on each of the ten wheels of the combination to prevent wheel lockup during braking; supplied by Bendix-Westinghouse.

·Trailer brake synchronization (Syncron) system, which effectively applies the trailer brakes as soon as the treadle valve is depressed; supplied by the Berg Manufacturing Company.

The operation of the Borg-Warner proportioning valves is described in detail in References (79) and (80). The Bendix-Westinghouse Adaptive Braking System is described in Reference (107). The Berg Syncron System is described in Appendix G.

During the course of the test program conducted with Vehicle 12, the basic air brake system of both the tractor and trailer was altered by installation of a high-capacity brake control (treadle) valve, quick release valves on each trailer brake actuator, larger capacity lines to the front tractor brakes, and replacement of connectors, fittings, and tees which tended to

FIGURE 23. ANTI-JACKKNIFE BUMPER, VEHICLE 12

TABLE 6

TEST PROGRAM SUMMARY

Vehicles Equipped with Standard Brake Systems and Disk Brake Truck

Test Vehicle Sequence No.	Prebuenish Effectiveness	Empty Effectiveness	Loaded Effectiveness	Front System Failure	Rear System Failure	Power Boost Failure	Service System Failure	Fade Test	Recovery Test	Brake Rating	Brake Balance	Brake Response Time	Pedal Force/Line Pressure	Static Parking Brake Test
Trucks														
2. Light Truck-Chevrolet C-30	X	X	X	X	X	X		X	X	X			X	
1. Medium Truck-International C-1700	X	X	X			X		X	X	X			X	
3. Heavy Truck-Chevrolet J70	X	X	X				X	X	X	X			X	
Buses														
4. School Bus-Ford B-750	X	X	X				X	X	X	X			X	
5. Intercity Bus-Motor Coach Ind. MC-7	X	X	X				X	X	X	X			X	
10. City Bus-GMC T6H-5305	X	X	X				X	X	X	X		X	X	
Tractor-Trailers														
6. Ford Tractor/35 ft Trailmobile Trailer	X	X	X					X		X		X	X	
7. Ford Tractor/40 ft Fruehauf Trailer	X	X	X				X	X		X	X	X	X	
8. Diamond Reo Tractor/40 ft Fruehauf Trailer	X	X	X				X	X		X	X	X	X	
9. International Tractor/Brown 27 ft Doubles	X	X	X				X	X		X	X	X	X	
Advanced System														
11. Disk Brake Truck-Ford F-1000	X	X	X	X	X		X	X		X			X	X

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TABLE 7

BRAKE EFFECTIVENESS VARIATION—VEHICLE 12

Designation	Tractor						Trailer (16-1/2 x 7)		For Combination Vehicle (Precentage of total brake force on axles)		
	Front Brakes (15 x 4)		Rear Brakes (16-1/2 x 7)						Tractor		Trailer
	Chamber Area, sq in.	Wedge Angle, deg	Chamber Area, sq in.	Slack Adj. Length, in.			Chamber Area, sq in.	Slack Adj. Length, in.	Front	Rear	
Baseline	9	14	30	5			30	6	11.7	44.3	44.0
Improved front brakes	12	12	30	5			30	6	17.4	38.4	44.2
Most effective brakes	12	12	30	6.5			30	6	15.4	47.9	36.7

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restrict air flow. A complete diagram of the brake system as tested is given in Appendix A.

To evaluate the braking performance of Vehicle 12 under the above specified conditions of loading and brake effectiveness, as modified by the various brake control systems and combinations of systems, 59 minimum-stopping-distance tests and 5 effectiveness tests were conducted. Procedures for these tests are specified in Appendix B. However, it should be noted that wheel locking was not allowed in the minimum-stopping-distance tests, even on the low coefficient surface. This restriction imposed severe demands on the driver's judgment, and introduced human factors into the test procedure to a degree which could well affect the reliability of certain results.

Ten static timing tests were run on Vehicle 12 to study the effect of the various devices and combinations of devices upon brake response time. Four brake balance tests were conducted to check the effectiveness of various combinations of brake hardware. A complete tabulation of the tests conducted on Vehicle 12 is given in Table 8.

The last vehicle tested in the program was a tractor-trailer combination, furnished by Eaton, Yale, and Towne. This vehicle (hereinafter designated Vehicle 14), a 4 x 2 Brockway COE tractor with an Arrow lowboy 35-ft trailer, was extensively modified for use in development of the Eaton, Yale, and Towne wheel antilock system, with which the vehicle was equipped. Three effectiveness tests, five minimum-stopping-distance tests, and four brake response time tests were conducted on Vehicle 14. A tabulation of the effectiveness and minimum-stopping-distance tests is given in Table 9.

3.4 TEST RESULTS—BASELINE VEHICLES

The test results for the ten baseline vehicles (vehicles equipped with standard braking systems) are given in this section. Included are results from effectiveness, brake failure, fade and recovery, brake rating, brake balance, and brake-timing tests.

3.4.1 EFFECTIVENESS TESTS. The vehicles were tested for brake effectiveness, in the loaded and empty condition to the point of wheel lockup if the brakes had sufficient torque capacity. If the vehicle remained stable beyond the point of first wheel lockup, pedal force was increased to lock up as many wheels as possible. Preburnish effectiveness tests for the loaded condition were run on all ten vehicles. Effectiveness test results for each of the ten vehicles are given in Figs. 24 through 33. Results for all the effectiveness tests are summarized in Table 10, and the deceleration capabilities of the baseline vehicles are summarized graphically in Fig. 34.

A comparison of the pre- and post-burnish effectiveness curves shows that burnishing did not greatly increase (or decrease) the overall effectiveness of trucks and buses, but dramatically improved the effectiveness of the tractor-semitrailer vehicles. Experience has shown that burnishing generally "wears in" and "cures" the brake linings, thus reducing any propensity the brake may have to "grab" or "pull" due to a misfit between lining and drum or nonuniformities on the lining surface. Inasmuch as braking performance for tractor-trailers is more dependent upon brake sensitivity and balance,
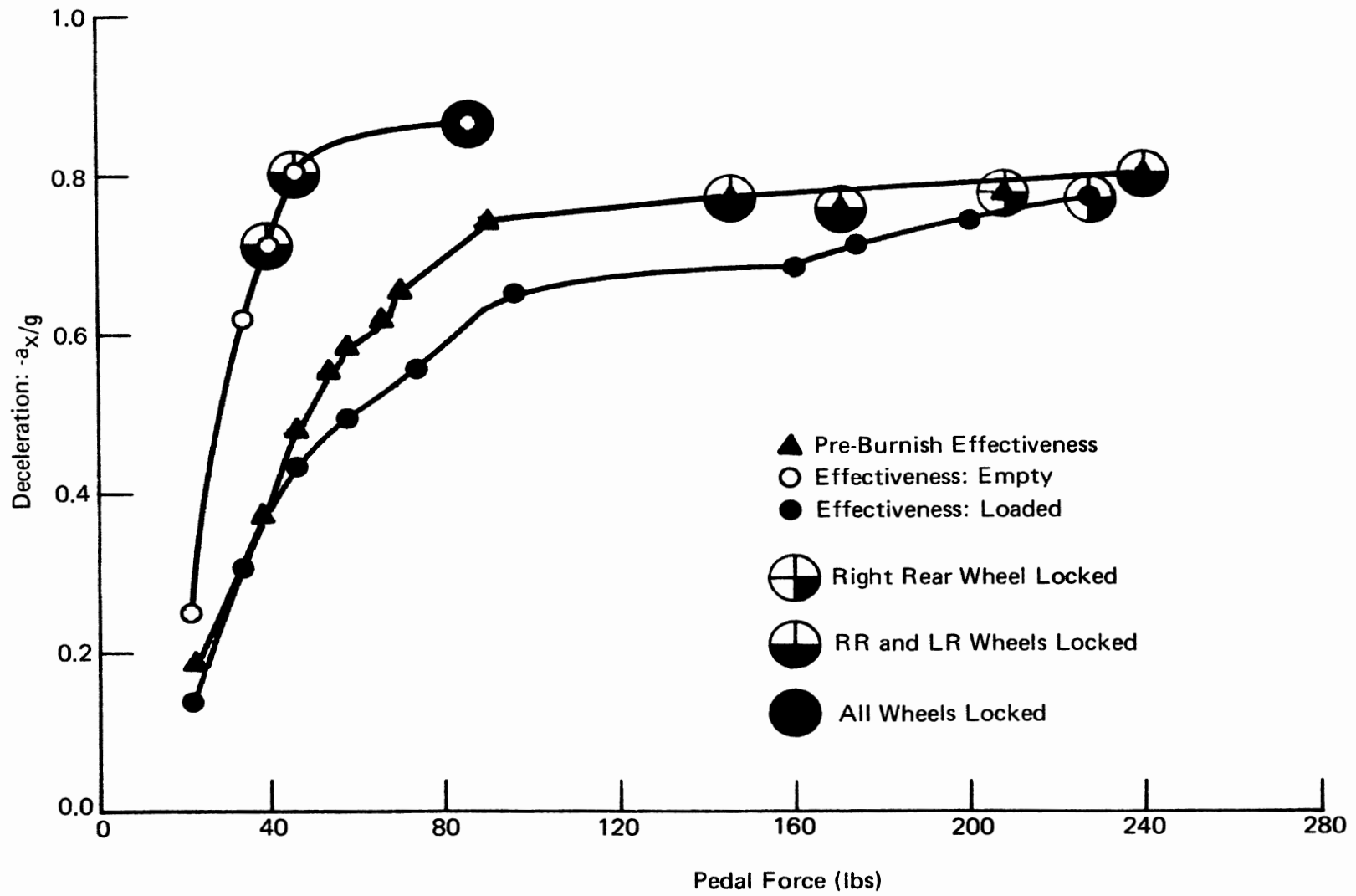
FIGURE 24. EFFECTIVENESS TEST RESULTS, LIGHT TRUCK
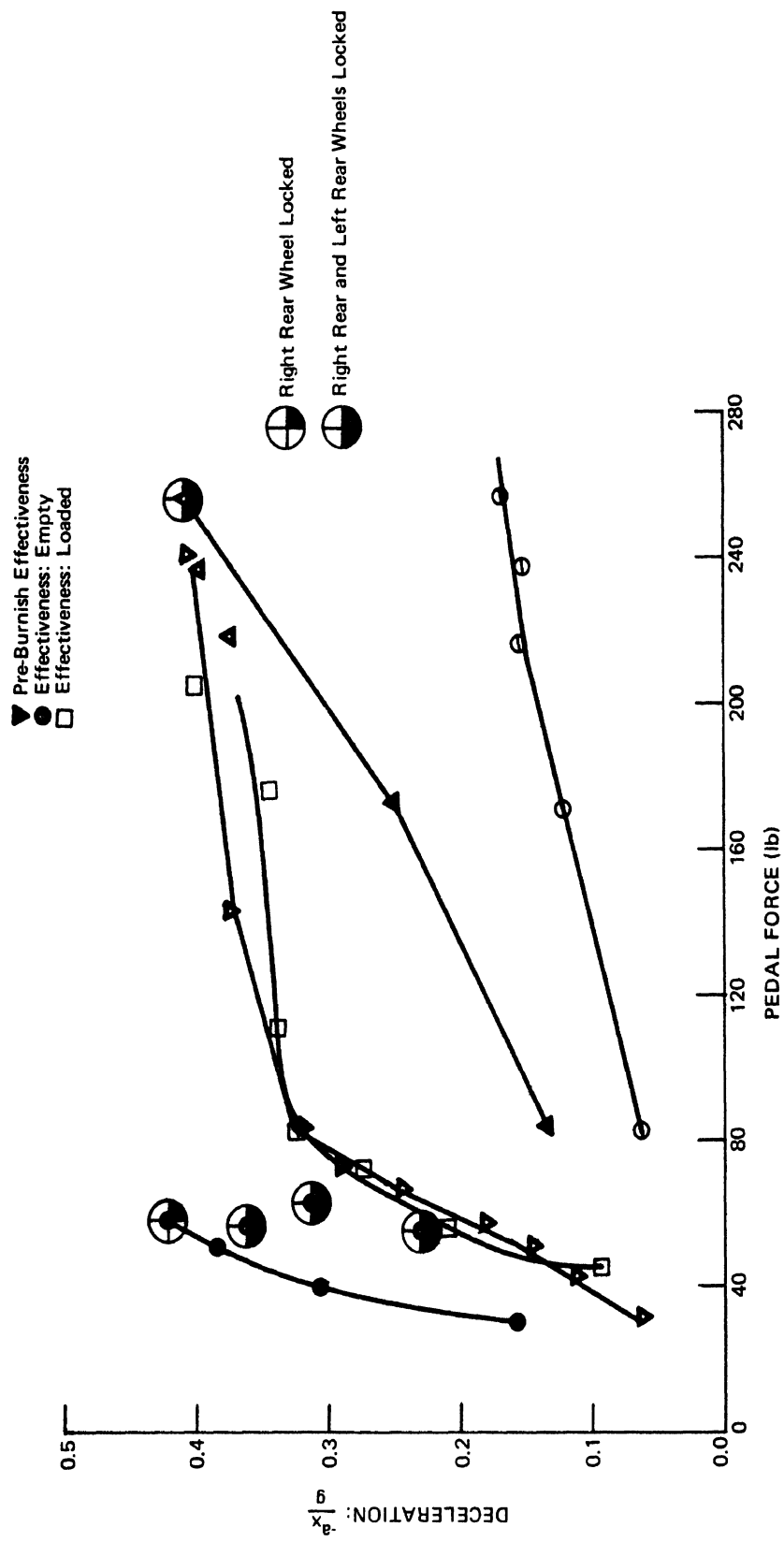
FIGURE 25. EFFECTIVENESS TEST RESULTS, MEDIUM TRUCK

71
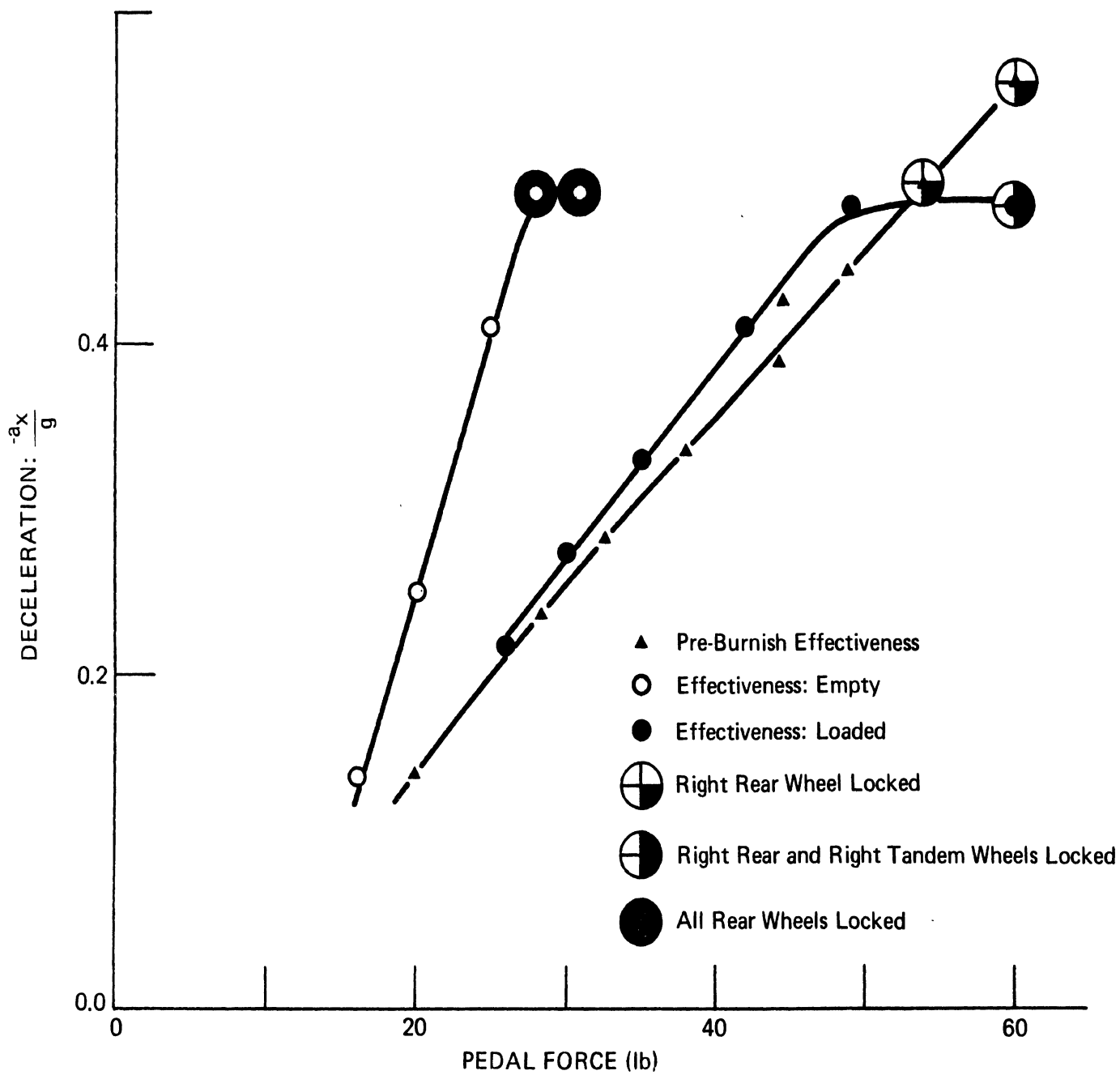
FIGURE 26. EFFECTIVENESS TEST RESULTS, HEAVY TRUCK
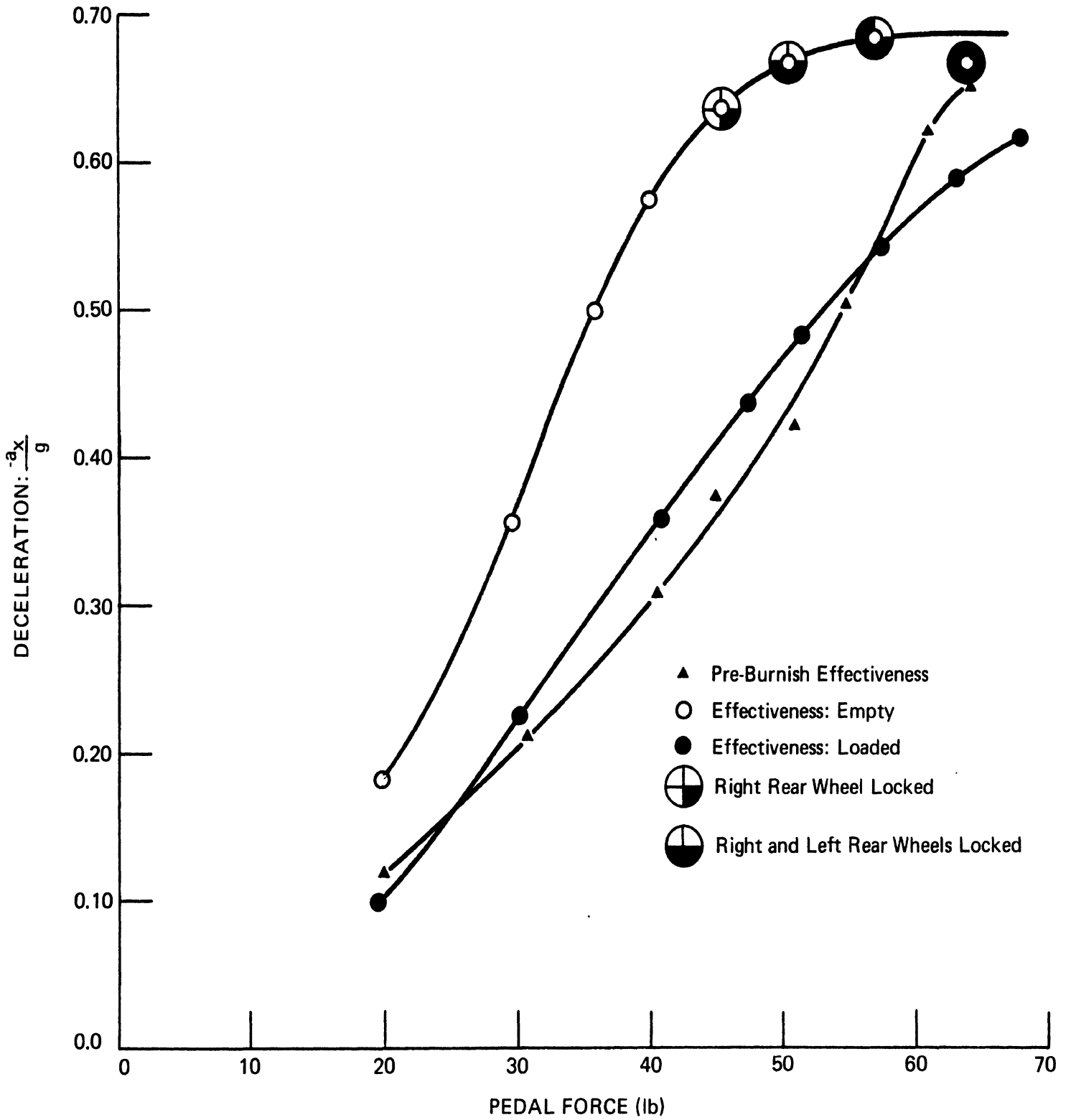
FIGURE 27. EFFECTIVENESS TEST RESULTS, SCHOOL BUS

73
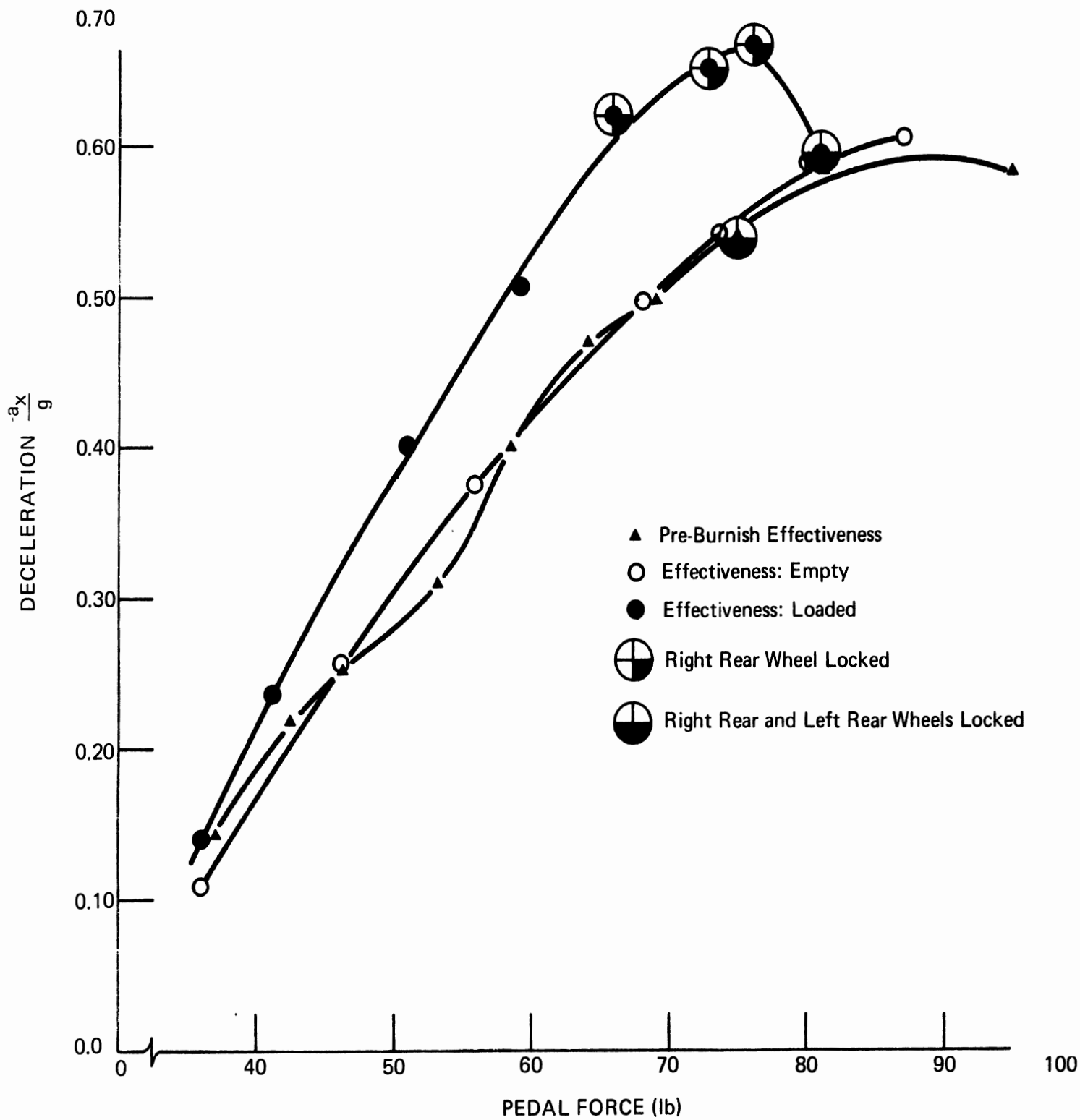
FIGURE 28. EFFECTIVENESS TEST RESULTS, INTERCITY BUS

74
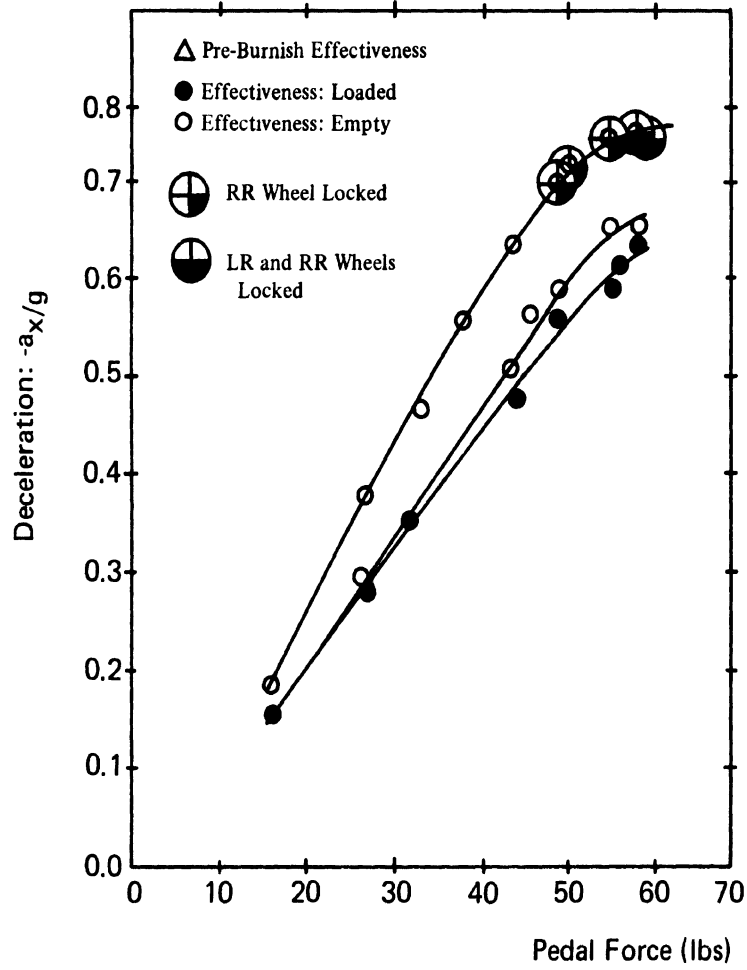
FIGURE 29. EFFECTIVENESS TEST RESULTS, CITY BUS
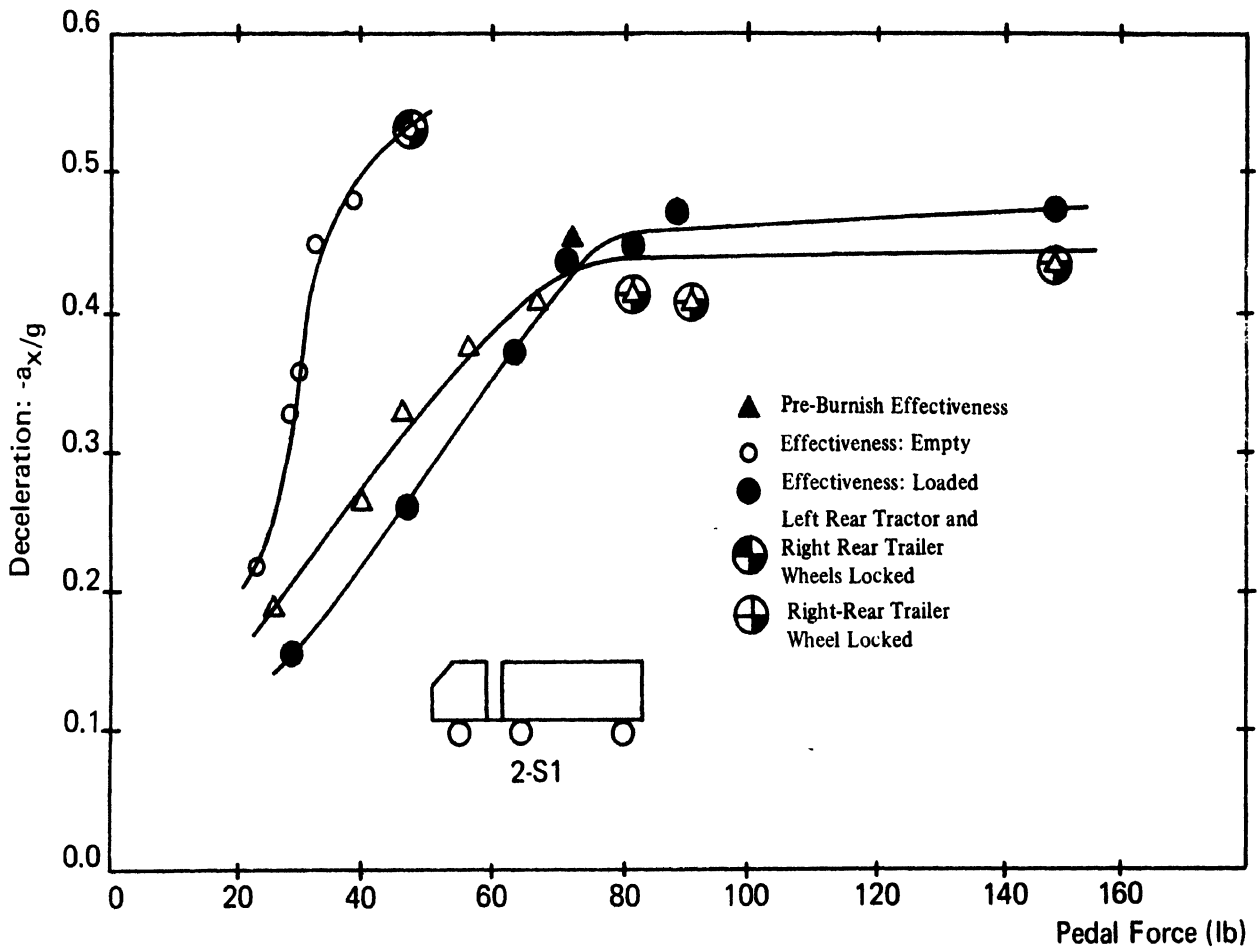
FIGURE 30. EFFECTIVENESS TEST RESULTS, TRACTOR-TRAILER 2-S1
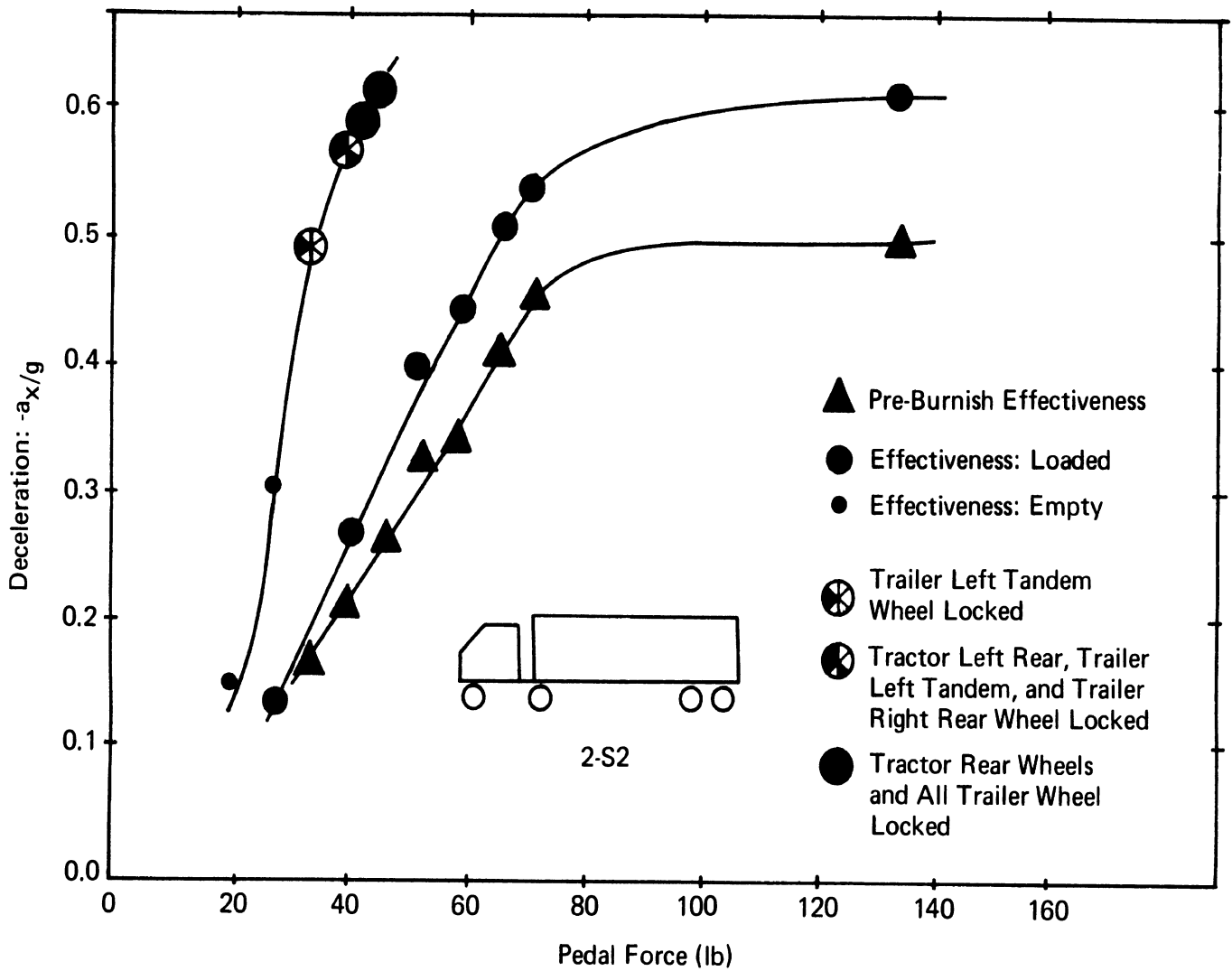
FIGURE 31. EFFECTIVENESS TEST RESULTS, TRACTOR-TRAILER 2-S2
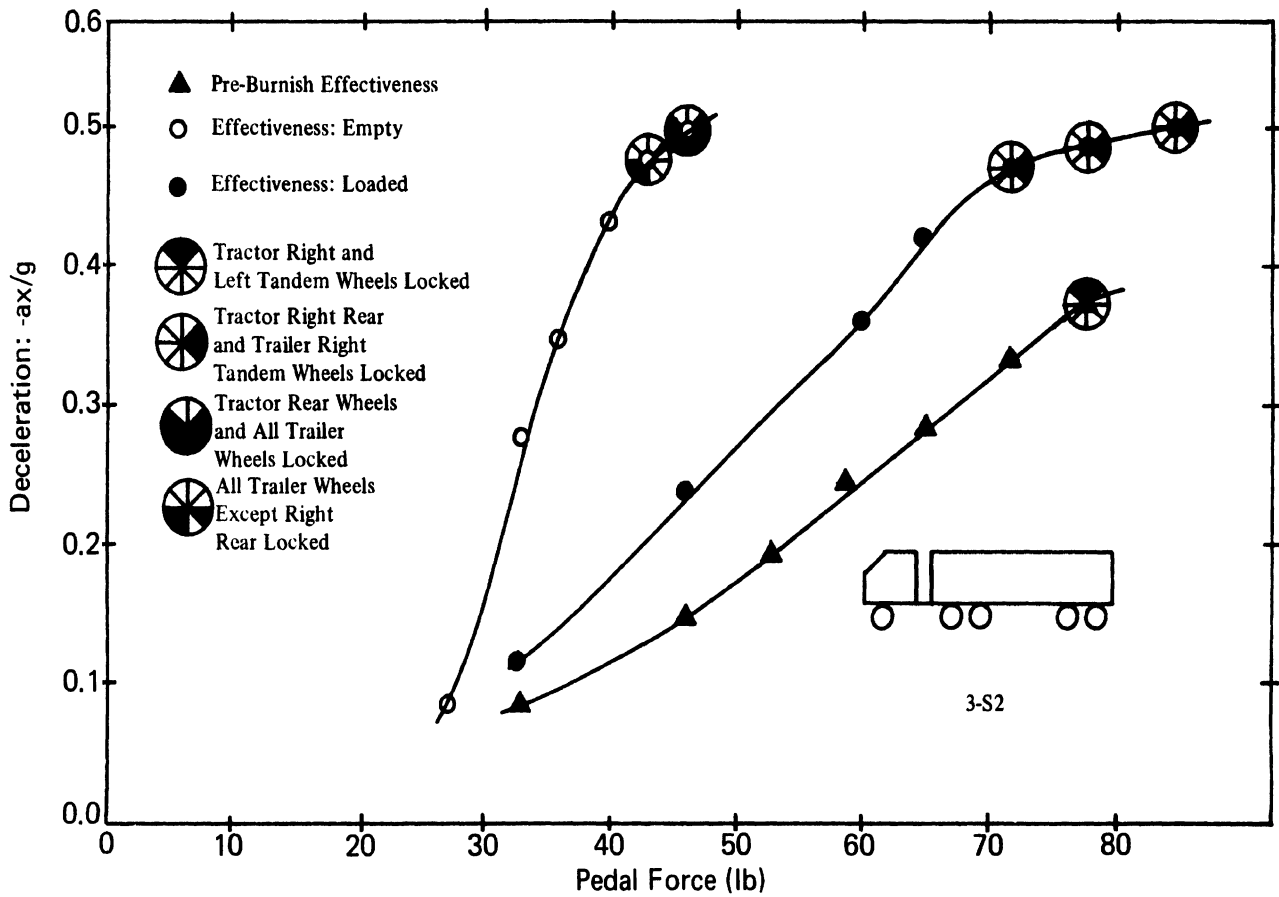
FIGURE 32. EFFECTIVENESS TEST RESULTS, TRACTOR-TRAILER 3-S2
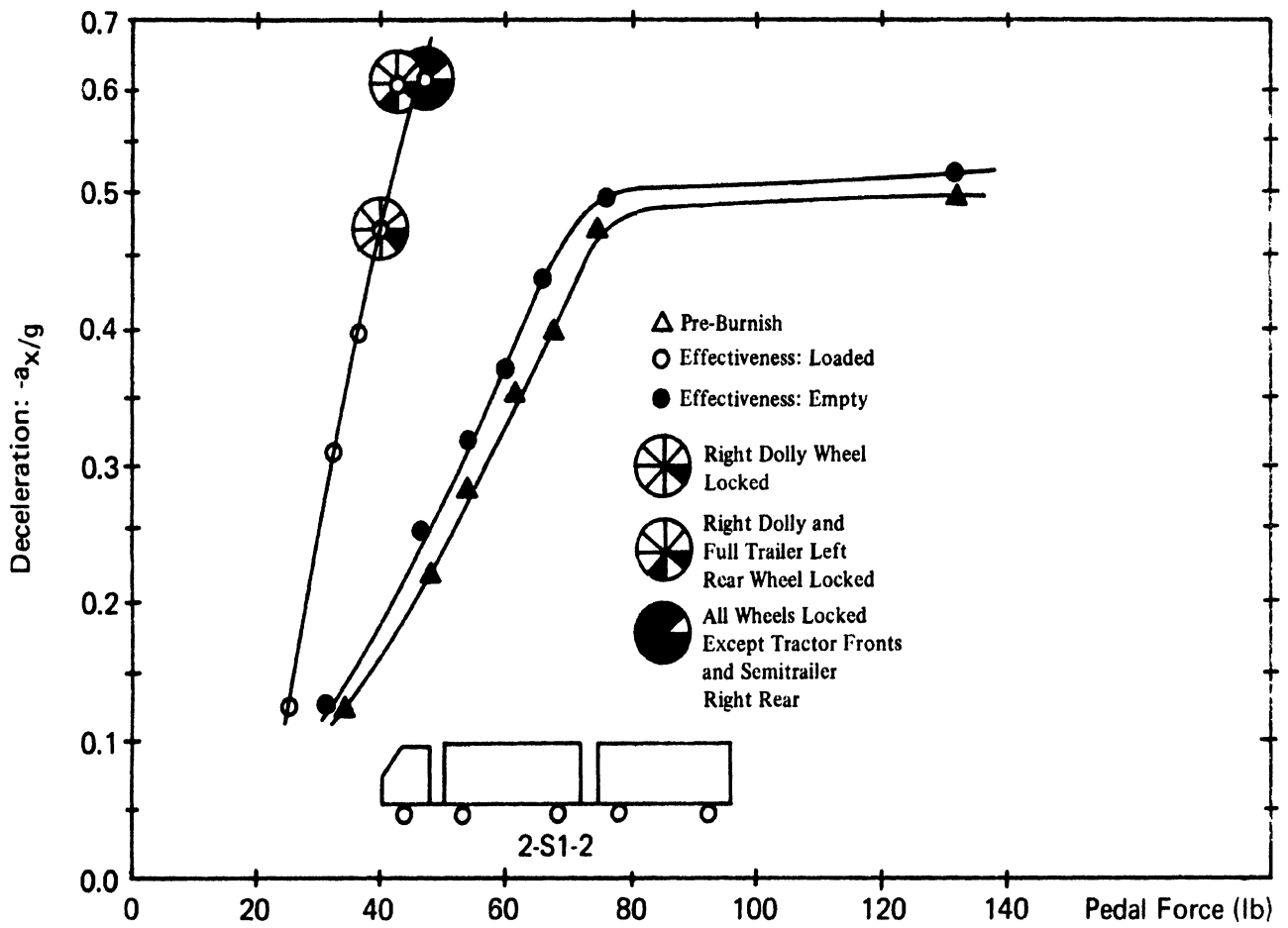
FIGURE 33. EFFECTIVENESS TEST RESULTS, DOUBLES COMBINATION 2-S1-2
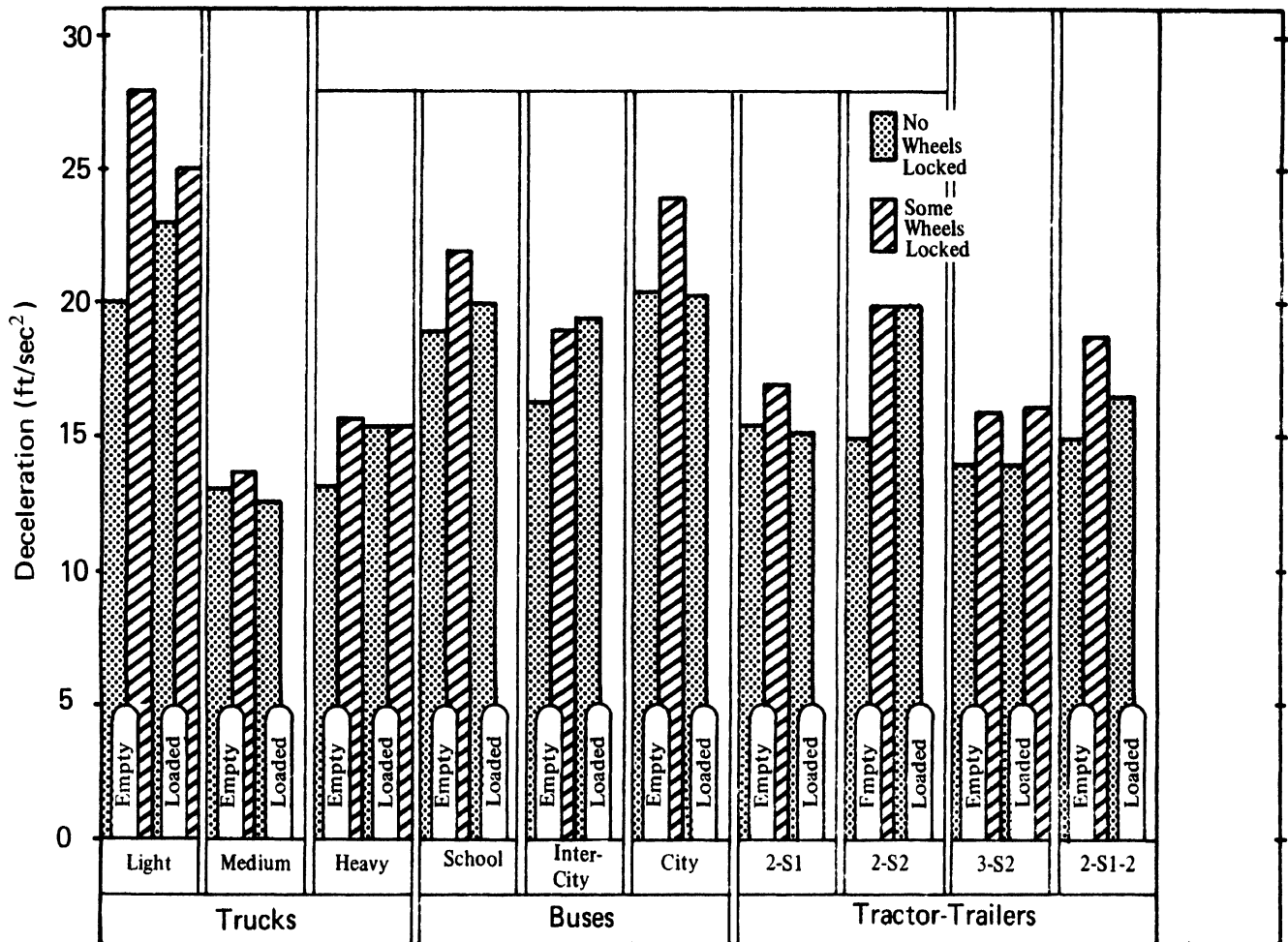
FIGURE 34. MAXIMUM DECELERATION CAPABILITY, BASELINE VEHICLES

TABLE 8

TEST PROGRAM—VEHICLE 12

	Configuration	V_o, mph	Track Condition	Load	CG	Brakes*	Advanced Systems	Test
1.	Tractor-Trailer	60	Dry	Empty	--	Baseline	None	Effectiveness
2.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Proportioning	Effectiveness
3.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Adaptive	Effectiveness
4.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Adaptive on Tractor Proportioning on Trailer	Effectiveness
5.	Tractor-Trailer	60	Dry	Loaded	Low	Most effective	None	Effectiveness
1.	Tractor-Trailer	60	Dry	Loaded	High	Baseline	None	Stopping Distance
2.	Tractor-Trailer	60	Dry	Loaded	High	Baseline	Syncron	Stopping Distance
3.	Tractor-Trailer	60	Dry	Loaded	High	Baseline	Adaptive	Stopping Distance
4.	Tractor-Trailer	60	Dry	Loaded	High	Baseline	Adaptive and Syncron	Stopping Distance
5.	Tractor-Trailer	60	Dry	Loaded	Low	Baseline	Syncron	Stopping Distance
6.	Tractor-Trailer	60	Dry	Loaded	Low	Baseline	Adaptive and Syncron	Stopping Distance
7.	Tractor-Trailer	60	Dry	Loaded	Low	Most effective	None	Stopping Distance
8.	Tractor-Trailer	60	Dry	Loaded	Low	Most effective	Adaptive and Syncron	Stopping Distance
9.	Tractor-Trailer	60	Dry	Loaded	Low	Most effective	None	Stopping Distance
10.	Tractor-Trailer	60	Dry	Empty	--	Improved Front Brakes	Adaptive	Stopping Distance
11.	Tractor-Trailer	60	Dry	Empty	--	Improved Front Brakes	Adaptive and Syncron	Stopping Distance
12.	Tractor-Trailer	60	Dry	Empty	--	Most effective	Adaptive and Syncron	Stopping Distance
13.	Tractor-Trailer	60	Dry	Empty	--	Baseline	None	Stopping Distance
14.	Tractor-Trailer	60	Dry	Empty	--	Baseline	None	Stopping Distance
15.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Proportioning	Stopping Distance
16.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Proportioning	Stopping Distance
17.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Antilock	Stopping Distance
18.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Adaptive on Tractor, Proportioning on Trailer	Stopping Distance
19.	Tractor-Trailer	60	Dry	Empty	--	Baseline	Adaptive on Tractor, Proportioning on Trailer	Stopping Distance
20.	Tractor-Trailer	40	Wet	Loaded	High	Baseline	None	Stopping Distance
21.	Tractor-Trailer	40	Wet	Loaded	High	Baseline	Syncron	Stopping Distance
22.	Tractor-Trailer	40	Wet	Loaded	High	Baseline	Adaptive	Stopping Distance
23.	Tractor-Trailer	40	Wet	Loaded	High	Baseline	Adaptive and Syncron	Stopping Distance
24.	Tractor-Trailer	40	Wet	Loaded	High	Improved Front Brakes	Adaptive	Stopping Distance

*Designations defined in Table 7.

TABLE 8 (Continued)

TEST PROGRAM SUMMARY—VEHICLE 12

	Configuration	V_o, mph	Track Condition	Load	CG	Brakes*	Advanced Systems	Test
25.	Tractor-Trailer	40	Wet	Loaded	Low	Most effective	Adaptive and Syncron	Stopping Distance
26.	Tractor-Trailer	40	Wet	Empty	--	Most effective	Adaptive and Syncron	Stopping Distance
27.	Tractor-Trailer	20	Dry	Loaded	High	Baseline	None	Stopping Distance
28.	Tractor-Trailer	20	Dry	Loaded	High	Baseline	Syncron	Stopping Distance
29.	Tractor-Trailer	20	Dry	Loaded	High	Baseline	Adaptive	Stopping Distance
30.	Tractor-Trailer	20	Dry	Loaded	High	Baseline	Adaptive and Syncron	Stopping Distance
31.	Tractor-Trailer	20	Dry	Loaded	Low	Baseline	Syncron	Stopping Distance
32.	Tractor-Trailer	20	Dry	Loaded	Low	Baseline	Adaptive and Syncron	Stopping Distance
33.	Tractor-Trailer	20	Dry	Empty	--	Improved Front Brakes	Adaptive	Stopping Distance
34.	Tractor-Trailer	20	Dry	Empty	--	Improved Front Brakes	Adaptive and Syncron	Stopping Distance
35.	Tractor-Trailer	20	Wet	Loaded	High	Baseline	None	Stopping Distance
36.	Tractor-Trailer	20	Wet	Loaded	High	Baseline	Syncron	Stopping Distance
37.	Tractor-Trailer	20	Wet	Loaded	High	Baseline	Adaptive	Stopping Distance
38.	Tractor-Trailer	20	Wet	Loaded	High	Baseline	Adaptive and Syncron	Stopping Distance
39.	Tractor-Trailer	20	Wet	Loaded	High	Improved Front Brakes	Adaptive	Stopping Distance
40.	Tractor-Trailer	20	Wet	Loaded	High	Improved Front Brakes	Adaptive and Syncron	Stopping Distance
41.	Tractor	60	Dry	Empty	--	Baseline	None	Stopping Distance
42.	Tractor	60	Dry	Empty	--	Baseline	Proportioning	Stopping Distance
43.	Tractor	60	Dry	Empty	--	Baseline	Adaptive	Stopping Distance
44.	Tractor	60	Dry	Loaded	Low	Most effective	None	Stopping Distance
45.	Tractor	60	Dry	Loaded	Low	Most effective	Adaptive	Stopping Distance
46.	Tractor	40	Wet	Empty	--	Baseline	None	Stopping Distance
47.	Tractor	40	Wet	Empty	--	Baseline	Proportioning	Stopping Distance
48.	Tractor	40	Wet	Empty	--	Baseline	Adaptive	Stopping Distance
49.	Tractor	40	Wet	Empty	--	Most effective	None	Stopping Distance
50.	Tractor	40	Wet	Empty	--	Most effective	Adaptive	Stopping Distance
51.	Tractor	40	Wet	Loaded	Low	Most effective	None	Stopping Distance
52.	Tractor	40	Wet	Loaded	Low	Most effective	Adaptive	Stopping Distance
53.	Tractor	20	Dry	Empty	--	Baseline	None	Stopping Distance
54.	Tractor	20	Dry	Empty	--	Baseline	Proportioning	Stopping Distance
55.	Tractor	20	Dry	Empty	--	Baseline	Adaptive	Stopping Distance
56.	Tractor	20	Dry	Empty	--	Improved Front Brakes	Adaptive	Stopping Distance
57.	Tractor	20	Wet	Empty	--	Baseline	None	Stopping Distance

*Designations defined in Table 7.

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TABLE 8 (Concluded)

TEST PROGRAM SUMMARY—VEHICLE 12

	Configuration	V_o, mph	Track Condition	Load	CG	Brakes*	Advanced Systems	Test
58.	Tractor	20	Wet	Empty	--	Baseline	Proportioning	Stopping Distance
59.	Tractor	20	Wet	Empty	--	Baseline	Adaptive	Stopping Distance
1.	Tractor-Trailer	--	--	--	--	Baseline	None	Brake Response Time
2.	Tractor-Trailer	--	--	--	--	Baseline	Syncron	Brake Response Time
3.	Tractor-Trailer	--	--	--	--	Baseline	Adaptive	Brake Response Time
4.	Tractor-Trailer	--	--	--	--	Baseline	Adaptive and Syncron	Brake Response Time
5.	Tractor-Trailer	--	--	Empty	--	Baseline	Proportioning	Brake Response Time
6.	Tractor-Trailer	--	--	Empty	--	Baseline	Proportioning and Syncron	Brake Response Time
7.	Tractor-Trailer	--	--	Empty	--	Baseline	Proportioning, Adaptive, and Syncron	Brake Response Time
8.	Tractor-Trailer	--	--	Loaded	--	Baseline	Proportioning	Brake Response Time
9.	Tractor-Trailer	--	--	Loaded	--	Baseline	Proportioning with Syncron	Brake Response Time
10.	Tractor-Trailer	--	--	--	--	Most effective	None	Brake Response Time
1.	Tractor-Trailer	20	Dry	Loaded	Low	Baseline	--	Brake Balance
2.	Tractor-Trailer	20	Dry	Loaded	Low	Improved Front Brakes	--	Brake Balance
3.	Tractor-Trailer	20	Dry	Loaded	Low	Most effective	--	Brake Balance
4.	Tractor-Trailer	20	Dry	Loaded	Low	Most effective	New pipings and fittings to front brakes	Brake Balance
1.	Tractor	--	Dry	Loaded	Low	Most effective	Berg Spring Brakes	Parking Brake
2.	Tractor	--	Dry	Loaded	Low	Most effective	Bendix Westinghouse Spring Brakes	Parking Brake

*Designations defined in Table 7.

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TABLE 9

TEST PROGRAM SUMMARY—VEHICLE 14

	Configuration	V_o, mph	Track Condition	Load	Brake System	Test
1.	tractor-trailer	60	dry	full	standard	effectiveness
2.	tractor-trailer	60	dry	full	antilock	effectiveness
3.	tractor-trailer	60	dry	empty	antilock	effectiveness
4.	tractor-trailer	40	wet	empty	standard	min. stop. dist.
5.	tractor-trailer	40	wet	full	antilock	min. stop. dist.
6.	tractor-trailer	60	wet	empty	antilock	min. stop. dist.
7.	tractor	60	dry	bobtail	antilock	min. stop. dist.
8.	tractor	40	wet	bobtail	antilock	min. stop. dist.

TABLE 10

EFFECTIVENESS TEST SUMMARY—BASELINE VEHICLES

| Vehicles | Nom. V_o, mph | Minimum Stopping Distance, ft | | | | Maximum Decel., fps^2 | | | |
| | | Empty | | Loaded | | Empty | | Loaded | |
		No Wheels Locked	Some Wheels Locked	No Wheels Locked	Some Wheels Locked	No Wheels Locked	Some Wheels Locked	No Wheels Locked	Some Wheels Locked
Light Truck	60	238	150	219	191	20.0	28.0	23.0	25.0
Medium Truck	60	322	282	307	–	13.0	13.7	12.6	–
Heavy Truck	60	316	248	263	262	13.2	15.8	15.5	15.5
School Bus	40	108	84	119	–	19.0	22.0	20.0	–
Intercity Bus	60	290	221	202	–	16.3	21.5	19.5	–
City Bus	40	93	72	89	–	20.5	24.0	20.3	–
Tractor-Trailer 2-S1	60	291	258	292	–	15.5	17.0	15.2	–
Tractor-Trailer 2-S2	60	428	220	222	–	15.0	20.0	20.0	–
Tractor-Trailer 3-S2	60	366	247	376	299	14.0	16.0	14.0	16.1
Tractor-Double Trailer	60	395	294	309	–	15.0	18.8	16.6	–

especially in preventing premature wheel lockup, than for straight trucks, it appears that adequately burnished brakes are necessary to obtain the best possible performance of combination vehicles.

The average maximum deceleration achieved (without wheel lock occurring) with the tested trucks is 15.5 ft/sec^2 in the empty condition and 17.0 ft/sec^2 in the loaded condition. If wheel locking is permitted, the figures become 19.2 and 17.7 ft/sec^2, respectively. These results show that brake torque capacity has been proportioned to give best performance for the loaded condition.

For the test buses, the average maximum deceleration, wheels unlocked, is 18.5 ft/sec^2 empty and 19.9 ft/sec^2 loaded, with the performance of the empty buses increasing to 21.7 ft/sec^2, when some wheels are locked. It should be noted that it was impossible to lock up wheels on any of the buses when fully loaded.

For the tractor-trailers tested, the average maximum deceleration, wheels unlocked, is 15.0 ft/sec^2 empty and 16.5 ft/sec^2 loaded, with the performance empty increasing to 19.4 ft/sec^2 when wheels are locked.

Although the trucks exhibit a wide range in performance, the loaded buses exhibit uniform performance. A rather remarkable variation in performance is noted for the tractor-trailer combinations, even though the same tractor was used in two of the combinations (2-S1 and 2-S2) and the same trailer was used in two combinations (2-S2 and 3-S2).

Overall, the ten vehicles achieved an average maximum deceleration, wheels unlocked, of 16.5 ft/sec^2, empty, and 17.7 ft/sec^2, loaded; and, if some wheels are permitted to lock during the stop, 19.4 ft/sec^2, empty, and 18.1 ft/sec^2, loaded. The measured range of performance is greatest for the trucks (12.6 to 28.0 ft/sec^2) and least for the tractor-trailers (14.0 to 20.0 ft/sec^2). Bus performance ranged from 16.3 to 24.0 ft/sec^2.

An important performance consideration associated with effectiveness tests has to do with lateral stability. In the tests conducted on the baseline vehicles, lateral stability was assessed only by noting whether or not the driver was able to keep the vehicle in a 12-ft lane for the duration of the stop. In the test specification, the driver was instructed to release the brakes immediately and to abort the test if the vehicle became directionally unstable in all the effectiveness tests, except as noted below:

Light Truck. In the pre-burnish effectiveness test, the vehicle pulled five feet out of the 12-ft lane on one stop. After burnishing, however, no problem with directional stability was encountered in any of the tests.

Medium Truck. The loaded vehicle exhibited a slightly erratic tendency to pull to one side or the other, which behavior could be controlled except for two stops in which the vehicle pulled 10 ft out of the 12-ft lane. When tested empty, the instability was accentuated, and in two instances the driver had to release the brakes to regain directional control. It was noted that the rear wheels locked (one a fraction of a second later than the other) at the point where the driver had to release the brakes. This observation seems to indicate that side-to-side imbalance and differential wheel lockup were the causes of vehicle instability. In both instances the vehicle pulled

10 ft out of the 12-ft lane.

Heavy Truck. This vehicle showed no directional instability except in the last effectiveness test made on the loaded vehicle, with maximum line pressure. The rear wheels locked, causing the vehicle to become unstable. The driver released the brakes, which caused the vehicle to veer off the track on the left side, with the right wheels lifting off the ground. Subsequent control actions by the driver brought the vehicle back onto the track for a more or less controlled stop.

Tractor-Trailer. (2-S1). No wheel lockup occurred on the empty vehicle until the line pressure was increased to 40 psi. At this pressure, the left rear tractor wheel, the right trailer wheel, and the right rear tractor wheel locked up, in rapid succession, in that order, causing an increasing articulation angle between the tractor and the trailer. The driver released the brakes and applied the accelerator, which reduced the articulation angle and restored stability. The test was aborted at this point. (Subsequently, an anti-jackknifing bumper, limiting the articulation angle between the tractor and trailer to 15 degrees, was mounted on all of the articulated vehicles to be tested.)

Doubles Combination. (2-S1 and 2-S2). In testing this combination, the anti-jackknife bumper was used together with restricting chains between the trailers to limit relative movement between vehicle units. In the effectiveness test conducted on the empty combination, a tendency toward instability was noted on the 40 psi line pressure stop. Although the combination did not leave the 12-ft lane, seven out of ten wheels locked, causing the vehicle to stop in a "modified Z" configuration. The test was aborted at this point.

It should be noted that seven out of the ten vehicles, when tested in the loaded condition, did not have sufficient torque capacity to lock up any wheels on the surface used in this test program. It should also be pointed out that seven of the ten vehicles, when empty, could not exceed the performance of the loaded vehicle without locking up some wheels, which locking causes either loss of steering control or leads to lateral instability.

3.4.2 BRAKE FAILURE TESTS. The brake failure modes selected for test depended upon the design of the brake system. In the light truck, three failure modes were possible: front hydraulic line failure, rear hydraulic line failure, and power boost failure. Effectiveness test results for these modes of failure for the light truck in the empty and loaded condition are given in Figs. 35 and 36. The medium truck was not designed with separate front and rear brake failure protection, and thus was tested only for power boost failure. Results for the effectiveness test with the power boost failed in the empty and loaded condition are given in Fig. 37. The heavy truck and the city bus, each equipped with air brakes, did not contain an emergency system except for a mechanical hand brake attached to the drive shaft. The school and intercity buses, however, had separate emergency brake systems, which, when actuated under failure of the service brake system, supplied air pressure to the rear brakes. (See Appendix A for details of brake piping.) Opening the rear door of the city bus also actuated the rear brakes through the normal service system.
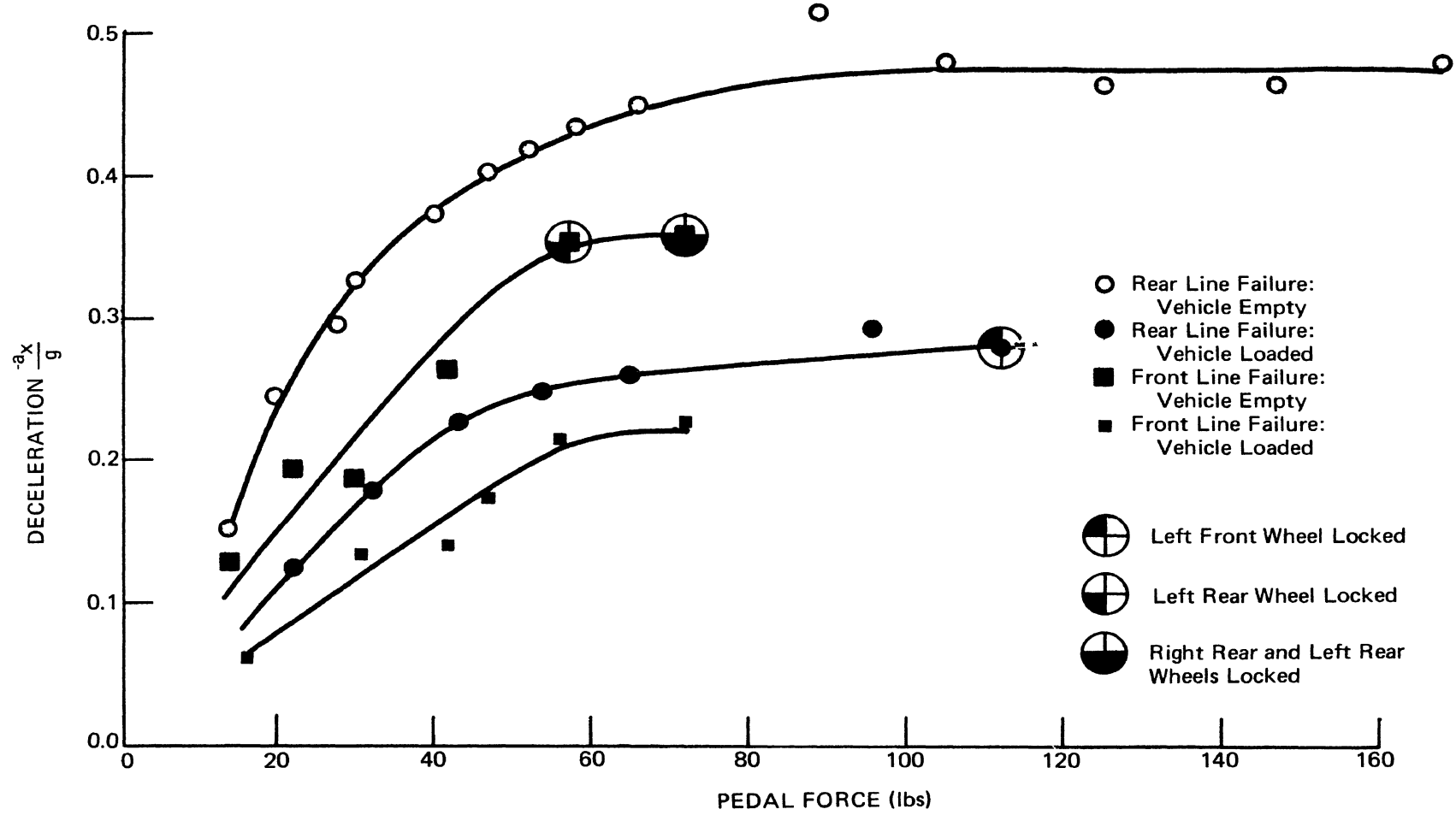
FIGURE 35. EFFECTIVENESS TEST RESULTS, FRONT AND REAR HYDRAULIC LINE FAILURE, LIGHT TRUCK

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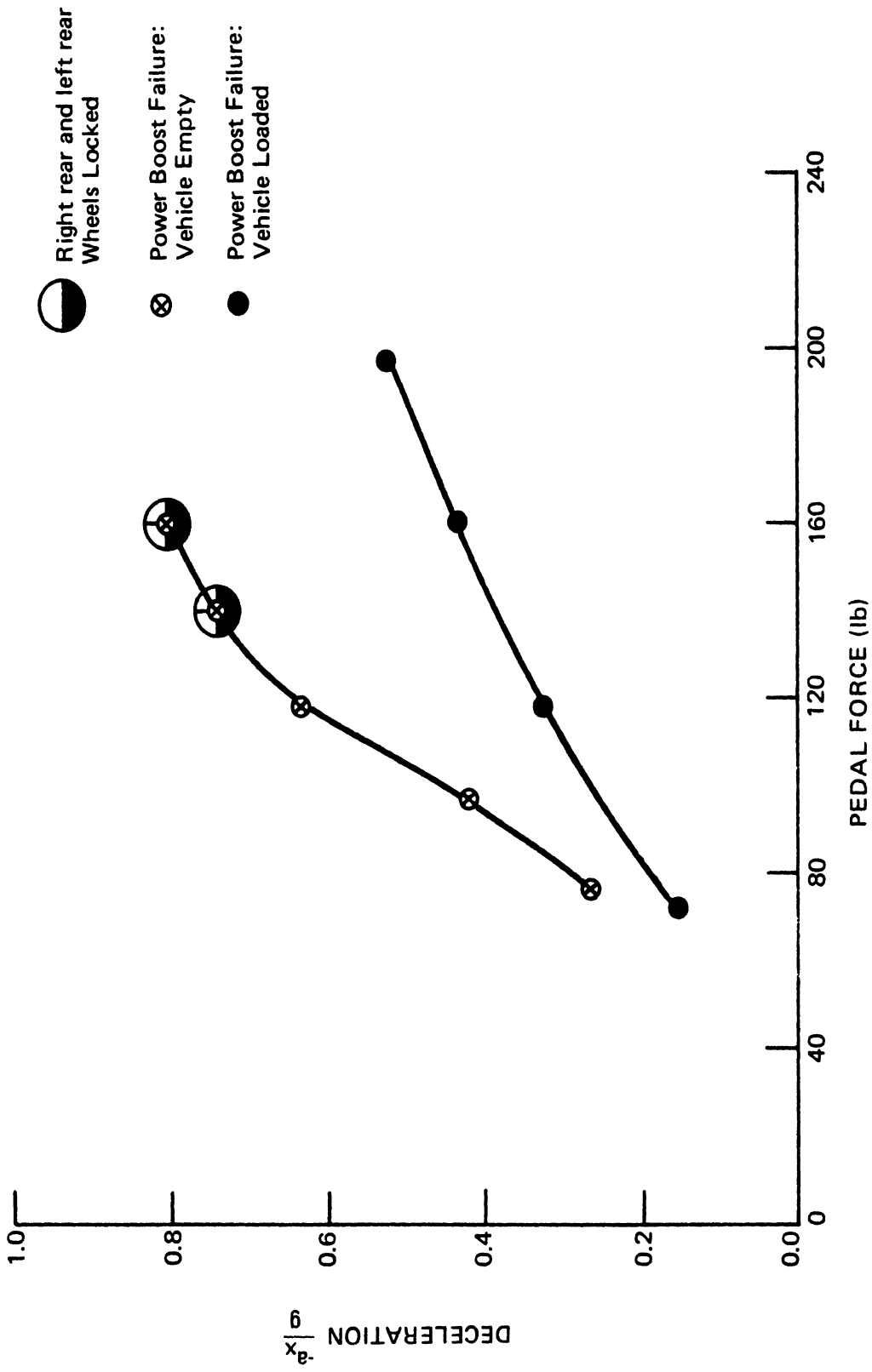
FIGURE 36. EFFECTIVENESS TEST RESULTS, POWER BOOST FAILURE, LIGHT TRUCK

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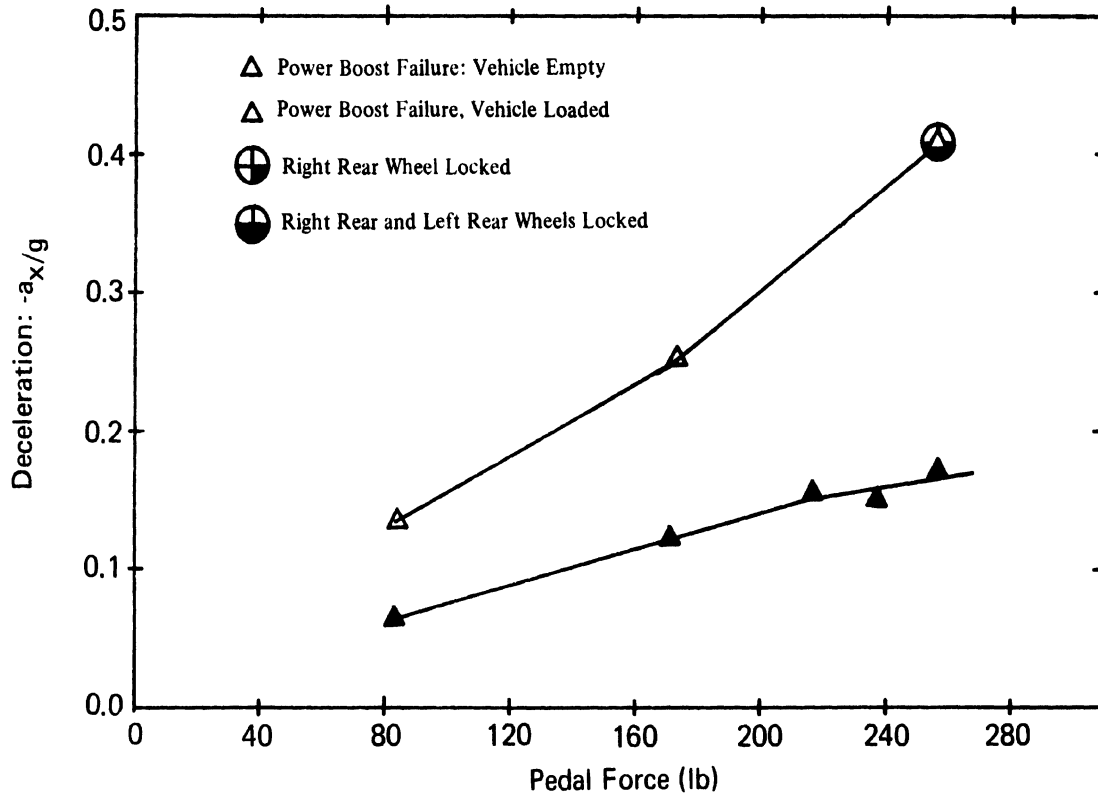
FIGURE 37. EFFECTIVENESS TEST RESULTS, POWER BOOST FAILURE, MEDIUM TRUCK

A summary of test results of all the failure modes tested for the trucks and buses is given in Table 11. Loss of the front brakes on the light truck (the only baseline vehicle tested with a split front/rear system) reduce the deceleration capability by 59% in the empty condition, while rear brake failure caused a reduction of 45% and 62%, empty and loaded, respectively. The amount of braking capability lost due to failure of front or rear brakes is dependent upon several factors, including weight distribution, brake-torque distribution before failure, pedal-force/line-pressure relationships, and brake fade effects. Power boost failure on the light truck reduced the deceleration capability only slightly in the empty condition, and by about 32% in the loaded condition. The loss in braking capability for power boost failure does not reduce the effectiveness of the vehicle's brakes, but requires very large pedal forces on the part of the driver. In the medium truck also, only slight reduction in maximum deceleration capability was noted in the emtpy vehicle, while in the loaded condition it was reduced by 56%.

From the decelerations and stopping distances given in Table 11, it is clear that the hand brake on the heavy truck is not an adequate means of stopping the vehicle in the event of a service-system failure. However, it should be noted that the hand brake on the city bus produces practically the same deceleration as does use of the rear brakes only. Yet, in the loaded condition, the hand brake faded considerably, causing a much longer stopping distance.

Deceleration performance was measured on tractor-trailer combinations in tests conducted at 20 mph in which brake failure was simulated by opening the trailer emergency air line at the tractor protection valve. It was not considered practical to conduct these tests from 60 mph because of the strong possibility that damage to the suspension and brake supporting structure would result from such a severe brake application at high speed. Two of the tractors were equipped with spring brakes. These brakes were also tested from 20 mph. Results obtained for the combinations in the empty and loaded conditions are given in Table 12.

3.4.3 FADE AND RECOVERY TESTS. Results of fade and recovery tests conducted on the trucks are given in Figs. 38 through 40, on the school and intercity bus in Figs. 41 and 42, and the fade test on the city bus in Fig. 43. No recovery test was conducted on the city bus since the vehicle did not have sufficient power to accelerate to test speed rapidly enough (after a fade snub) to prevent the brakes from cooling. The acceleration capabilities of the tractor-trailer combinations were also insufficient to prevent cooling of the brakes between snubs, as indicated in Figs. 44 through 47. A summary of fade test results is given in Table 13.

Using pedal force as a measure, it took 12 snubs at 15 ft/sec^2 deceleration to fade the brakes on the light truck such that the pedal force exceeded 100 lb.

On the medium truck, a pedal force of 250 lb was required to achieve even 12 ft/sec^2 deceleration due to the saturation of the power boost system at the pedal force required to produce about 0.3 g deceleration in the loaded condition. Thus any decrease in brake effectiveness at a deceleration higher
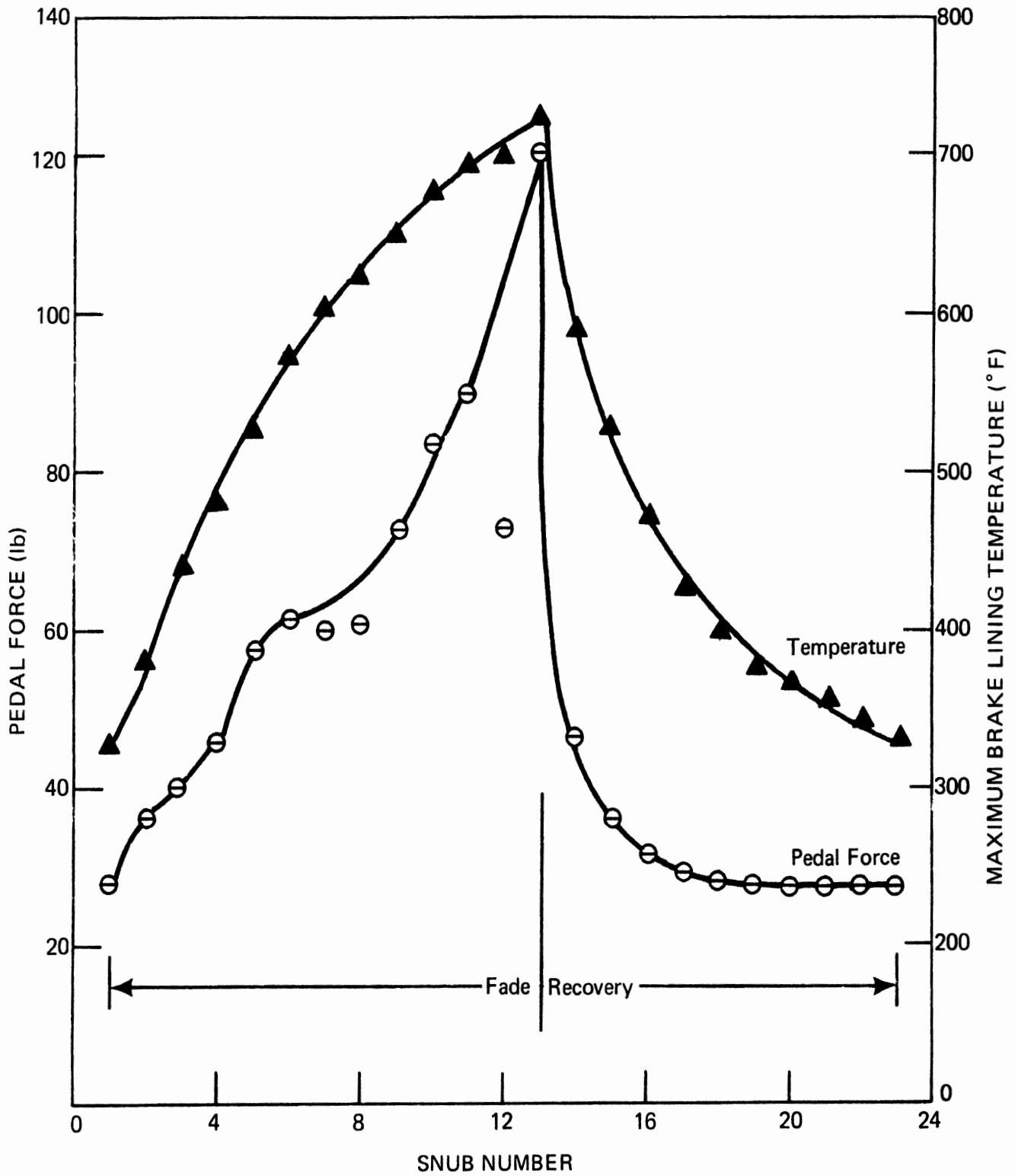
FIGURE 38. FADE AND RECOVERY TEST RESULTS, LIGHT TRUCK
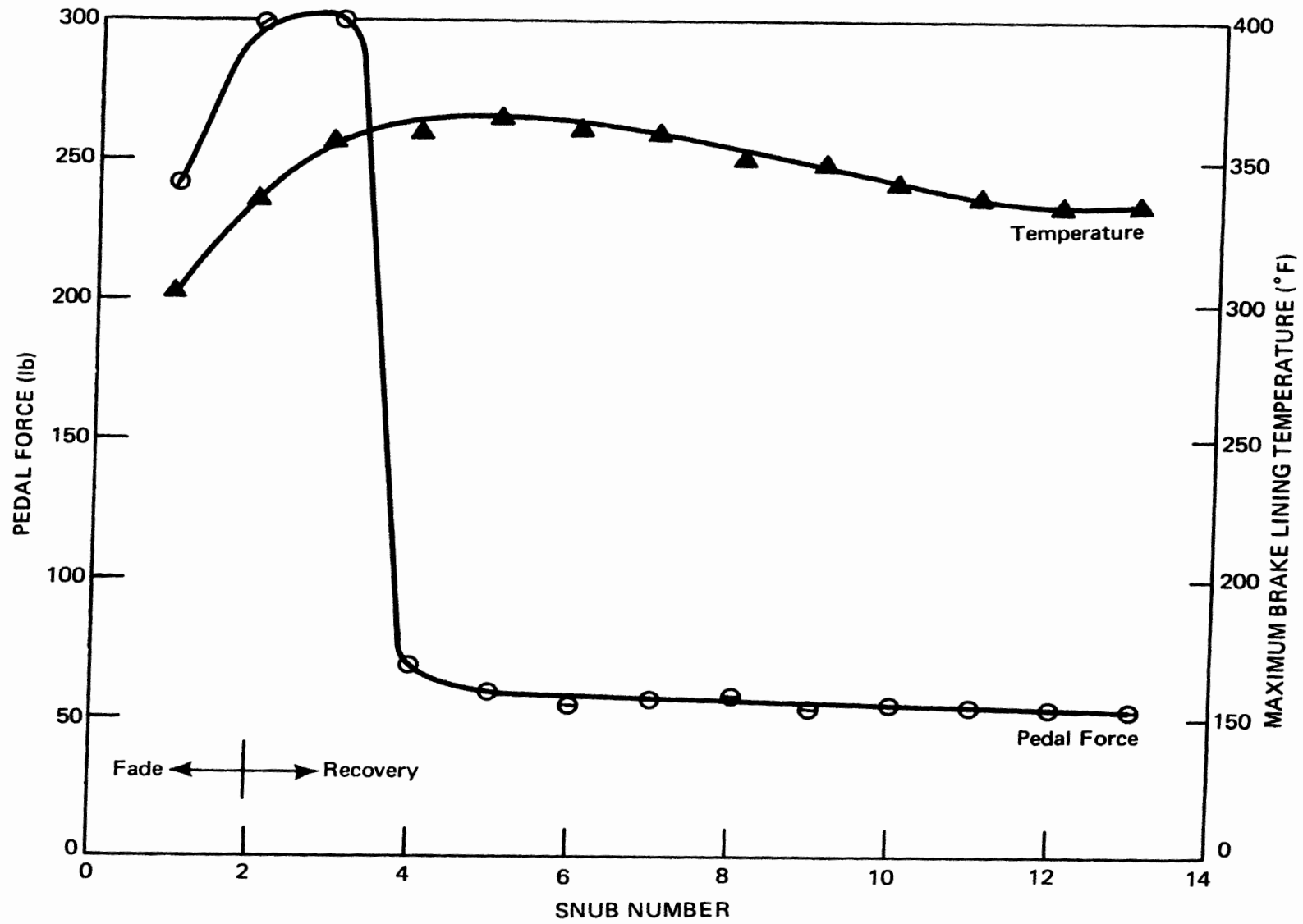
FIGURE 39. FADE AND RECOVERY TEST RESULTS, MEDIUM TRUCK
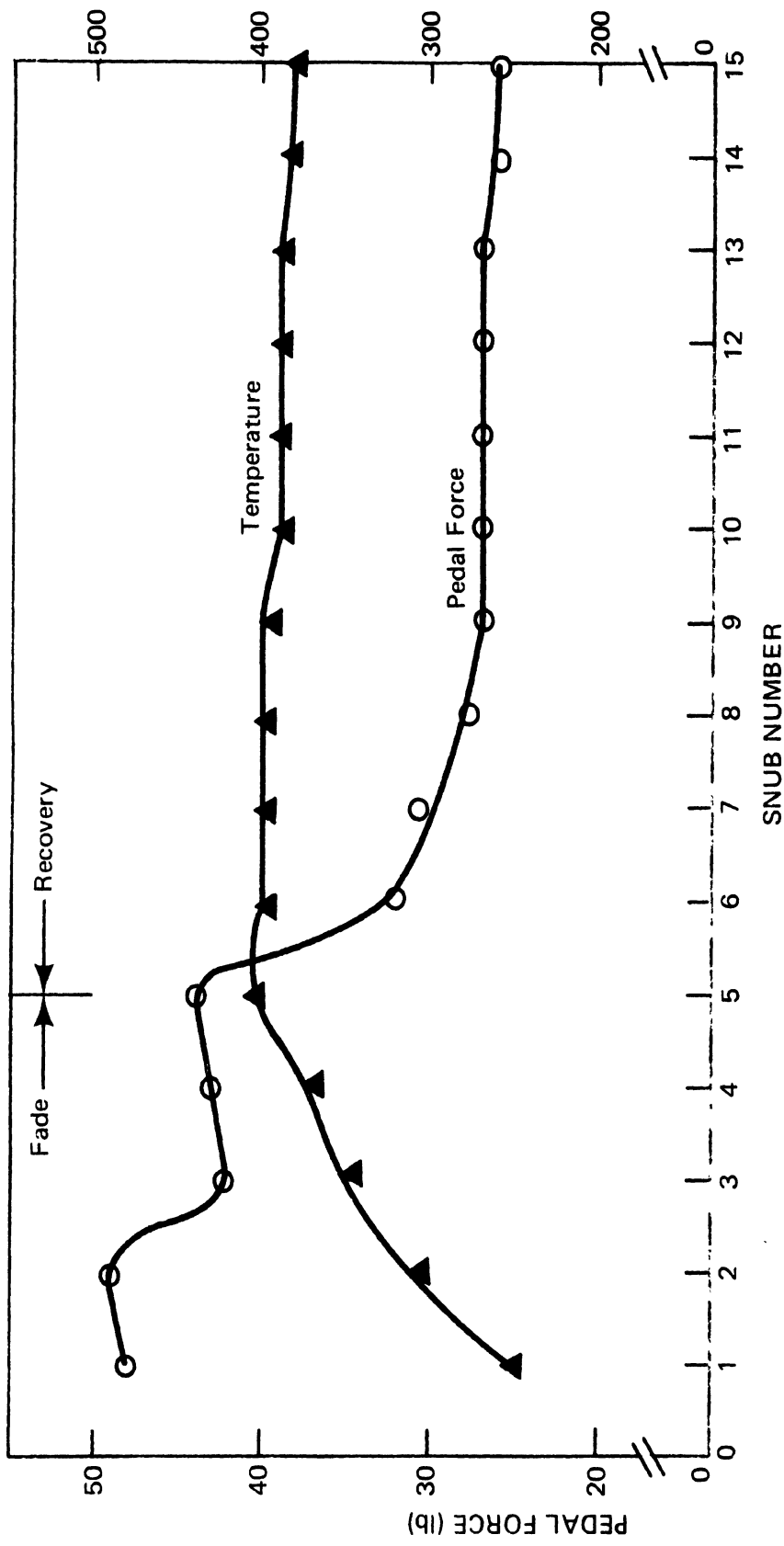
FIGURE 40. FADE AND RECOVERY TEST RESULTS, HEAVY TRUCK
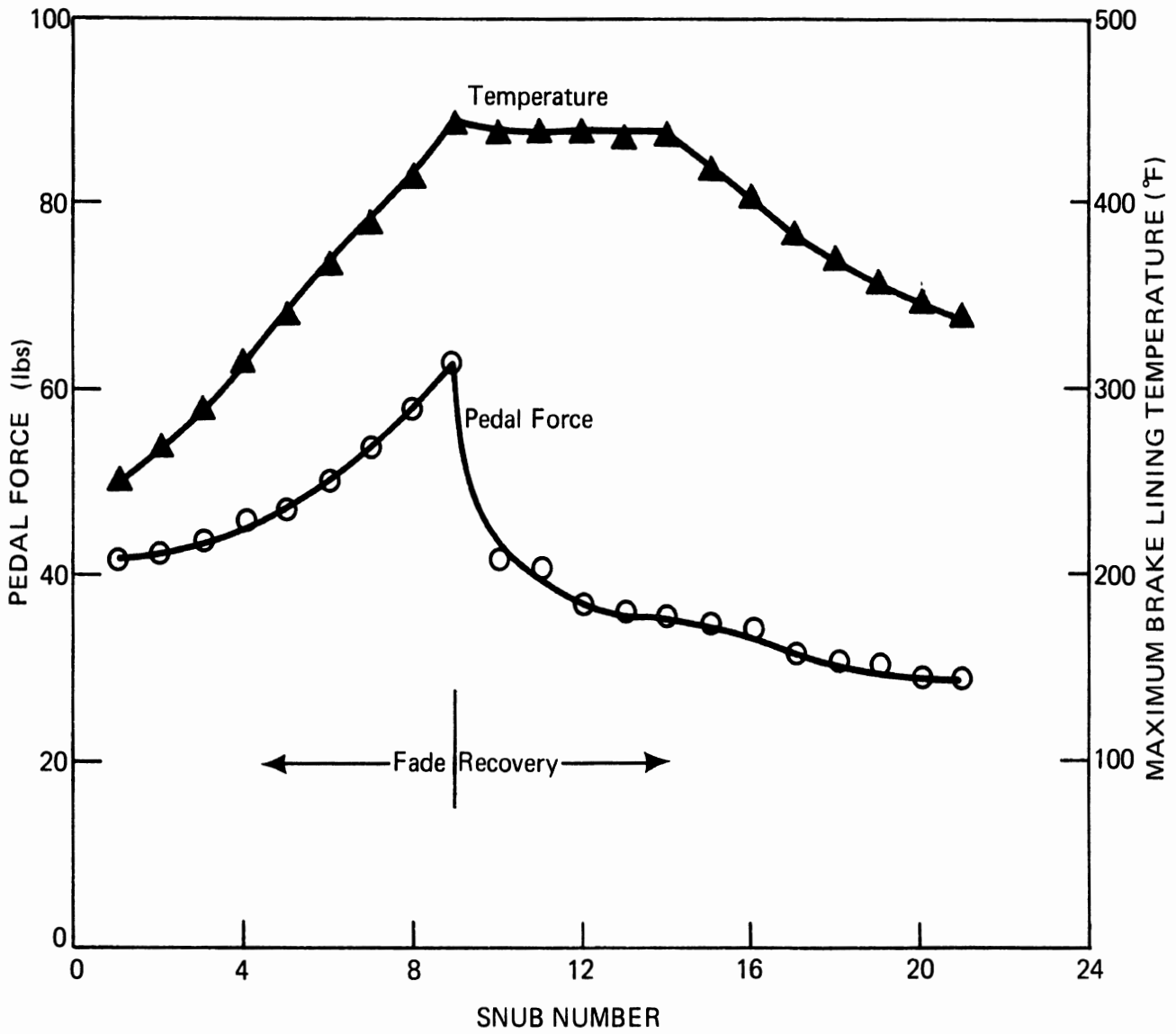
FIGURE 41. FADE AND RECOVERY TEST RESULTS, SCHOOL BUS
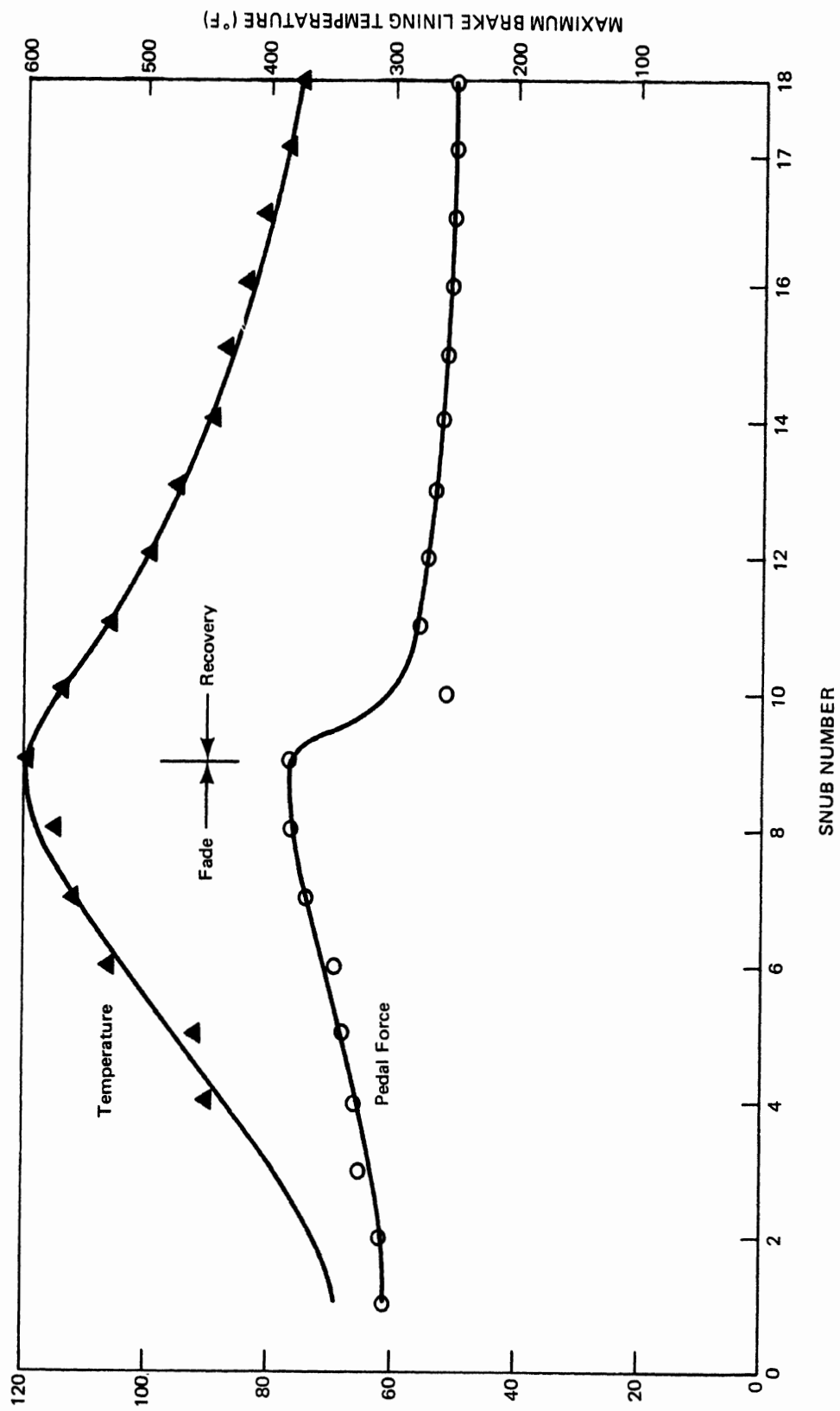
FIGURE 42. FADE AND RECOVERY TEST RESULTS, INTERCITY BUS
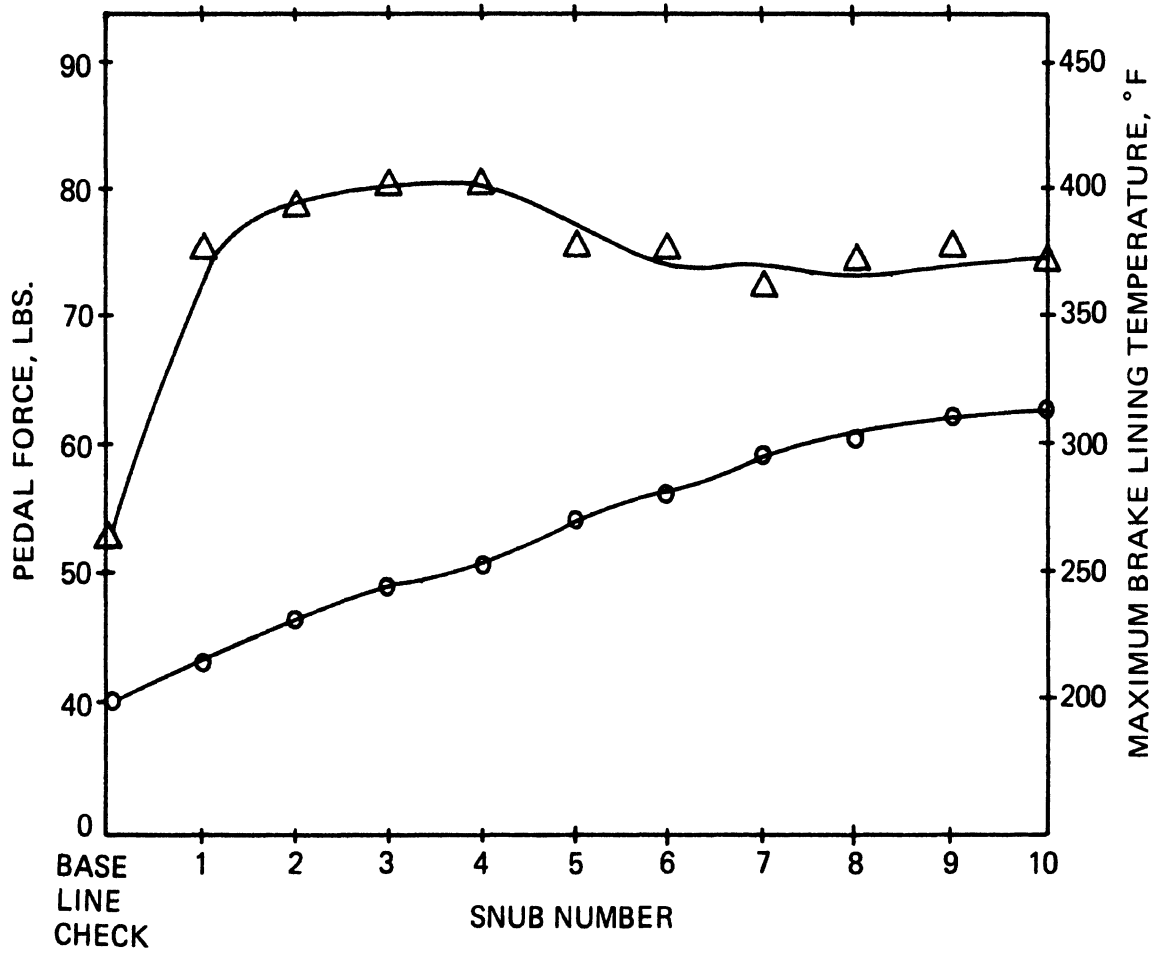
FIGURE 43. FADE TEST RESULTS, CITY BUS
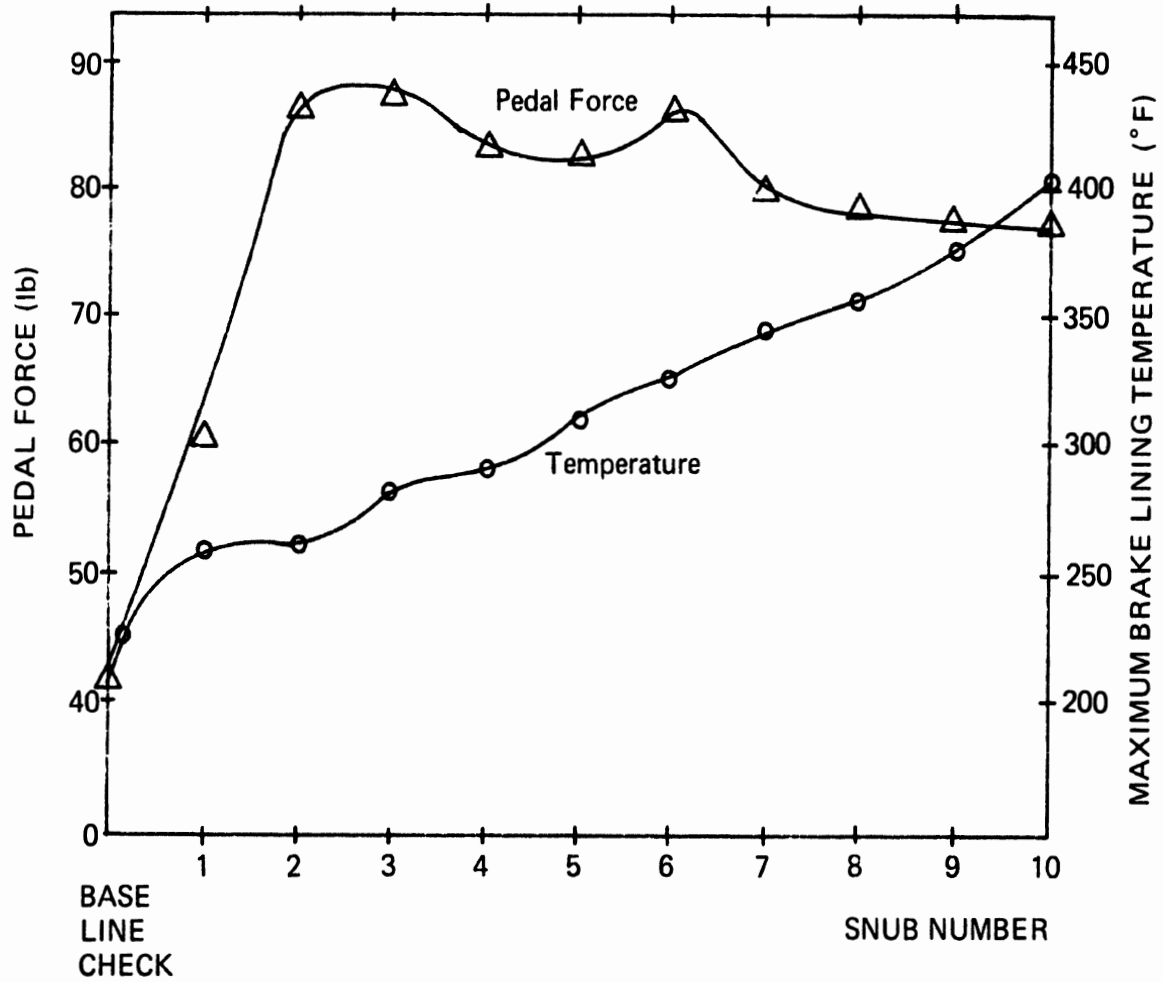
FIGURE 44. FADE TEST RESULTS, TRACTOR-TRAILER 2-S1
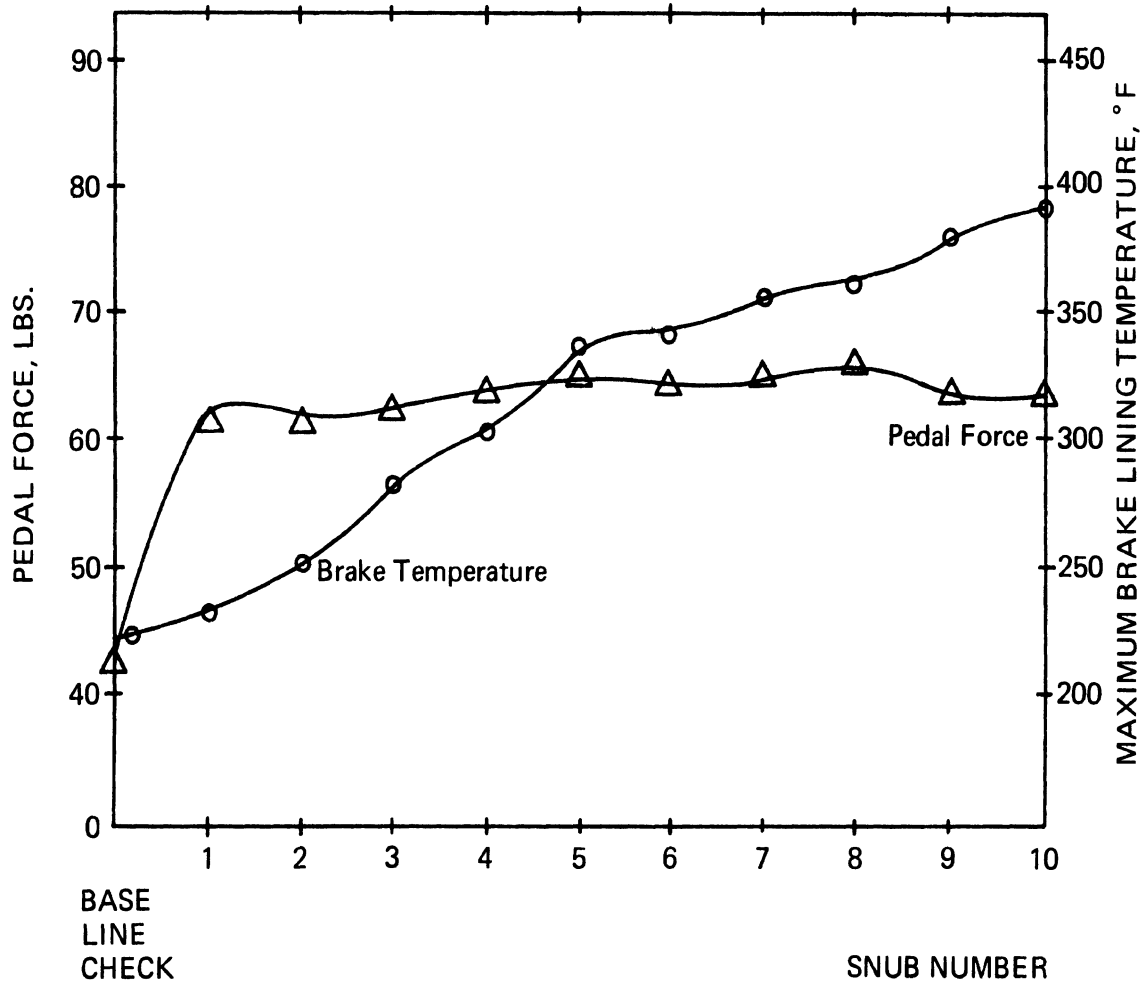
FIGURE 45. FADE TEST RESULTS, TRACTOR-TRAILER 2-S2

99
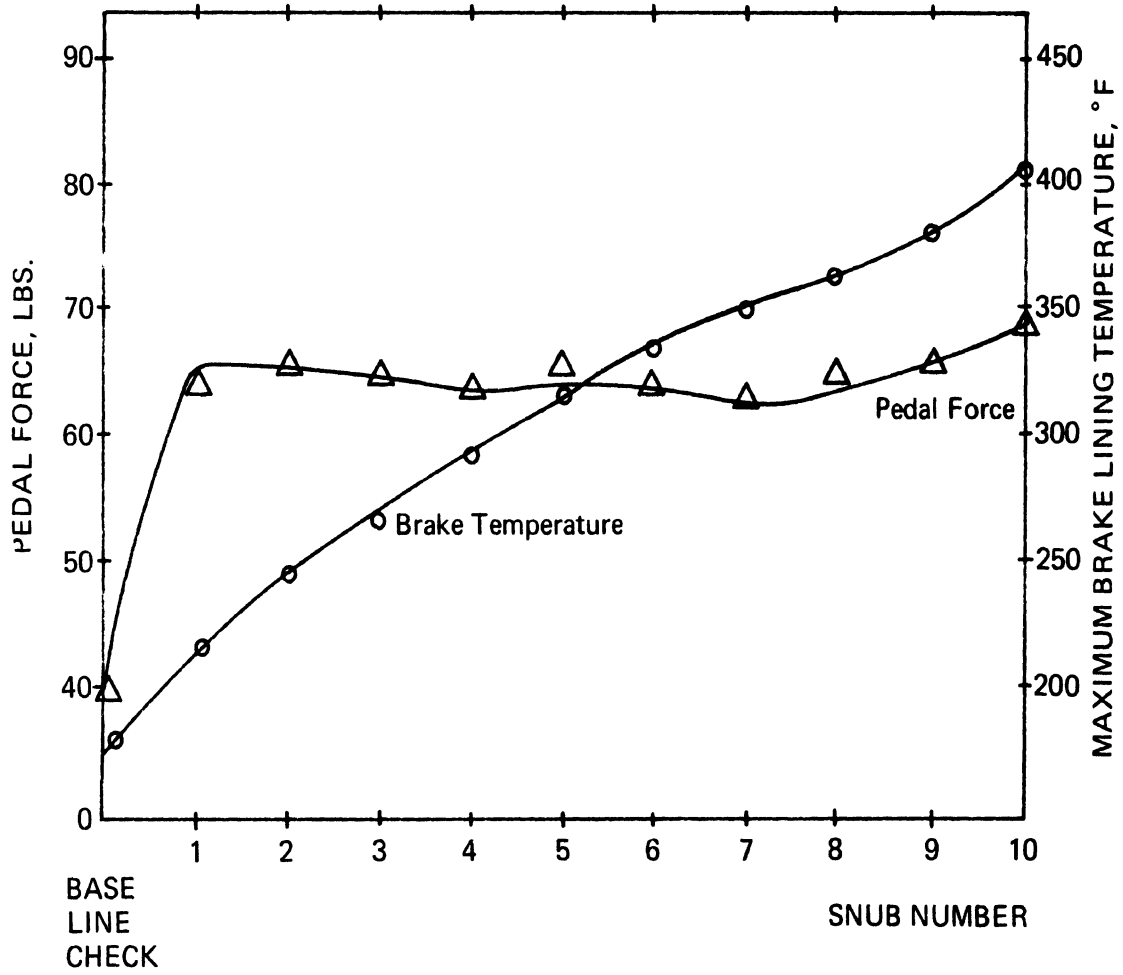
FIGURE 46. FADE TEST RESULTS, TRACTOR-TRAILER 3-S2

100
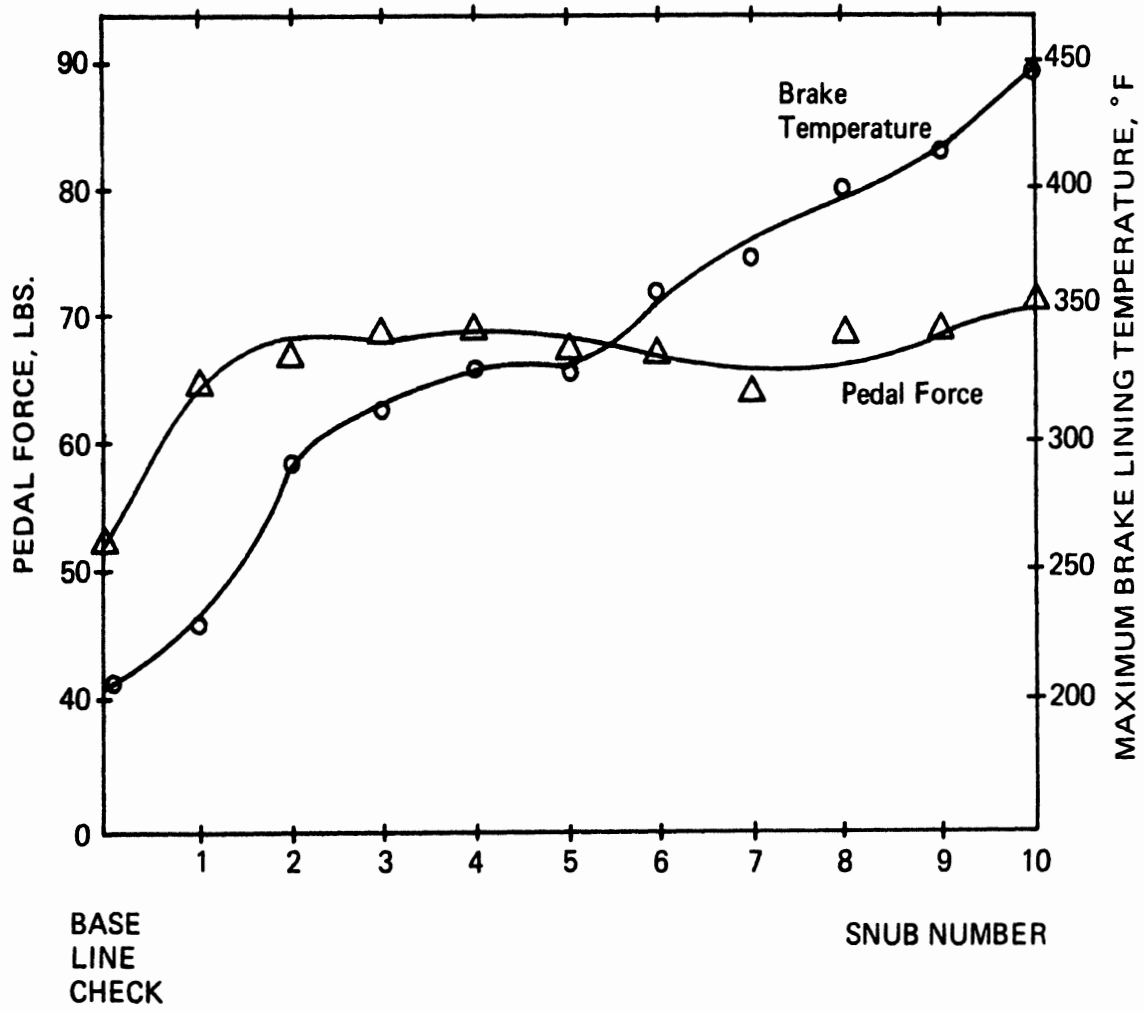
FIGURE 47. FADE TEST RESULTS, DOUBLES COMBINATION 2-S1-2

101

TABLE 11

SUMMARY OF PERFORMANCE OF TRUCKS AND BUSES UNDER FAILURE CONDITIONS

Vehicles	Type of Failure	Brakes Used	V_o, mph	Max. Decel., fps^2		Min. Stopping Dist., ft	
				Empty	Loaded	Empty	Loaded
Trucks							
light	front brakes	rear brakes only	60	11.5*	7.3	380	620
light	rear brakes	front brakes only	60	15.5	9.0	281	484
light	power boost	service brakes	60	26.0*	17.0	160	238
medium	power boost	service brakes	60	13.0*	5.5	352	744
heavy	service brakes	hand brake	60	1.5	1.0	1594	2200
Buses							
school	service brakes	emergency	40	9.2	6.0	188	287
intercity	service brakes	emergency	60	6.8	5.8	571	710
city	service system	hand brake	40	8.1	5.0	212	437
city	rear door opening	rear brakes	40	8.0	5.1	235	350

*Indicates lock-up of one or more wheels.

TABLE 12

SUMMARY OF PERFORMANCE OF TRACTOR-TRAILERS UNDER FAILURE CONDITIONS

Vehicle	Type of failure	Brakes Used	V_o, mph	Loading Condition	Max Decel., ft/sec^2	Stopping Dist, ft	Wheel Lockup
2-S1	opening trailer emergency air line	trailer	20	empty	6.5	94	both trailer wheels
	opening trailer emergency air line	trailer	20	loaded	7.5	85	none
2-S2	opening trailer emergency air line	trailer	20	empty	9.5	70	all trailer wheels
	opening trailer emergency air line	trailer	20	loaded	12.0	45	left tandem, right rear
3-S2	opening trailer emergency air line	trailer	20	empty	7.0	102	all trailer wheels
	opening trailer emergency air line	trailer	20	loaded	9.0	100	left tandem, right tandem
	service system	tractor spring brakes	20	empty	8.0	83	all tractor drive wheels
	service system	tractor spring brakes	20	loaded	6.2	110	none
2-S1-2	opening trailer emergency air line	trailer	20	empty	13.3	48	right front and left rear on full trailer
	opening trailer emergency air line	trailer	20	loaded	7.8	91	none
	service system	tractor spring brakes	20	empty	5.4	100	both rear tractor wheels
	service system	tractor spring brakes	20	loaded	3.2	155	none

TABLE 13

SUMMARY OF FADE TEST RESULTS

Vehicle	Test Decel., ft/sec²	Number of Snubs Achieved	Velocity Initial/Final, sec	Average		Highest Lining Temp., °F	Hottest Brake
				Δt_{2-1}*, sec	Δt_{1-2}**, sec		
Trucks							
light	15	12	60/10	5.9	55	620	left rear
medium	12	3	60/10	7.5	125	335	right rear
heavy	14	5	60/10	5.2	215	405	left tandem
Buses							
school	15	9	40/10	3.8	29	445	left front
inter-city	15	9	60/10	6.3	85	600	left drive
city	15	10	40/10	3.4	48	315	right front
Tractor-Trailers							
2-S1	15	10	50/10	5.6	222	390	left rear tractor
2-S2	15	10	45/10	4.3	105	380	left rear tractor
3-S2	14.5	10	50/10	5.3	137	405	right rear tractor
2-S1-2	12.0	10	50/10	7.0	144	460	left rear tractor

*Δt_{2-1} is the time from initial brake pedal application at initial velocity to brake pedal release at lower snub velocity.

**Δt_{1-2} is the acceleration time from lower snub velocity back to initial test velocity.

than 0.3 g required an inordinate increase in pedal force to maintain the given deceleration. Although the temperature of the brakes did not increase significantly during this test, the slight loss in effectiveness due to the 100 degree temperature rise was enough to increase the pedal force required to maintain the 12 ft/sec^2 deceleration beyond the capability of the driver. During the recovery sequence, the 10 ft/sec^2 deceleration was low enough to allow the power boost to be effective, and the pedal force required was less than 60 lb.

The fade test on the heavy truck was difficult to execute since, in attempting a maximum deceleration of 14 ft/sec^2, the rear wheels locked up on the second snub. At this point the pedal force was decreased slightly, giving a deceleration of 13, 12.5, and 10 ft/sec^2 on the third, fourth, and fifth snubs, respectively. The recovery sequence was started after the fifth snub.

The fade test was successful on the school bus and the intercity bus. Nine snubs were required to fade the brakes on each vehicle.

In conducting the fade test on the city bus and the tractor-trailers, it soon became obvious that the specified procedure would not fade the brakes even after ten snubs. The tests were terminated at this point. The test procedure also proved impractical in that it was not possible to maintain the required 15 ft/sec^2 deceleration on the 3-S2 tractor-trailer combination or on the doubles combination because of incipient wheel lockup.

3.4.4 BRAKE RATING TEST. There are two types of road tests which have been used to rate the horsepower capacity of brakes: a series of fade snubs (such as outlined in SAE J880), and a drawbar dynamometer test in which the vehicle to be rated is towed with its brakes applied. The drawbar test procedure was specified by the NHTSA in this program. The test procedure, described in Appendix B, required that the test vehicle be towed on a flat surface at a constant velocity with brakes applied at a line pressure equivalent to that required to maintain constant vehicle velocity on a seven percent descending grade. The vehicle was to be towed with line pressure held constant until the towbar force decreased (due to fade) by 15%. The original specification called for a test velocity of 40 mph. However, no tow vehicle with sufficient power capacity was available and the velocity for this procedure had to be reduced to the range of 22 to 30 mph. (The only exception was the light truck which was tested at 40 mph because of the relatively small amount of power required.)

The brake rating test was extremely difficult to execute. Although the driver of the towed vehicle had no difficulty maintaining a constant brake-pedal force such that the towbar force initially equalled the sum of the rolling resistance and the specified brake force, other factors (i.e., undulations in the test track, increased resistance in the curves, and steering corrections to keep the towed vehicle in proper alignment with the tow vehicle) caused the towbar force to vary so much that it was difficult to determine just when the brake force had dropped off by 15%. Despite all these factors tending to produce gross inaccuracies in the test data, an attempt was made to average the towing force over the duration of the test, subtract the rolling resistance, and calculate the average braking force, from which the energy

absorption rate of the brake (horsepower) and the total energy absorbed were calculated. Two measures were formulated for rating the brakes. The first measure is based on horsepower and lining area giving some indication of the rate of energy absorption, the second is based upon total energy absorbed and the weight of the vehicle. A summary of test data for the trucks and buses is given in Table 14 and for tractors and trailers in Table 15. If all the tests had been conducted at the same velocity with the brake force, initially at least, equal to seven percent of the vehicle weight, then the horsepower into the brakes could be calculated by:

$$HP = \frac{0.07WV}{550}$$

where

W = weight of the test vehicle, in pounds

V = the test velocity in ft/sec

The total energy absorbed would then be:

$$E_t = 0.07WS_t$$

where

E_t = total energy, in ft-lb

S_t = distance travelled during the test

The energy measure, E/W, would then be merely 0.07S, giving distance travelled during the test as the ultimate measure. However, due to the variations in drawbar force during the test, the velocity variations from test to test, the energy and power ratings, such as they are, seem to be the best measures that can be calculated from the test data. These measures are plotted for each vehicle in Figs. 48 and 49.

Because of the unreliability of these test results, no attempt will be made at this point to assess the brake thermal performance of these vehicles based upon the results of the brake rating test. For example, when the Ford F-7000 tractor was rated in combination with a single-axle trailer (2-S1 combination), Table 15 shows that the tractor brakes held without fading for 195 seconds at 22 mph with a tow force such that the energy rate into the brakes was equivalent to 60 hp. However, when tested again, this same tractor (in the 2-S2 combination) showed that its brakes held without fading for 200 seconds at 30 mph and an energy absorption rate of 150 hp. This anomaly can be explained partially by the fact that before commencement of testing of the 2-S1, the brake burnishing after relining consisted of only 200 stops whereas in the 2-S2, the burnishing after relining consisted of 300 stops. Variation
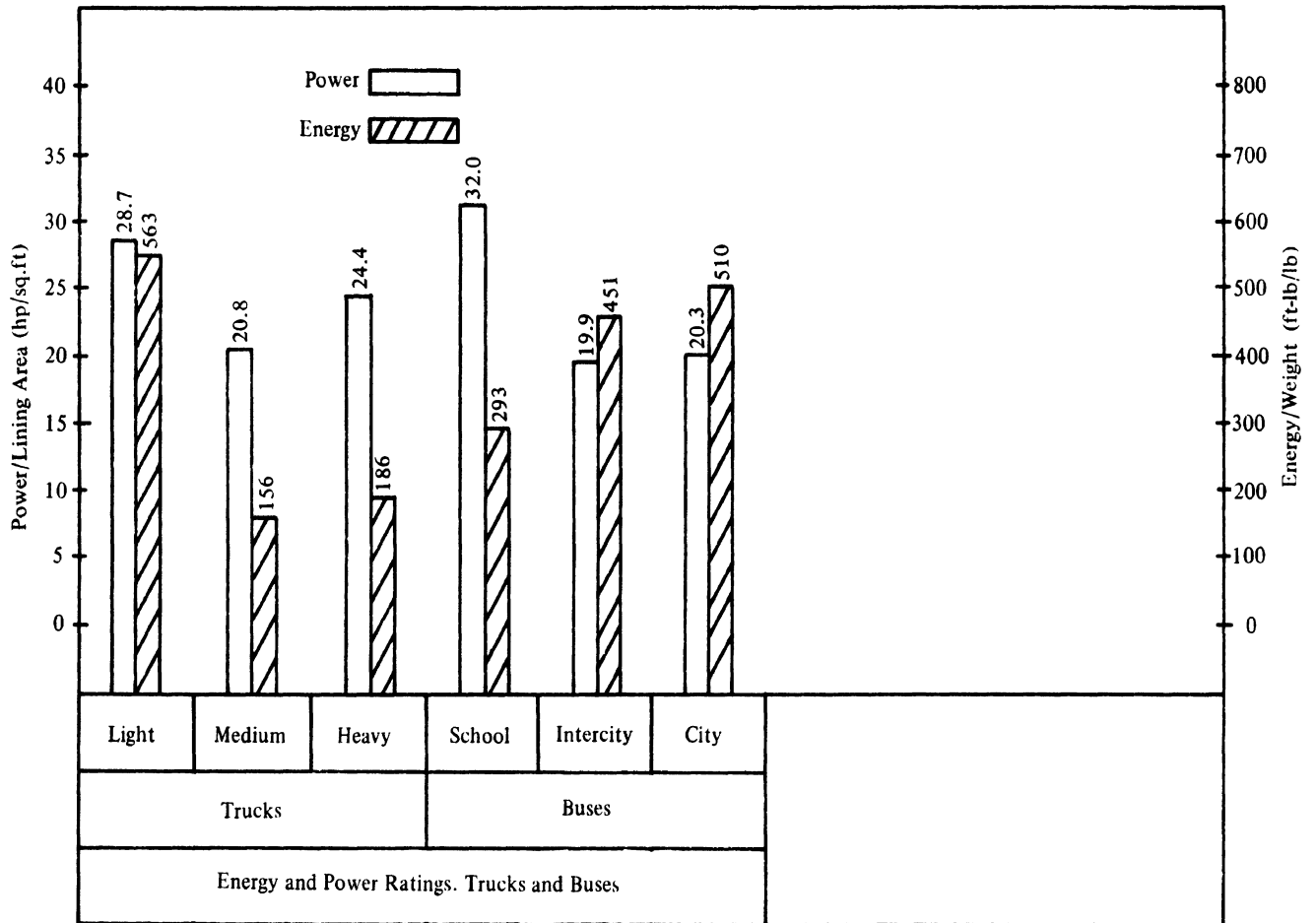
FIGURE 48. ENERGY AND POWER RATINGS—TRUCKS AND BUSES
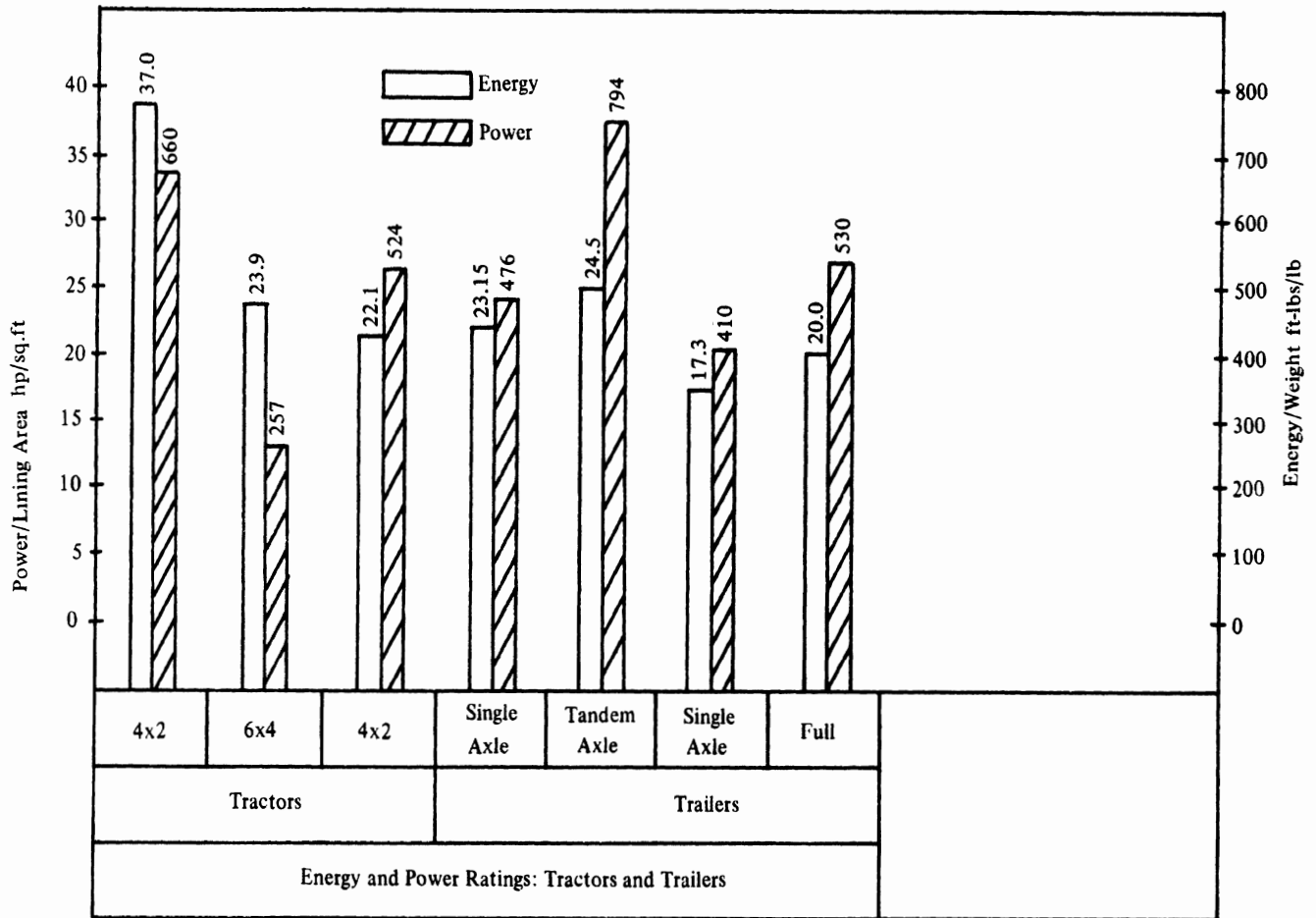
FIGURE 49. ENERGY AND POWER RATINGS—TRACTORS AND TRAILERS

TABLE 14

SUMMARY OF BRAKE RATING TEST RESULTS—TRUCKS AND BUSES

Vehicle	Vehicle Weight, lb	Test Velocity, mph	Test Duration, sec	Average Brake Power, hp	Power/ Lining Area, hp/ft^2	Energy Weight, ft-lb/lb	Max. Brake Temperature, °F	Hottest Brake
Trucks								
light	10,732	45	165	66.5	28.7	563	470	left front
medium	25,500	22	85	85.)	20.8	156	275	right rear
heavy	39,000	23	90	146 0	24.4	186	425	left tandem
Buses								
school	24,500	25	120	109.0	32.0	293	455	left front
inter- city	35,940	23	195	151.0	19.9	451	435	left drive
city	32,145	24	240	124.0	20.3	510	425	left front

TABLE 15

SUMMARY OF BRAKE RATING TEST RESULTS—TRACTOR-TRAILERS

Vehicle	Weight On Test Axles, lb	Test Velocity, mph	Test Duration, sec	Average Brake Power, hp	Power/ Lining Area, hp/ft^2	Energy/ Weight, ft-lb/lb	Max. Brake Temperature, °F	Hottest Brake
2-S1 tractor	23,830	22	195	60	14.8	275	550	right rear
2-S1 trailer	18,010	22	205	78	23.15	476	480	right trailer
2-S2 tractor	24,930	30	200	150	37.0	660	480	right rear
2-S2 trailer	31,990	30	280	165	24.5	794	400	left rear
3-S2 tractor	43,660	21	142	146	23.9	257	405	right rear
3-S2 trailer	31,990	23	238	123	18.3	463	425	right rear
2-S1-2 tractor	20,120	23	240	105.4	22.1	524	435	right rear
2-S1-2 semi-trailer	17,940	23.5	229	58.2	17.3	410	325	left trailer
2-S1-2 full trailer	36,205	23	265	134.5	20.0	530	375	right front

in composition of the lining may also have been a factor. The data for the tandem axle Fruehauf trailer tested in the 2-S2 and 3-S2 configurations show a similar anomaly. Therefore it can be concluded that although the test as specified may produce meaningful results when conducted under ideal conditions, the test when conducted under normal conditions produces results which deviate in excess of normal variation.

3.4.5 BRAKE BALANCE TEST. The brake balance test was used to determine if the brakes of a tractor-trailer combination are balanced properly to axle loads. Table 16 gives, for each tractor-trailer tested, the load distribution, the actual brake distribution as determined from the brake balance test, and the brake distribution as calculated using vehicle and brake system design data.

3.4.6 BRAKE TIMING TESTS. To measure the brake response time of the tractor-trailer combinations pressure transducers were fitted to the output of the brake-control (treadle) valve, and at each axle of the vehicle on which brakes were mounted, except only one transducer was mounted at a tandem axle. Results of the response-time tests both for application and release are given in Figs. 50 through 55 and are summarized in Table 17.

The brake application times shown in Table 17 were measured from the instant pressure started to rise at the output of the treadle valve to the instant at which the pressure reached 60 psi at a given axle.* Release times were measured from the instant pressure began to drop at the output of the treadle valve to the instant at which the pressure at a given axle dropped to 5 psi. These measurements indicate that the average response time for the rear axle of the tractor (when the tractors were tested in combination with a semitrailer) was 0.24 sec. Average response time for the axle on the semi-trailers was 0.28 sec. Release times were considerably longer, averaging 0.55 sec for the rear axle on the tractor and 0.79 sec for the axle on the semitrailers. Application and release times were considerably longer on the doubles combination due to the larger volumes of air that had to be moved through the system.

3.4.7 SYSTEM FAILURES. Several vehicle suspension and brake system failures occurred during the testing of the baseline vehicles. None of these failures was considered extraordinary since the vehicles and systems were subjected to a series of high-level decelerations that generated stresses and cycles of stress not normally encountered in service. Details on these failures are noted below:

Light Truck. During effectiveness tests, loaded vehicles, the brake cylinder in the right rear wheel brake failed, causing contamination of the linings from the brake fluid. The wheel cylinder was replaced, and brakes were relined and burnished before testing was continued. During an effectiveness check of the new brakes, severe wheel hop in the rear brakes occurred with a mild brake application (500 psi line pressure), causing the drive shaft to pull out and drop. This problem was caused by a break in the right rear

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*This definition is slightly different from that given in SAE J982, which specifies that application time be measured "from the start of pedal movement to a pressure buildup of 60 psi."
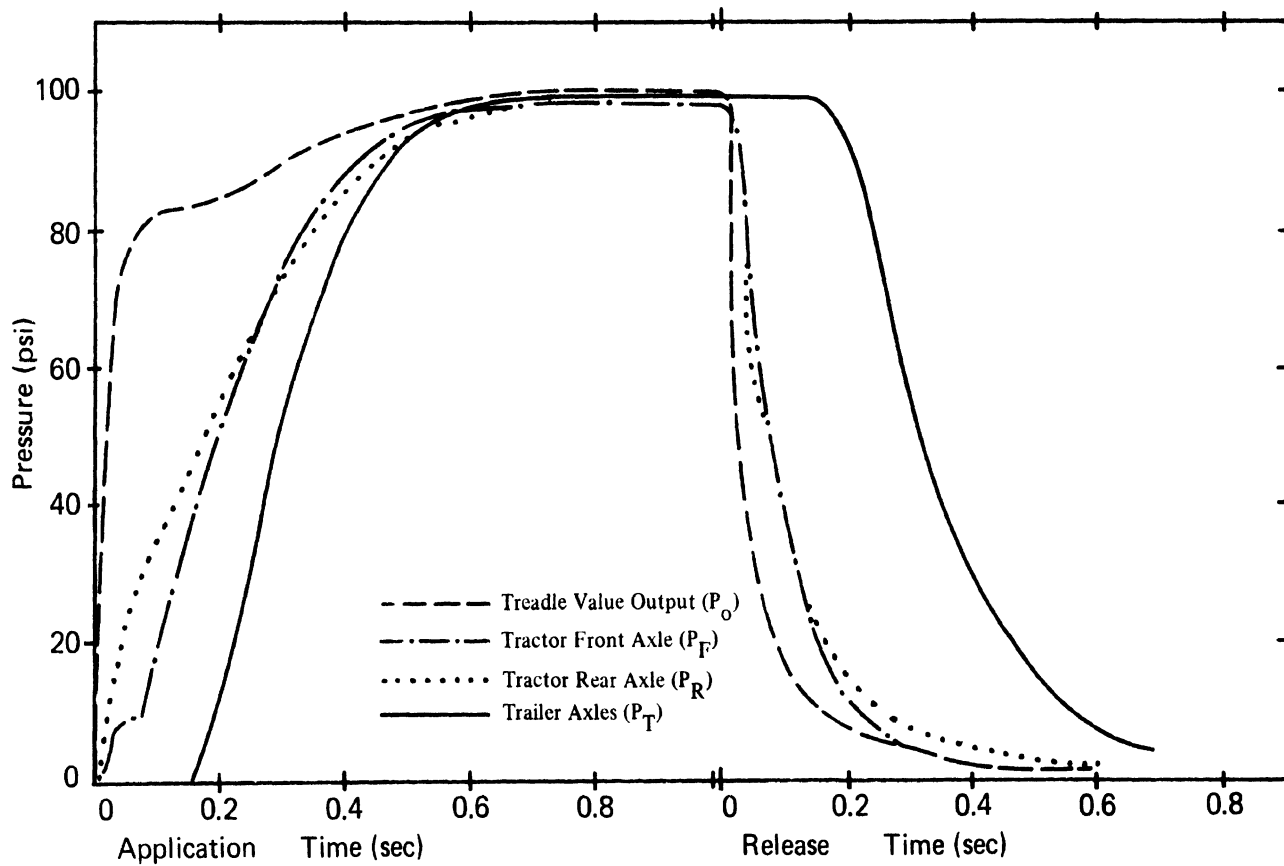
FIGURE 50. BRAKE RESPONSE TIME TEST, TRACTOR-TRAILER 2-S1

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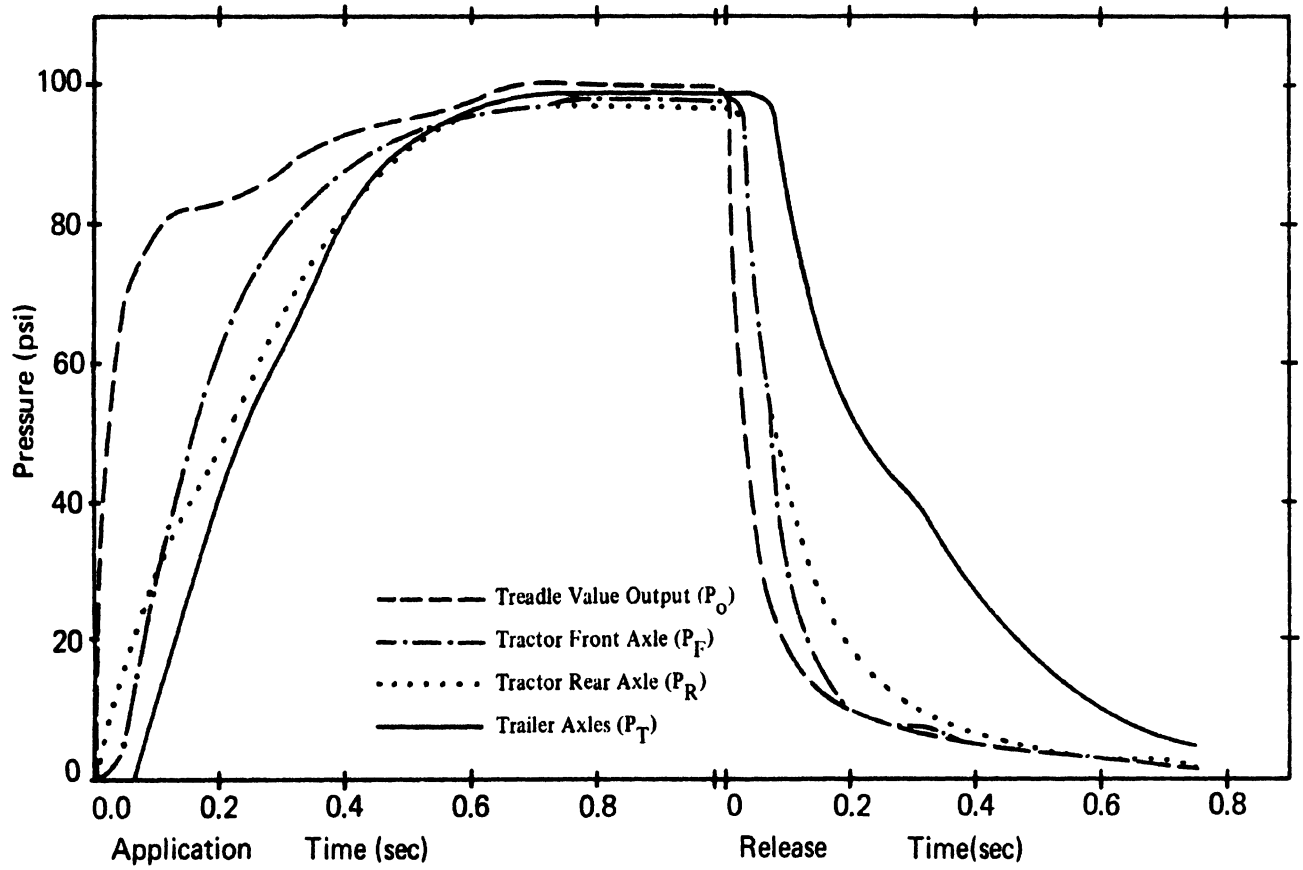
FIGURE 51. BRAKE RESPONSE TIME TEST, TRACTOR-TRAILER 2-S2
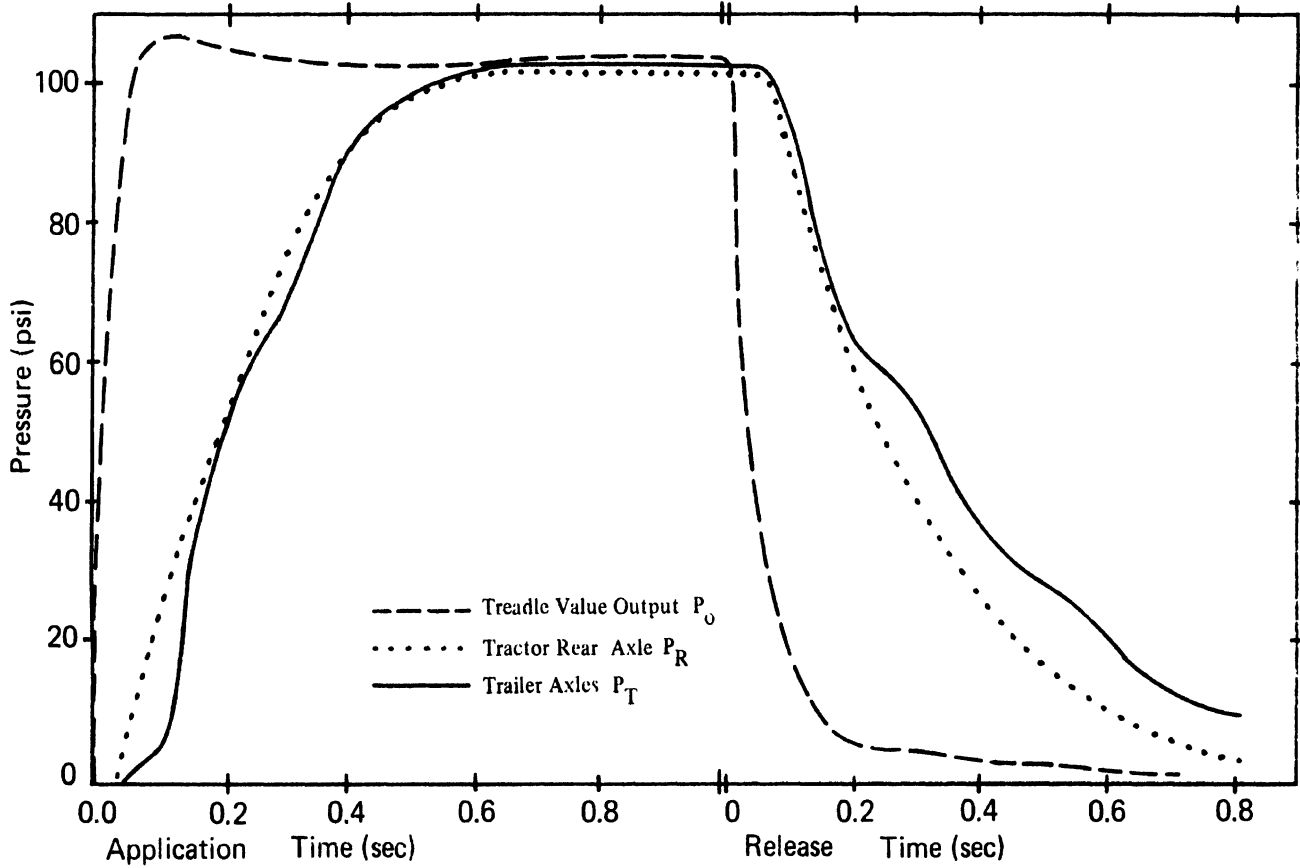
FIGURE 52. BRAKE RESPONSE TIME TEST, TRACTOR-TRAILER 3-S2
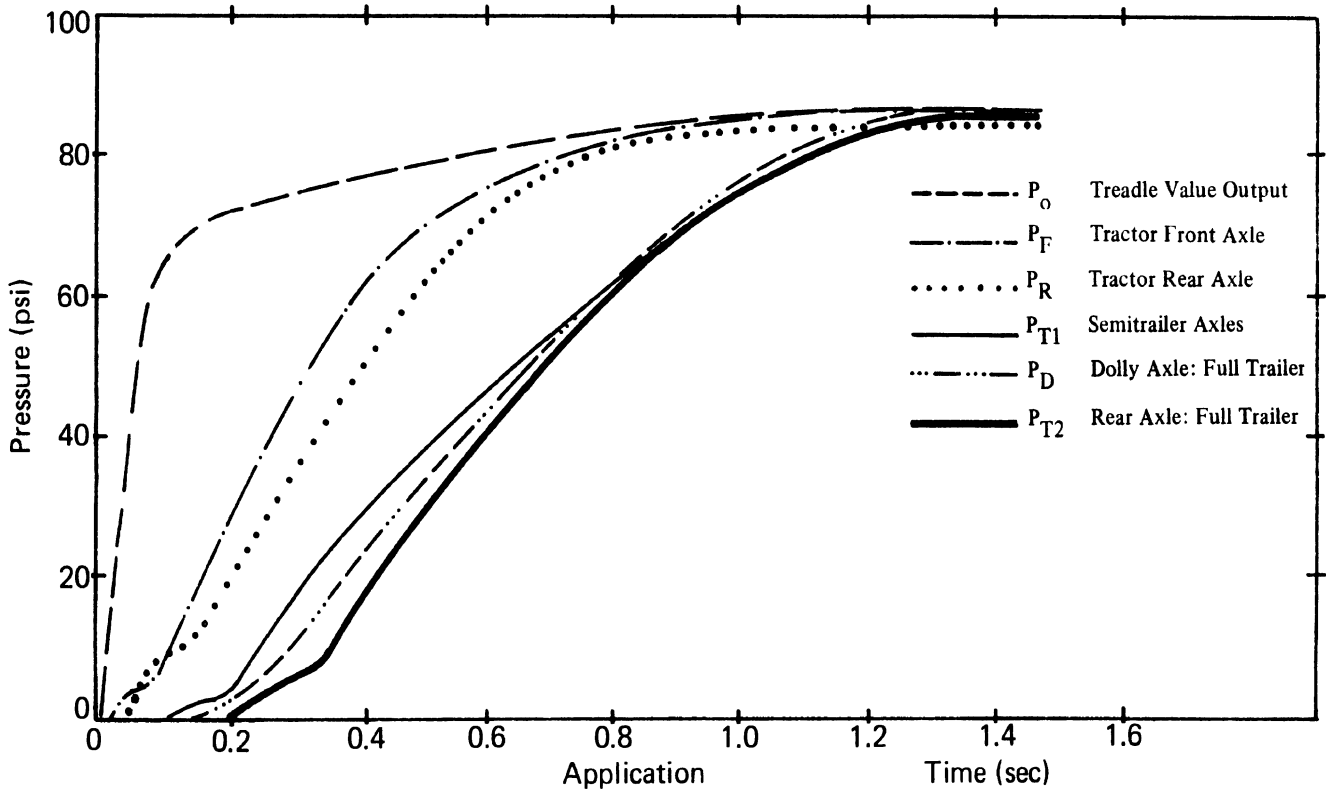
FIGURE 53. BRAKE APPLICATION TEST, DOUBLES COMBINATION 2-S1-2

Press
PSI

— — —	P_o	Treadle Valve Output
— · — · —	P_f	Tractor Front Axle
— · · — · · —	P_R	Tractor Rear Axle
··········	P_{T1}	Semitrailer Axle
————	P_D	Dolly Axle Full Trailer
— · · · —	P_{T2}	Rear Axle: Full Trailer
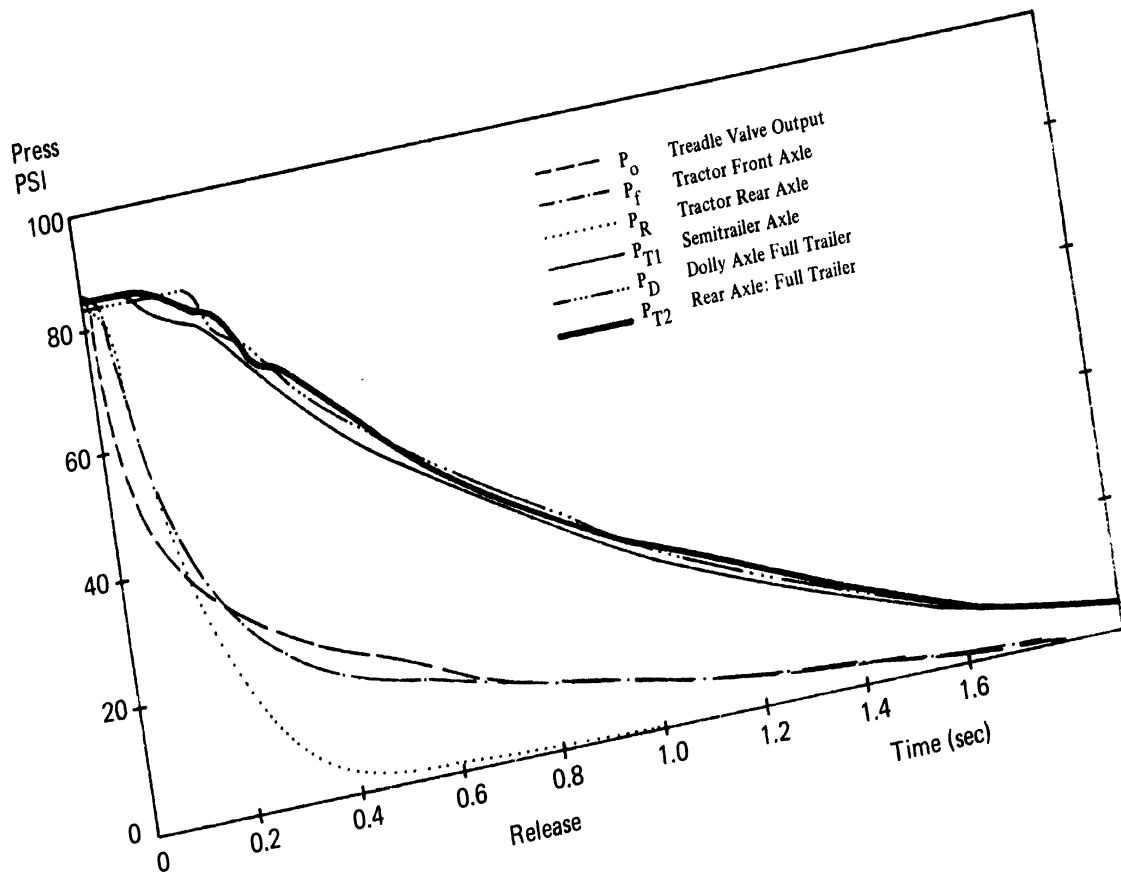
FIGURE 54. BRAKE RELEASE TIME TEST, DOUBLES COMBINATION 2-S1-2

116
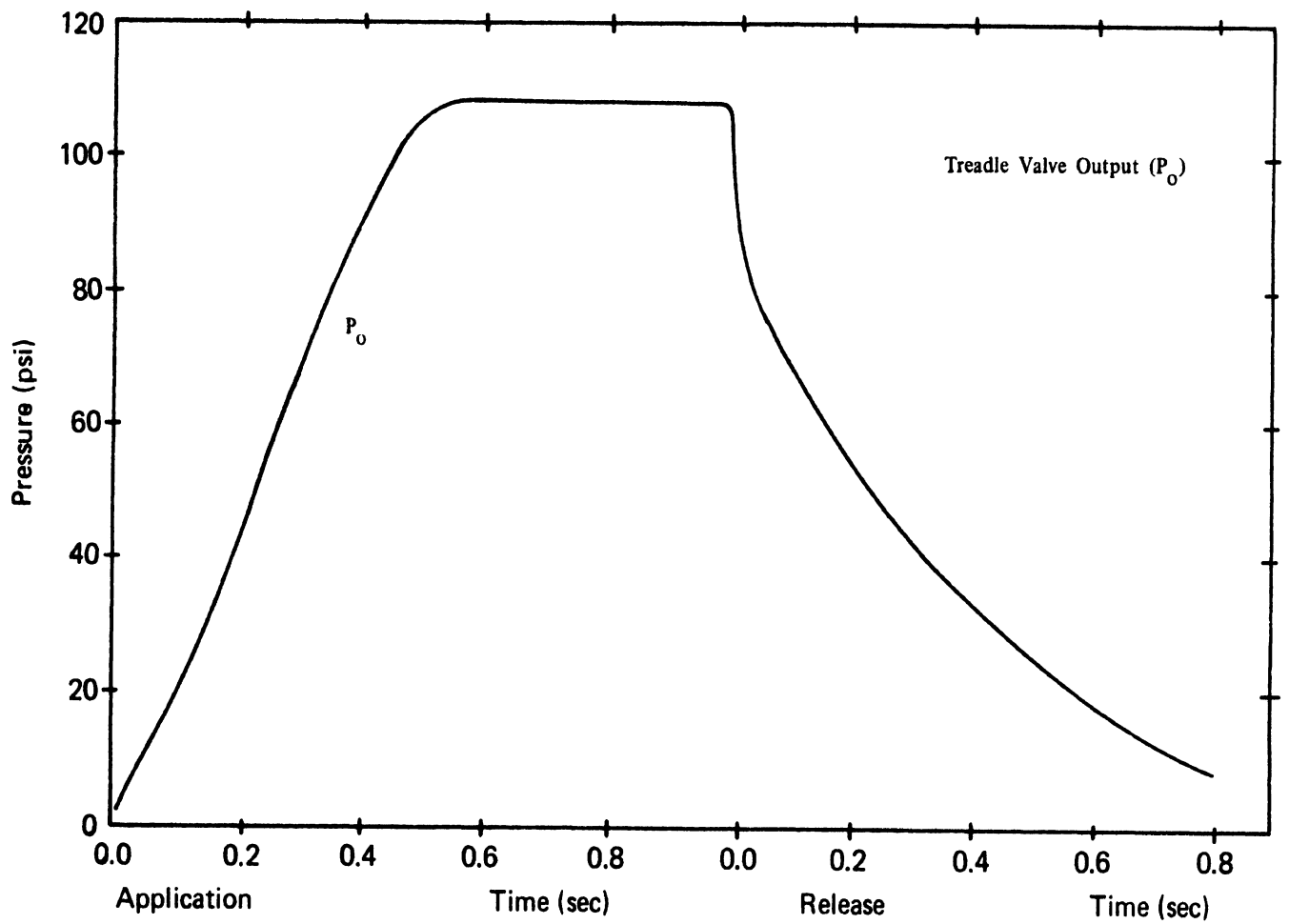
FIGURE 55. BRAKE RESPONSE TIME TEST, CITY BUS

TABLE 16

SUMMARY OF RESULTS FROM BRAKE BALANCE TESTS—TRACTOR-TRAILERS

Vehicle	Weight Distribution (Percent of Total Weight)			Actual Brake Distribution (Percent of Total Brake Force)			Cal. Brake Force Distribution (Percent of Total Brake Force)		
	Tractor	Semi-Trailer	Full Trailer	Tractor	Semi-Trailer	Full Trailer	Tractor	Semi-Trailer	Full Trailer
2-S1	57.0	43.0		62.8	37.2		54.6	45.4	
2-S2	45.2	56.8		50.0	50.0		50.7	49.3	
3-S2	57.7	42.3		47.0	53.0		47.6	52.4	
2-S1-2	35.0	22.0	43.0	41.7	20.4	37.9	35.8	21.4	42.8

TABLE 17

SUMMARY OF AIR BRAKE RESPONSE TIME TESTS—TRACTOR-TRAILER COMBINATIONS

Vehicle	Application Time to 60 psi, sec			Release Time to 5 psi, sec		
	Tractor Front	Tractor Rear	Trailer Axle	Tractor Front	Tractor Rear	Trailer Axle
2-S1	0.24	0.23	0.32	0.30	0.40	0.65
2-S2	0.20	0.26	0.29	0.44	0.50	0.75
3-S2		0.24	0.24		0.75	0.97
2-S1-2	0.40	0.49	0.76	1.30	0.37	1.90
	full trailer front	full trailer rear		full trailer front	full trailer rear	
	0.77	0.79		1.90	1.90	

leaf spring just at the point where it is fastened to the vehicle frame. The vehicle was subsequently repaired and testing successfully concluded.

Ford F-7000 Tractor. Despite the fact that this vehicle was run only 4000 miles prior to delivery to BADC for testing, inspection of the brake drums revealed severe hot-spotting and heat-checking in all the drums. After relining the brakes, new drums were installed. No problems occurred in the preburnish effectiveness test of the tractor-trailer combination (2-S1), but during burnishing, at the 125th stop, the left rear tractor drum cracked. The crack was clearly a tensile failure, extending from the edge of the flange and continuing across the entire rim. The drum was replaced and burnishing continued. No further problems were encountered.

Diamond Reo C11464DF Tractor. This vehicle when received at BADC for testing showed an odometer reading of 191 miles. It was tested in the 3-S2 combination with the Fruehauf trailer. After running the preburnish effectiveness test, burnishing the brakes, a brake balance test was conducted. Following this an effectiveness check from 20 mph was made. On the fourth stop, at maximum pedal force applications, the tractor rear springs pulled out of the tandem axle equalizer, causing the rear driveshaft to be pulled out of place. Repairs were effected immediately and stops were installed to prevent recurrence of the failure. No further problems were encountered in the testing of this vehicle.

3.5 TEST RESULTS—VEHICLES EQUIPPED WITH ADVANCED BRAKING SYSTEMS

The test results for the three vehicles equipped with advanced braking systems are given in this section. These vehicles include:

Vehicle 11. The medium weight truck equipped with full power hydraulic disk brakes

Vehicle 12. The 3-S2 tractor-trailer combination equipped with brake proportioning valves, an adaptive braking system, and a trailer brake synchronization system

Vehicle 14. The 2-S2 tractor-trailer combination equipped with a wheel antilock system.

3.5.1 TRUCK EQUIPPED WITH DISK BRAKES. Specifications for this truck and a description of its brake system are given in Appendix A. This vehicle was subjected to effectiveness tests, brake failure tests, fade tests, the brake rating test, and a static parking brake test.

3.5.1.1 Effectiveness Test. The results of the effectiveness test are given in Fig. 56. These data point up two important factors:

(1) The vehicle has a maximum deceleration capability (with no wheels locked on the given test surface) of better than 21 ft/sec^2, which is better than nine out of the ten baseline vehicles, and considerably better than both the medium and heavy truck.

(2) The vehicle has the capability of locking up all four wheels in the loaded condition, demonstrating that it does indeed have brake torque sufficient to utilize the frictional forces in the tire-road interface to the maximum extent.

3.5.1.2 Brake Failure Tests. With separately powered front and rear
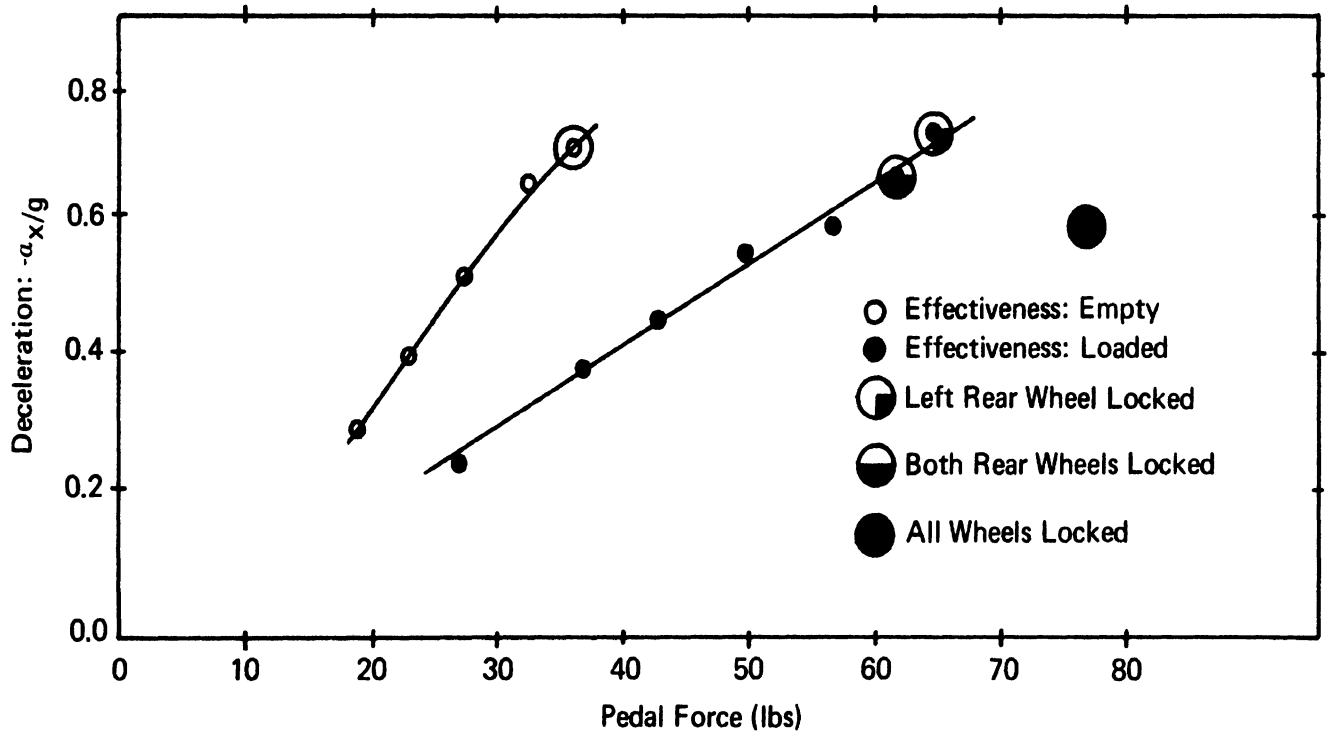
FIGURE 56. EFFECTIVENESS TEST RESULTS, DISK BRAKE TRUCK

brake systems, it was possible to fail either system and maintain braking capability on one axle. Effectiveness test results are given in Fig. 57. Table 18 summarizes the minimum stopping distances and maximum decelerations achieved in the brake effectiveness and brake failure tests. Loss of the rear brakes decreases the maximum deceleration capability of the empty vehicle by 35% and of the loaded vehicle by 43%, while loss of the front brakes decreases the maximum deceleration capability by 47% and 43%, empty and loaded, respectively.

3.5.1.3 Fade Tests. Two fade tests were conducted on this vehicle. The results, shown in Fig. 58, demonstrate the superior fade resistance of disk brakes. In the first test the pedal force required to maintain 15 ft/sec^2 deceleration from 60-10 mph increased from 47 lb on the first snub to 65 lb on the 15th snub. In the second test the pedal force increased from 47 lb on the first snub to 84 lb on the 15th snub. In both cases the brake temperature, as measured by thermocouples in the lining pads, was in excess of 1270°F. To make a fair comparison with the results of the baseline vehicles, it should be noted that for the two tests the average snub deceleration time was 5.8 sec and 5.7 sec and the average time to accelerate back to test speed was 76.4 sec and 81.8 sec.

3.5.1.4 Brake Rating Test. In the brake rating test an average power input to the brakes of 91.3 hp was maintained for 670 sec with less than a 20% decrease in brake force at a constant pedal pressure. The test was terminated after 670 sec because, by this time, the temperatures of all of the brakes exceeded 1200°F. In comparison with the baseline vehicles, this vehicle produced an energy and power rating of 1215 ft-lb of thermal energy absorbed per pound of vehicle weight and 71.4 horsepower for square foot of lining, respectively.

3.5.1.5 Parking Brake Tests. Spring actuators were mounted on each of the rear axle brakes and tested according to the procedure specified in Appendix B. Tested with the brakes at ambient temperature, a towing force of 6750 lb was required to start motion of the truck, which force corresponds to a K-factor* of 0.37. When the brakes were tested at 250°F minimum brake temperature, the force required to start motion increased to 9230 lb, corresponding to a K-factor of 0.5. Once rolling in this condition, however, the average towbar force required to sustain motion against the force of the spring brakes was 3200 lb, corresponding to a K-factor of 0.18. These brakes have the capacity to restrain the vehicle on a 19.5° grade if the brakes were hot, and a 13.8° grade if the brakes were cold, presuming the necessary tire-road friction forces exist to prevent sliding of the rear wheels.

3.5.2 TEST RESULTS, VEHICLE 12.

3.5.2.1 Effectiveness Tests. Results from the five effectiveness tests conducted on the Vehicle 12 tractor-trailer are given in Figs. 59 through 63. The purpose of the first four effectiveness tests was to assess the

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*The K-factor as used here is defined as the force capability of the brake as measured at the tire-road interface divided by the rated static axle load.
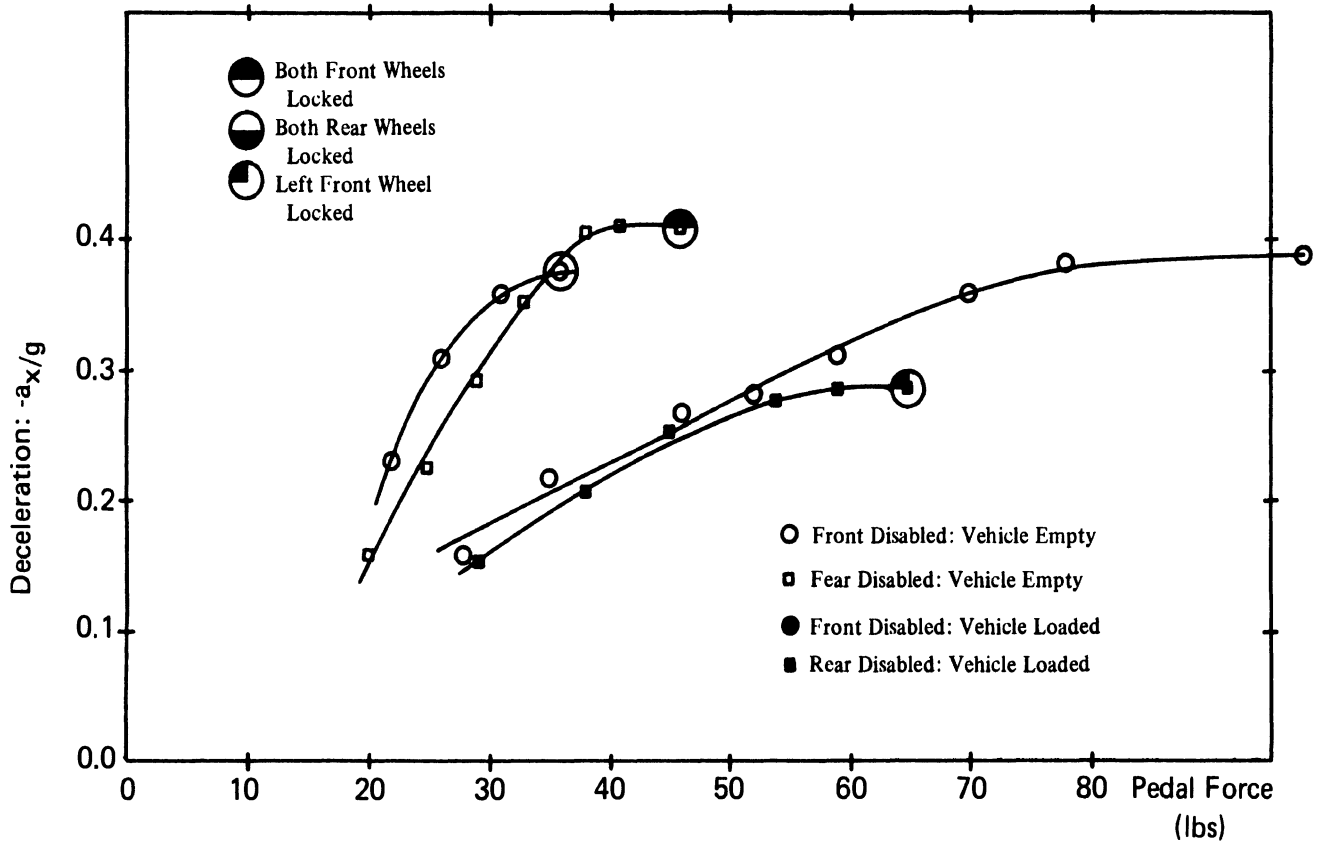
FIGURE 57. EFFECTIVENESS TEST RESULTS, FRONT AND REAR HYDRAULIC LINE FAILURE, DISK BRAKE TRUCK
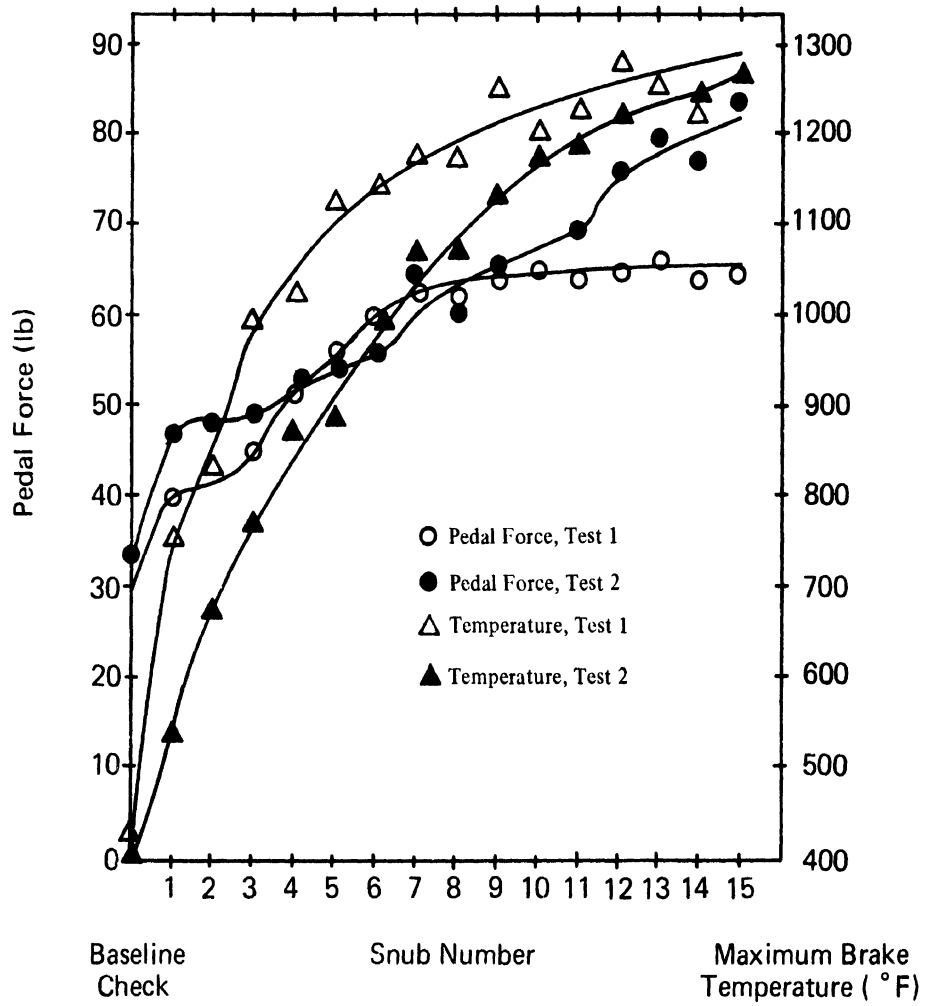
FIGURE 58. FADE TEST RESULTS, DISK BRAKE TRUCK

TABLE 18

EFFECTIVENESS TEST SUMMARY—DISK BRAKE TRUCK (NOMINAL TEST SPEED 60 MPH)

Systems Operational	Minimum Stopping Distance, ft				Maximum Deceleration, ft/sec^2			
	Empty		Loaded		Empty		Loaded	
	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked
Full system	255	198	204	181	21.8	22.8	21.3	23.8
Front system only	321	323	438	436	13.2	13.0	9.1	9.1
Rear system only	351	322	345	---	11.5	12.2	12.4	---

improvement in performance of the empty vehicle with the use of brake proportioning valves, an adaptive braking system, and a combination of antilock and proportioning. The purpose of the fifth test was to determine the performance of the loaded vehicle using the most effective brakes specified for the program.

Figure 59 shows that the standard braking system produces a rather steep pedal force versus deceleration curve for the empty combination indicating a comparatively "sensitive pedal." Wheels lock up on the leading axle of the trailer at decelerations as low as 17 ft/sec^2. Maximum deceleration capability of the vehicle as indicated by this test was 19.6 ft/sec^2. On the last stop of the test, all wheels locked, except the tractor fronts, and the vehicle jackknifed to the limit of the restraining bumper.

The use of proportioning valves decreased the pedal sensitivity, but, as is shown in Fig. 60, wheels started to lockup on the leading axle of the trailer at decelerations as low as 11.8 ft/sec^2. Lockup of wheels on any other axles was prevented by use of the proportioning valves, but at the expense of decreasing the maximum deceleration capability to 15.7 ft/sec^2. It can be shown that load transfer within the two elliptic spring suspension, such as exists on the trailer of this combination, causes the leading axle to be more severely unloaded than the trailing axle (4). Since one valve was used to proportion brakes for both axles, the setting of the valve had to be based on the average load carried by the suspension, which allowed premature lockup of the wheels on the leading axle. If the setting were cut down to prevent this premature lockup, the effectiveness of the trailer brakes would have been further reduced. If proportioning valves are to be used effectively with trailers having such suspensions, some means of preventing premature lockup of the wheels on the leading axle must be employed.

The effectiveness curve obtained with the adaptive system operational is given in Fig. 61. The maximum deceleration achieved is 18.9 ft/sec^2, a deceleration that is 96% of the maximum value obtained with the unaugmented vehicle in stops that included wheel lockups (see Fig. 59). Notwithstanding these moderately high decelerations with the empty vehicle, stability problems were not encountered when the adaptive system was used.

When the adaptive system was operational on the tractor with a proportioning valve used only on the trailer, wheel lock on the trailer leading axle was prevented up to a deceleration of 18.1 ft/sec^2, as is indicated in Fig. 62. For the empty combination, this scheme proved to be almost as effective in preventing wheel lock as the fully adaptive (antilock) system. A maximum deceleration of 19.5 ft/sec^2 was achieved with lockup of 3 wheels on the trailer.

Figure 63 shows the effectiveness curve obtained with the loaded vehicle after installing the most effective brakes specified for the program. When it was discovered that the original hoses did not have adequate capacity to supply air to the larger front-brake chambers fast enough to keep the response time of the front brakes to a reasonable minimum, larger diameter hoses were installed. This change decreased the response time of the front brakes from 0.29 sec to 0.11 sec. Maximum deceleration achieved in this test was 15.1 ft/sec^2 with lockup occurring on some wheels.

3.5.2.2 <u>Minimum Stopping Distance Tests</u>. Minimum stopping distance tests
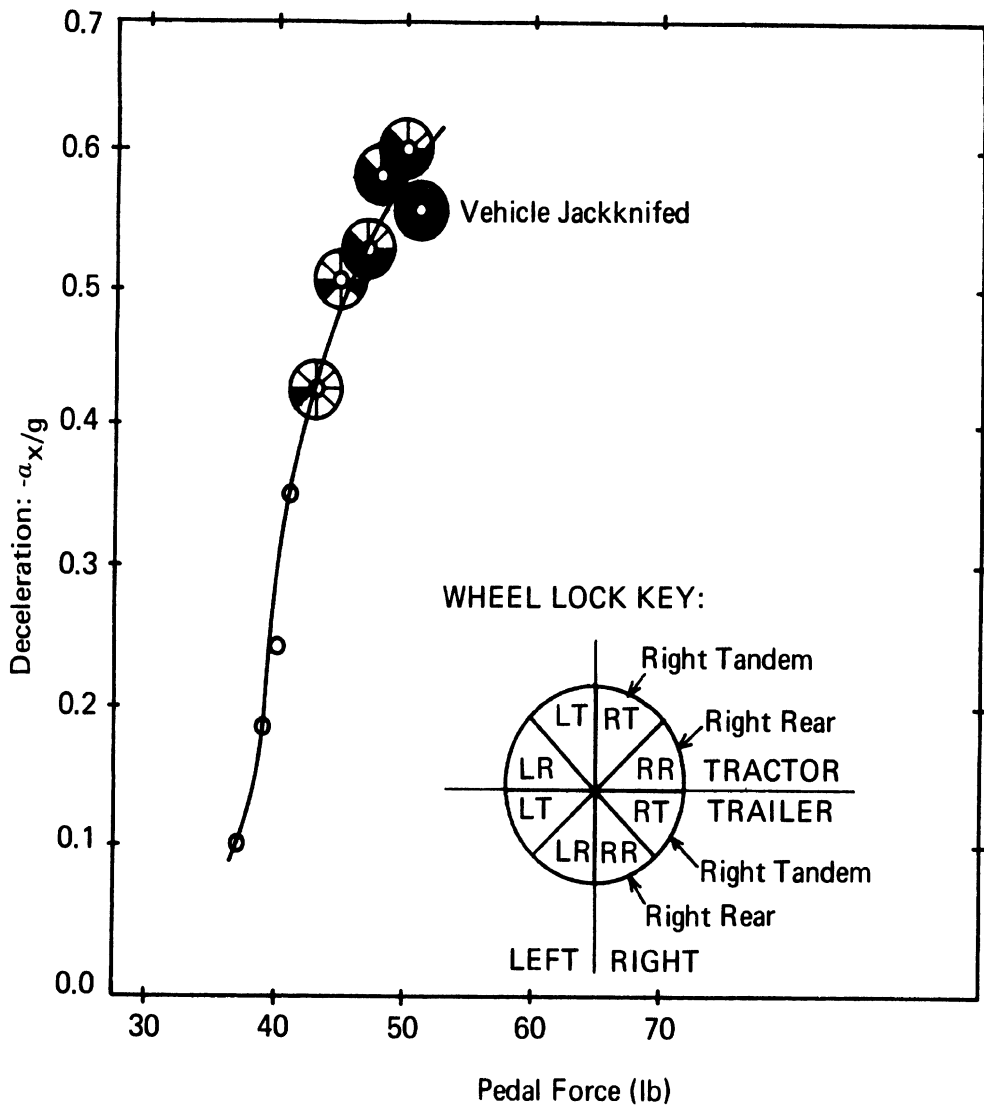
FIGURE 59. EFFECTIVENESS TEST, VEHICLE 12, WITH STANDARD SYSTEM
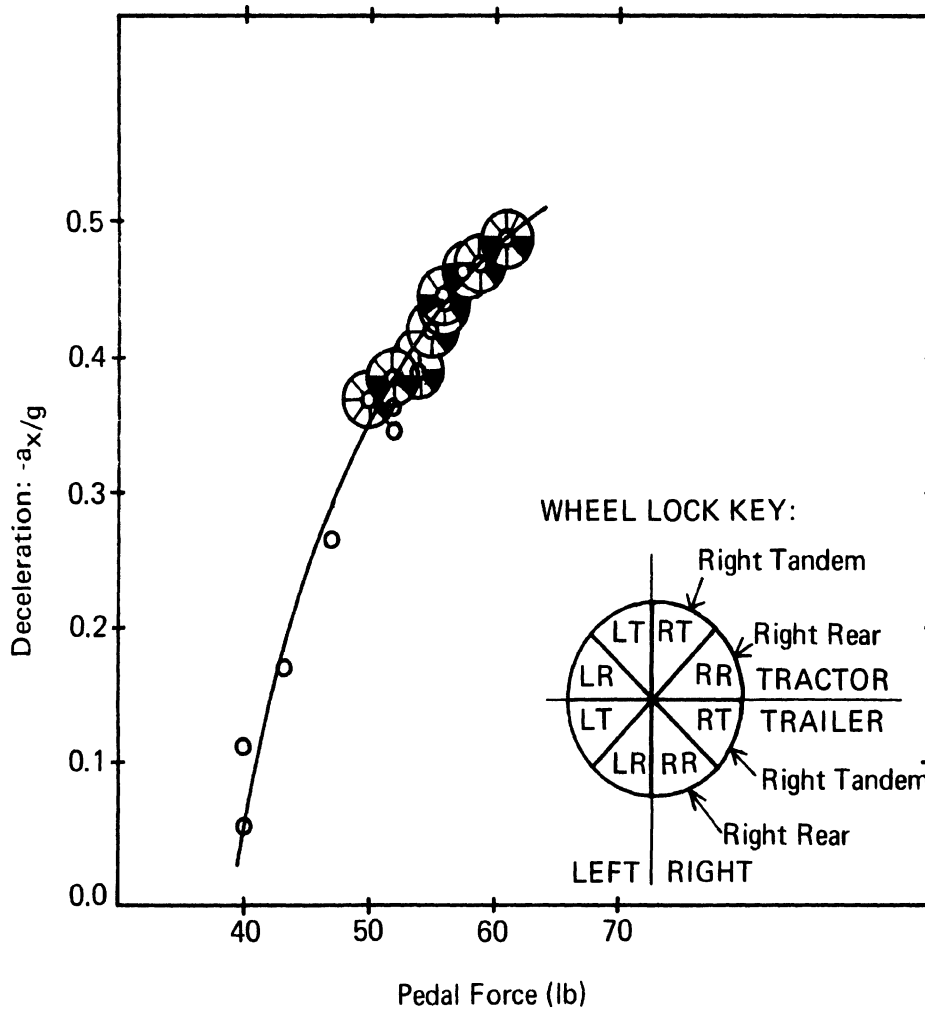
FIGURE 60. EFFECTIVENESS TEST, VEHICLE 12, WITH PROPORTIONING VALVES
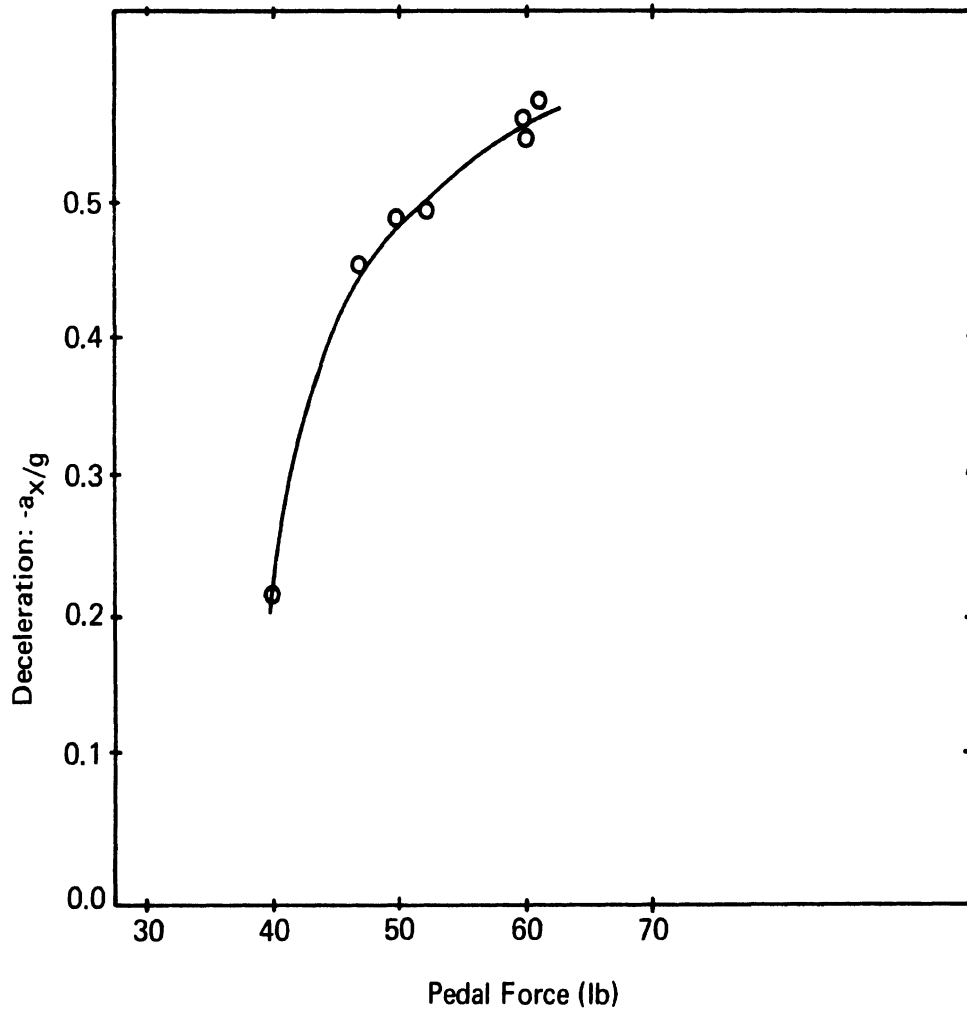
FIGURE 61. EFFECTIVENESS TEST, VEHICLE 12, WITH ADAPTIVE SYSTEM
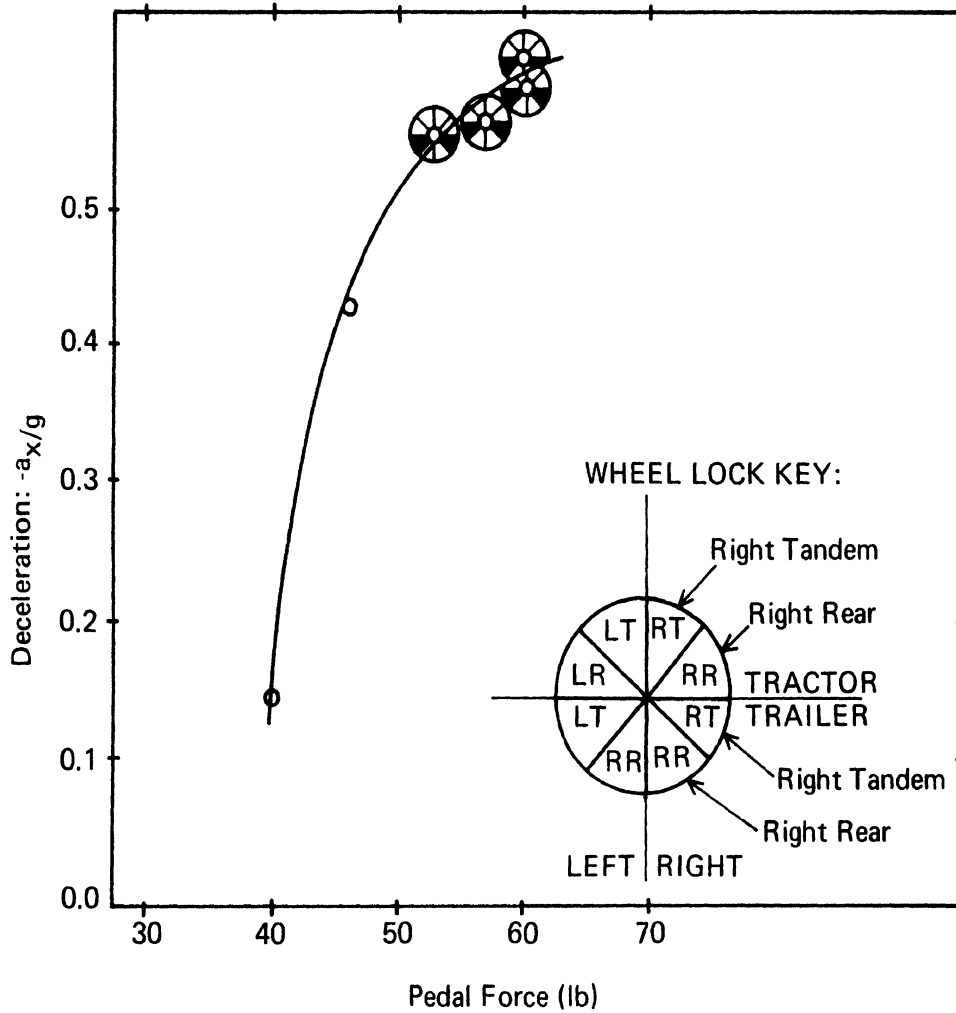
FIGURE 62. EFFECTIVENESS TEST, VEHICLE 12, WITH ADAPTIVE SYSTEM ON TRACTOR AND PROPORTIONING VALVE ON TRAILER

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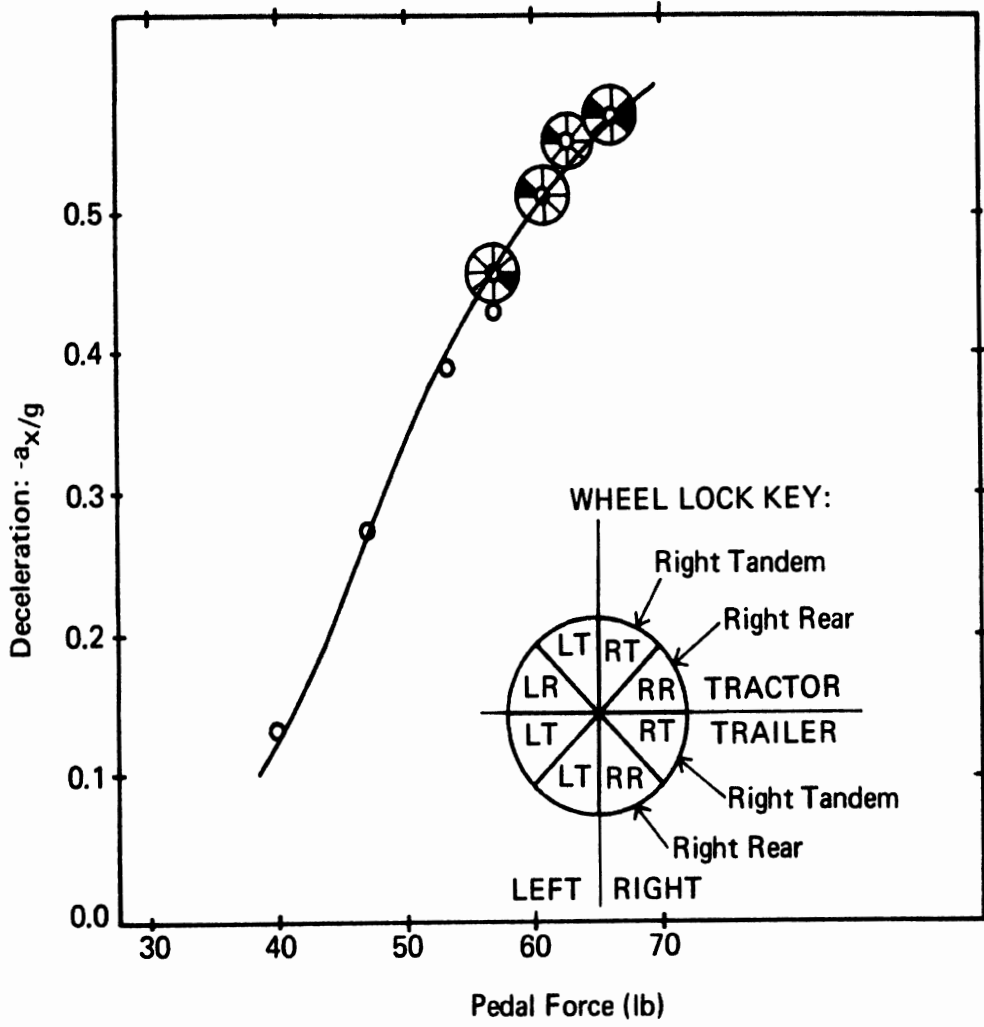
FIGURE 63. EFFECTIVENESS TEST, VEHICLE 12, WITH MOST EFFECTIVE BRAKES

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were run to determine the improvement in braking performance yielded by the various advanced systems and combinations of advanced systems that were employed on Vehicle 12 in a wide variety of test conditions. Tables 19 through 27 provide a complete compilation of the data from these tests. Stopping distance data have been averaged and corrected to represent stops conducted from nominal initial speeds of 60 mph or 20 mph; the results are shown graphically in Figs. 64 through 72.* The designations "baseline," "improved front," and "most effective" brakes used in these tables and graphs are defined in Tables 7 and 29.

Some of the results obtained in effectiveness tests have been included in these figures for comparison purposes. Whenever wheel lockup occurred on an effectiveness test, this occurrence is indicated by an asterisk. Otherwise no wheel lock was permitted in the minimum-stopping-distance tests. The following conclusions can be drawn from an examination of these data:

(1) For the tractor-trailer combination, best performance (as measured both by stopping distance and maximum deceleration) is obtained when the adaptive system is used with the trailer brake synchronization system. This finding holds for both wet and dry surfaces, and for both the empty and loaded vehicles.

(2) Proportioning valves were not as effective on the tractor-trailer combination as would have been the case if premature lockup of the wheels on the leading axle of the trailer had been eliminated.

(3) The dry surface braking performance of the single-element tractor, both loaded and in the bobtail configuration, was improved to an equal degree by use of the proportioning valve or the adaptive system. On the low coefficient surface, however, the deceleration performance of the empty tractor was severely degraded by the proportioning valve, which set the brake-line pressure ratio (front to rear) at 5 to 1. In these latter tests, the line pressure had to be kept below 30 psi to prevent the front wheels from locking on the low coefficient surface. Since the proportioning valve kept the pressures

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*In certain loading configurations, the vehicles could not attain 60 mph on the test track for the high-speed tests. Speed on the wet skid pad surface was limited to 40 mph for most of the tests. In order to compare data, all stopping distance values for the high-speed tests were corrected to an initial speed of 60 mph, using the following equation:

$$S_{60} = S_1 \frac{3600}{V_1^2}$$

where

V_1 = actual test velocity, mph

S_2 = actual stopping distance, ft

S_{60} = stopping distance, ft, corrected to 60 mph initial speed.
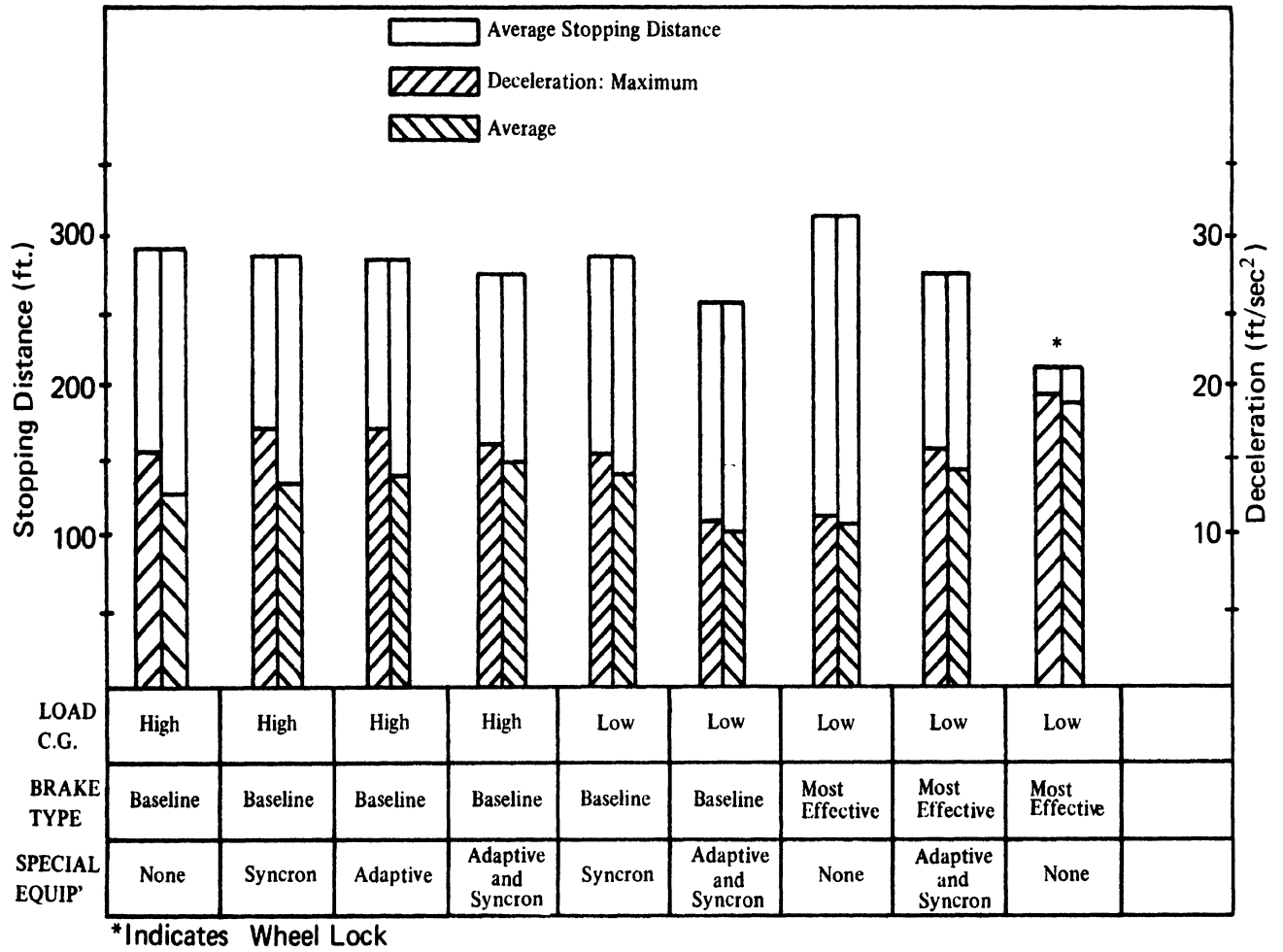
FIGURE 64. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR-TRAILER LOADED, DRY TRACK, FROM 60 MPH
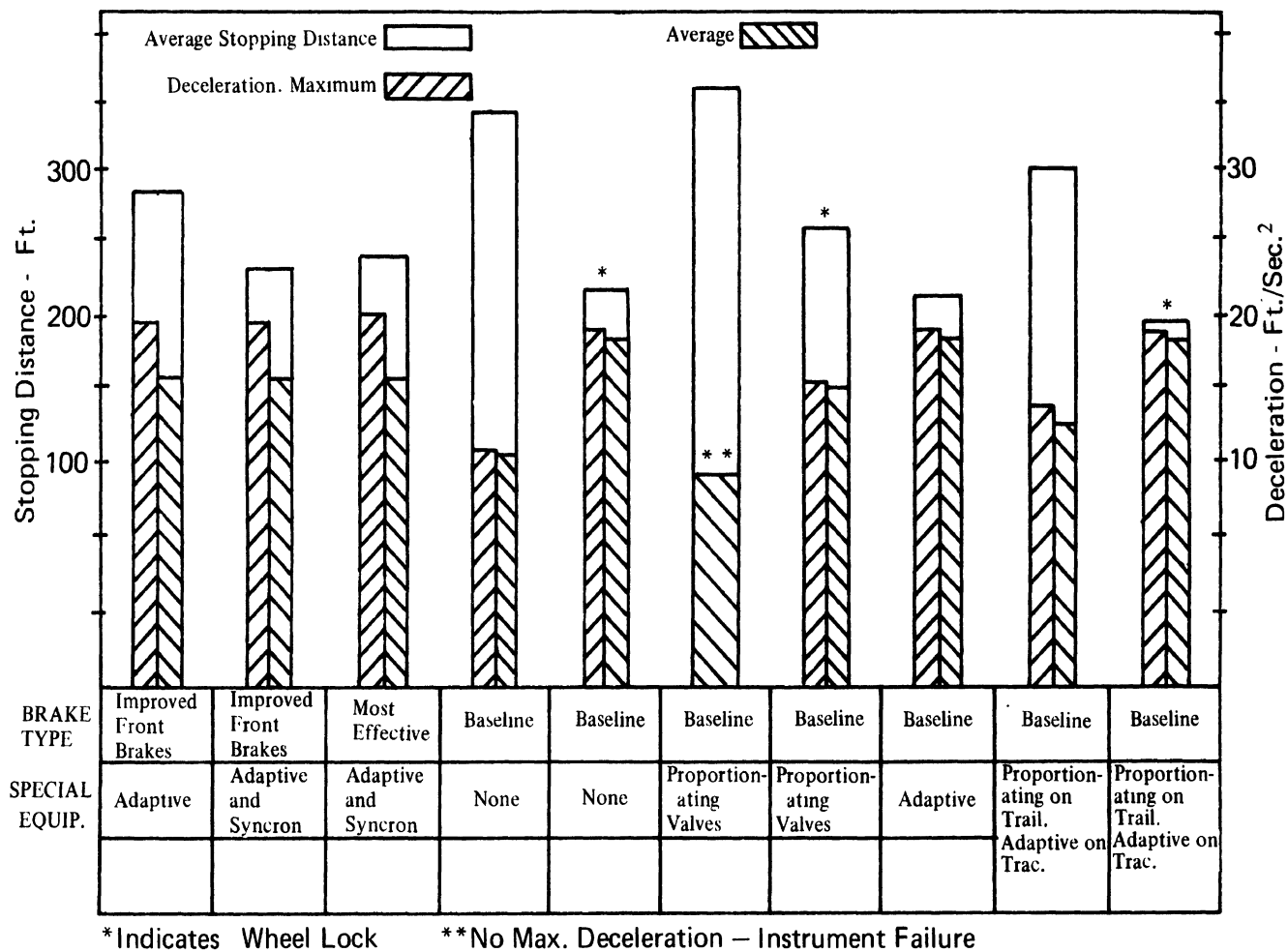
FIGURE 65. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR-TRAILER, EMPTY, DRY TRACK, FROM 60 MPH

Legend:
- Average Stopping Distance
- Deceleration: Maximum
- Average

LOAD C.G.	High	High	High	High	High	Low	Empty
BRAKE TYPE	Baseline	Baseline	Baseline	Baseline	Adaptive	Most Effective	Most Effective
SPECIAL EQUIP.	None	Syncron	Adaptive	Adaptive and Syncron	Improved Front Brakes	Adaptive and Syncron	Adaptive and Syncron
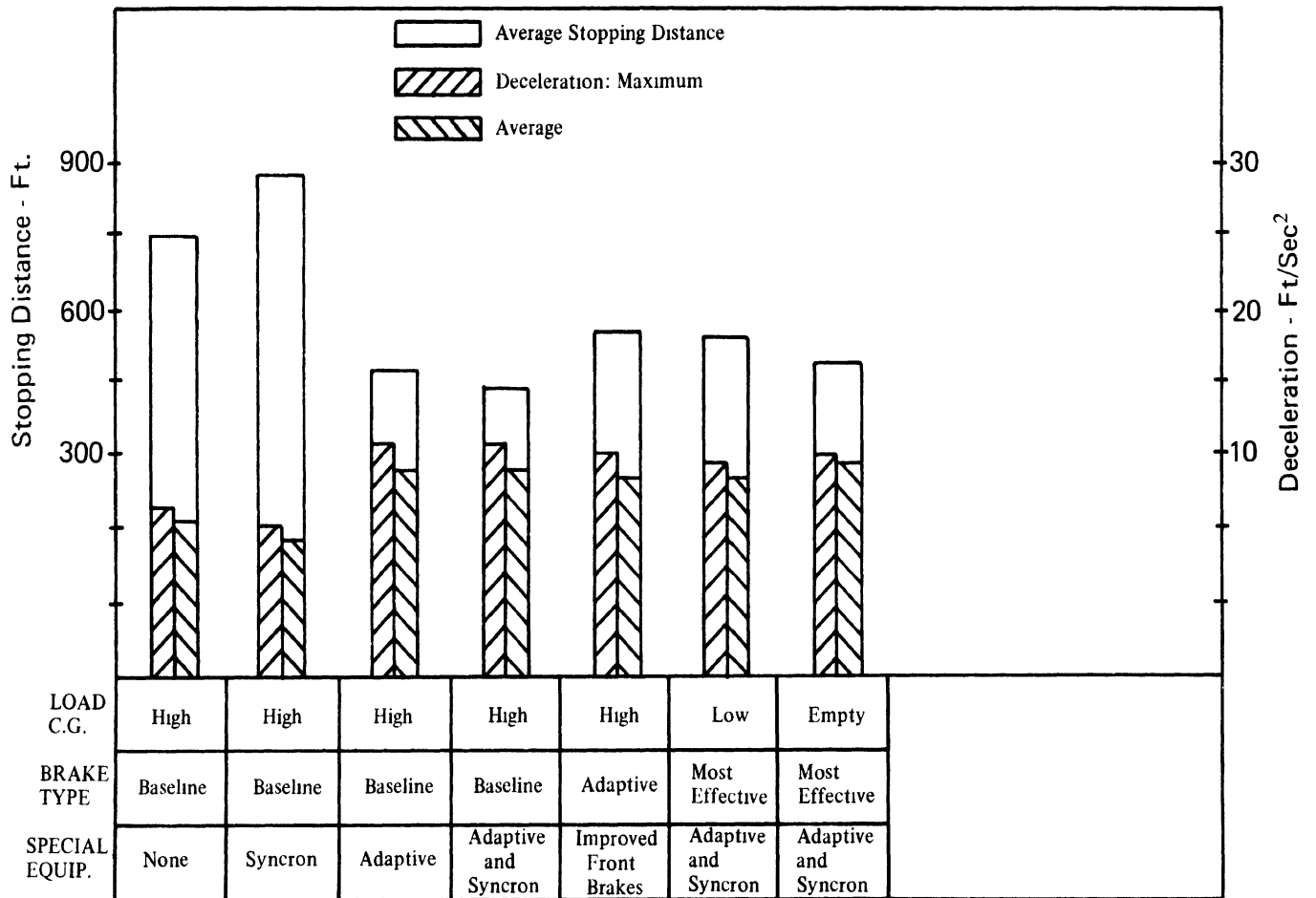
FIGURE 66. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR-TRAILER, WET TRACK, CORRECTED TO 60 MPH
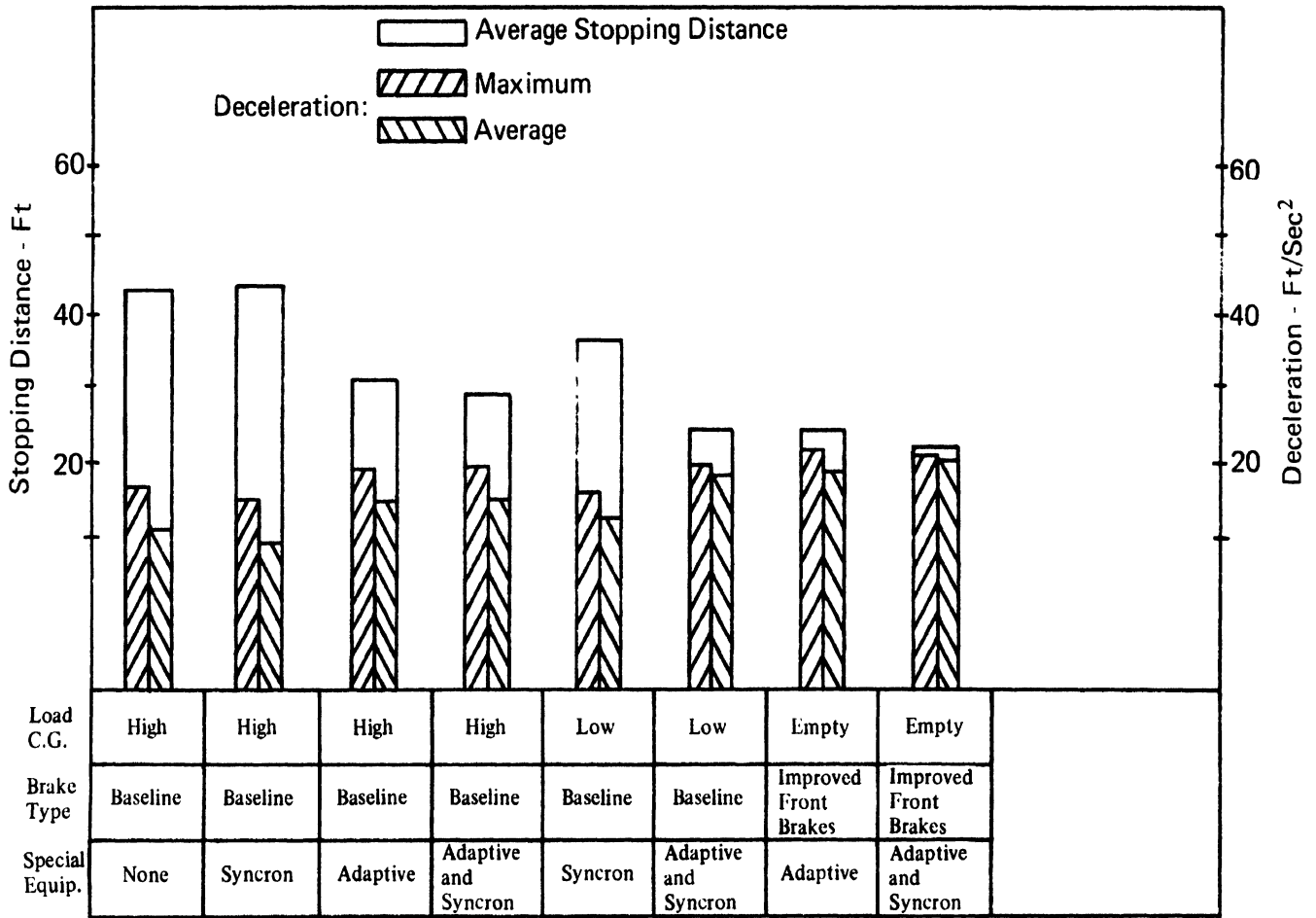
FIGURE 67. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR-TRAILER, DRY TRACK, FROM 20 MPH
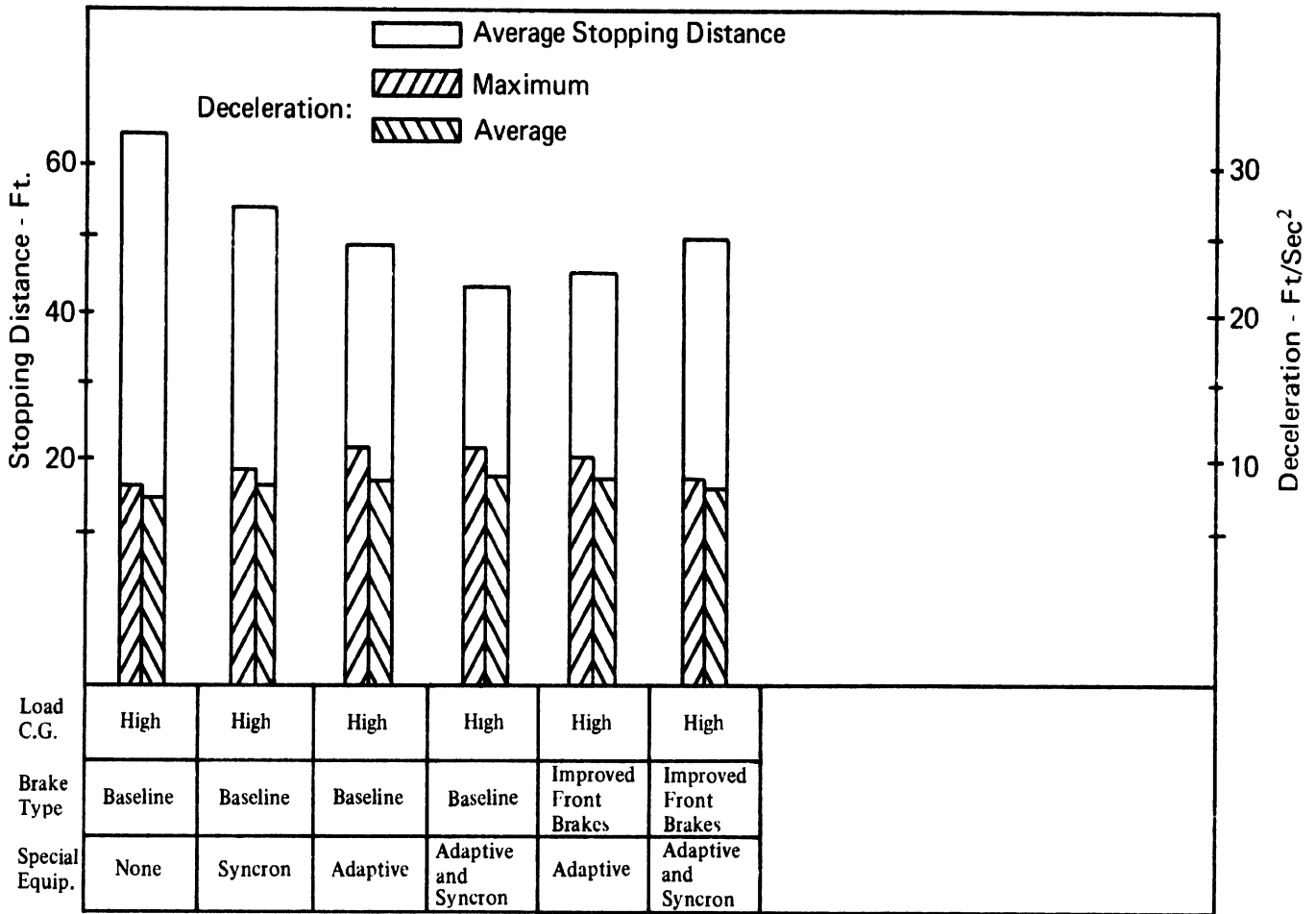
FIGURE 68. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR-TRAILER, LOADED, WET TRACK, FROM 20 MPH
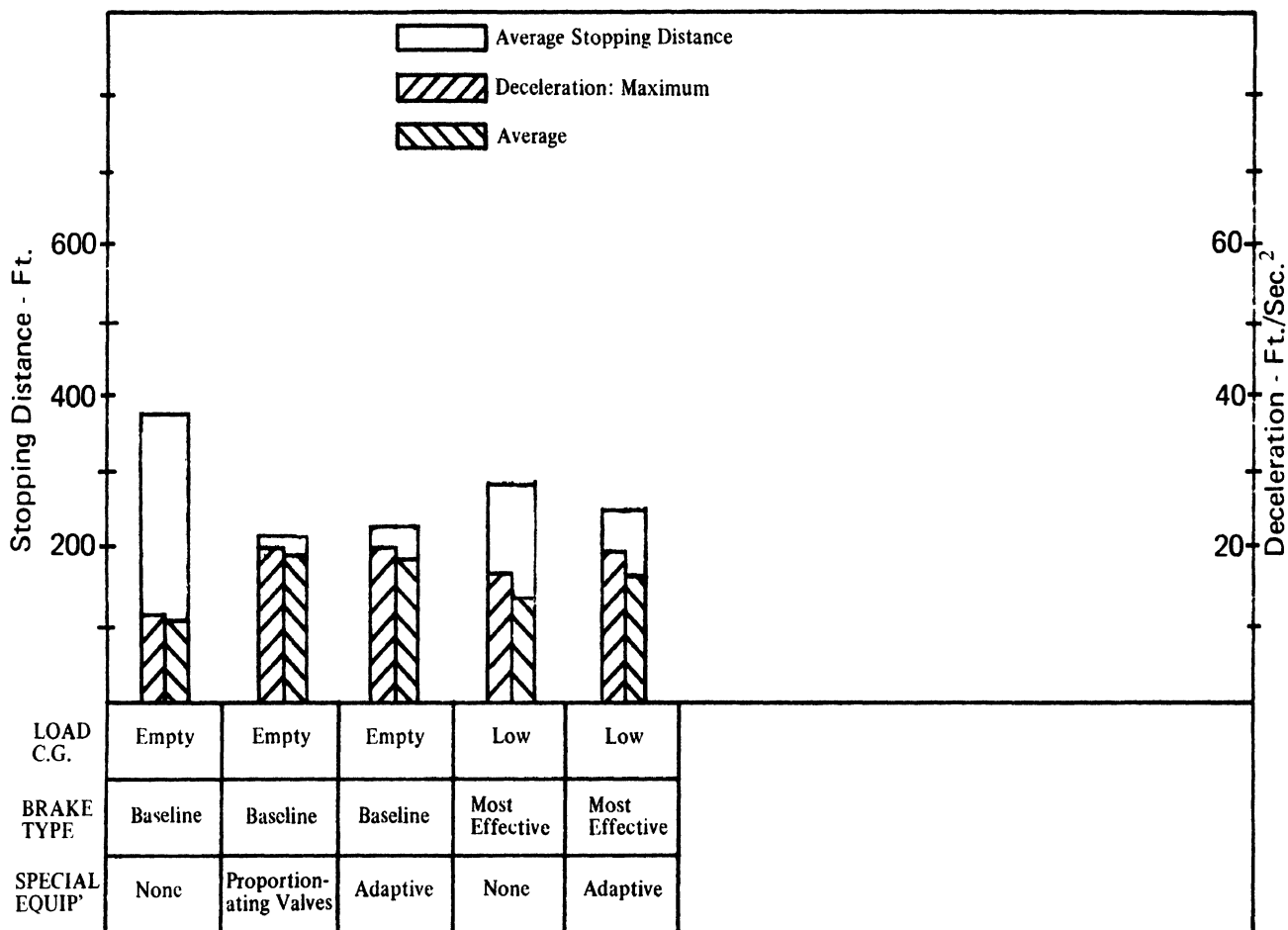
FIGURE 69. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR ONLY, DRY TRACK, FROM 60 MPH
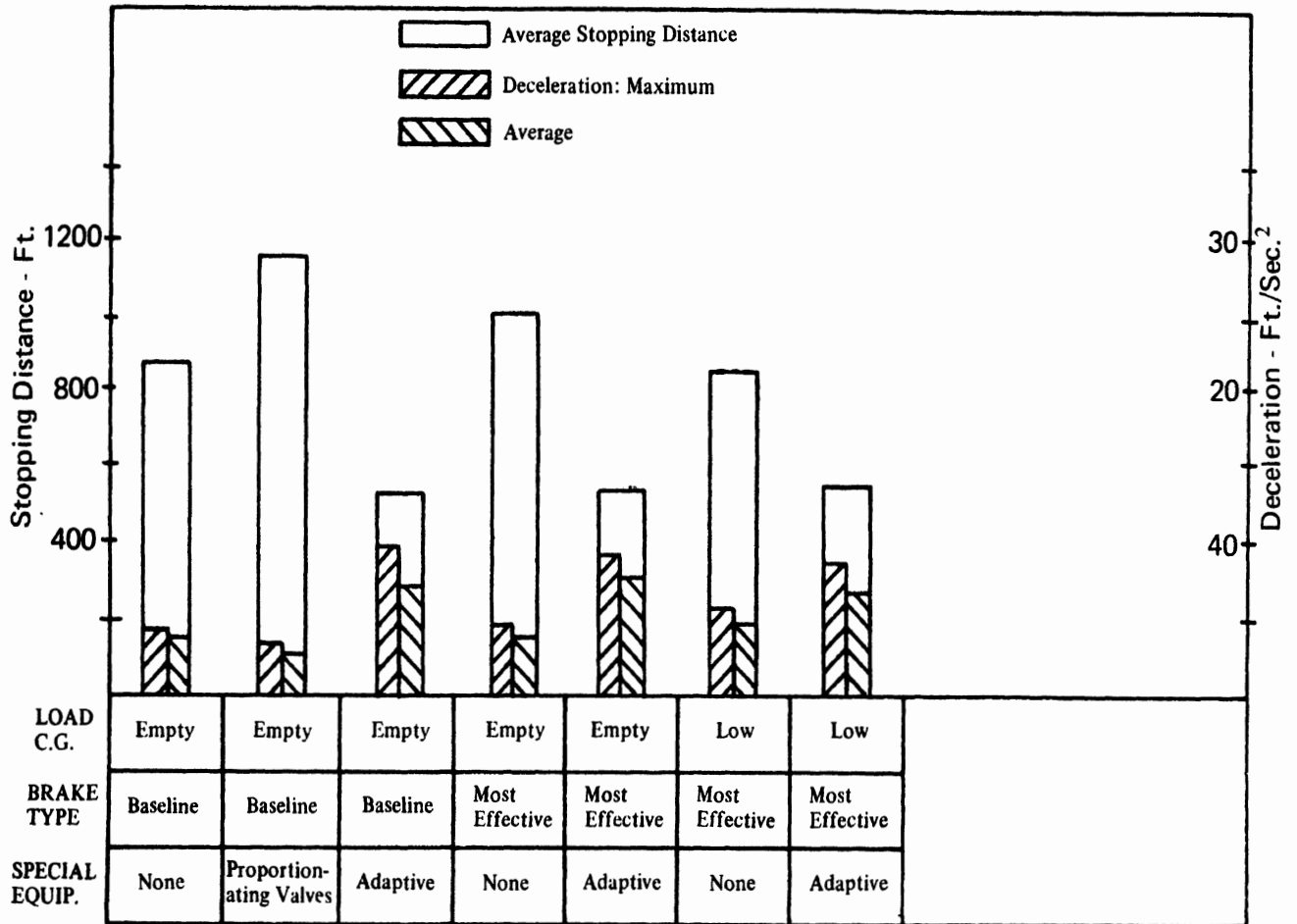
FIGURE 70. MINIMUM STOPPING DISTANCE TEST RESULTS, TRACTOR ONLY, WET TRACK, FROM 60 MPH

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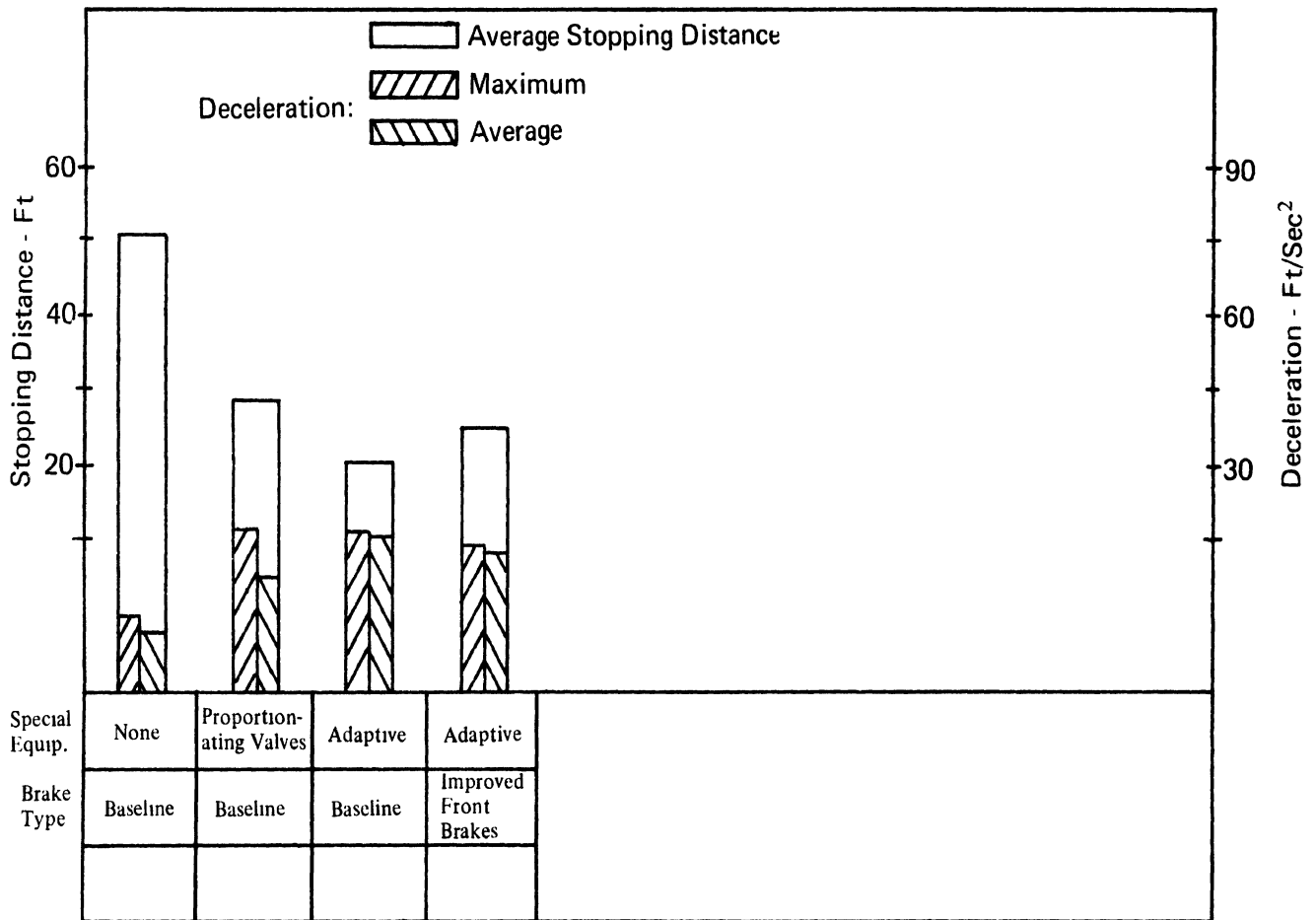
FIGURE 71. MINIMUM STOPPING DISTANCE TEST RESULTS, BOBTAIL TRACTOR, DRY TRACK, FROM 20 MPH
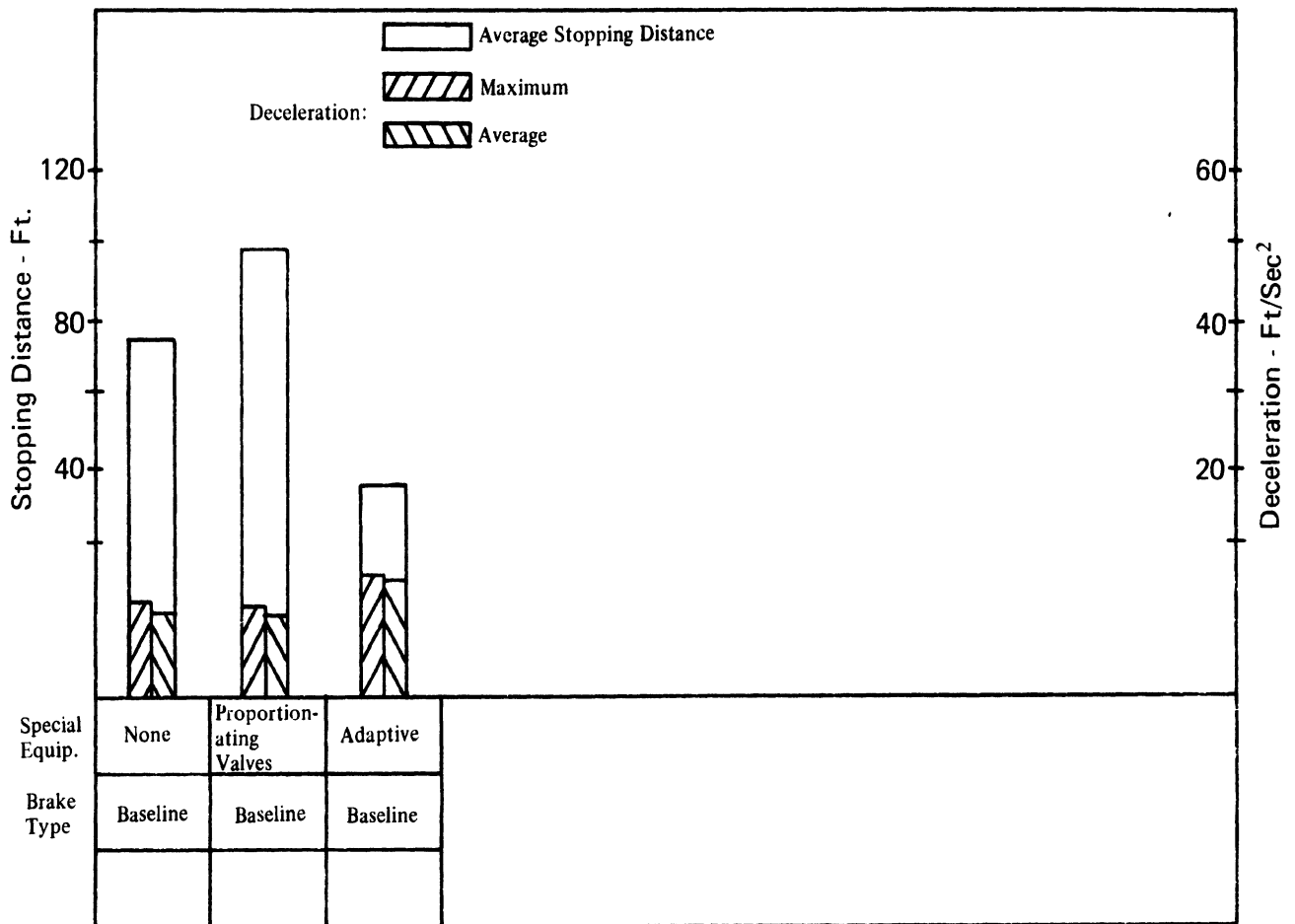
FIGURE 72. MINIMUM STOPPING DISTANCE TEST RESULTS, BOBTAIL TRACTOR, WET TRACK, FROM 20 MPH

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TABLE 19

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 64

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	57	260	60	288	16.0	13.4
	56	258	60	296	16.4	13.1
Syncron	57	251	60	278	17.3	14.0
	55	246	60	293	17.2	13.2
Adaptive Braking	53	218	60	280	16.9	13.9
	53	229	60	283	17.1	13.7
	54	227	60	280	16.9	13.8
Adaptive Braking, Syncron	55	236	60	281	16.4	13.8
	54	220	60	272	17.0	14.3
	54	223	60	275	16.9	14.1
Syncron	56	255	60	292	14.7	13.3
	56	238	60	273	14.9	14.2
	55	235	60	280	15.9	13.9
Adaptive Braking, Syncron	57	231	60	256	16.0	15.2
	56	232	60	266	15.9	14.6
	57	233	60	258	15.9	15.0
Most Effective Brakes	54	257	60	317	12.8	11.5
Adaptive Syncron, Most Effective Brakes	54	220	60	272	16.0	14.3
	54	225	60	278	16.0	14.0
	53	216	60	276	16.2	14.0
Most Effective Brakes*	57	191	60	212	18.6	18.3

*Wheels locked.

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TABLE 20

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 65

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
Adaptive Braking Improved Front Brakes	59	238	60	246	18.9	15.8
Adaptive, Syncron, Improved Front Brakes	60	246			18.8	15.8
	60	224			19.1	17.3
	60	221			19.2	17.5
Adaptive, Syncron, Most Effective Brakes	57	225	60	249	19.5	15.6
	58	222	60	238	20.2	16.3
	60	220			20.4	17.6
None	60	346			11.6	11.2
None*	61	214	60	207	18.2	17.0
Proportioning Valves	59	242	60	354	*	8.7
Proportioning Valves*	60	259			15.7	15.0
Adaptive Braking	60	216			18.9	18.0
Proportioning on Trailer, Adaptive on Tractor	61	299	60	289	14.0	13.4
Proportioning on Trailer, Adaptive on Tractor*	61	205	60	198	19.8	19.5

*Wheels locked, instrument failure.

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TABLE 21

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 66

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	50	512	60	737	6.2	5.3
	50	530	60	763	6.1	5.1
	50	520	60	748	6.1	5.2
Syncron	48	570	60	890	5.2	4.4
	48	543	60	848	5.2	4.6
	48	588	60	919	5.0	4.2
Adaptive Braking	50	318	60	458	10.4	8.5
	50	304	60	438	10.7	8.9
	50	319	60	459	10.6	8.4
	50	333	60	479	10.1	8.1
Adaptive Braking, Syncron	47	281	60	458	10.2	8.5
	49	302	60	453	10.8	8.6
	50	297	60	428	11.2	9.1
	50	305	60	440	10.8	8.8
Adaptive Braking, Improved Front Brakes	48	371	60	580	9.7	6.7
	46	304	60	517	10.3	7.5
	48	360	60	563	10.0	6.9
Adaptive Syncron, Most Effective Brakes	38	224	60	558	8.8	6.9
	38	223	60	555	8.4	7.0
	39	227	60	538	9.0	7.2
Adaptive Syncron, Most Effective Brakes*	40	212	60	476	9.0	8.1
	41	218	60	466	9.3	8.3
	39	206	60	487	9.4	8.0

*Empty, not loaded.

TABLE 22

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 67

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	21	50	20	45	15.0	9.5
	21	48	20	44	18.4	9.9
	21	40	20	36	18.2	11.8
	21	51.5	20	47	15.0	9.2
Syncron	20	48			15.0	9.0
	20	42			15.3	10.3
	20	41			15.8	10.5
Adaptive Braking	20	31			19.8	13.9
	20	30.5			19.6	14.2
	20	29			19.5	14.9
Adaptive Braking, Syncron	20	28			19.6	15.4
	20	28			19.7	15.4
	20	27			19.8	16.0
Syncron	20	36.5			16.7	11.8
	20	33.5			16.8	12.1
	20	32.5			17.0	13.3
Adaptive Braking, Syncron	21	26	20	23.6	20.0	18.2
	21	24.5	20	22.2	19.5	19.3
	20	26			19.5	16.5
Adaptive Braking, Improved Front Brakes*	20	24.5			21.0	17.6
	21	26	20	23.5	21.2	18.2
	21	25	20	22.5	21.3	19.0
Adaptive Syncron, Improved Front Brakes*	21	24	20	22	21.0	19.8
	21	23	20	21	21.1	20.6
	21	23	20	21	21.1	20.6

*Empty, not loaded.

TABLE 23

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 68

Special Equipment	Actual Vel, mph	Actual Dist, ft	Corrected Vel, mph	Corrected Dist, ft	Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
None	20	63			9.5	6.8
	20	64			8.8	6.7
	20	68			8.4	6.4
Syncron	20	58			9.0	7.4
	20	52			9.1	8.3
	20	54			9.1	8.1
Adaptive Braking	21	51.5	20	47	11.6	9.2
	20.5	49	20	47	12.0	9.2
	20	49			11.2	8.8
Adaptive Braking, Syncron	20	46			11.0	9.4
	20	44			10.7	9.8
	20	43			11.0	10.0
	20	41			11.1	10.5
Adaptive Braking, Improved Front Brakes	20	46	20	41	10.4	9.4
	21	45			10.8	10.5
	20	47			10.2	9.2
Adaptive Braking, Syncron, Improved Front Brakes	20	48			10.1	9.0
	20	49			9.9	8.8
	20	50			9.6	8.6

TABLE 24

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 69

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	60	387			10.8	10.0
	60	377			11.0	10.3
	60	366			10.7	10.6
	60	354			11.8	10.9
Proportioning Valve	60	206			19.1	18.8
	60	197			20.8	19.7
	60	205			19.6	18.9
Adaptive Braking	60	199			19.8	19.5
	60	207			19.7	18.7
	60	205			19.6	18.9
Most Effective Brakes*	56	243	60	279	17.0	13.9
	57	266	60	294	17.3	13.2
	57	263	60	291	16.3	13.3
Adaptive Braking, Most Effective Brakes*	56	212	60	243	19.0	16.0
	59	212	60	219	19.5	17.7
	60	218.5			19.7	17.7

*Loaded.

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TABLE 25

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 70

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	51	662	60	914	4.3	4.2
	49	713	60	814	3.9	3.6
	50	648	60	894	4.2	4.2
Proportioning Valve	50	774	60	1115	3.6	3.5
	50	821	60	1174	3.5	3.3
	49	905	60	1304	3.1	2.9
Adaptive Braking	61	532	60	514	8.8	7.5
	60	456			9.6	8.5
	58	486	60	520	9.1	7.5
	59	533	60	551	8.8	7.0
Most Effective Brakes	35	423	60	1243	4.6	3.1
	35	371	60	1090	5.0	3.6
	39	343	60	812	5.2	4.8
Adaptive Braking, Most Effective Brakes	36	190	60	528	8.9	7.3
	35	186	60	546	8.8	7.1
	37	177	60	465	9.6	8.3
Most Effective Brakes*	40	352	60	791	5.8	4.9
	39	345	60	816	5.2	4.7
	39	360	60	851	5.1	4.6
Adaptive Braking, Most Effective Brakes*	39	331.5	60	546	8.3	7.1
	40	214	60	481	9.0	8.0
	39	222	60	525	8.6	7.4

*Loaded.

TABLE 26

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 71

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	19	48	20	53	11.0	8.1
	19	51	20	56	9.7	7.6
	19	39	20	43	11.4	10.0
	19	44	20	49	11.3	8.8
Proportioning Valve	20	29			22.6	14.8
	20	27.5			23.2	15.7
	20	28			24.2	15.4
Adaptive Braking	19	18	20	20	21.6	21.6
	19	18	20	20	21.6	21.6
	19	20	20	20	21.8	19.4
Adaptive Braking, Improved Front Brakes	20	23			20.0	18.7
	20	25			19.8	17.2

TABLE 27

MINIMUM STOPPING DISTANCE DATA FOR FIGURE 72

Special Equipment	Actual		Corrected		Maximum Deceleration, ft/sec^2	Average Deceleration, ft/sec^2
	Vel, mph	Dist, ft	Vel, mph	Dist, ft		
None	20	71			7.5	6.1
	20	79			7.0	5.5
	20	73			7.2	5.9
Proportioning Valve	20	95			7.0	4.5
	20	101			6.7	4.3
	20	97			6.2	4.4
Adaptive Braking	20	37			12.3	11.7
	20	35			12.3	12.3
	20	37			11.5	11.7
	20	38			11.5	11.4

at the rear brakes below 6 psi, that is, at or near the pushout pressure, the rear brakes did little or no work in these stops, resulting in low decelerations and long stopping distances. This problem could have been eliminated had the valve been biased such that proportioning did not occur on the rear brakes until after the brake line pressure exceeded the pushout pressure.

(4) The influence of load c.g. height on the performance of the tractor-trailer combination was indeterminate, because the minimum-stopping-distance test procedure introduced human factors that confounded the test data. The rate of initial pedal application and the pressure level maintained throughout the test was subject to driver judgment. Table 28 compares test data obtained in 60 and 20 mph stops under both high and low c.g. loading conditions with baseline brakes plus syncron. At 60 mph, with the high load, the driver was inclined to be very careful and did not apply the brakes rapidly. Consequently, the time from initial movement of the pedal until the final pressure was fully applied averaged out at 1.55 sec compared to 0.80 sec for the low c.g. load. At 20 mph, the figures are 1.10 sec and 0.50 sec, respectively. For some unknown reason, however, the driver elected to use a sustained line pressure of 94 psi when stopping with the high c.g. load from 60 mph, while for the low c.g. load he used only 80 psi. The reverse was true for the 20 mph stops, where with the high c.g. the driver selected a pressure of 68 psi, while for the low c.g., 75 psi. It appears clear that the slightly shorter stopping distances achieved with the low c.g. in the tests conducted without the antilock system operational resulted from the driver applying the brakes more rapidly than he did with the high c.g. When the antilock system was operational, the driver took practically the same time to apply the brakes in both loading conditions. Further, the influence of loading on system effectiveness could not be determined exactly because the adaptive system worked differently under each loading condition. All that can be concluded is that driver factors and the peculiarities of the adaptive system undoubtedly masked whatever differences occurred in dynamic axle loading, as caused by load transfer during the stops that were made under the two load conditions. It appears, however, that the measured stopping distance is consistently (if not much) shorter with the low c.g. load.

3.5.2.3 Brake Balance Tests. In addition to supplying the tractor for this test program, White Motors provided the hardware necessary to alter brake effectiveness. The results of the brake balance tests, as run with "baseline," "improved front," and "most effective" brakes, are given in Table 29. It should be noted that whereas the "baseline" brakes were balanced in accordance with static axle loads, the brakes in the "most effective" condition were best balanced for higher deceleration stops.

3.5.2.4 Response Time Tests. Results from the brake response time tests are given in Figs. 73 through 82 and are summarized in Table 30. The following points should be noted:

(1) Proportioning valves and/or an adaptive system do not seem to affect either application or release times.

(2) Syncron improves (i.e., decreases) trailer brake application time by about 25% and the release time by better than 40%.
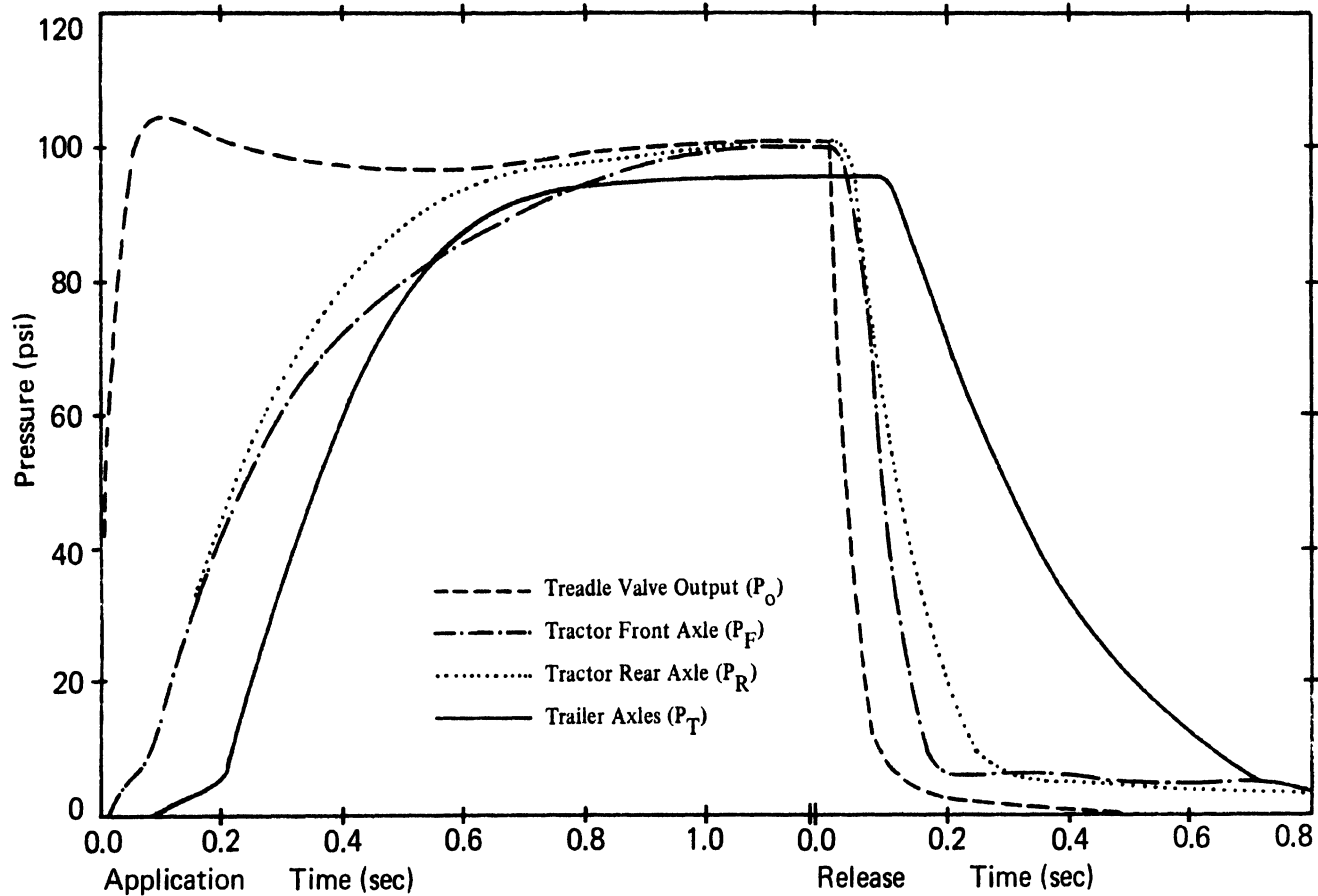
FIGURE 73. BRAKE RESPONSE TIME TEST, VEHICLE 12, STANDARD SYSTEM
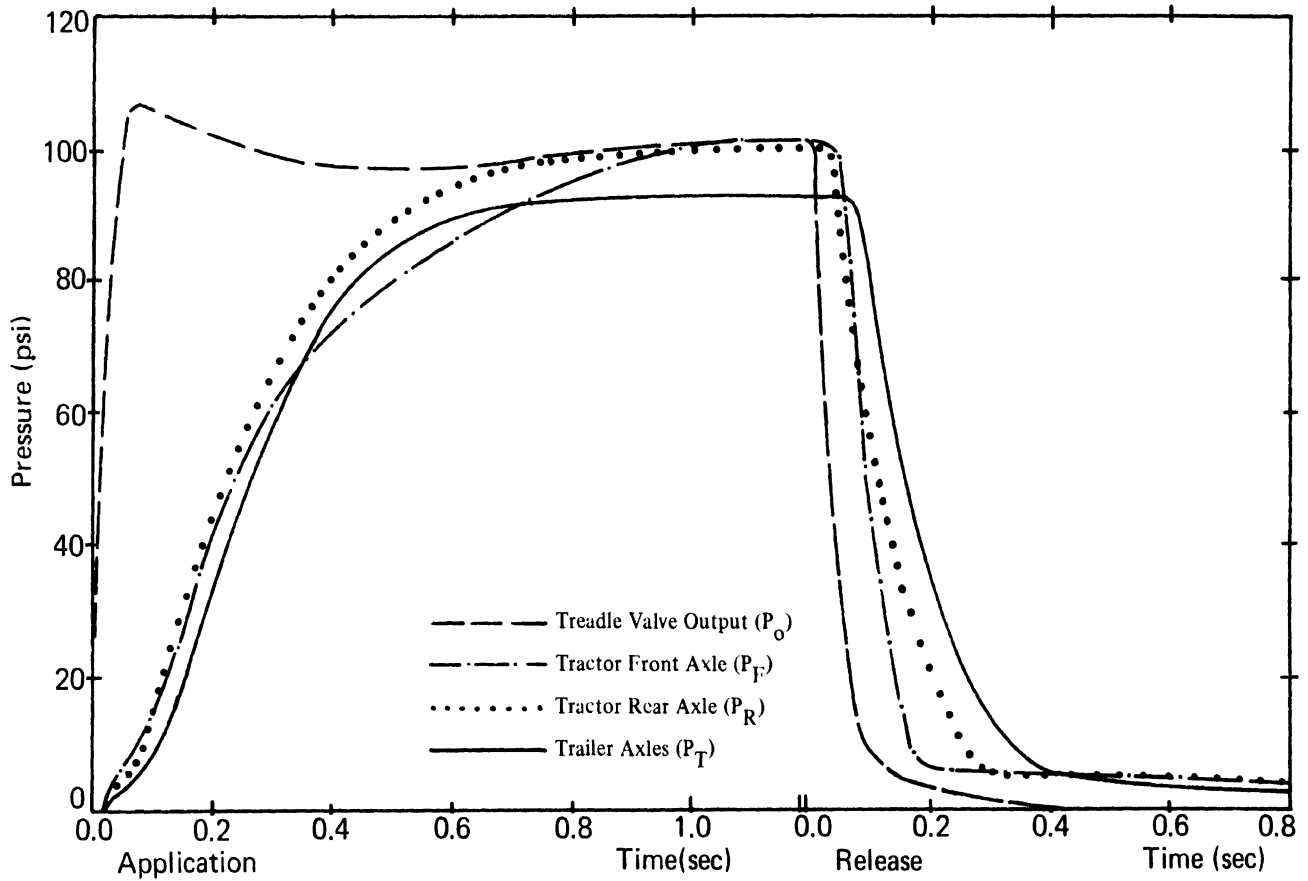
FIGURE 74. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH SYNCRON
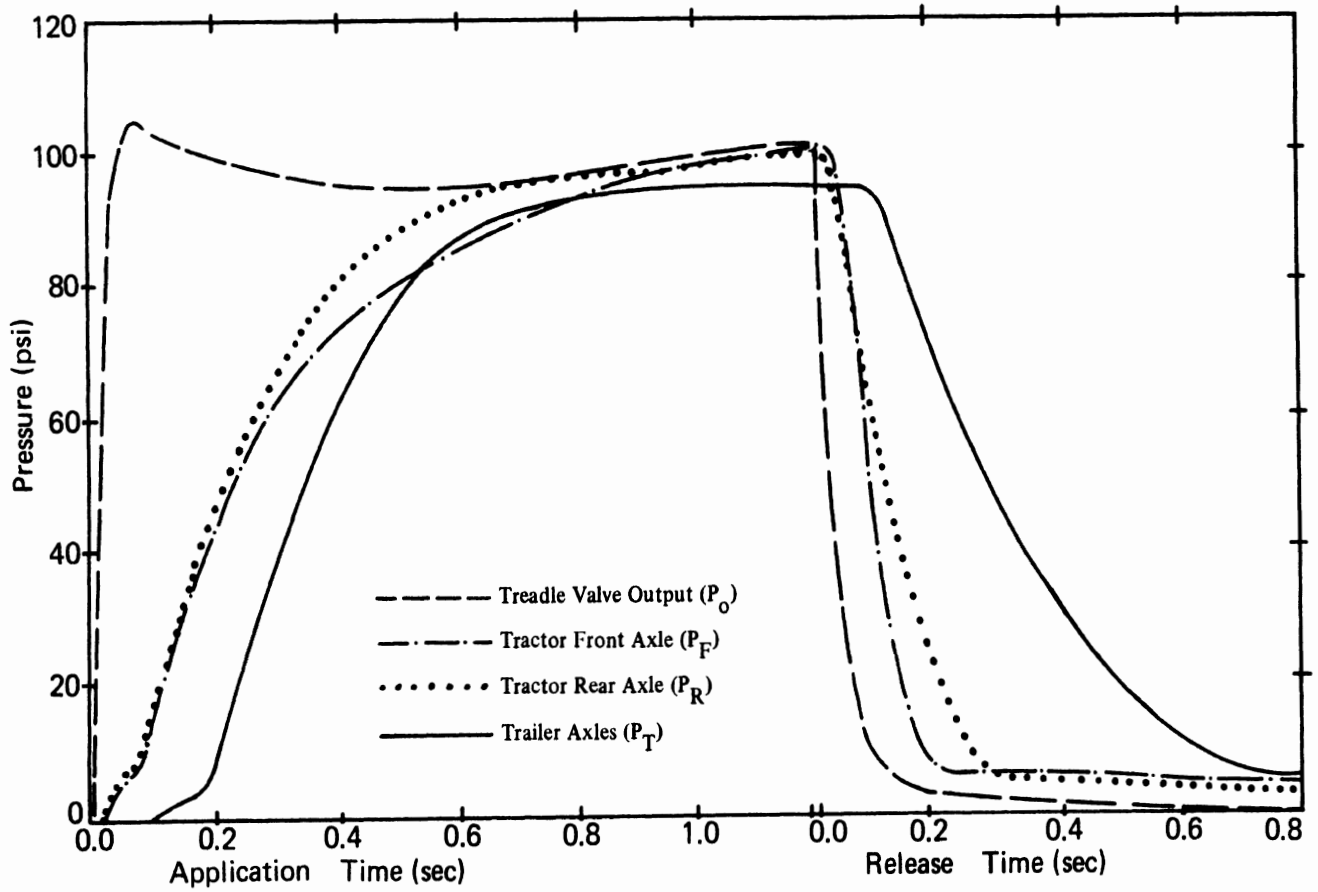
FIGURE 75. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH ADAPTIVE BRAKING

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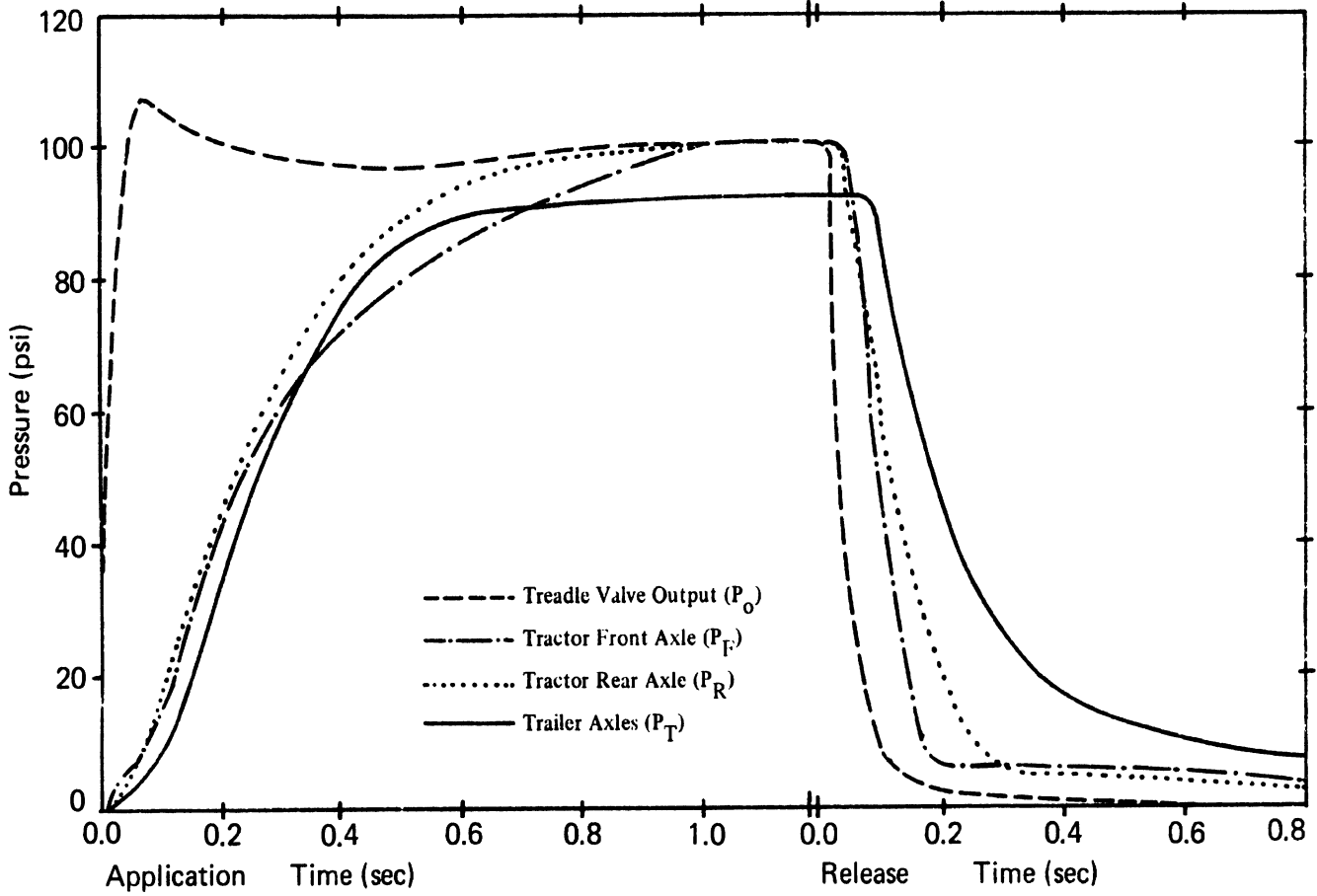
FIGURE 76. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH ADAPTIVE BRAKING AND SYNCRON

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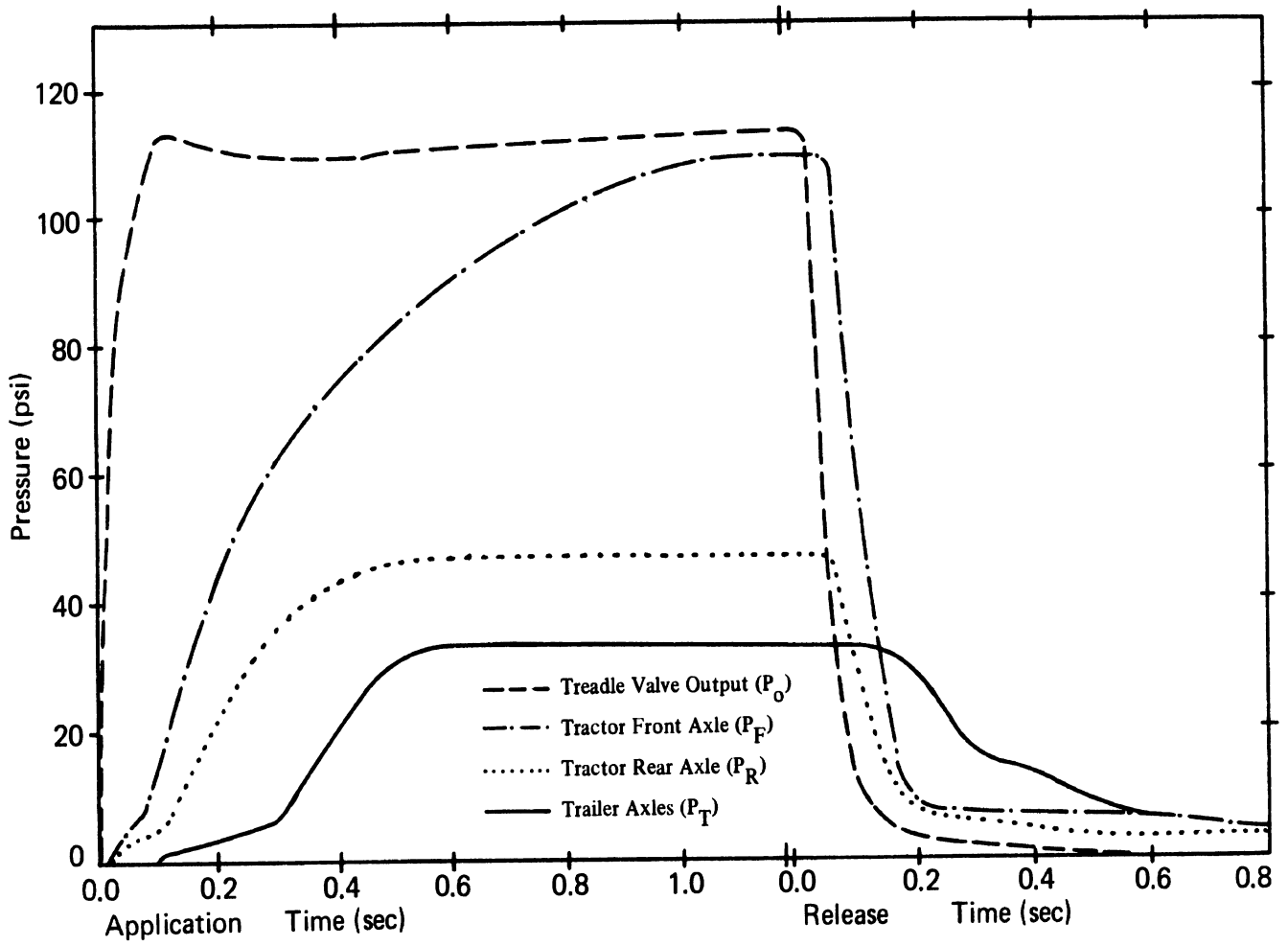
FIGURE 77. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH PROPORTIONING (EMPTY)

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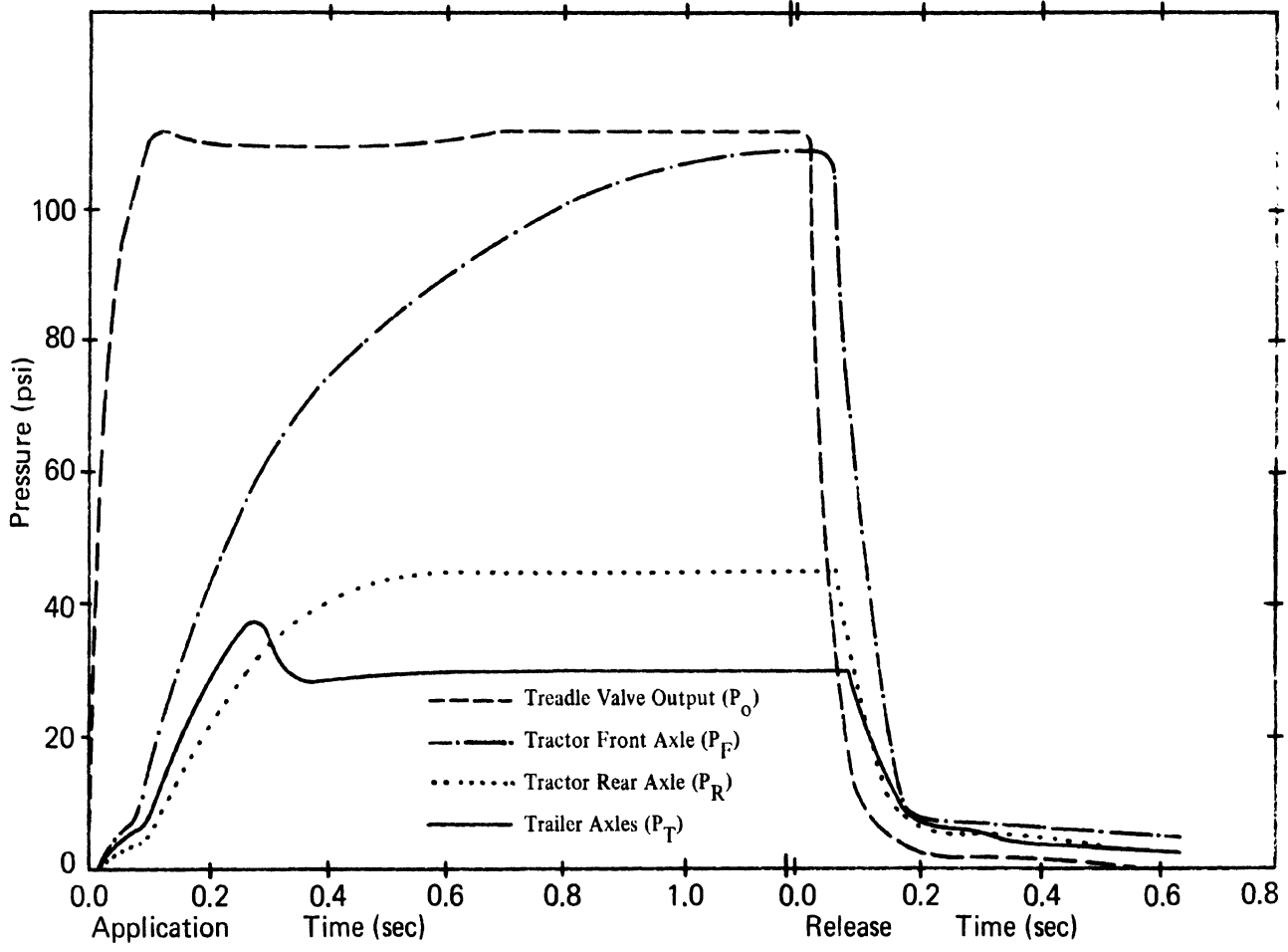
FIGURE 78. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH PROPORTIONING (EMPTY) AND SYNCRON

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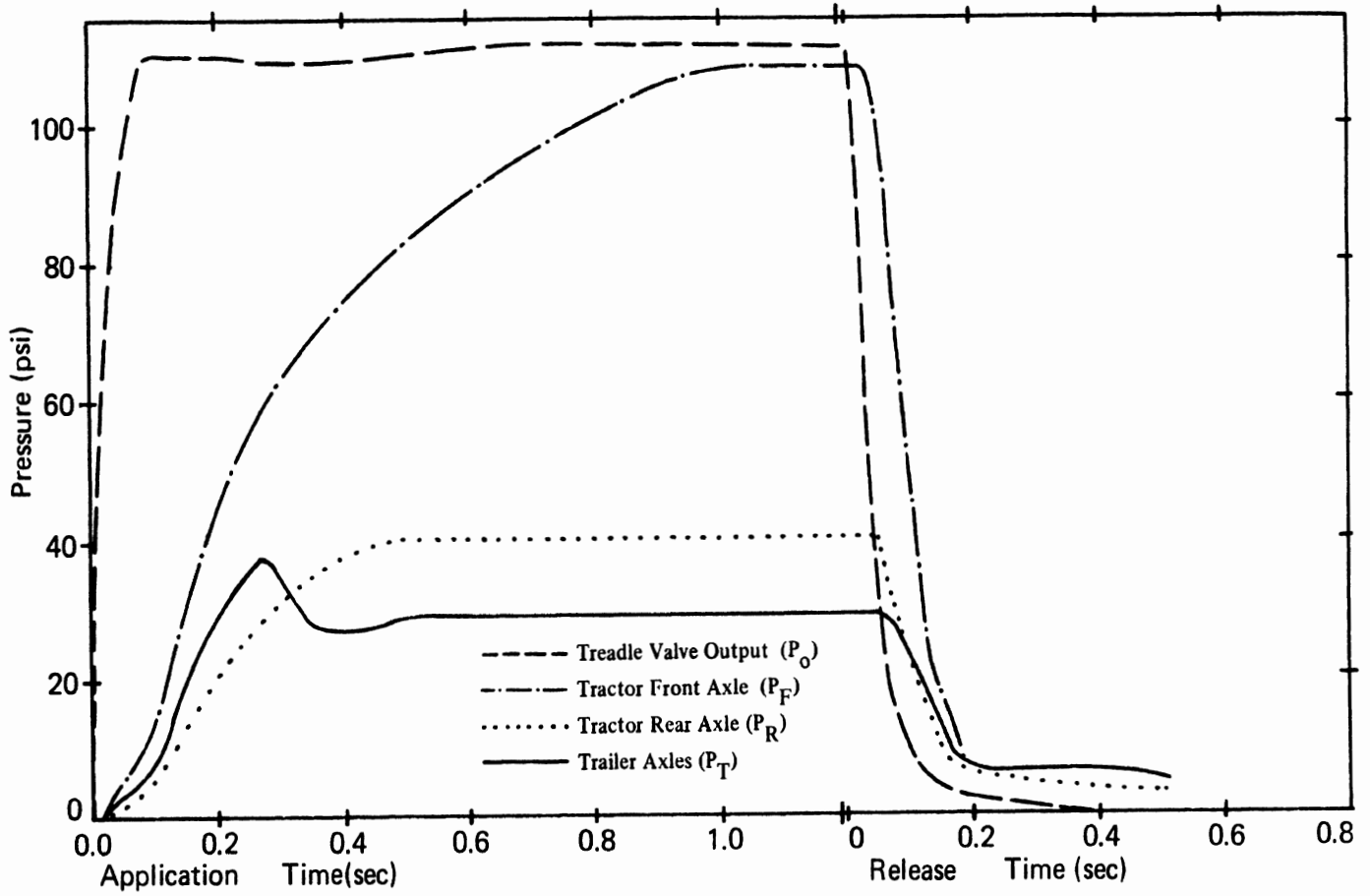
FIGURE 79. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH SYNCRON, ADAPTIVE SYSTEM, AND PROPORTIONING (EMPTY)
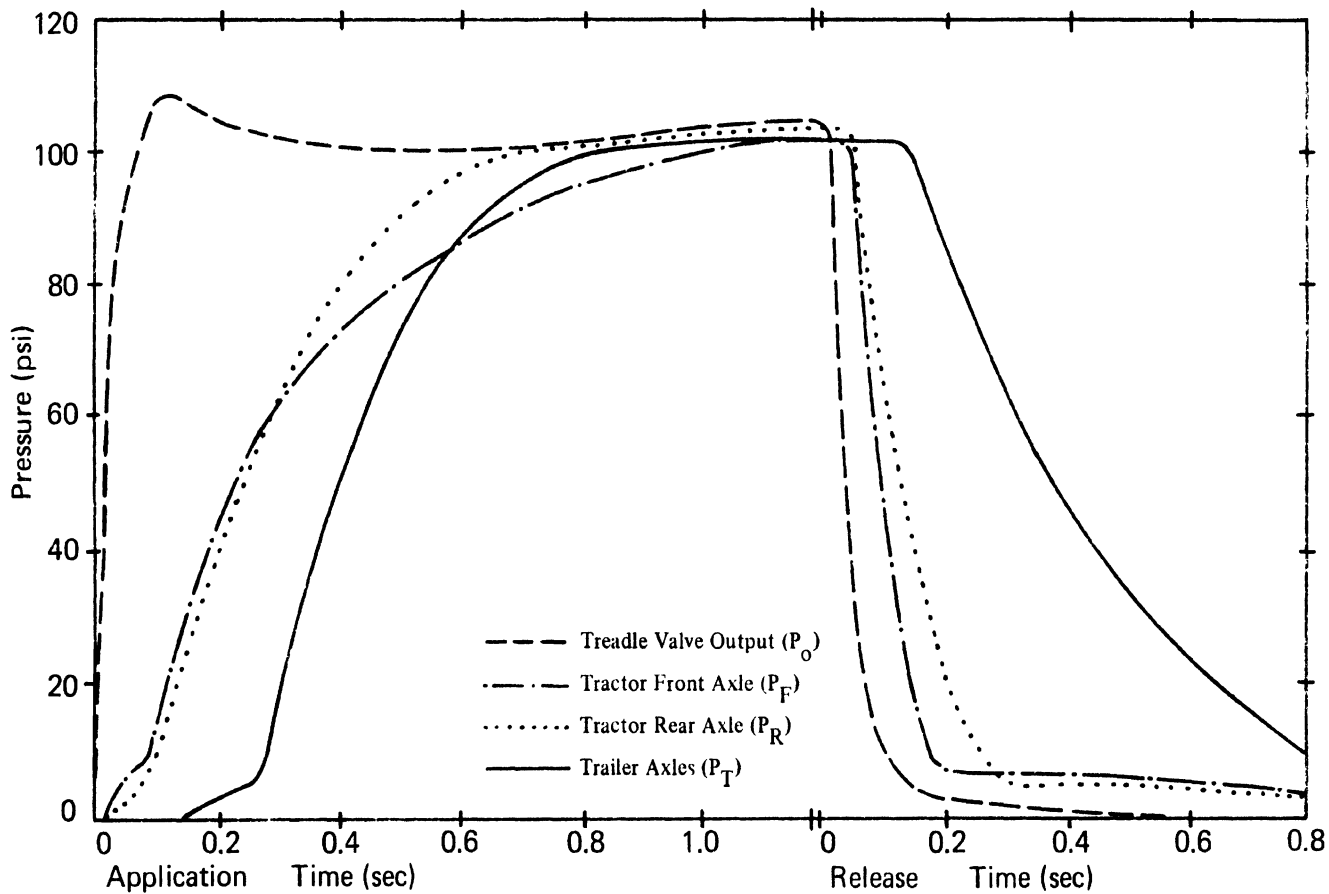
FIGURE 80. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH PROPORTIONING (FULL LOAD)
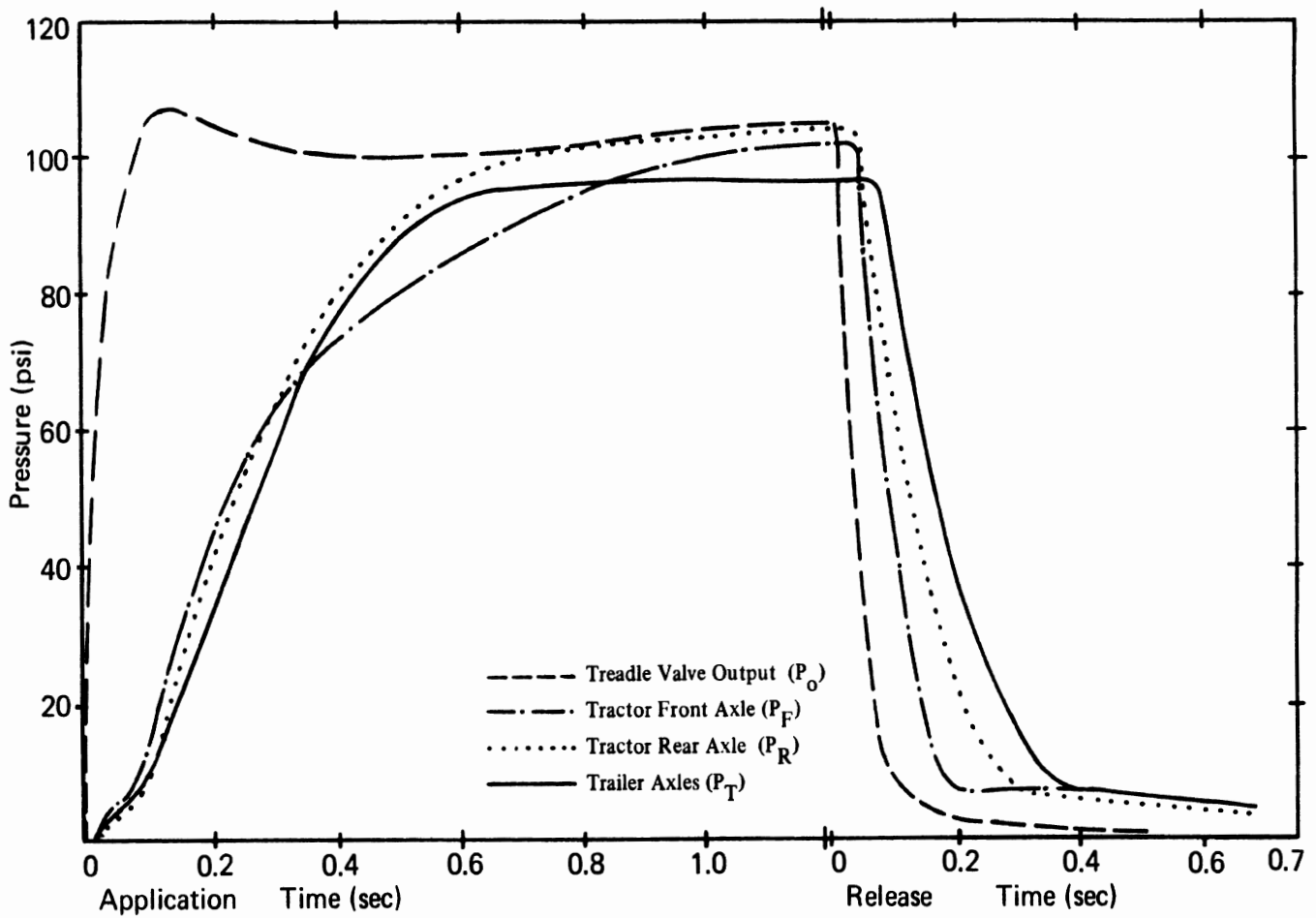
FIGURE 81. BRAKE RESPONSE TIME TEST, VEHICLE 12, WITH PROPORTIONING (FULL LOAD) AND SYNCRON

Legend:
- — — — Treadle Valve Output (P_o)
- —·—· Tractor Front Axle (P_F)
- ······· Tractor Rear Axle (P_R)
- ——— Trailer Axles (P_T)

Y-axis: Pressure (psi)
X-axis (left): Application Time (sec)
X-axis (right): Release Time (sec)
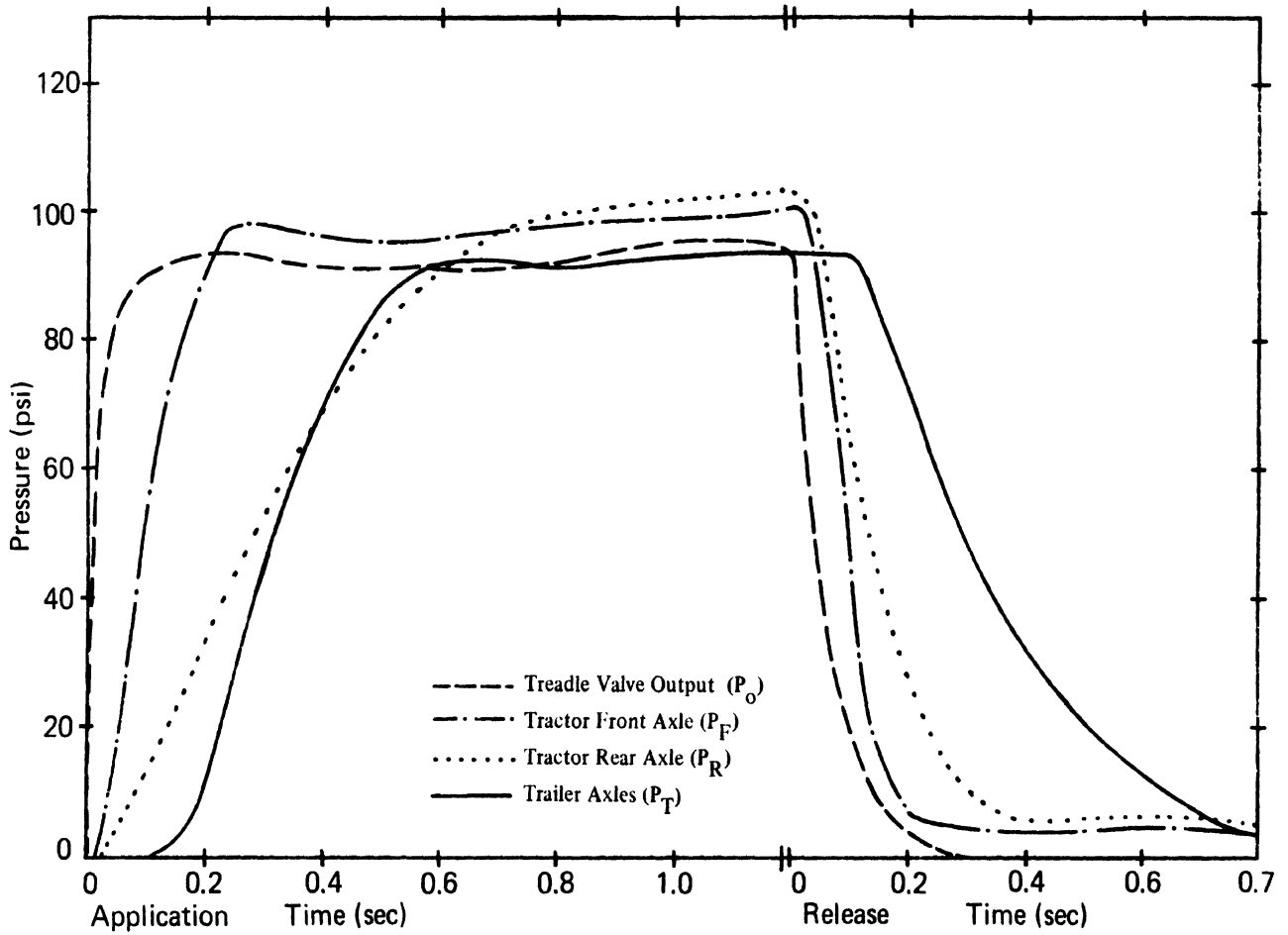
FIGURE 82. BRAKE RESPONSE TIME TEST, VEHICLE 12, MOST EFFECTIVE BRAKES

TABLE 28

DRIVER/VEHICLE PERFORMANCE AS INFLUENCED BY HIGH AND
LOW LOAD CONFIGURATIONS

System	Test Velocity, mph	System	High Load, c.g.	Low Load, c.g.
Syncron	60	Time to Apply Brakes, sec	1.55	0.80
		Sustained Line Pressure, psi	94	80
		Stopping Distance, ft	285	282
		Average Deceleration, ft/sec^2	13.60	13.80
		Maximum Deceleration, ft/sec^2	17.20	15.20
Syncron	20	Time to Apply Brakes, sec	1.10	0.50
		Sustained Line Pressure, psi	68	75
		Stopping Distance, ft	43.70	34.50
		Average Deceleration, ft/sec^2	9.90	12.40
		Maximum Deceleration, ft/sec^2	15.30	16.80
Syncron and Antilock	60	Time to Apply Brakes, sec	0.25	0.28
		Sustained Line Pressure, psi	100	100
		Stopping Distance, ft	276	260
		Average Deceleration, ft/sec^2	14.10	14.90
		Maximum Deceleration, ft/sec^2	16.80*	15.90**
Syncron and Antilock	20	Time to Apply Brakes, sec	0.13	0.15
		Sustained Line Pressure, psi	100	100
		Stopping Distance, ft	28.70	23.60
		Average Deceleration, ft/sec^2	15.60	18.20
		Maximum Deceleration, ft/sec^2	19.70*	20.20***

*Tractor rear wheel antilock cycling during stops.
**Tractor rear and trailer wheel antilock cycling during stops.
***Tractor front and trailer wheel antilock cycling during stops.

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TABLE 29

SUMMARY OF RESULTS OF BRAKE BALANCE TESTS—VEHICLE 12

Brakes	Weight Distribution (Percent Total Weight)			Actual Brake Distribution (Percent of Total Brake Force)			Calculated Brake Force Distribution (Percent of Total Brake Force)		
	Tractor Axles		Trailer Axles	Tractor Axles		Trailer Axles	Tractor Axles		Trailer Axles
	Front	Rear		Front	Rear		Front	Rear	
Baseline	13.2	43.5	43.3	12.3	40.5	47.2	11.7	44.3	44.0
Improved Front Brakes	13.2	43.5	43.3	15.4	47.1	47.5	17.4	38.4	44.2
Most Effective	13.2	43.5	43.3	18.7	43.8	37.5	15.4	47.9	36.7

Note:

   Baseline Brakes - tractor equipped with dual 9 sq in. chambers and 14° wedges on front brakes and 5 in. slack adjusters on rear brakes.

   Improved Front Brakes - tractor equipped with dual 12 sq in. chambers and 12° wedges on front brakes and 5 in. slack adjusters on rear brakes.

   Most Effective Brakes - tractor equipped with dual 12 sq in. chambers and 12° wedges on front brakes and 6-1/2 in. slack adjusters on rear brakes.

For all tests, tractor rear brakes were equipped with 30 sq in. chambers and trailer brakes had 30 sq in. chambers and 6 in. slack adjusters.

TABLE 30

SUMMARY OF AIR BRAKE RESPONSE TIME TESTS, VEHICLE 12

Brake System	Application Time to 60 psi, sec*				Release Time to 5 psi, sec			
	Treadle Valve Output	Tractor Axles Front	Tractor Axles Rear	Trailer Axles	Treadle Valve Output	Tractor Axles Front	Tractor Axles Rear	Trailer Axles
Standard	0.02	0.29	0.27	0.4	0.14	0.20	0.30	0.7
Syncron	0.015	0.29	0.27	0.31	0.14	0.20	0.30	0.4
Adaptive	0.02	0.29	0.27	0.4	0.14	0.20	0.30	0.7
Syncron and Adaptive	0.02	0.29	0.27	0.31	0.13	0.20	0.30	1.0
Proportioning (Empty Setting)	0.02	0.29	0.24	0.4	0.14	0.20	0.20	0.6
Proportioning (Empty Setting) and Syncron	0.02	0.29	0.25	0.15	0.14	0.20	0.20	0.2
Proportioning (Empty Setting), Antilock and Syncron	0.02	0.28	0.23	0.18	0.14	0.20	0.20	0.2
Proportioning (Loaded Setting)	0.02	0.28	0.28	0.44	0.14	0.20	0.30	0.8
Proportioning (Loaded Setting and Syncron)	0.02	0.29	0.29	0.32	0.15	0.20	0.30	0.4
Most Effective Brakes	0.02	0.12	0.33	0.35	0.19	0.26	0.35	0.7

*Time to reach 60% of proportioned pressure when using proportioning valves.

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(3) Data apply to "baseline" brakes except those values given in the last row of Table 29 which data apply to the "most effective" brakes. When the "most effective" brakes were installed, the front brake air lines were replaced with larger lines and the connectors to the front brakes were replaced with less restrictive connectors, resulting in brake application time being reduced by better than 50%.

3.5.2.5 Parking Brake Tests. Five sets of spring brakes were tested on the leading rear tractor axle of Vehicle 12. The towbar forces (both static and rolling) required to move the vehicle, with the brakes set on this one axle, are given in Table 31. The resulting K-factors are given, but it should be noted that the spring brakes were designed for the baseline brakes which had 5-in. slack adjusters on the rear tractor axles. Since the force output of the spring brake depends upon the amount the spring extends from its caged length, which in turn is dependent upon slack adjuster length and initial brake adjustment, it is impossible to deduce from test results what the K-factors would be if 5-in. slack adjusters were used. However, if it is assumed that the same angular rotation is required to set the brakes with 5-in. slack adjusters as was required for the 6.5-in. slacks, i.e., the brakes in both cases would have been exactly the same adjustment, the K-factors can be calculated using nominal brake design data from the manufacturers. Such a calculation results in a 12 to 13% reduction in K-factor from the value determined with 6.5-in. slacks if 5-in. slacks are used:

$$K_f' = K_f \frac{\ell'}{\ell} [1 + \frac{K(X_2 - X_1)}{F_2} (1 - \frac{\ell'}{\ell})]$$

where

K_f = K-factor
ℓ = slack adjuster length, inches
K = spring rate, lb/inch
X_1 = caged length of spring
X_2 = length of spring when spring brakes are set
F_2 = spring force at extended length.

Unprimed values are for condition as tested, i.e., 6.5-in. slacks. Primed values are for conditions as calculated, i.e., 5-in. slacks.

3.5.3 TEST RESULTS, VEHICLE 14

3.5.3.1 Effectiveness and Minimum Stopping Distance Tests. Results from three effectiveness tests conducted on Vehicle 14 are given in Fig. 83. Because of the highly sensitive pedal, resulting from a high line pressure/pedal force gradient, and the very effective brakes, making it possible to lock the wheels at a relatively low line pressure, effectiveness tests could not be conducted on the empty vehicle without the antilock system being operational. In addition, it was deemed inadvisable to conduct effectiveness tests on the loaded vehicle beyond the point which wheel lock first occurs. The
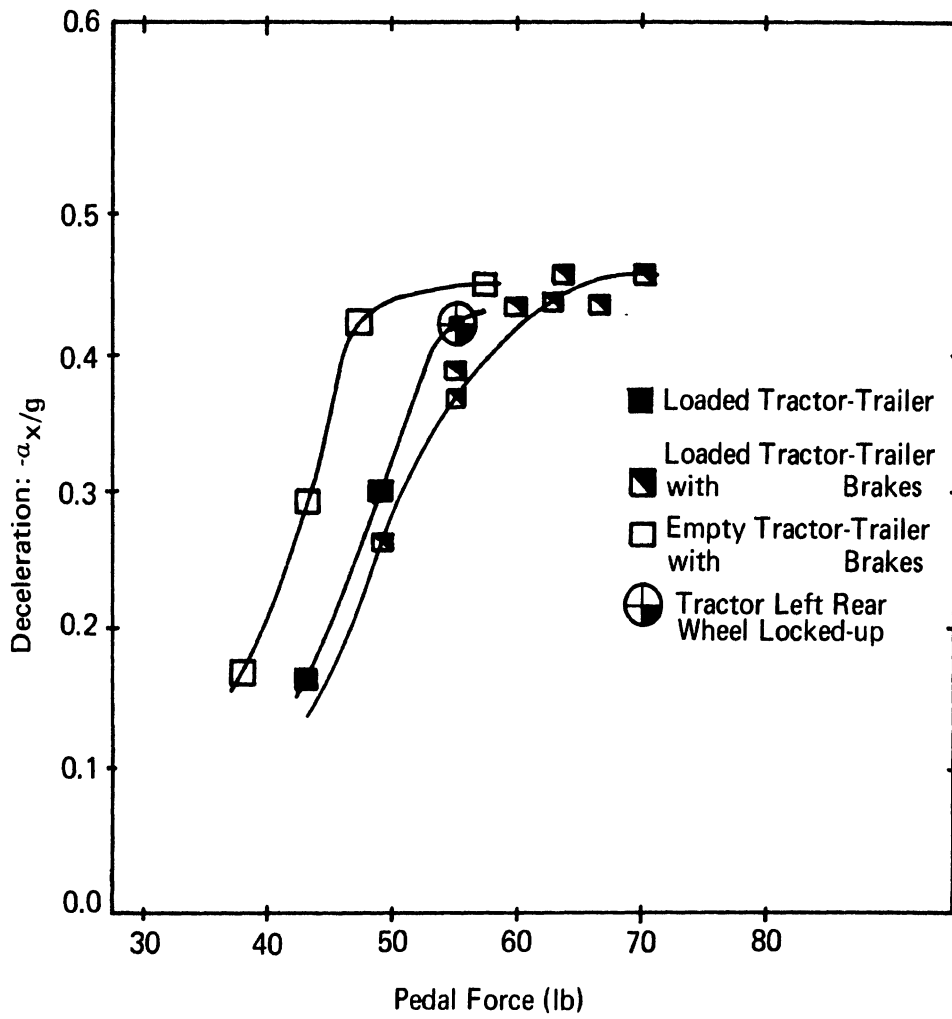
FIGURE 83. EFFECTIVENESS TESTS, VEHICLE 14

TABLE 31

SPRING BRAKE TESTS, VEHICLE 12

Brakes Tested	Tow Bar Force (Static, lb)	K-Factor* (6.5 in. Slacks)	Tow Bar Force (Rolling, lb)	K-Factor* (6.5 in. Slacks)
Berg Set 1	9600	0.565	6800	0.400
Berg Set 2	9800	0.576	6870	0.404
Berg Set 3	8800	0.518	7100	0.418
Berg Set 4	8800	0.518	7300	0.430
Bendix-Westinghouse	9320	0.543	7900	0.465

*K-Factor based upon rated axle load of 17,000 lb.

left rear wheel of the tractor locked at a pedal force of 55 lb and a deceleration of 14.0 ft/sec^2, with the test halted at this point because of the very real probability that lockup of both tractor wheels at higher decelerations would result in severe instability. The effectiveness test performed with the loaded vehicle, with the antilock system operating, produced a maximum deceleration capability of 15.2 ft/sec^2. The effectiveness test conducted on the empty vehicle, with the antilock system operational, was discontinued when the deceleration reached 14.8 ft/sec^2 because of excessive vibration in the front brakes resulting in a broken pipe fitting.

Data for the minimum stopping distance tests are shown graphically in Figs. 84 and 85, and summarized in Table 32.

3.5.3.2 <u>Response Time Tests</u>. Brake response time measurements were made on the vehicle as tested, both with and without the antilock system operational. On completion of the specified test sequence, the brakes were adjusted, and the response time tests were made again. Results of these tests are given in Figs. 86 through 89 and are summarized in Table 33. The maximum pressures measured by the transducers mounted on the vehicle were between 60-80 psi. Consequently, the brake application times, as given in Table 33, were measured from the instant pressure started to rise at the output of the treadle valve to the instant that pressure reached 60% of maximum value, rather than 60 psi as was used previously. Two findings are worthy of comment:

(1) When the antilock system is turned on, little change is noted in brake application response time, except on the rear axle where it is decreased significantly, i.e., from 0.35 to 0.26 sec. Little change is noted in release times, except that the rear axle brakes on the tractor are somewhat improved in this regard.

(2) Brake adjustment does not seem to affect the application times significantly, but did decrease the release times of the brakes on the tractor.

3.5.4 SUMMARY OF PERFORMANCE OF VEHICLES EQUIPPED WITH ADVANCED SYSTEMS. Table 34 summarizes the performance exhibited by the three test vehicles on the dry track, and Table 35 summarizes the performance exhibited by Vehicles 12 and 14 on the low coefficient surface. On the low coefficient surface, direct comparisons can be made since both vehicles were tested under the same conditions. Dry-surface tests on the truck with disk brakes and on Vehicle 12 were conducted on the east straightaway; Vehicle 14 was tested on the approach to the skid pad. Tire-road interface tests indicated that both peak and sliding tire-road interface coefficients are less on the approach road than on the east straightaway. Thus any comparison of the dry surface performance of Vehicle 14 with the other two vehicles must take this difference into account.

3.5.5 FAILURES ENCOUNTERED IN TESTS OF VEHICLES EQUIPPED WITH ADVANCED SYSTEMS. A number of problems were encountered in testing the vehicles equipped with the advanced systems which are detailed below. These problems could be attributed primarily to the fact that the systems under test were in the prototype or developmental stages.

<u>Disk Brake Truck</u>. No actual structural or brake system failures occurred during the testing of this vehicle. Developmental problems were encountered,
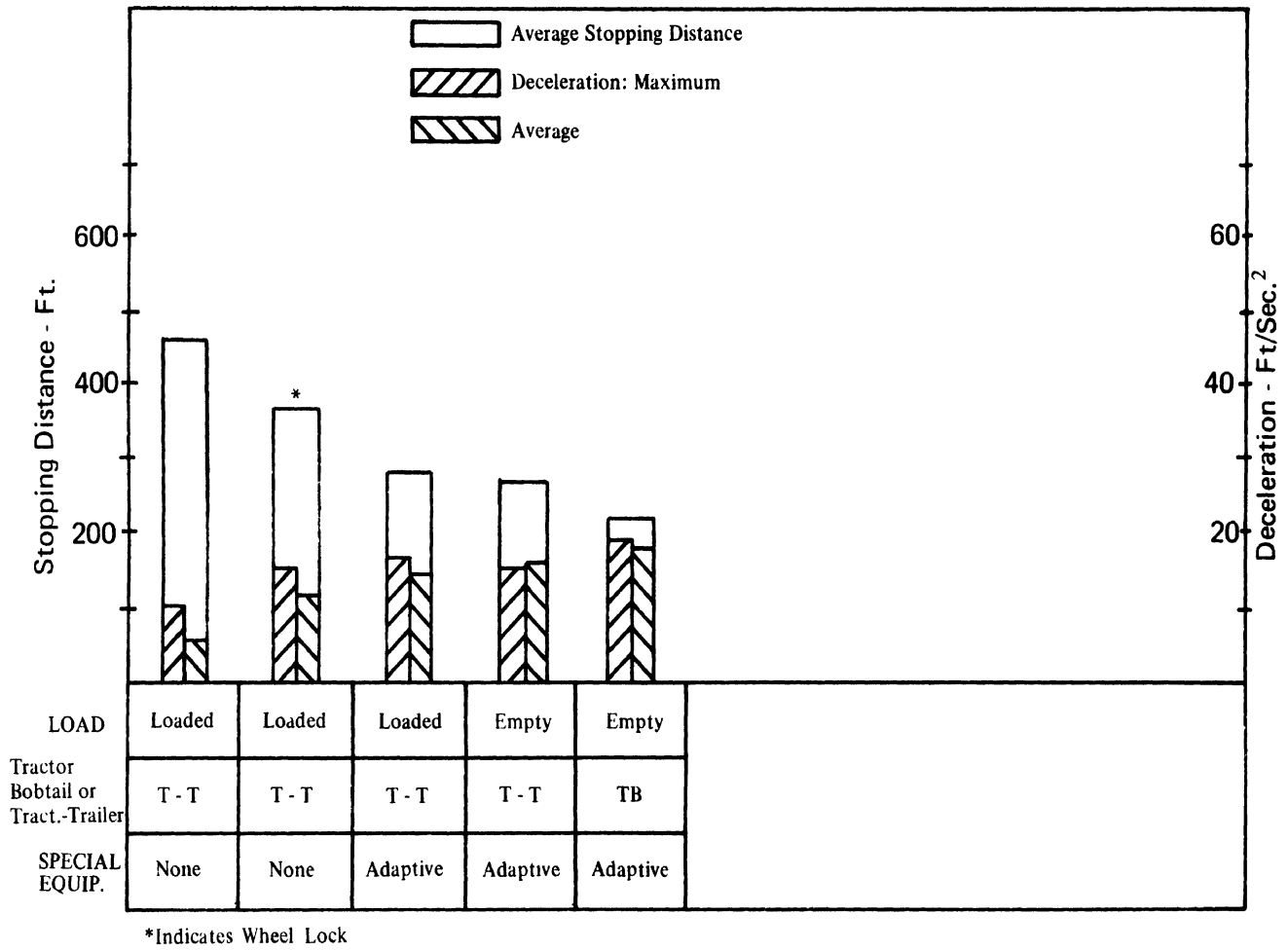
FIGURE 84. MINIMUM STOPPING DISTANCE TEST RESULTS, VEHICLE 14, DRY TRACK
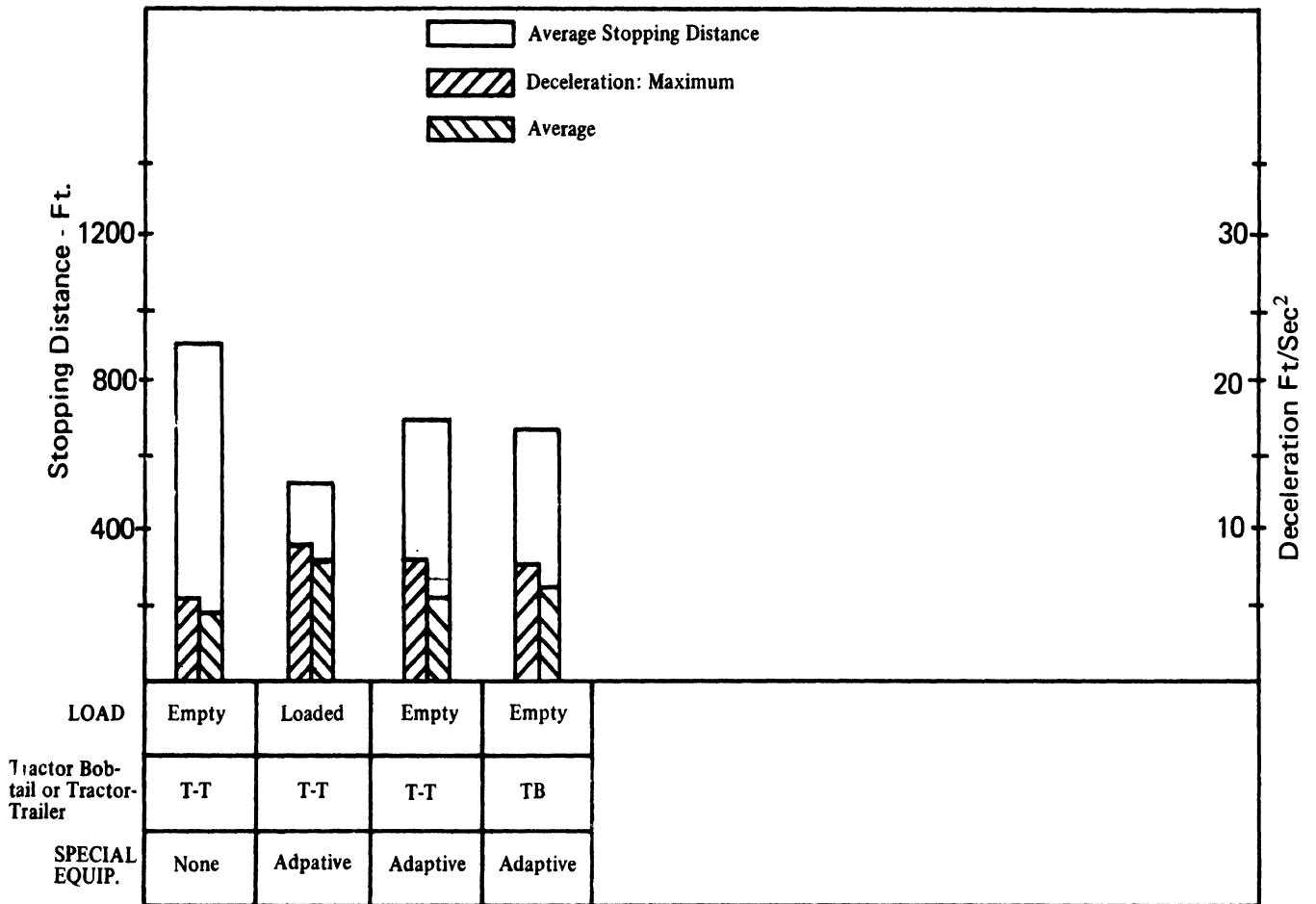
FIGURE 85. MINIMUM STOPPING DISTANCE TEST RESULTS, VEHICLE 14, WET TRACK
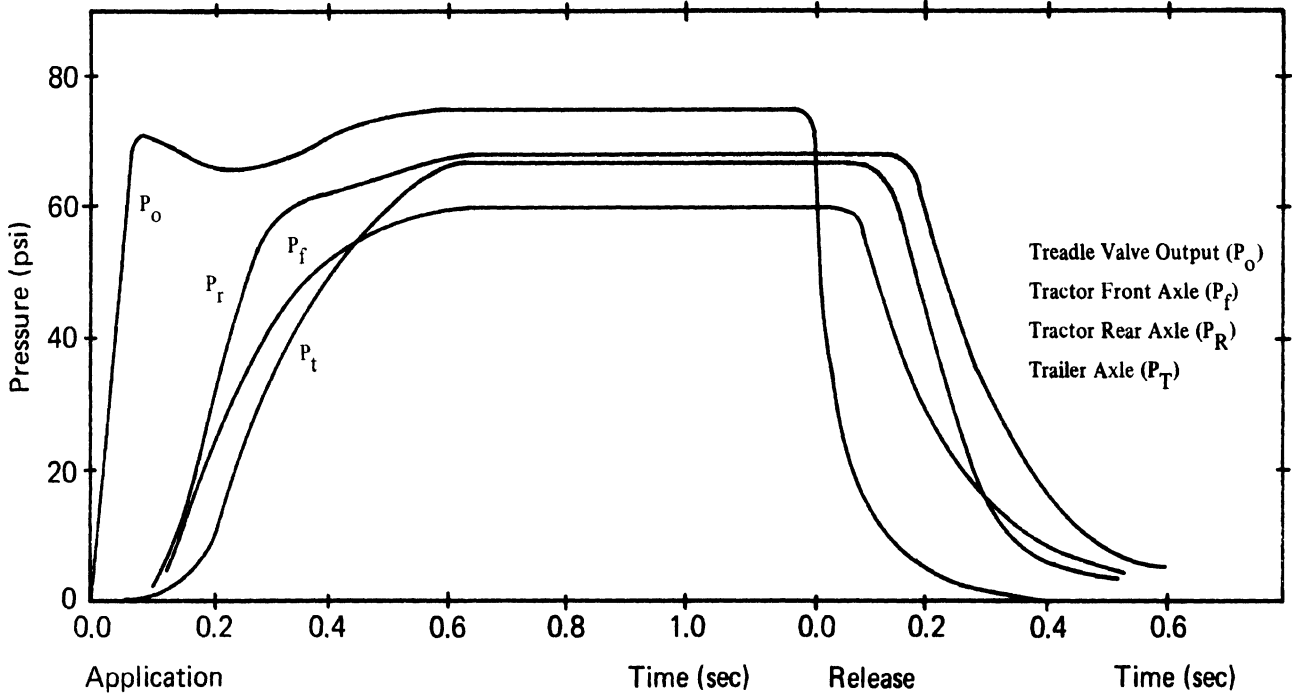
FIGURE 86. BRAKE RESPONSE TIME TEST, VEHICLE 14, STANDARD SYSTEM
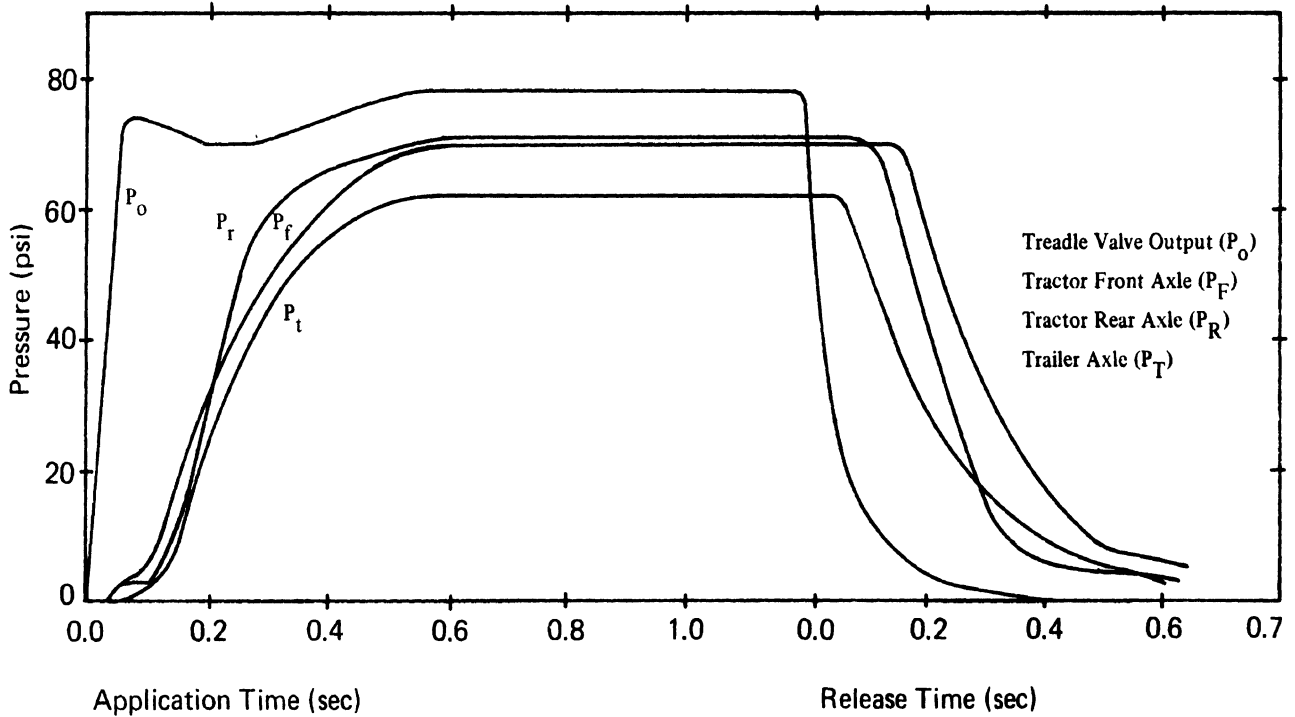
FIGURE 87. BRAKE RESPONSE TIME TEST WITH ANTILOCK SYSTEM ON

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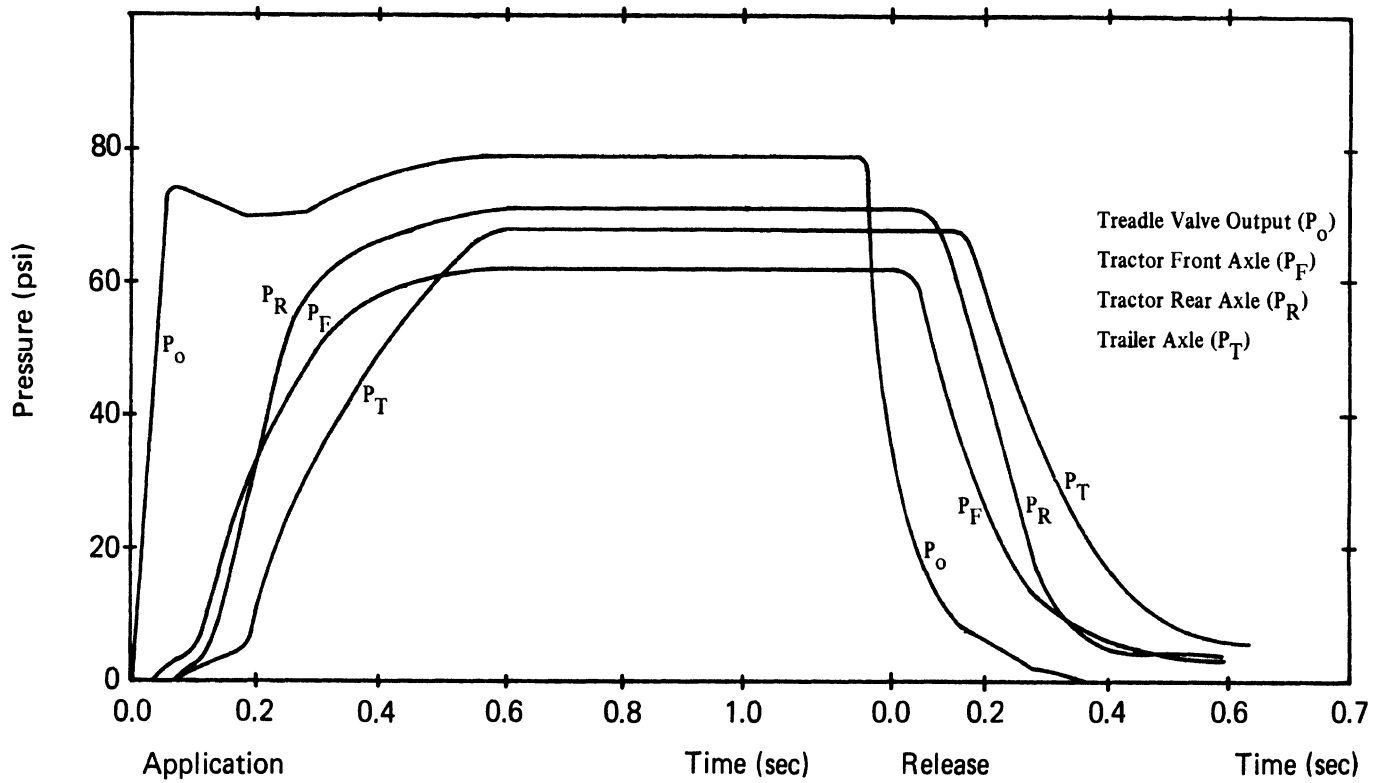
FIGURE 88. BRAKE RESPONSE TIME TEST, VEHICLE 14, AFTER ADJUSTMENT, STANDARD SYSTEM
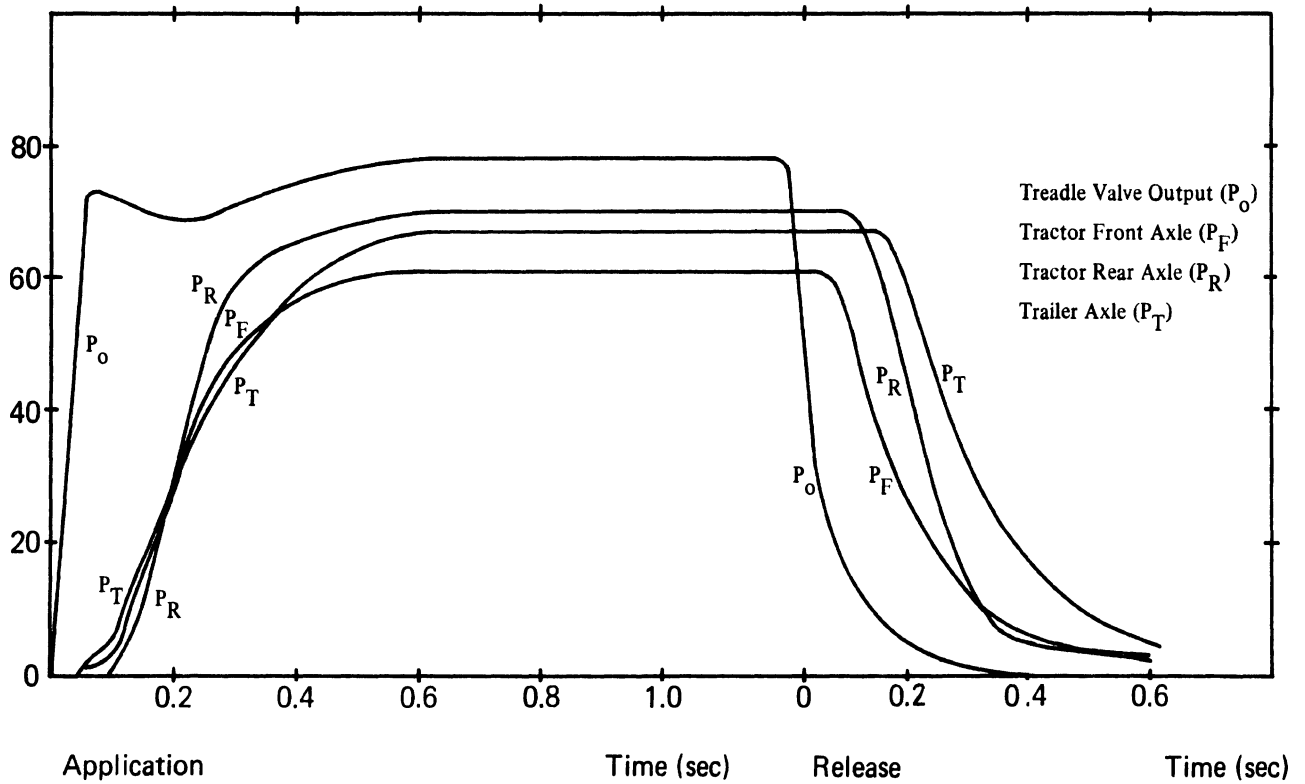
FIGURE 89. BRAKE RESPONSE TIME TEST, VEHICLE 14, AFTER ADJUSTMENT, WITH ANTI-LOCK

TABLE 32

MINIMUM STOPPING DISTANCE TEST DATA, VEHICLE 14

Test No.	Vehicle	Load	Track	Special Equipment	Actual		Corrected		Max. Decel., ft/sec²	Avg. Decel., ft/sec²
					Vo, mph	S, ft	Vo, mph	S, ft		
1	Tractor-Trailer	Loaded	Dry	None	51	329	60	455	10.0	8.5
					52*	269	60	358	14.0	10.8
2	Tractor-Trailer	Loaded	Dry	Adaptive Braking	53	211	60	270	15.2	14.3
3	Tractor-Trailer	Empty	Dry	Adaptive Braking	60	267	--	--	14.8	14.5
4	Tractor-Trailer	Empty	Wet	None	40	360	60	810	5.5	4.8
					39	451	60	1067	5.0	3.6
					39	424	60	1003	5.1	3.9
					39	455	60	1077	4.9	3.6
					39	336	60	795	5.5	4.9
5	Tractor-Trailer	Loaded	Wet	Adaptive Braking	41	240	60	514	8.0	7.5
					40	232	60	522	8.0	7.4
					40	227	60	510	8.5	7.6
6	Tractor-Trailer	Empty	Wet	Adaptive Braking	40	300	60	675	7.6	5.7
					40	315	60	708	7.6	5.5
					41	320	60	685	7.4	5.6

*Indicates wheel lockup.

TABLE 32 (Concluded)

MINIMUM STOPPING DISTANCE TEST DATA, VEHICLE 14

Test No.	Vehicle	Load	Track	Special Equipment	Actual		Corrected		Max. Decel., ft/sec²	Avg. Decel., ft/sec²
					V_o, mph	S, ft	V_o, mph	S, ft		
7	Tractor (Bobtail)	Empty	Dry	Adaptive Braking	60	202			19.4	19.2
					60	197			19.8	19.7
					60	214			18.2	18.1
					62	225	60	211	--	18.4
8	Tractor (Bobtail)	Empty	Wet	Adaptive Braking	40	295	60	663	7.0	6.8
					40	285	60	641	7.4	6.0
					39	279	60	660	7.1	5.9

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TABLE 33

SUMMARY OF AIR BRAKE RESPONSE TIME TESTS, VEHICLE 14

Brake System	Application Time to 60% of Max. Press.				Release Time to 5 psi, sec			
	Treadle Valve Output	Tractor Axles Front	Tractor Axles Rear	Trailer Axles	Treadle Valve Output	Tractor Axles Front	Tractor Axles Rear	Trailer Axles
Standard System as Tested	0.04	0.27	0.24	0.35	0.20	0.50	0.52	0.60
With Antilock System as Tested	0.04	0.26	0.24	0.26	0.20	0.52	0.44	0.64
Standard System After Adjustment	0.04	0.23	0.23	0.36	0.21	0.44	0.40	0.64
With Antilock System After Adjustment	0.04	0.22	0.23	0.27	0.20	0.44	0.40	0.60

TABLE 34

TEST SUMMARY, PERFORMANCE OF VEHICLES EQUIPPED WITH ADVANCED SYSTEMS ON DRY TRACK
(All Stopping Distances Corrected to 60 mph Initial Test Velocity)

| Vehicle | Minimum Stopping Distance, ft | | | | Maximum Deceleration, ft/sec^2 | | | |
| | Empty | | Loaded | | Empty | | Loaded | |
	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked	No Wheels Locked	Some Locked
Disk Brake Truck	255	198	204	181	21.8	22.8	21.3	23.8
Vehicle 12 Tractor-Trailer								
Standard Brakes	346	207	292	-	11.6	19.6	16.2	-
With Proportioning	354	259	-	-	11.3	15.7	-	-
With Adaptive System	216	-	281	-	18.0	-	17.0	-
With Adaptive System and Syncron	-	-	260	-	-	-	15.9	-
With Adaptive System on Tractor and Proportioning on Trailer	289	198	-	-	14.0	19.8	-	-
Vehicle 12 Tractor-Trailer								
Most Effective Brakes	-	-	317	212	-	-	15.1	18.6
With Adaptive System and Syncron	236	-	275	-	20.0	-	16.1	-
Vehicle 12 Tractor								
Standard Brakes	371	-	-	-	11.1	-	-	-
With Proportionings	203	-	-	-	19.8	-	-	-
With Adaptive System	204	-	-	-	19.7	-	-	-
Vehicle 12 Tractor								
Most Effective Brakes	-	-	288	-	-	-	16.9	-
With Adaptive System	-	-	227	-	-	-	19.4	-
Vehicle 14 Tractor-Trailer								
Standard Brakes*	-	-	455	358	-	-	10.0	14.0
With Antilock*	267	-	270	-	14.8	-	15.2	-
Vehicle 14 Tractor								
With Antilock*	210	-	-	-	19.1	-	-	-

*Vehicle 14 was tested on this skid pad approach road, which had a lower skid number than the East straightaway, where all the other dry tests were conducted.

TABLE 35

TEST SUMMARY, PERFORMANCE OF VEHICLES EQUIPPED WITH ADVANCED SYSTEMS
ON LOW COEFFICIENT SURFACE

(All Stopping Distances Corrected to 60 mph Initial Test Velocity)

Vehicle	Minimum Stopping Distance, ft		Maximum Deceleration, ft/sec^2	
	Empty	Loaded	Empty	Loaded
Vehicle 12 Tractor-Trailer				
Standard Brakes	--	749	--	6.1
With Adaptive System and Syncron	--	445	--	11.0
Vehicle 12 Tractor-Trailer				
Most Effective Brakes with Antilock and Syncron	476	550	9.2	8.7
Vehicle 12 Tractor				
Standard Brakes	874	--	4.1	--
With Proportioning	1194	--	3.4	--
With Adaptive System	510	--	9.1	--
Vehicle 12 Tractor				
Most Effective Brakes	1048	819	4.9	5.4
With Adaptive System	513	517	9.1	8.6
Vehicle 14 Tractor-Trailer				
Standard Brakes	939	--	5.2	--
With Antilock	689	516	7.5	8.2
Vehicle 14 Tractor				
With Antilock	655	--	7.2	--

however. After the split front/rear system was installed just prior to test-
ing, it was found that the full power hydraulic system did not have the capac-
ity to lock up all four wheels in tests with the loaded vehicle. Installation
of more accumulators and a higher capacity pump alleviated the problem.

Vehicle 12. No problems were encountered with the vehicle structure or
basic brake system except that midway through the tests all brake drums had
to be replaced due to severe hot spotting and heat checking. Brakes were re-
lined at the time of drum replacement. Several times during testing grease
seals on various axles had to be replaced due to faulty installation after
repair of wheel speed sensors for the adaptive braking system.

Vehicle 12, Adaptive Braking System. On the adaptive braking system,
the following components failed at least once: wheel speed sensors, sole-
noids on the modulating valves, and components in the electronic control
modules. The largest problem was encountered with failure of speed sensors,
which were developmental units, and were not designed to take the high brake
temperatures encountered in burnishing.

Vehicle 12, Syncron System. Air leaks in the system were detected
shortly after installation. Failure of the brake pedal mechanical stoplight
switch which triggers the syncron system also occurred shortly after instal-
lation.

Vehicle 14. Problems were encountered with the front wheel brakes on
this vehicle throughout the period of testing, including: structural failure
of the brake actuator support bracket, and a broken pipe fitting. Vibration
in the front brakes caused the brake chambers to work loose periodically, and
the pressure transducer on the chamber to work loose and blow out. No prob-
lems were encountered with the antilock system during testing.

4. ANALYTICAL PROGRAM

4.1 SCOPE OF PROGRAM

The purpose of the analytical study was to establish mathematical procedures for predicting vehicle braking performance, specifically: brake effectiveness, braking efficiency, pedal force gain, and thermal response. Analytical methods found in the literature were adapted for use in the study.

The test results indicated that the loaded vehicles experienced considerable brake fade when being decelerated from speeds of 60 mph. The test data also showed that wheel lockup occurred prematurely on axles of a tandem-axle pair mounted on either a tractor or trailer. The analysis developed to predict maximum deceleration capability necessarily had to include fade effects as well as dynamic load transfer occurring on tandem-axle suspensions. Since this analysis resulted in a rather extensive set of algebraic equations, a digital computer program was developed to perform the calculations. A description of this performance prediction scheme is given below.

4.2 THEORETICAL PREDICTION OF BRAKING PERFORMANCE

For vehicles equipped with unassisted hydraulic brake systems, the pedal force is the following function of line pressure:

$$F_p = \frac{p_h \cdot A_{MC}}{\ell_p \cdot \eta}$$

where p_h = line pressure

A_{MC} = master cylinder area

ℓ = pedal lever ratio

η = mechanical efficiency   (4-1)

The mechanical efficiency is the product of the lever efficiency η_1 and the cylinder efficiency η_c, viz.:

$$\eta = \eta_1 \cdot \eta_c \qquad\qquad (4\text{-}2)$$

Typical values of lever efficiency η_1 range from 0.92 to 0.94. Cylinder efficiency η_c is typically 0.96 and 0.88 for single-piston and tandem master cylinders, respectively.

For vacuum-assisted hydraulic brake systems, Equation 4-1 becomes

$$F_p = \frac{p_h \cdot A_{MC}}{\ell_p \cdot \eta \cdot B^*} \qquad\qquad (4\text{-}3)$$

where   B* = vacuum assist characteristics, defined by the ratio of pushrod
          force upon the master cylinder piston over the pedal force
          multiplied by the pedal lever ratio.

In air brake systems, the applied pedal force merely controls the air
flow through the brake valve and does not serve as a work producing element.
Thus, no attempt was made to predict the pedal force/line pressure relation-
ship in air braked vehicles.

The actual brake force per axle created by a hydraulic brake system is
given by (3,50):

$$F_{x,h} = 2(p_{\ell} - p_o)A_{wc}\eta_c(BF)\frac{r}{R} \qquad (4\text{-}4)$$

where

p_{ℓ} = brakeline pressure
p_o = pushout pressure
A_{wc} = wheel cylinder area
BF   = brake factor, defined as the ratio drum drag over the actua-
        ting force of one brake shoe (169)
r   = effective drum or disk radius
R   = effective tire radius
η_c = mechanical efficiency $\approx$ 0.96

A similar expression serves to compute the brake force per axle of a
vehicle equipped with air brakes:

$$F_{x,a} = 2(P_{\ell} - p_o)A_c\eta_m(BF)\frac{r}{R}\rho \qquad (4\text{-}5)$$

where

ρ = lever ratio between brake chamber and brake shoe
A_c = brake chamber area
η_m = mechanical efficiency between brake chamber and shoe actuation
For S-cam brakes, the lever ratio is given by

$$\rho = \frac{\ell_s}{2\ell_c} \qquad (4\text{-}6)$$

where

ℓ_s = effective slack adjuster length
ℓ_c = effective cam radius
For wedge brakes, the lever ratio is related to the wedge angle α, viz.,

$$\rho = \frac{1}{2\tan(\alpha/2)} \qquad (4\text{-}7)$$

If the brakes are in good mechanical condition, the mechanical efficiencies exhibited by S-cam and wedge brakes range from 0.70 to 0.75 and 0.80 to 0.88, respectively.

Analytical expressions for predicting the brake factor of various types of drum and disc brakes as a function of brake geometry and the friction coefficient of the lining have been developed by Strien (169). These techniques were deemed to be the most accurate available and have been used exclusively in this study. Typical brake factor/lining friction coefficient relationships for a leading-trailing shoe, a two-leading shoe, and a duo-servo brake are presented in Fig. 90.

The "sensitivity" of a brake is defined by Strien as (169,185):

$$\varepsilon = \frac{d(BF)}{d(\mu_L)} \qquad\qquad (4\text{-}8)$$

where

$\quad \mu_L$ = lining friction coefficient

Note that as defined, the brake sensitivity ε is the change in brake factor (and hence brake torque) with respect to a change in the friction coefficient of the brake lining. Brake sensitivity is included in the analysis by computing the theoretical brake factor as a function of lining friction coefficient for each brake using design information. An example computation is shown in Fig. 91. A non-faded operating design point is established on this curve by the value for μ (designated as μ_{Lh}) provided by the vehicle or brake manufacturer. (Typical values of μ_{Lh} for S-cam and wedge brakes are 0.35 to 0.38 and 0.45 to 0.48, respectively.) On drawing a straight line tangent to the curve at the point corresponding to μ_{Lh} (as indicated in Fig. 91) a brake sensitivity is established at the operating design point. The value for brake sensitivity determined in this manner can then be used to calculate a brake factor (approximate) as a function of values of μ_L that depart from the design operating point as a result of fade.

The lining friction coefficient μ_L is a function of temperature, velocity, and pressure (109,182,183). Since all vehicles were tested under almost identical conditions of velocity and ambient temperature, and since the brake temperatures did not very significantly from vehicle to vehicle, the coefficient of friction of the lining can be expressed as a function of pressure between lining and drum, viz.:

$$\mu_L = \mu_{L\ell} + (\mu_{Lh} - \mu_{L\ell})e^{-fp_m} \qquad\qquad (4\text{-}9)$$

where

$\quad p_m$ is the mean pressure between brake shoe and drum

$\quad \mu_{L\ell}$ is the lining coefficient of friction at an arbitrarily specified "faded" condition (see Fig. 91)
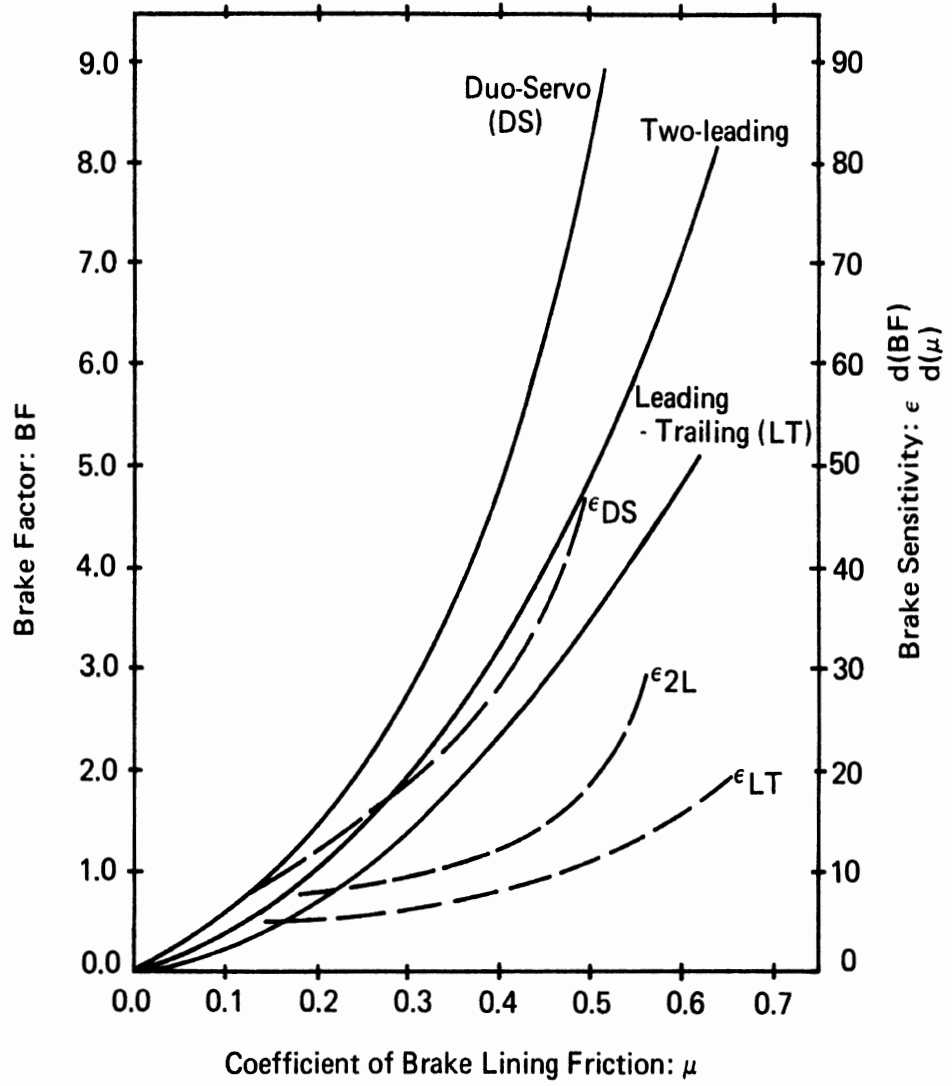
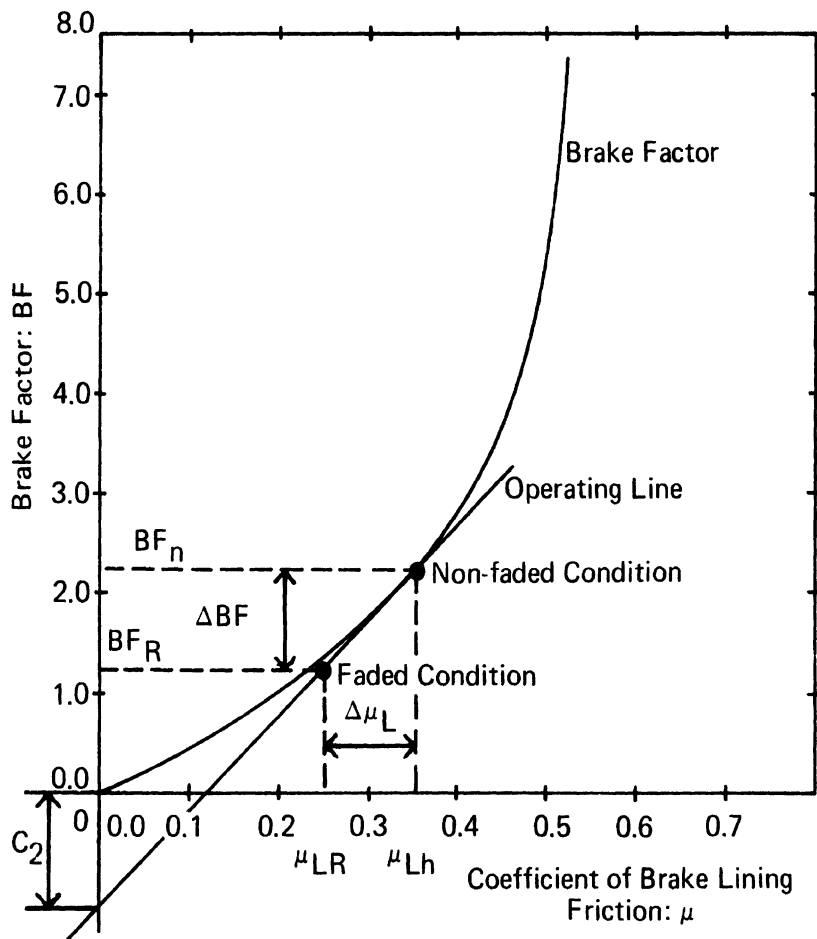FIGURE 90. BRAKE FACTOR VS LINING FRICTION COEFFICIENT

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FIGURE 91. EXACT AND APPROXIMATE BRAKE FACTOR

Since a comparative calculation has shown that the product $\mu_L p$ does not vary considerably from one test vehicle to another, the mean pressure p_m between shoe and drum can be replaced by the line pressure at the individual brake chambers. The fade coefficient f as well as the low limit for the lining friction coefficient $\mu_{L\ell}$, were determined from test data and curve fitting procedures. Vehicles with air brake systems yielded fade coefficients of 0.018 [in.^{2}/lb] when loaded and 0.003 [in.^{2}/lb] when empty. For vehicles equipped with hydraulic brake systems, the corresponding values for f are 0.00088 [in.^{2}/lb] and 0.00028 [in.^{2}/lb]. Analysis of the experimental data indicated that the maximum reduction in lining friction coefficient could be approximated reasonably well by assuming that $\mu_{L\ell} \doteq 0.70\ \mu_{Lh}$.

If vehicles were equipped with proportioning or limiting valves, line pressure was introduced into Equation 4-4 or 4-5 as a variable determined by the proportioning or limiting used on a particular axle. The brake force per axle can be expressed as:

$$F_{x,a} = 2(p_\ell - p_o)A_c \eta_m \{C_1[\mu_{L\ell} + (\mu_{Lh} - \mu_{L\ell})]e^{-f(p_\ell - p_o)} - C_2\}\frac{r}{R}\rho \quad (4\text{-}10)$$

where
$$C_1 = \frac{\Delta(BF)}{\Delta(\mu_L)} \text{ and } C_2 \text{ as shown in Fig. 91.}$$
On evaluating Equation 4-10 for each axle as a function of line pressure, the total retarding force acting on the vehicle is obtained from

$$F_{x,total} = \sum^{i} F_{x,i} \quad (4\text{-}11)$$

where i designates the number of braked axles.

For two-axle trucks, the equations describing dynamic axle loading may be expressed as (19,21,26):

$$F_{z,Front/dyn} = [(1 - \psi) + \chi a]W \quad (4\text{-}12a)$$

$$F_{z,Rear/dyn} = [\psi - \chi a]W \quad (4\text{-}12b)$$

where
$$\psi = \frac{F_{z,Rear/static}}{W}$$

$$\chi = \frac{center\ of\ gravity\ height}{wheelbase}$$

W = vehicle weight

$$a = \frac{F_{x,total}}{W}$$

For tandem-axle trucks and tractor-semitrailer combinations, the above expressions are considerably more complicated (3), but are nevertheless amenable to computer solution. For a given brake-line pressure, Equations 4-10, 4-11, and 4-12 serve to define the deceleration of the vehicle, and the dynamic axle loads existing at this deceleration. The tire-roadway friction coefficient required to prevent wheel lockup is given by

$$\mu_{Road,i} = \frac{F_{x,i}}{F_{z,i}/dyn} \tag{4-13}$$

and the resulting braking efficiency achieved by individual axles is defined as

$$E_i = \frac{a}{\mu_{Road,i}} \tag{4-14}$$

If wheel lockup is found to occur on some but not all axles at a given line pressure, the retarding force produced by the axle with locked wheels is assumed to be given by the following relationship:

$$F_{x,i/slide} = \mu_{road} F_{z,i/dyn} \tag{4-15}$$

where μ_{road} corresponds to the actual coefficient of friction existing at tire-road interface.

For eleven of the thirteen vehicles tested, braking performance diagrams were constructed, in which by use of analytical expressions, relationships between pedal force, brake line pressure, vehicle deceleration capability (loaded and unloaded) and tire-road friction coefficient required to prevent wheel lockup on a given axle are depicted.* Figures 92 through 104 show the braking performance diagrams for these vehicles. Corresponding braking efficiency diagrams are given in Figs. 105 through 116. In these diagrams good agreement is shown between theoretical and test results, indicating that accurate predictions of braking performance can be made based upon vehicle and brake system design data for buses, trucks, and tractor-trailers.

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*Exceptions were the heavy truck and the city bus, for which diagrams could not be constructed because necessary design information was not available from the vehicle manufacturers.
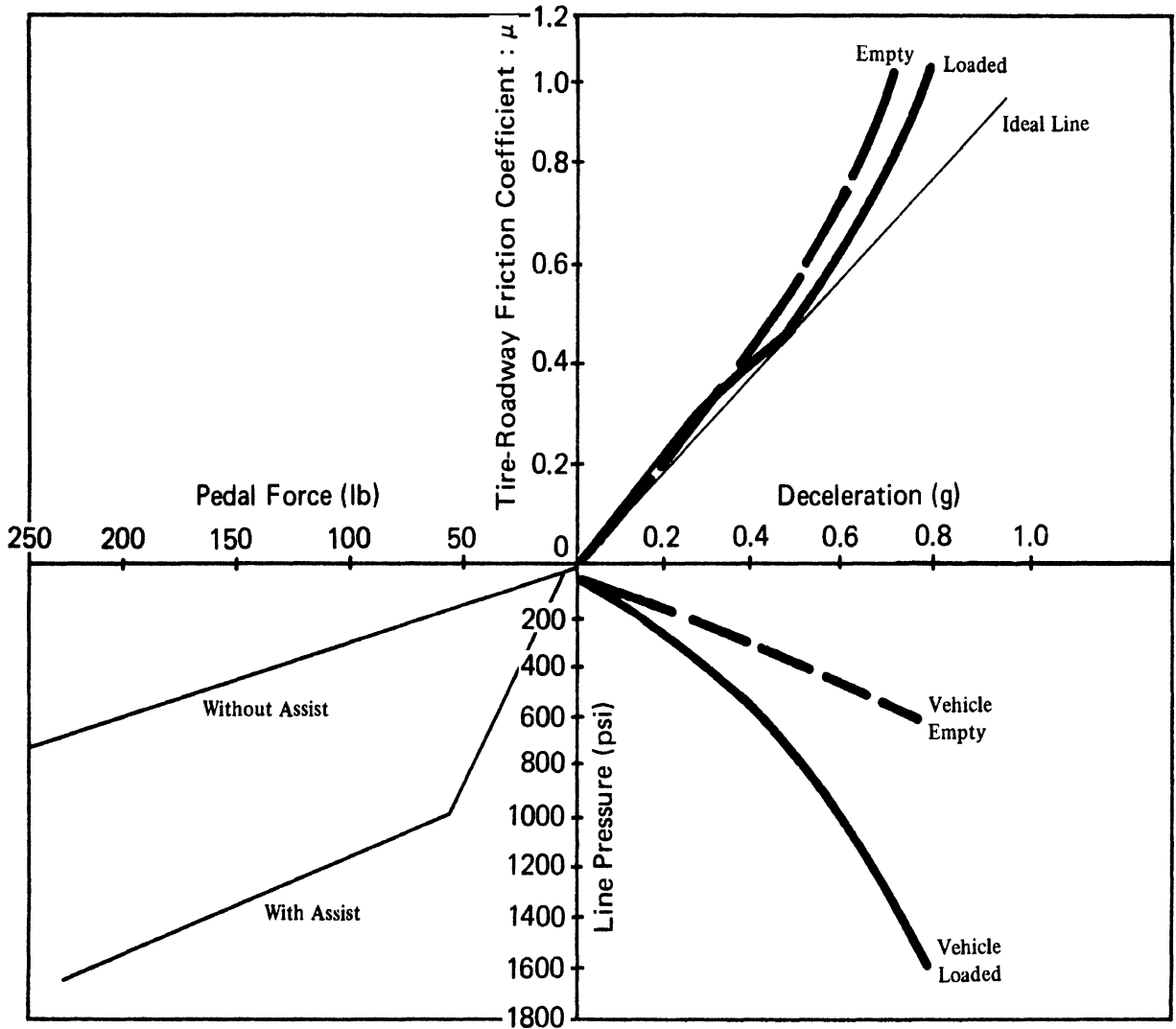
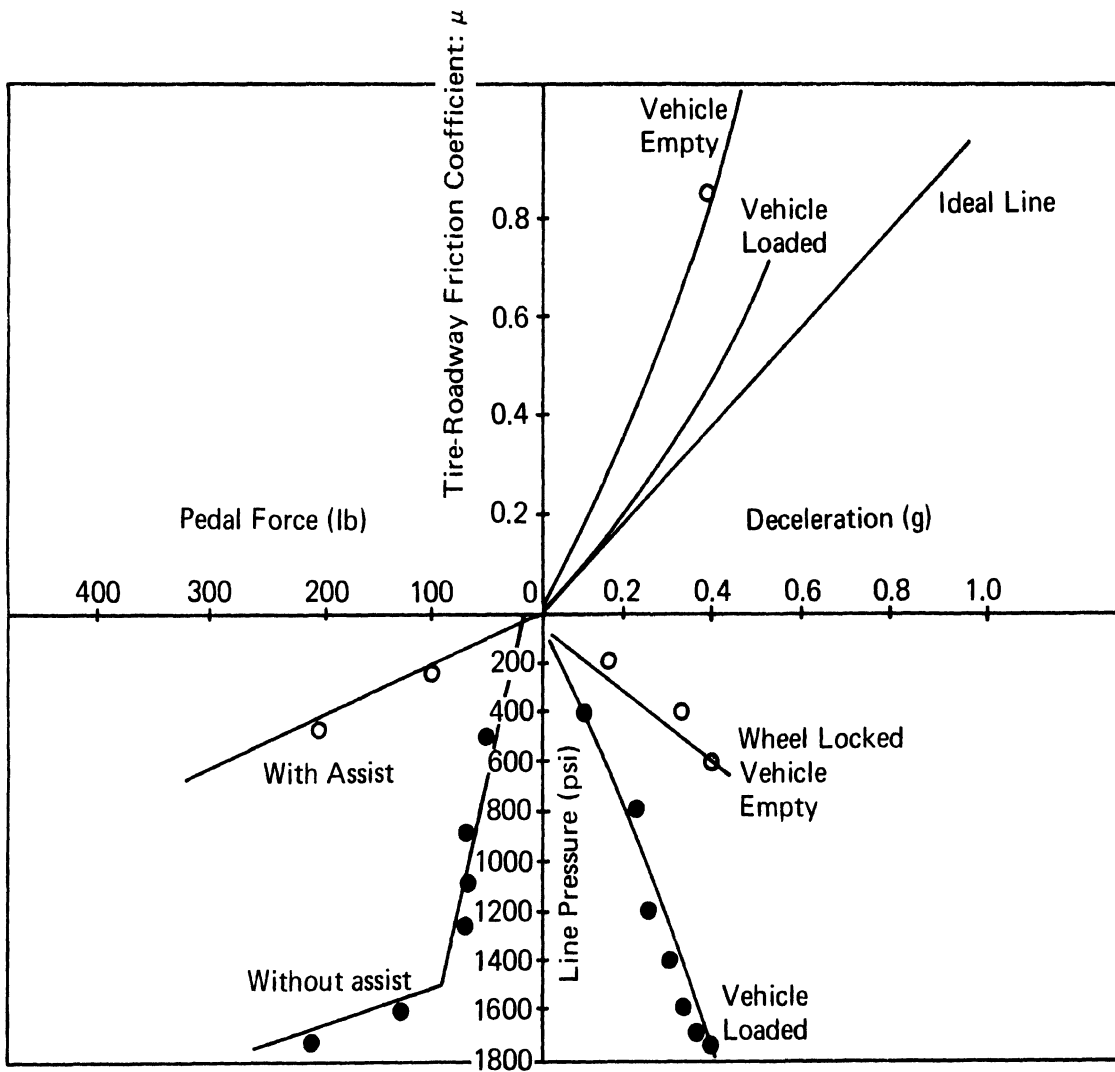FIGURE 92. BRAKING PERFORMANCE DIAGRAM FOR LIGHT TRUCK

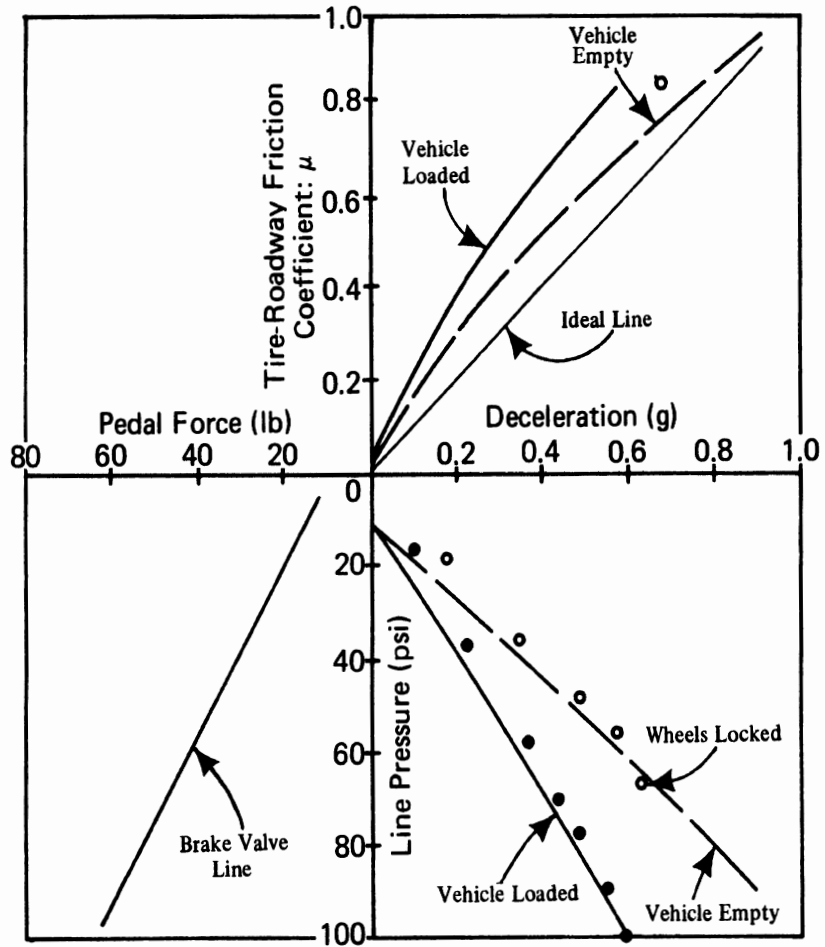FIGURE 93. BRAKING PERFORMANCE DIAGRAM FOR MEDIUM TRUCK

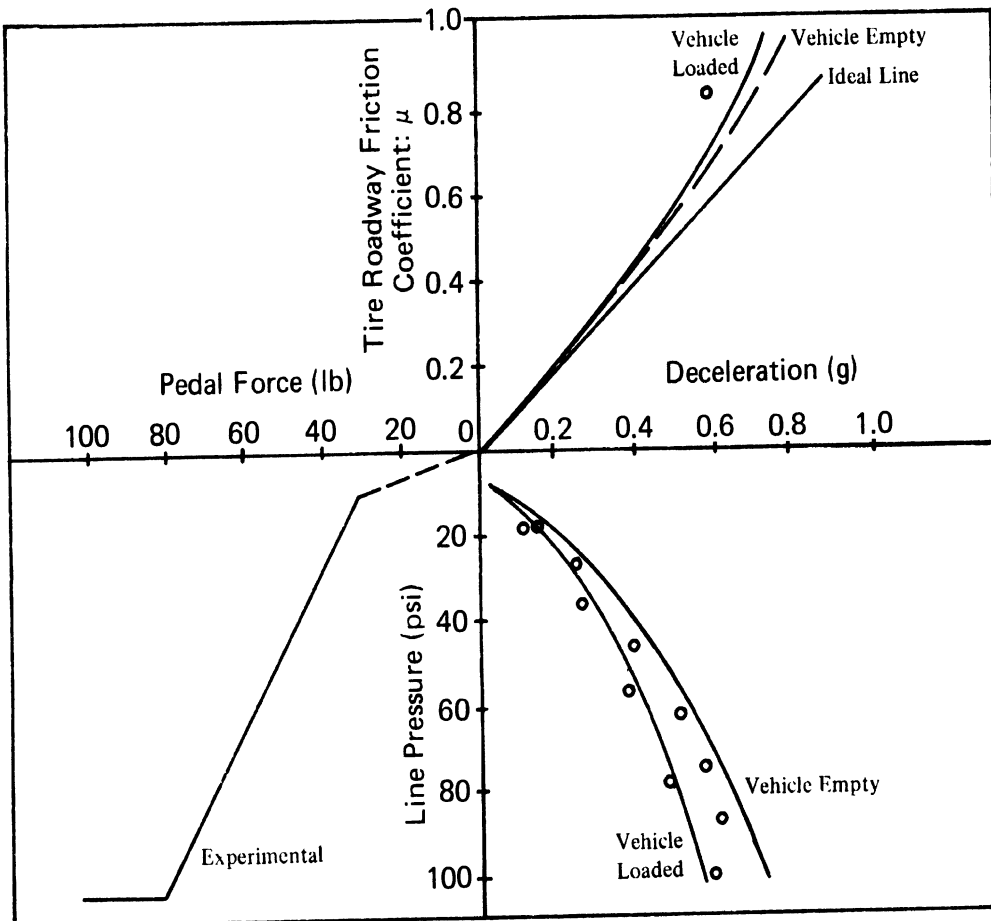FIGURE 94. BRAKING PERFORMANCE DIAGRAM FOR SCHOOL BUS
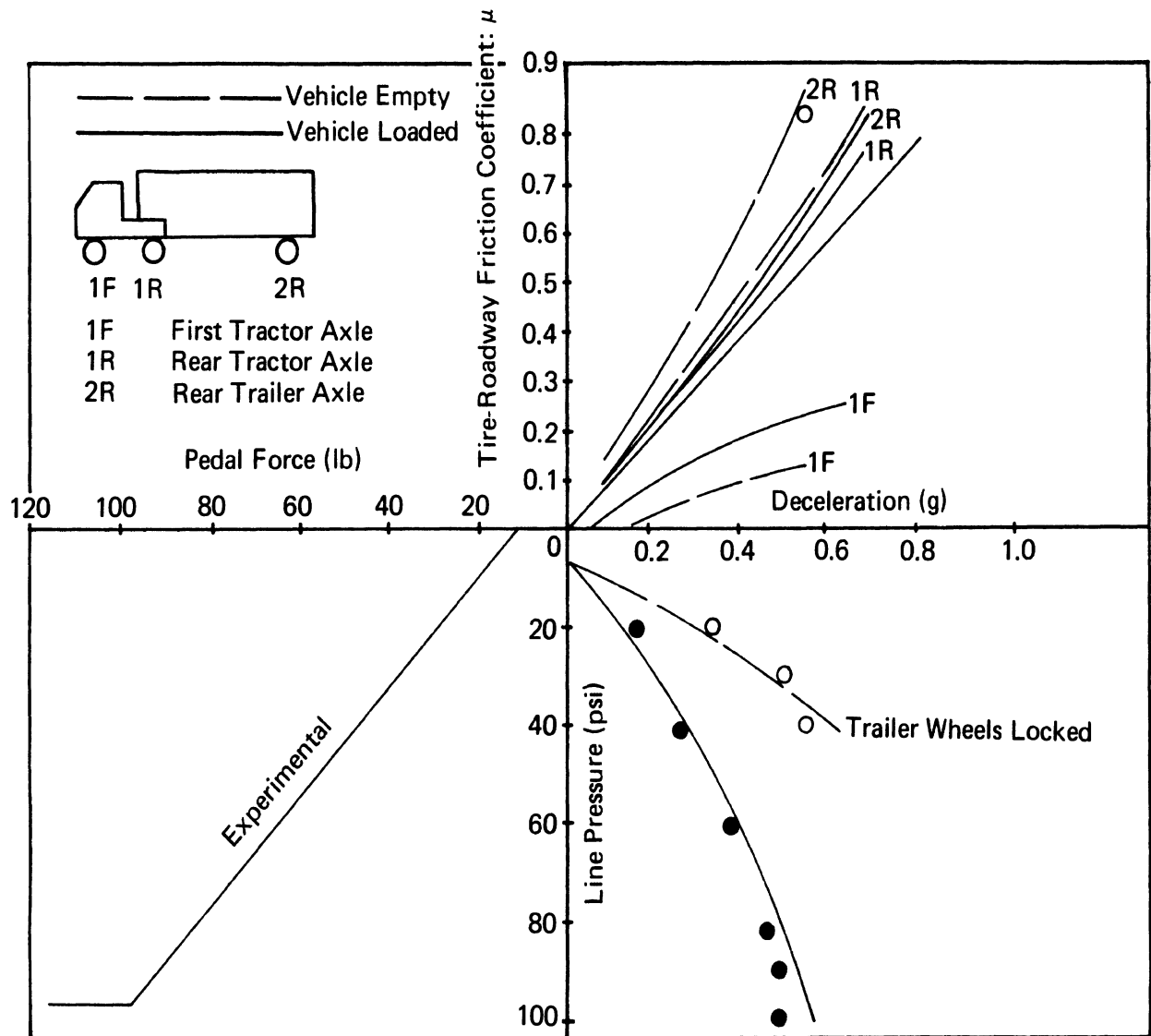
FIGURE 95. BRAKING PERFORMANCE DIAGRAM FOR INTERCITY BUS

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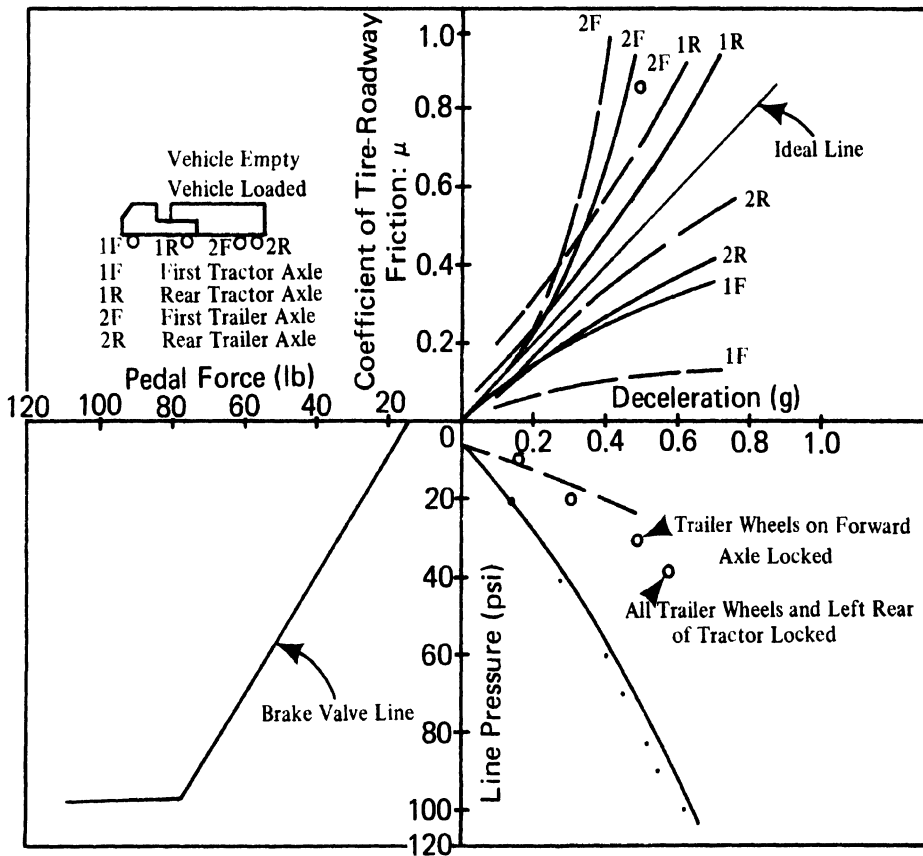FIGURE 96. BRAKING PERFORMANCE DIAGRAM FOR TRACTOR-TRAILER 2-S1

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FIGURE 97. BRAKING PERFORMANCE DIAGRAM FOR TRACTOR-TRAILER 2-S2
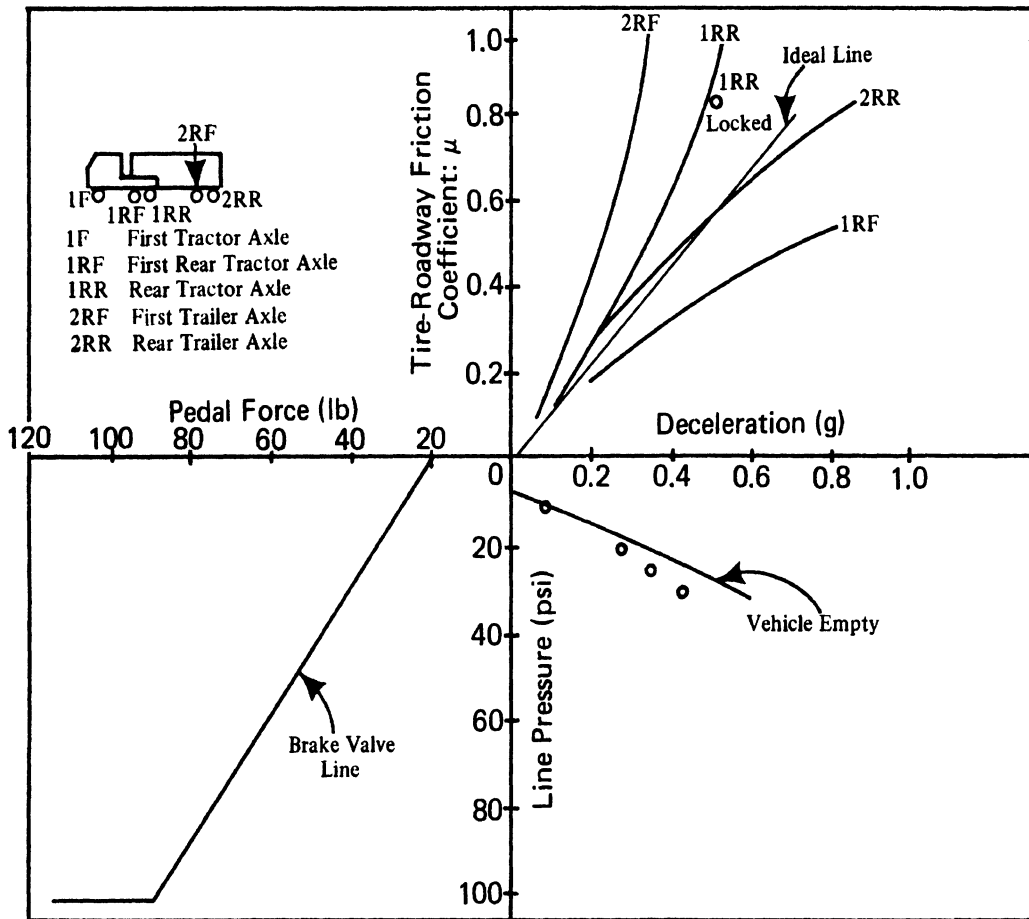
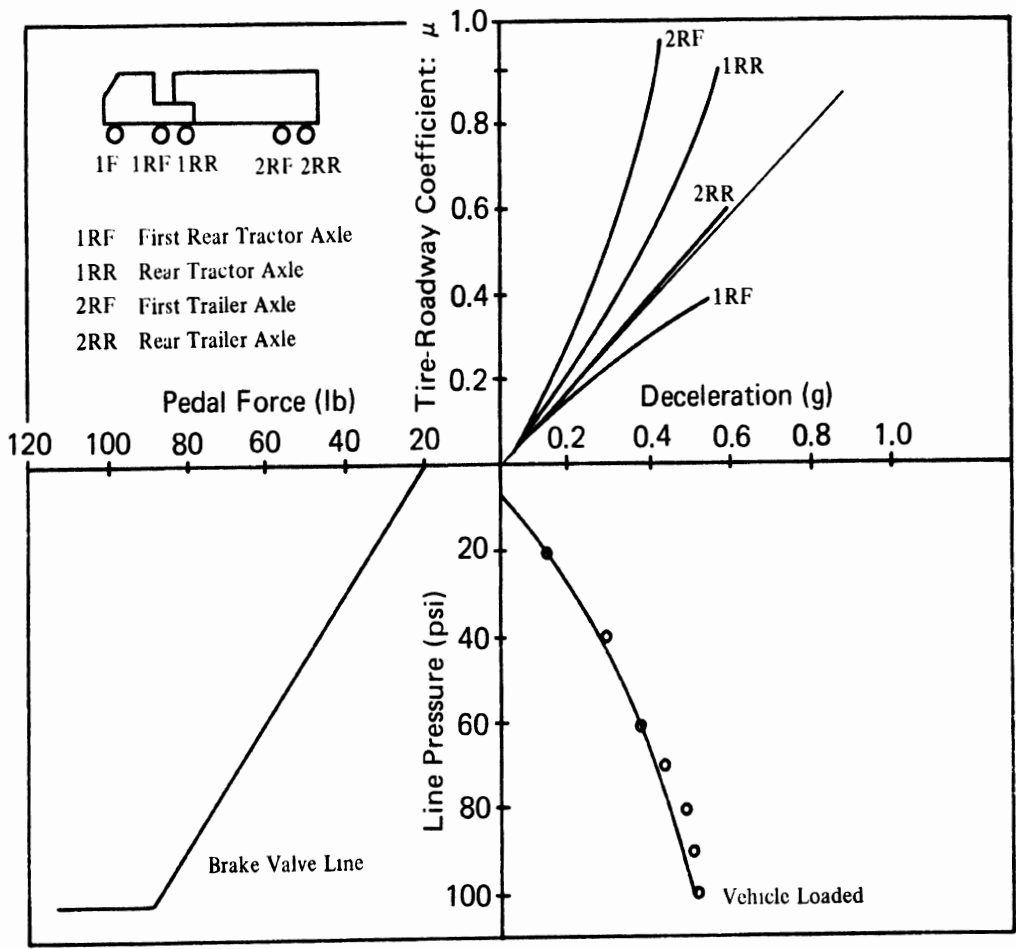FIGURE 98. BRAKING PERFORMANCE DIAGRAM FOR EMPTY TRACTOR-TRAILER 3-S2

FIGURE 99. BRAKING PERFORMANCE DIAGRAM FOR LOADED TRACTOR-TRAILER 3-S2
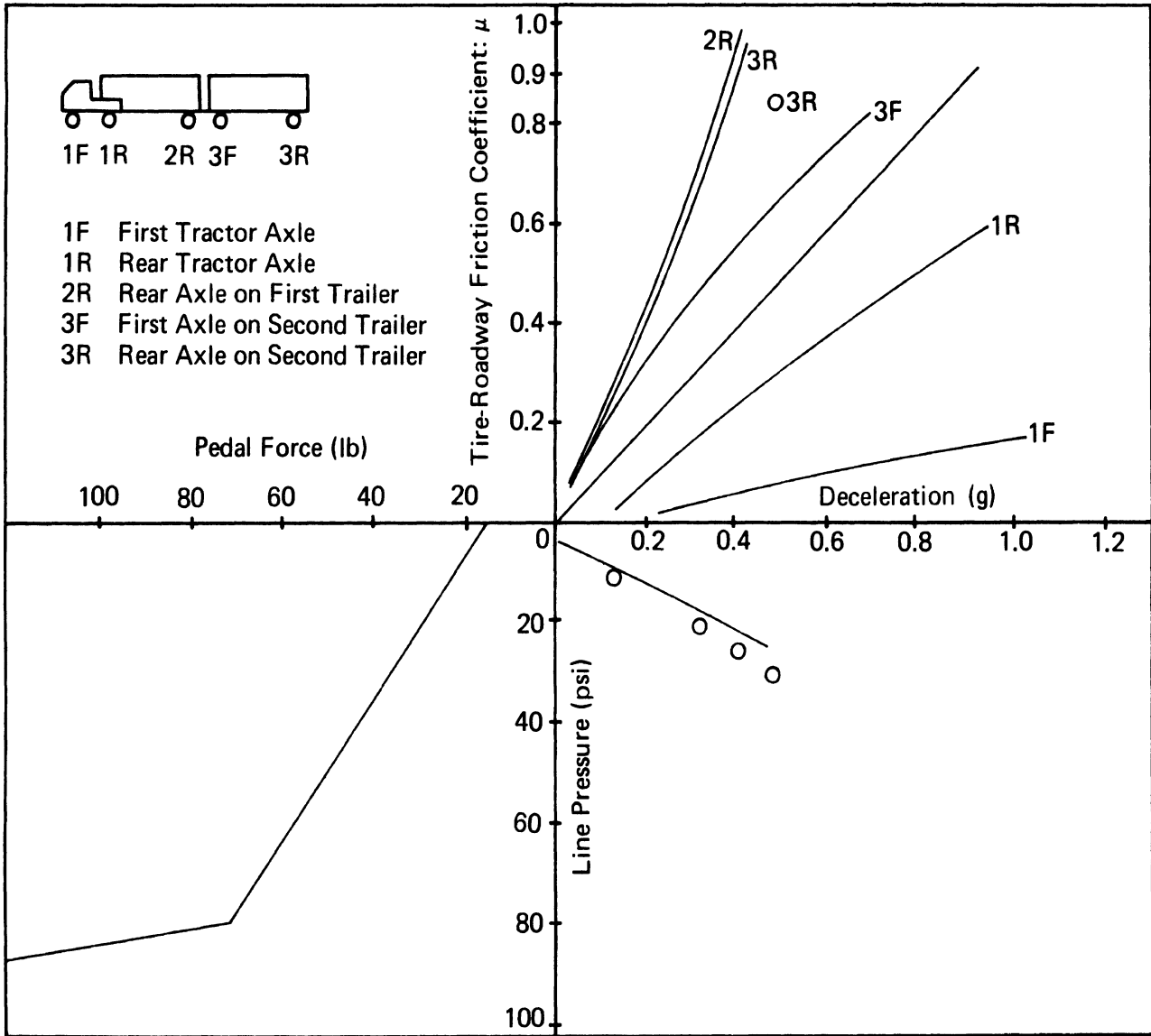
FIGURE 100. BRAKING PERFORMANCE DIAGRAM FOR EMPTY DOUBLES COMBINATION 2-S1-2

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FIGURE 101. BRAKING PERFORMANCE DIAGRAM FOR LOADED DOUBLES COMBINATION 2-S1-2

FIGURE 102. BRAKING PERFORMANCE DIAGRAM FOR DISK BRAKE TRUCK
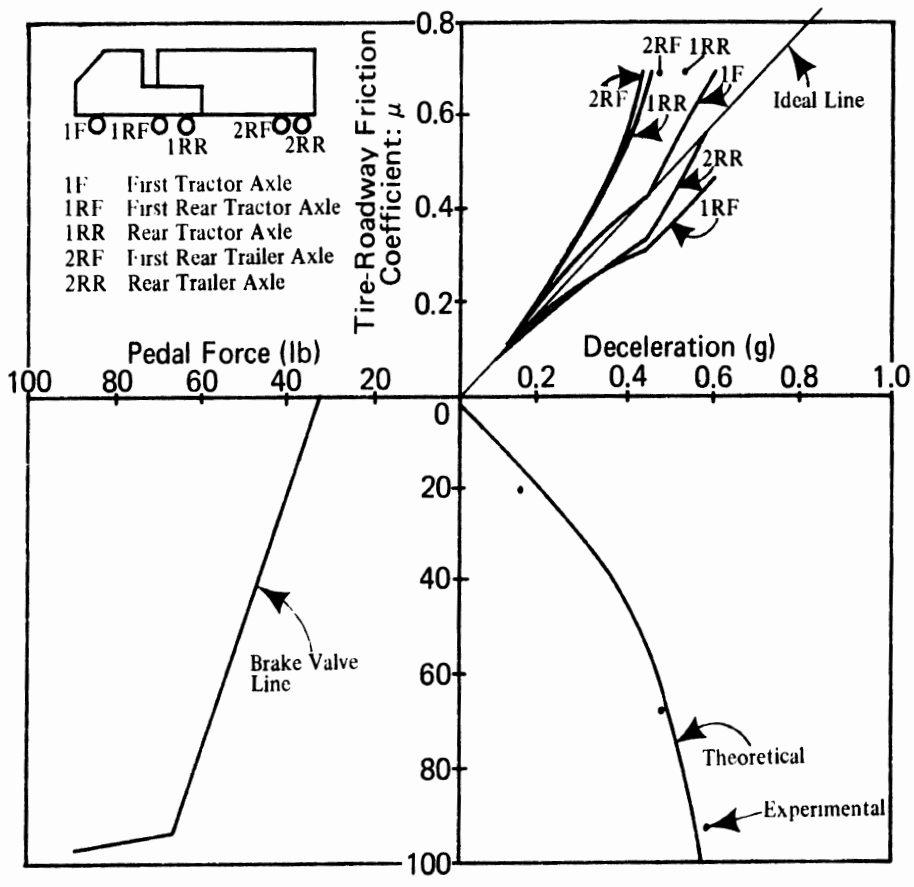
FIGURE 103. BRAKING PERFORMANCE DIAGRAM FOR VEHICLE 12, LOADED, WITH MOST EFFECTIVE BRAKES

FIGURE 104. BRAKING PERFORMANCE DIAGRAM FOR VEHICLE 14
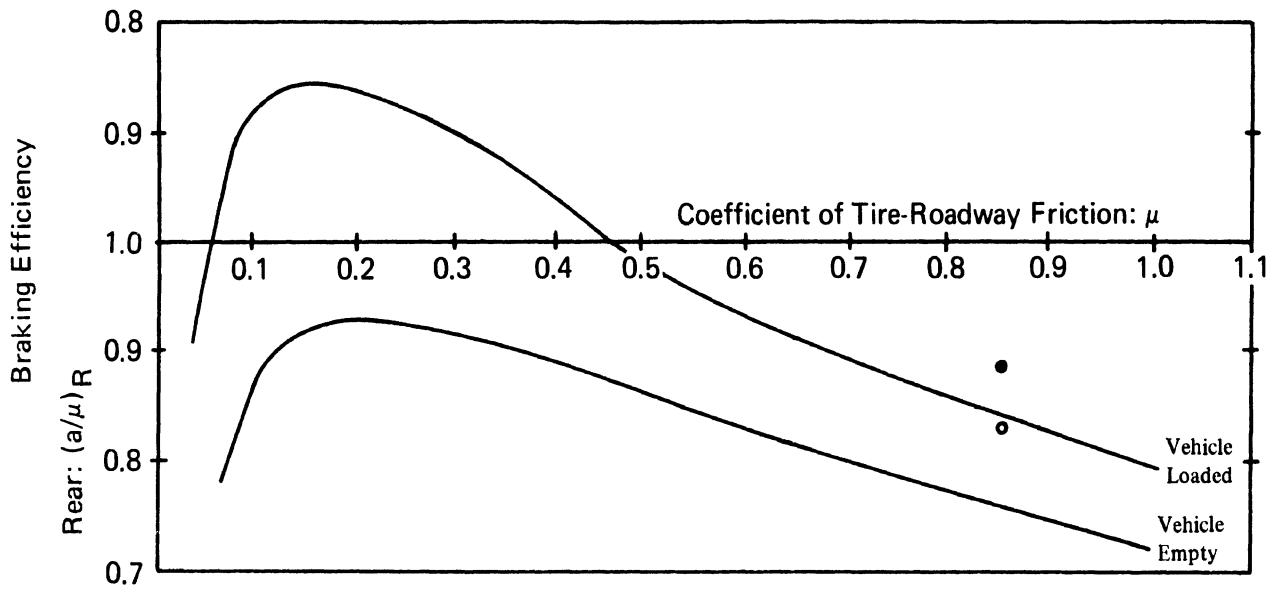
FIGURE 105. BRAKING EFFICIENCY FOR LIGHT TRUCK
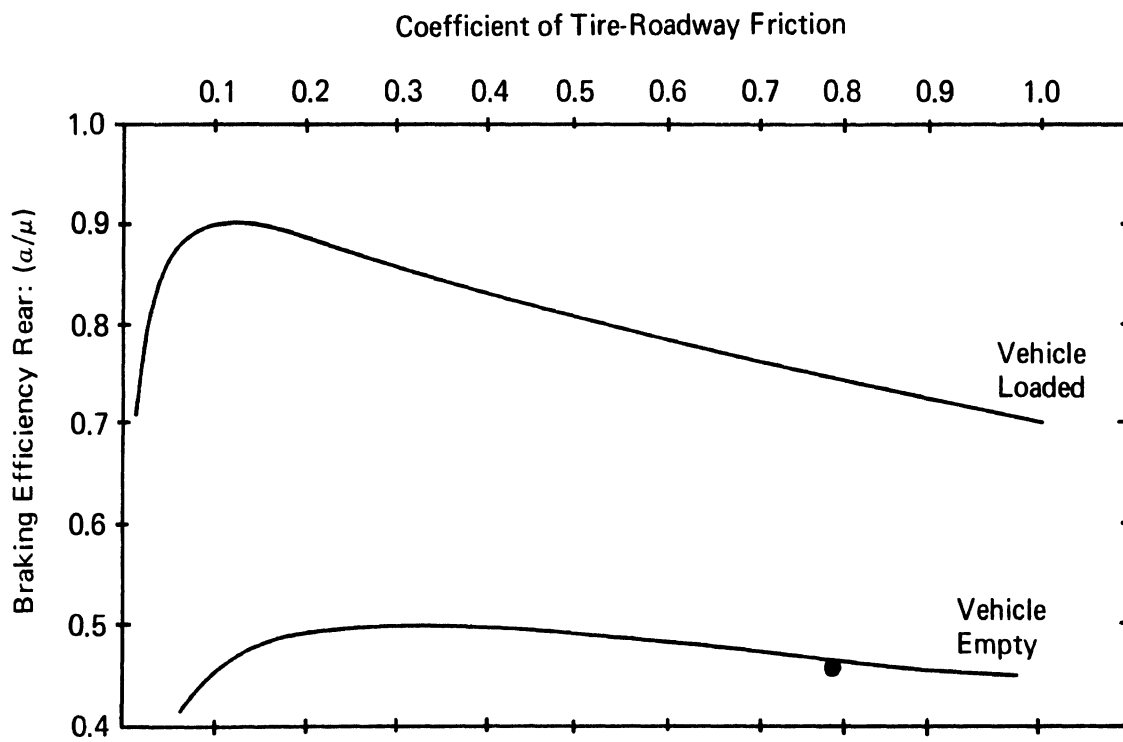
FIGURE 106. BRAKING EFFICIENCY FOR MEDIUM TRUCK
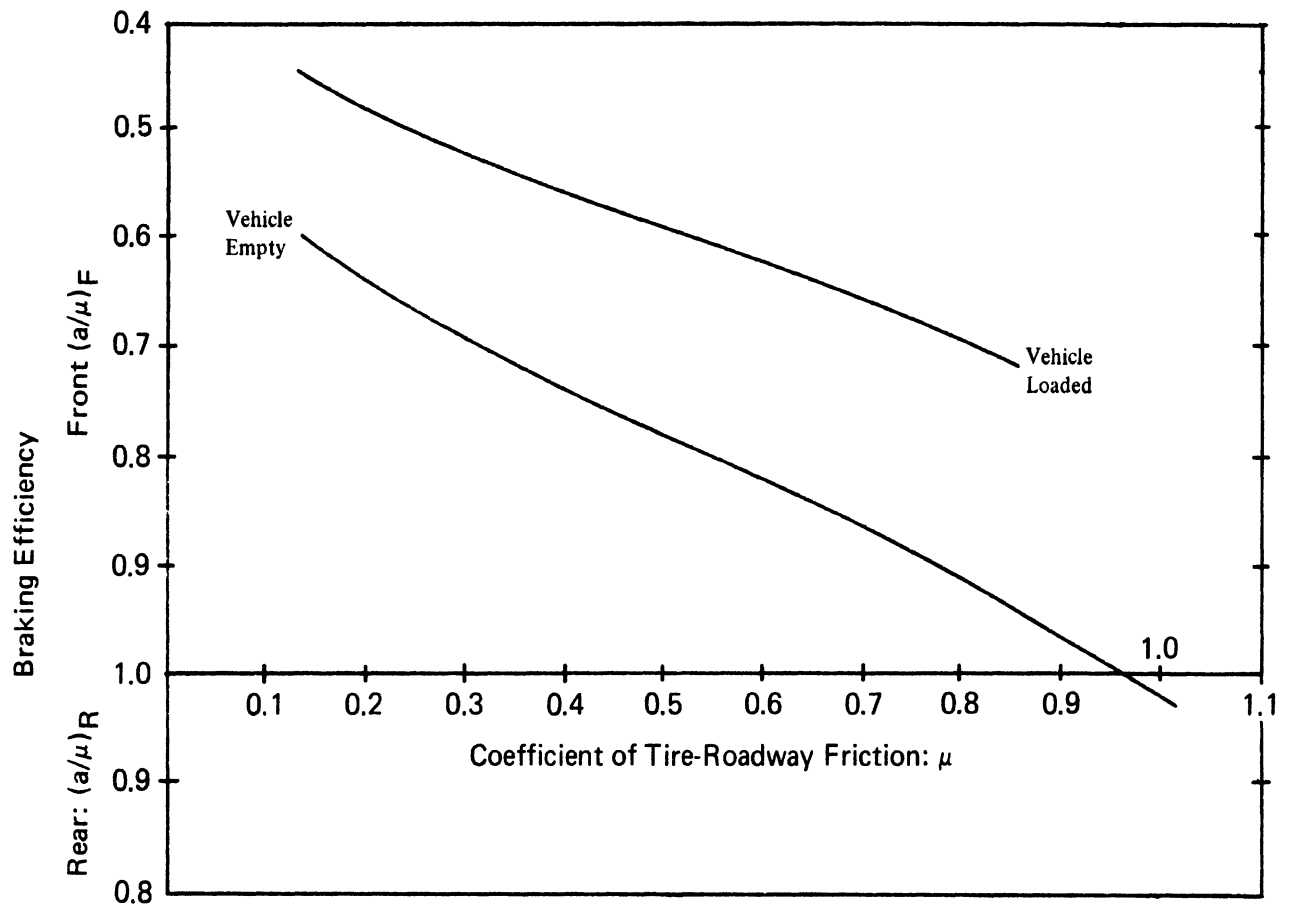
FIGURE 107. BRAKING EFFICIENCY FOR SCHOOL BUS
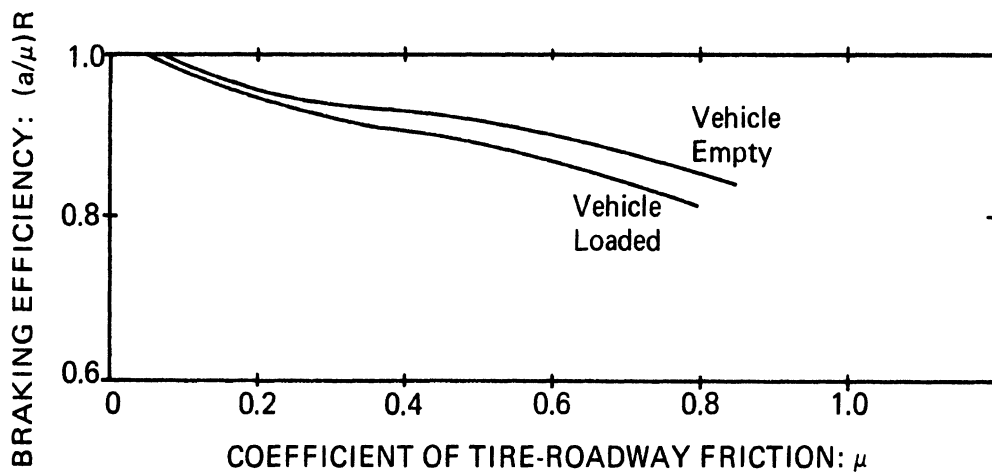
FIGURE 108. BRAKING EFFICIENCY FOR INTERCITY BUS
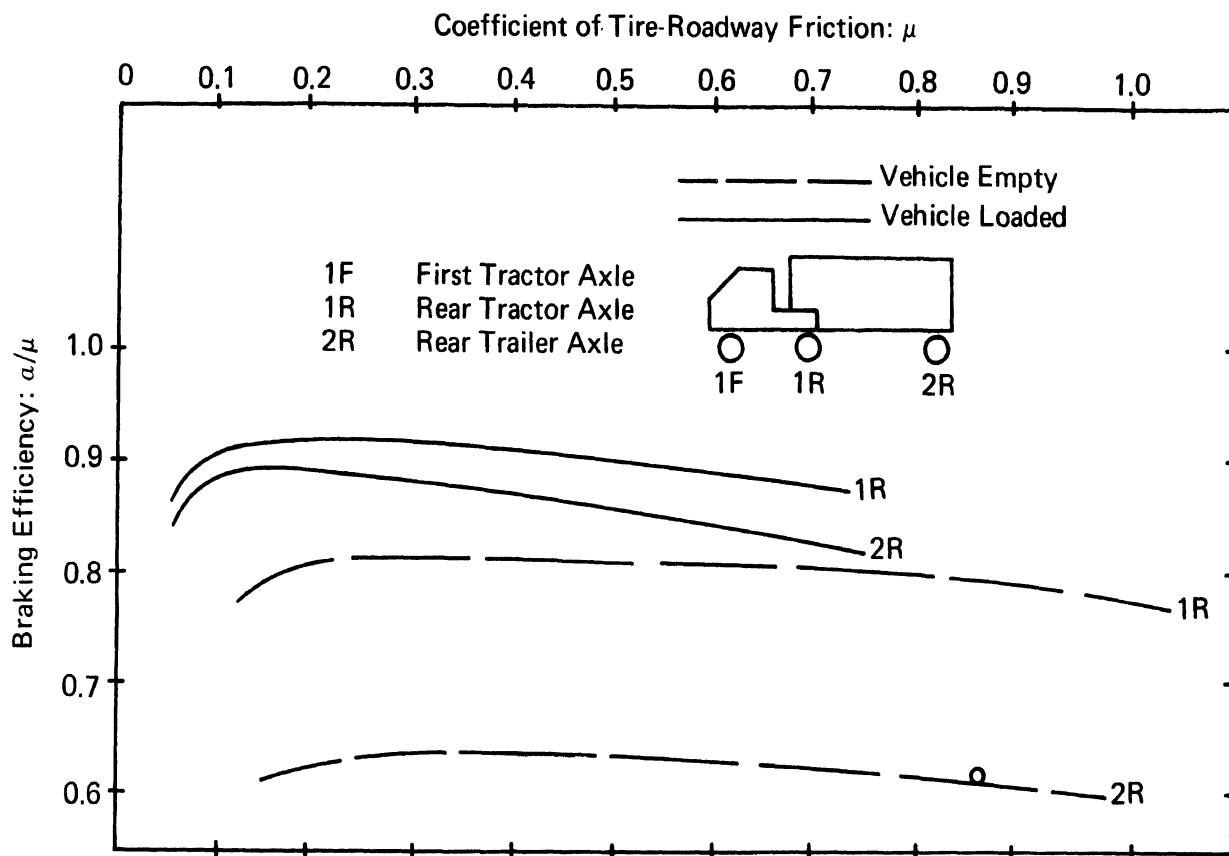
FIGURE 109. BRAKING EFFICIENCY FOR TRACTOR-TRAILER 2-S1
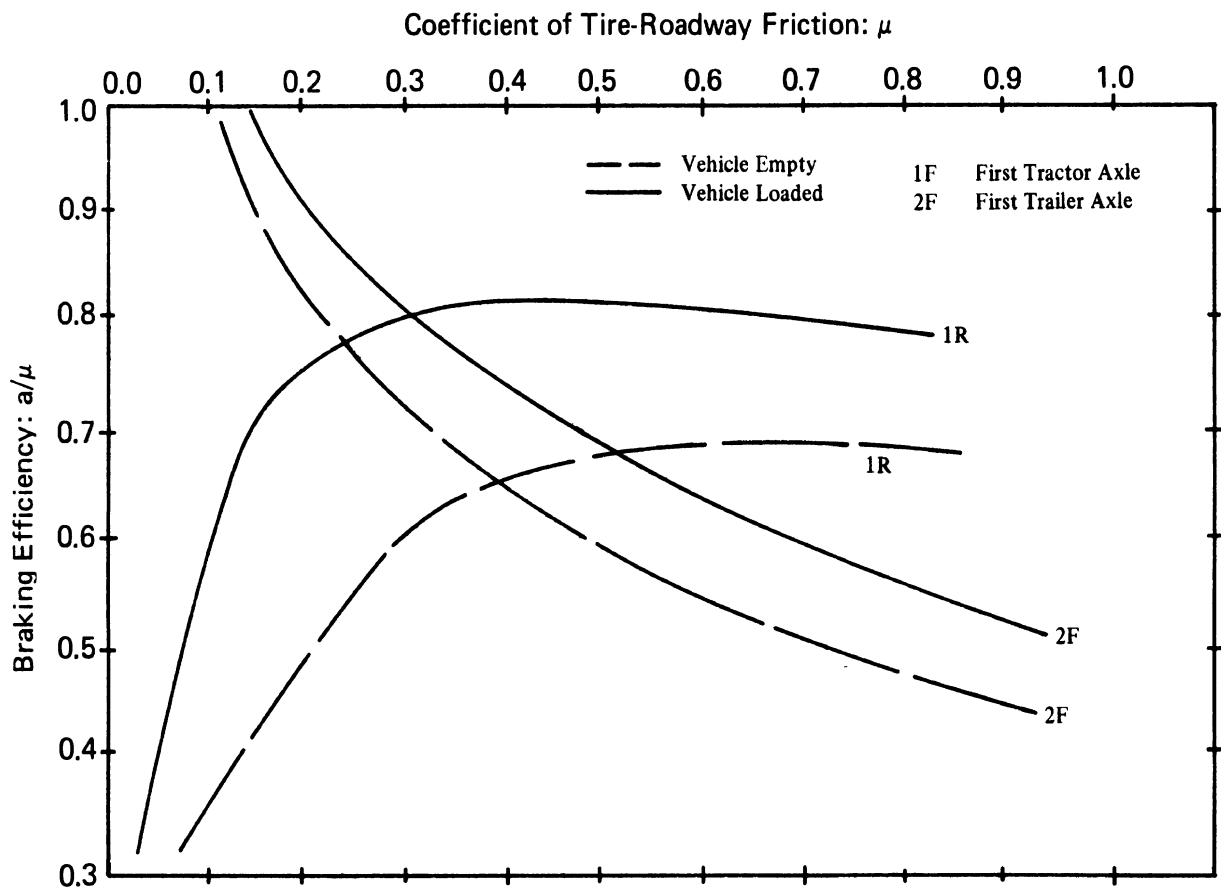
FIGURE 110. BRAKING EFFICIENCY FOR TRACTOR-TRAILER 2-S2
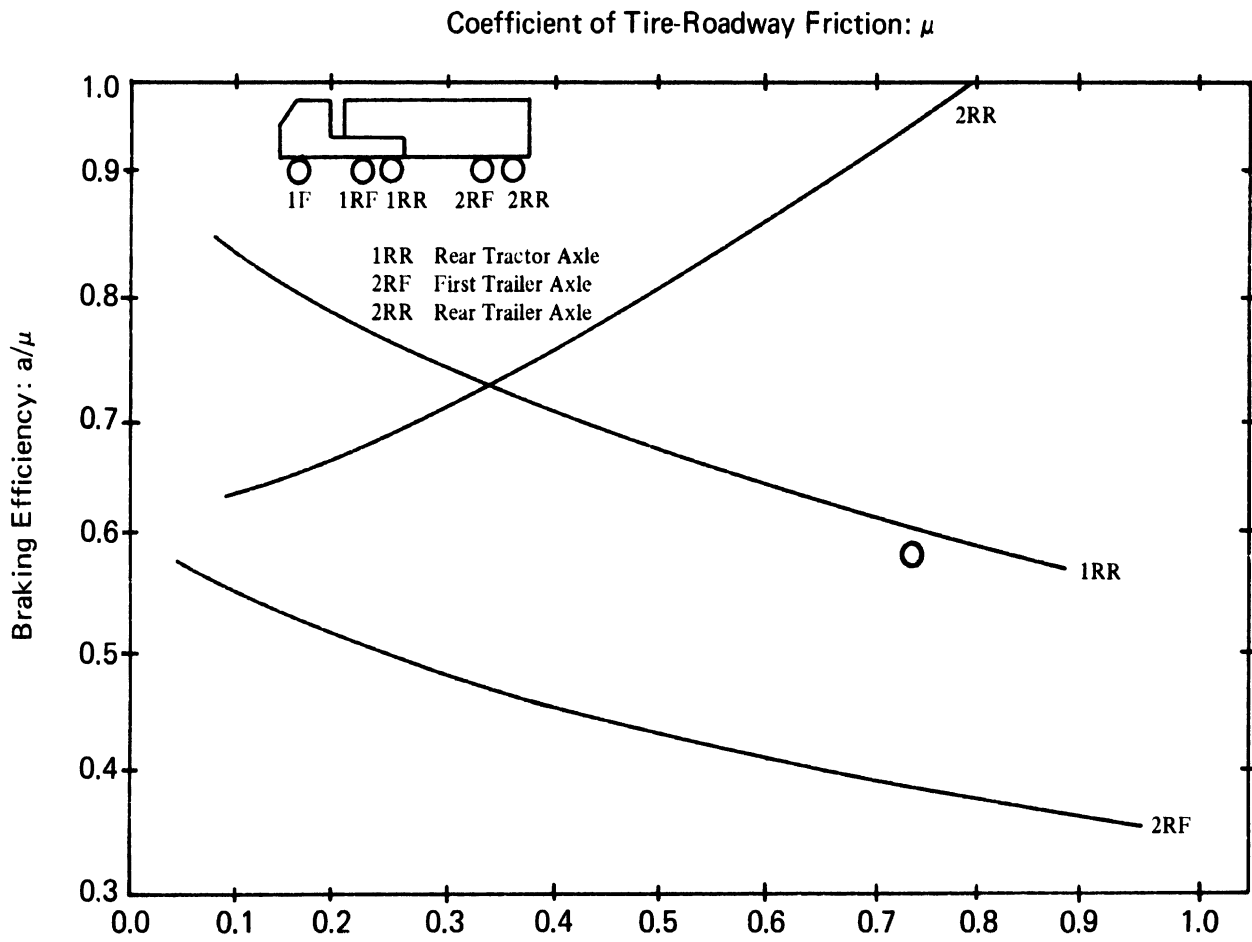
FIGURE 111. BRAKING EFFICIENCY FOR EMPTY TRACTOR-TRAILER 3-S2

Coefficient of Tire-Roadway Friction: μ
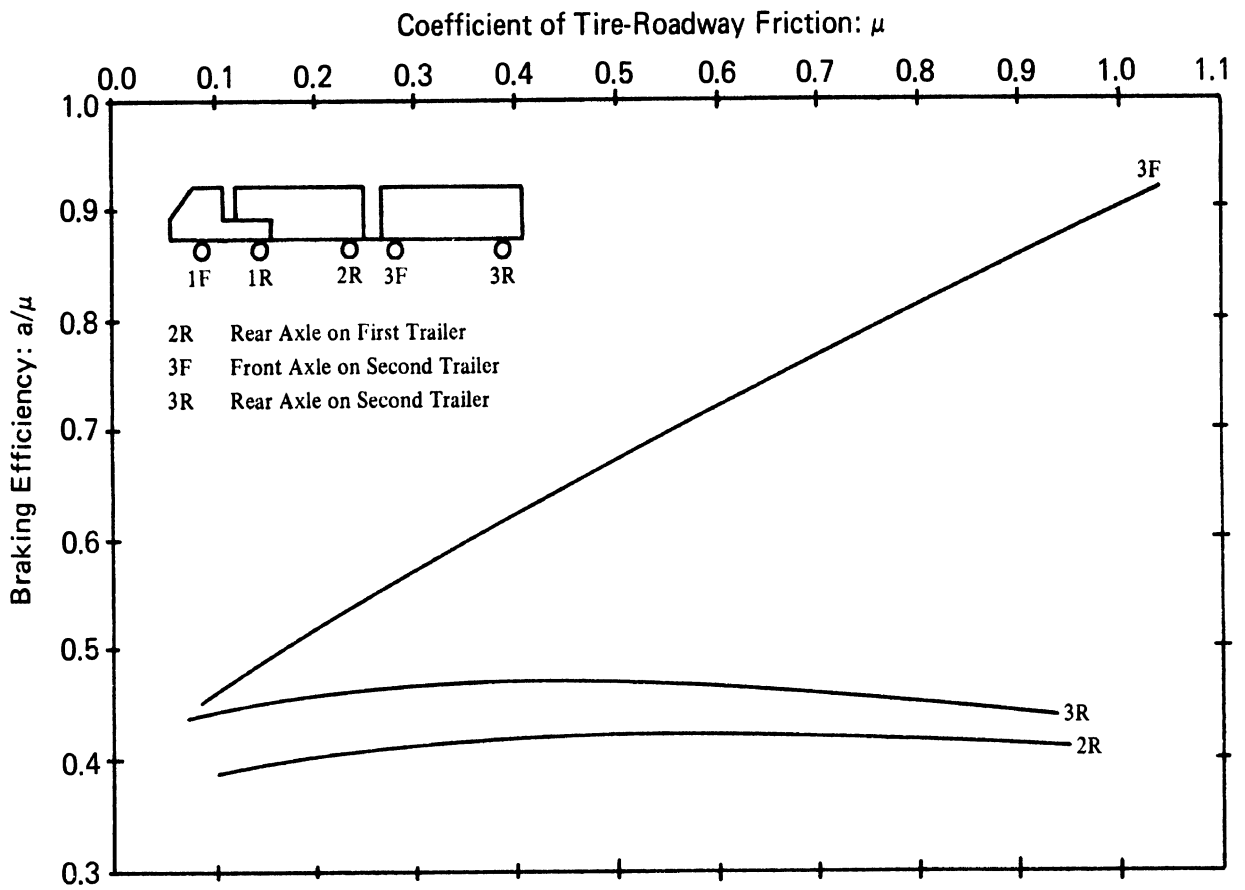
FIGURE 112. BRAKING EFFICIENCY FOR EMPTY DOUBLES COMBINATION 2-S1-2

Coefficient of Tire-Roadway Friction: μ
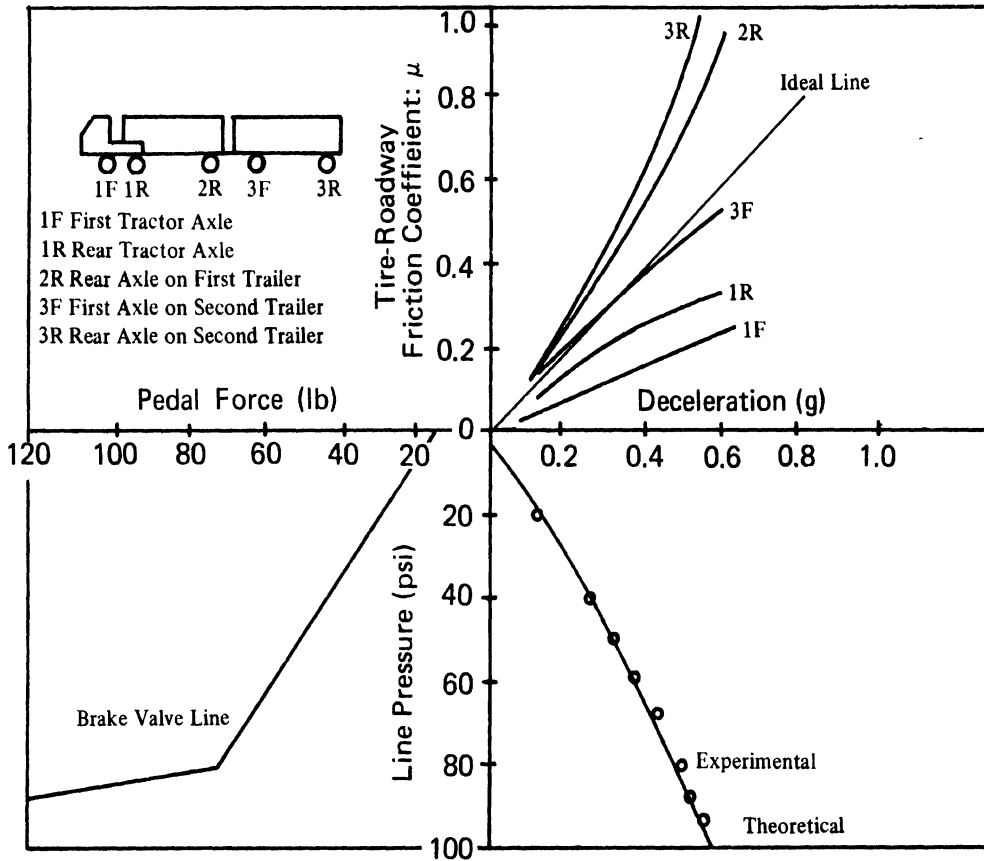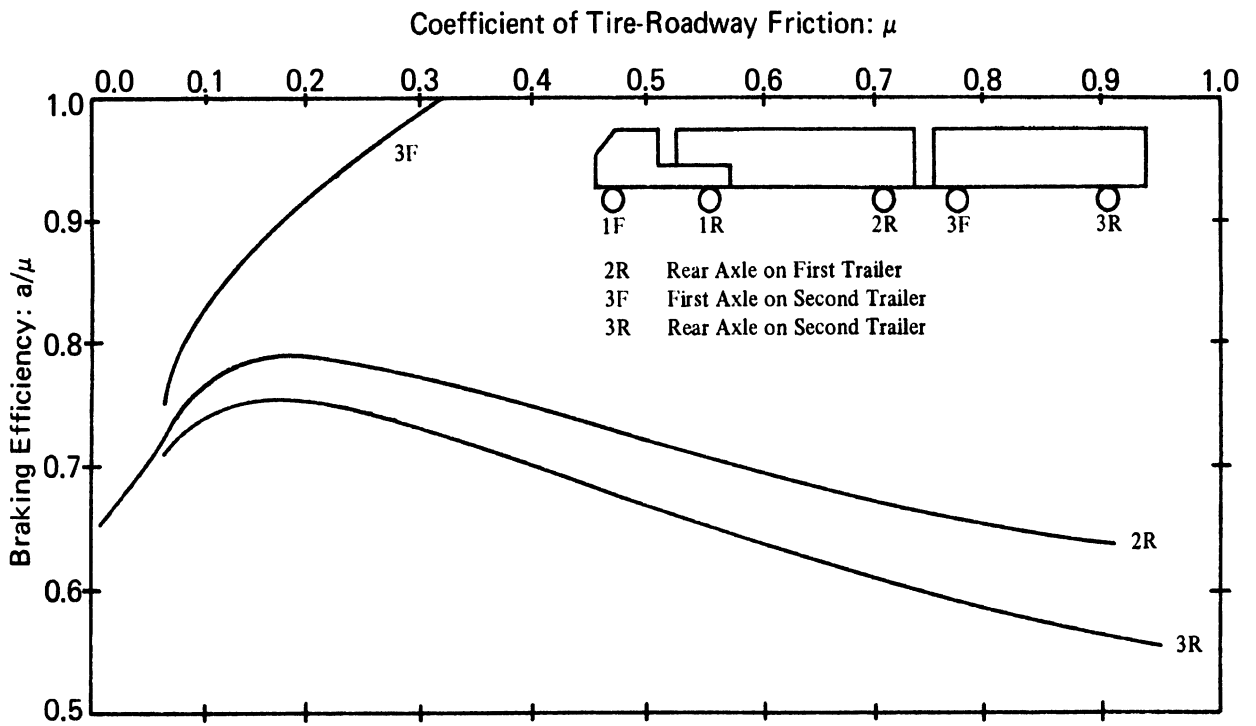

FIGURE 113. BRAKING EFFICIENCY FOR LOADED DOUBLES COMBINATION 2-S1-2
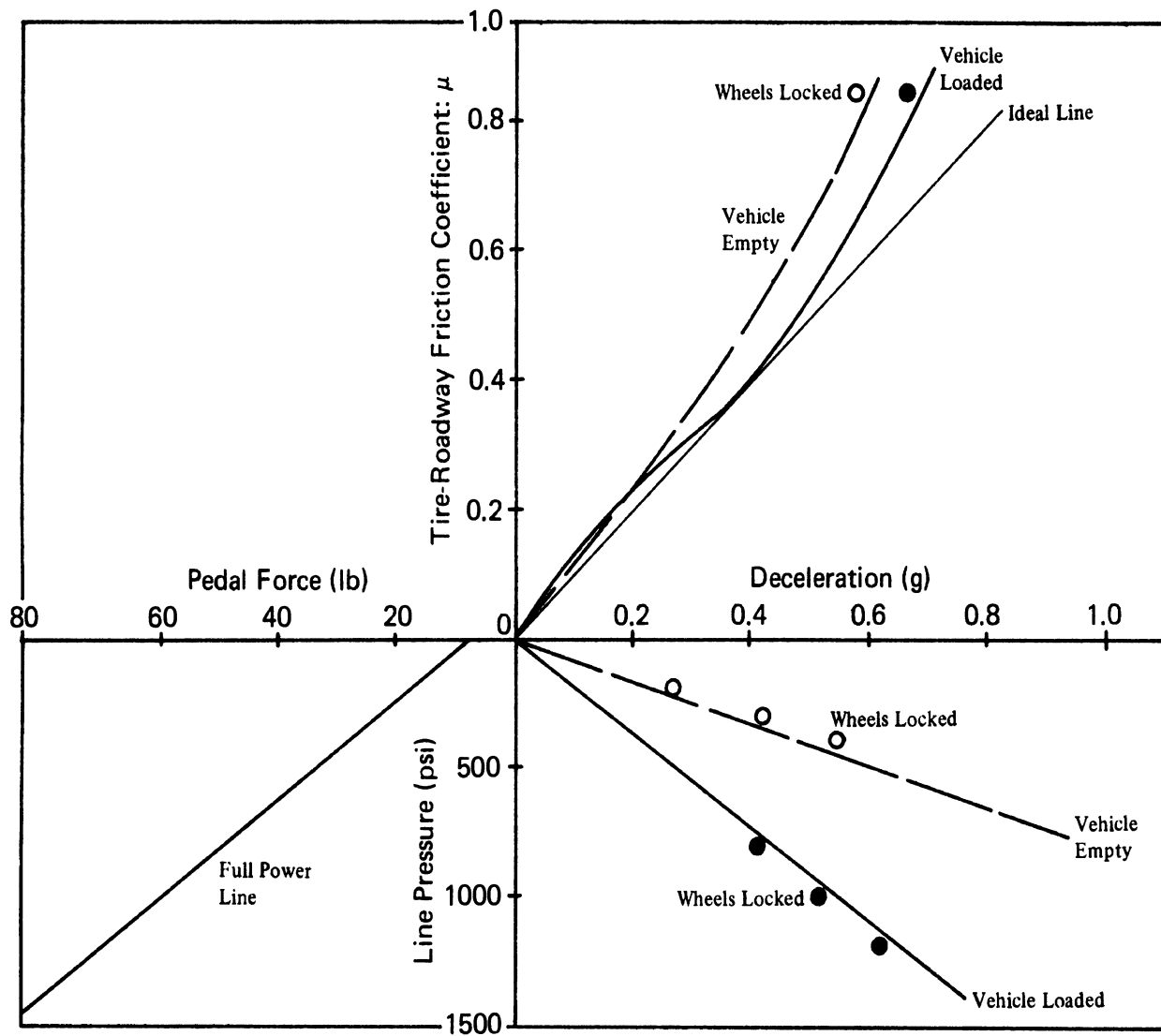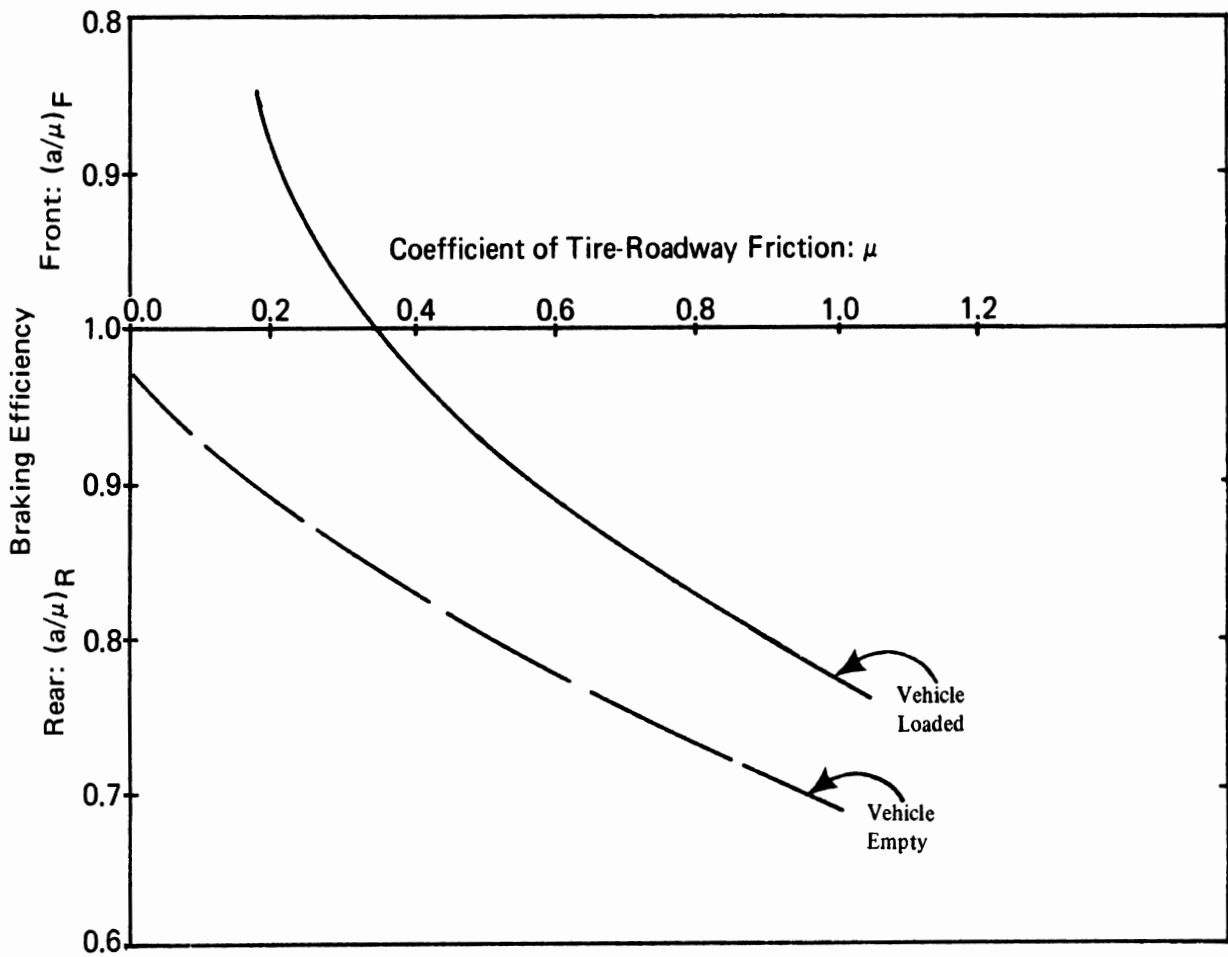
FIGURE 114. BRAKING EFFICIENCY DIAGRAM FOR DISK BRAKE TRUCK
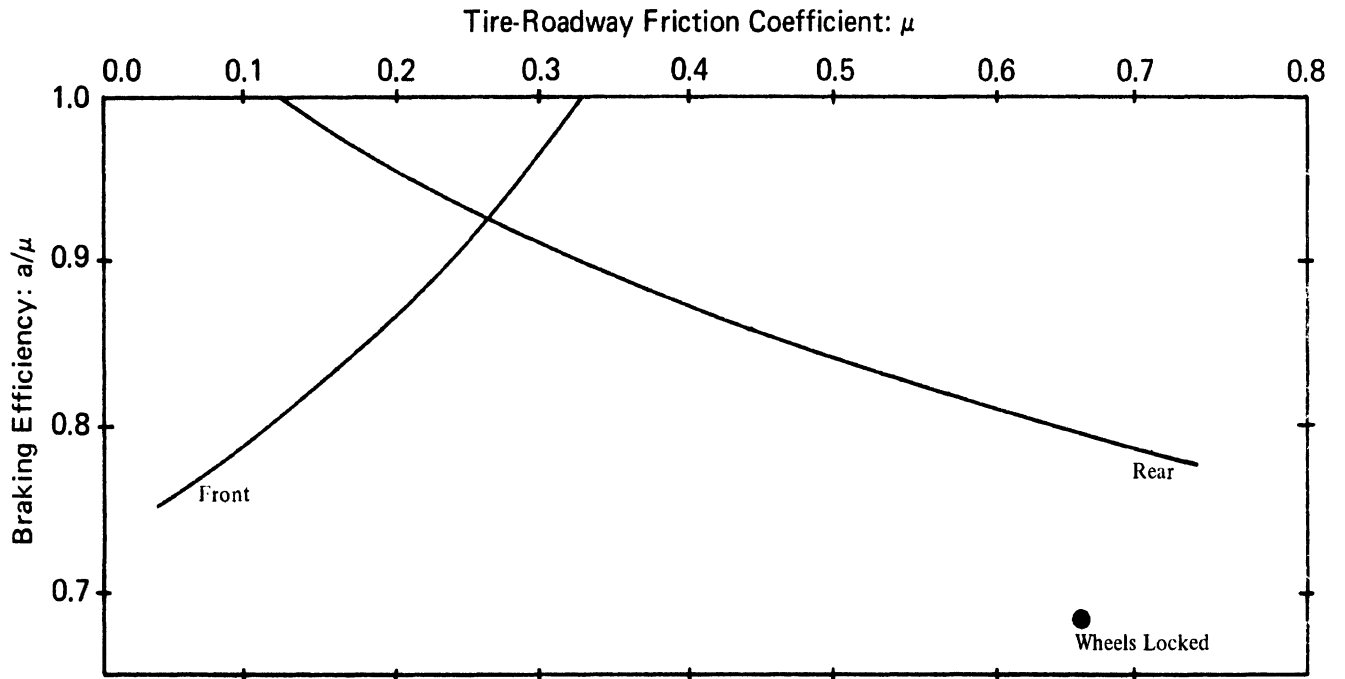
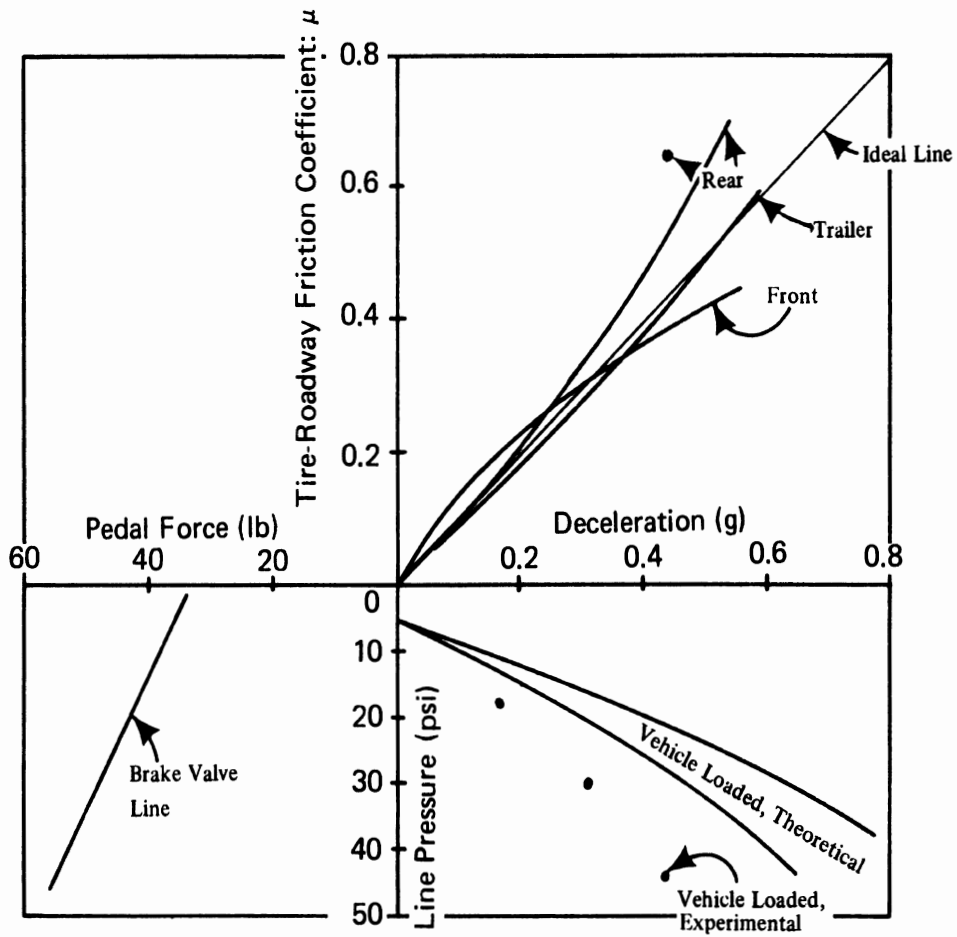FIGURE 115. BRAKING EFFICIENCY FOR VEHICLE 12, LOADED, WITH MOST EFFECTIVE BRAKES
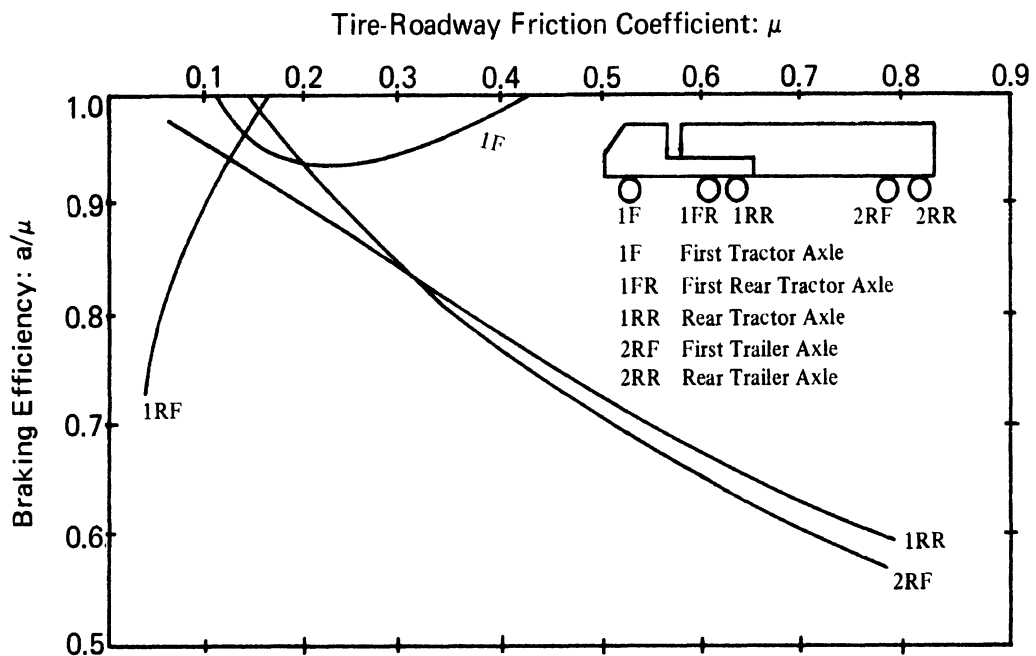

FIGURE 116. BRAKING EFFICIENCY DIAGRAM FOR VEHICLE 12

4.3 INFLUENCE OF BRAKE FORCE DISTRIBUTION ON PERFORMANCE

Analyses have also been conducted to examine the extent to which brake force distribution constituted an influential factor determining the braking performance measured in this program. The literature shows that it is possible to define an ideal brake force distribution which takes into account the friction available at the tire-road interface and the vehicle properties influencing load transfer during braking (3,50). For the two-axle vehicle, the following limiting condition can be defined for the brake-force distribution, ϕ:

$$\left(1 - \mu\chi - \frac{1 - \psi}{E_{min}}\right) \leq \phi \leq \left(\frac{\psi}{E_{min}} - \mu\chi\right) \tag{4-16}$$

where

ϕ = brake force on the rear axle divided by total brake force

μ = the tire-road friction coefficient

ψ = static rear axle load divided by the total vehicle weight

χ = height of center of gravity divided by wheelbase

E = braking efficiency

The above inequality serves to define an envelope for values of ϕ that constitute an optimum design compromise when maximum wheels-unlocked deceleration performance is being sought for the loaded and unloaded vehicle operating over a specified range of tire-road friction coefficients.

4.3.1 BRAKE DISTRIBUTION AND BRAKING PERFORMANCE OF TRUCKS AND BUSES.

On assuming a maximum and minimum value of μ equal to 0.8 and 0.2, respectively, application of condition 4-16 to the light truck results in a theoretical value for ϕ = 0.51, as contrasted with the actual brake force distribution in which ϕ = 0.53. The computed distribution (ϕ = 0.51) yields a minimum braking efficiency of 77% for the loaded vehicle operating on slippery road surfaces (μ = 0.2). For a dry road surface, the computed braking efficiencies are 0.80 and 0.87 for the empty and loaded cases, respectively. These braking efficiencies would produce wheels-unlocked decelerations of 20.6 ft/sec² for the empty vehicle and 22.6 ft/sec² for the loaded vehicle on a road surface having a tire-roadway friction coefficient of 0.8. On comparing these theoretical values to test data (20 ft/sec², unloaded, and 23 ft/sec² loaded), it appears that the braking system of the light truck was operating near an optimum condition.

Application of condition 4-16 to the school bus yields values of ϕ = 0.50 to 0.55, whereas the vehicle actually had a brake force distribution of ϕ = 0.42. If the bus had a brake force distribution of ϕ = 0.55, braking efficiencies equal to 0.72 and 0.93 could be expected for the empty and loaded vehicle, respectively, operating on slippery roadways (μ = 0.2), while on dry surfaces (μ = 0.8) efficiencies of 0.92 and 0.96 could be expected. It appears that the deceleration capability of this vehicle can be improved by changing the brake force distribution. A proportional braking system will

not, however, significantly increase braking performance because of the small change in relative static rear axle load from the empty to loaded condition $(\Delta\psi = 0.105)$.

The small change in relative static axle load on the intercity bus produced by the addition of payload $(\Delta\psi \cong 0)$ indicates that a fixed brake force distribution should yield near optimim braking performance. Condition 4-16 indicates that $\phi = 0.61$ is near optimum whereas the intercity bus actually had a brake force distribution of $\phi = 0.66$. Test results indicate that neither the front nor the leading tandem axle approached the wheel slide condition, either empty or loaded. Improvements in the maximum braking performance of the intercity bus can apparently be achieved by increasing the brake effectiveness, front and rear, and by changing the brake force distribution in a proper manner.

The brake force distribution ϕ of the medium truck was 0.74, and the change in relative static axle load from the empty to loaded condition was rather large, namely, $\Delta\psi = 0.29$.

Application of condition 4-16 to the medium truck produces the following requirements for ϕ:

$$0.27 \leq \phi \leq 0.52 \text{ for } \mu = 0.2, \text{ empty vehicle}$$
$$0.12 \leq \phi \leq 0.36 \text{ for } \mu = 0.8, \text{ empty vehicle}$$
$$0.58 \leq \phi \leq 0.83 \text{ for } \mu = 0.2, \text{ loaded vehicle}$$
$$0.43 \leq \phi \leq 0.68 \text{ for } \mu = 0.8, \text{ loaded vehicle}$$

These results indicate that the actual brake force distribution $(\phi = 0.74)$ is near optimum only for the loaded vehicle operating on slippery road surfaces. These findings indicate that a brake force distribution of $\phi = 0.47$ would probably yield better braking performance for all road surface and loading conditions than can be expected from $\phi = 0.74$. Braking efficiencies computed with $\phi = 0.47$ are 0.83 and 0.61 for the empty and loaded vehicle on road surfaces having a tire-road friction coefficient of $\mu = 0.2$, and 0.67 and 0.90 for the empty and loaded vehicle, respectively, on road surfaces with $\mu = 0.8$. On assuming a roadway friction coefficient of 0.8, the empty vehicle would decelerate at approximately 17 ft/sec^2 and the loaded vehicle would decelerate at 23 ft/sec^2, provided the brake effectiveness is increased to a level such that wheel slide conditions can be approached. A further increase in braking capability could be accomplished only by means of a proportional braking system, which, based on calculations, could increase the maximum wheels unlocked deceleration of the medium truck to 23 ft/sec^2 for both loading conditions, if $\mu = 0.8$.

4.3.2 BRAKE DISTRIBUTION AND BRAKING PERFORMANCE OF TRACTOR-TRAILERS. An inequality comparable to condition 4-16 has been derived for the combination vehicle equipped with tandem axles (3), but is not presented here, in view of its length. Computations have nevertheless been performed for three of the combinations tested.

For the 2S-1 combination (Vehicle 6), a fixed distribution of brake force

can be determined by fitting the criterion which was applied to the integral
vehicles considered above. This distribution is given by:

$$\phi_{1F}:\phi_{1R}:\phi_{2R} = 0.20:0.42:0.38$$

where ϕ_{1F} = brake force on front axle/total brake force

ϕ_{1R} = brake force on rear axle/total brake force

ϕ_{2R} = brake force on trailer axle/total brake force.

The actual brake force distribution on Vehicle 6 was 0.11:0.44:0.45, indica-
ting insufficient braking effort on the front axle and too much on the trailer
axle. Test data support this analysis in that one tractor rear wheel and one
trailer wheel approached wheel slide when the empty vehicle decelerated at
17 ft/sec^2. Although better performance can be achieved by modifying the
brake force distribution, a fixed distribution should not be expected to pro-
duce braking efficiencies much higher than 75%.

The other two tractor-trailer combinations for which the relationship
between brake force distribution and braking performance was investigated are
Vehicle 7 (2-S2) and Vehicle 8 (3-S2), both of which used the same trailer.
Vehicle 7 was capable of decelerations (no wheels locked) of 20 ft/sec^2 in
the loaded condition while Vehicle 8 was capable of only 14.1 ft/sec^2 (no
wheels locked) and 16.1 ft/sec^2 (some wheels locked in the trailer). Analysis
of the equations for the dynamic load on the trailer axles indicates that
load transferred from the trailer axles when the vehicle is decelerated is
directly proportional to the total weight of the trailer plus load, the trailer
center of gravity height, and the percent of the total braking effort concen-
trated on the trailer axles. Values for the above mentioned variables for
Vehicles 7 and 8 are as follows:

	Vehicle 7	Vehicle 8
Weight of trailer plus load	45,000 lb	53,530 lb
Trailer plus load C.G. height	16 in.	16 in.
Percent of total brake effort on trailer	47.5%	53.0%

Plots showing the load on each axle as a function of vehicle deceleration
are given in Fig. 117 for Vehicle 7, and in Fig. 118 for Vehicle 8. For a
deceleration of 0.5 g, at which the trailer wheels of Vehicle 8 started locking
up, the line pressure for Vehicle 7 was 75 psi and for Vehicle 8 was 100 psi.
The calculations show that the load transferred from the trailer axles at
this deceleration was 12% more for Vehicle 8 than for Vehicle 7. This clearly
indicates that on Vehicle 8 braking effort was not balanced to load, and that
the tractor (without front wheel brakes) was not doing its share of the work
causing the trailer axles to be overbraked. This example illustrates that
unlimited interchange of tractors and trailers will have to be severely cur-
tailed, or some means found to easily, quickly, and accurately balance brakes
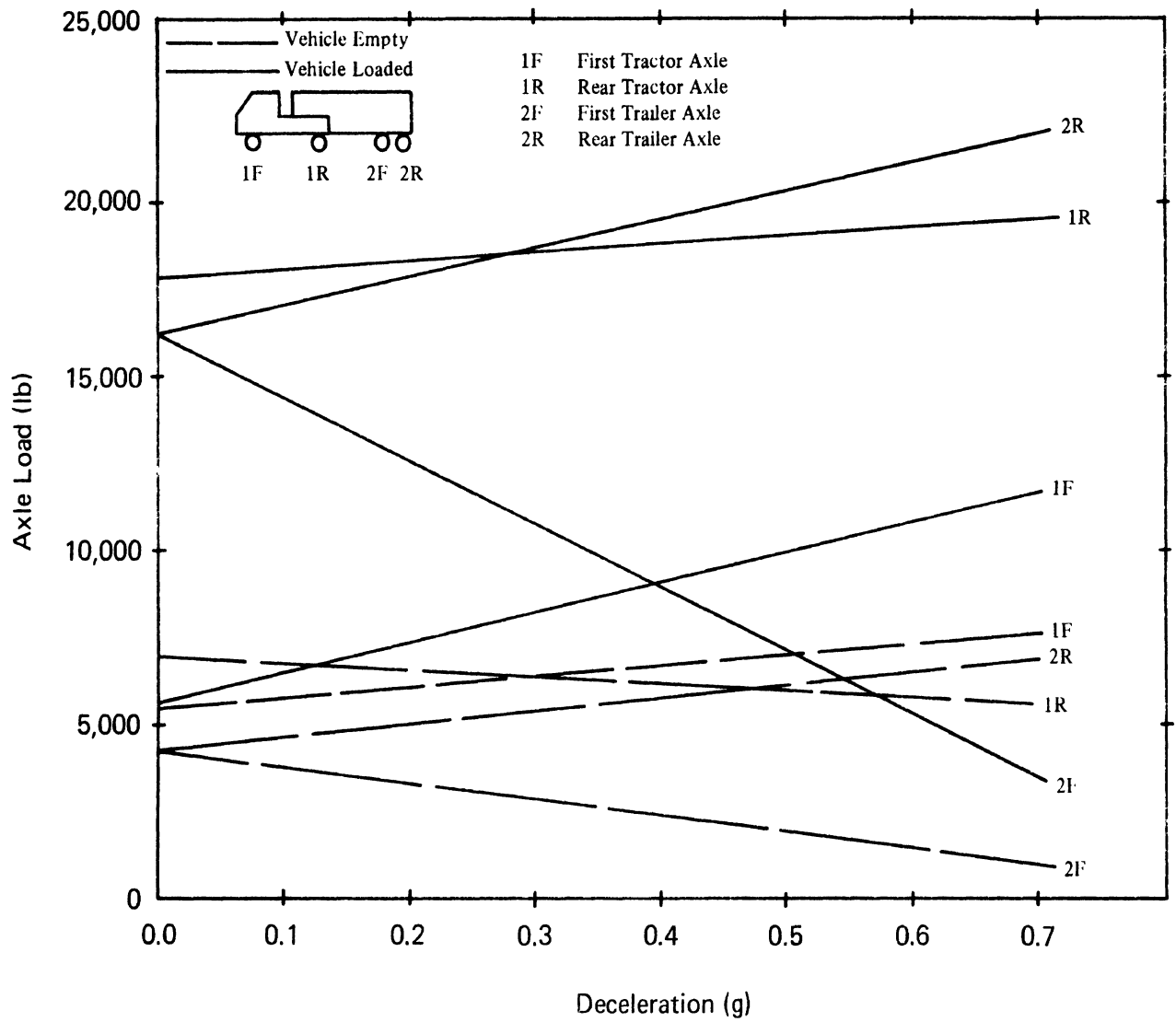
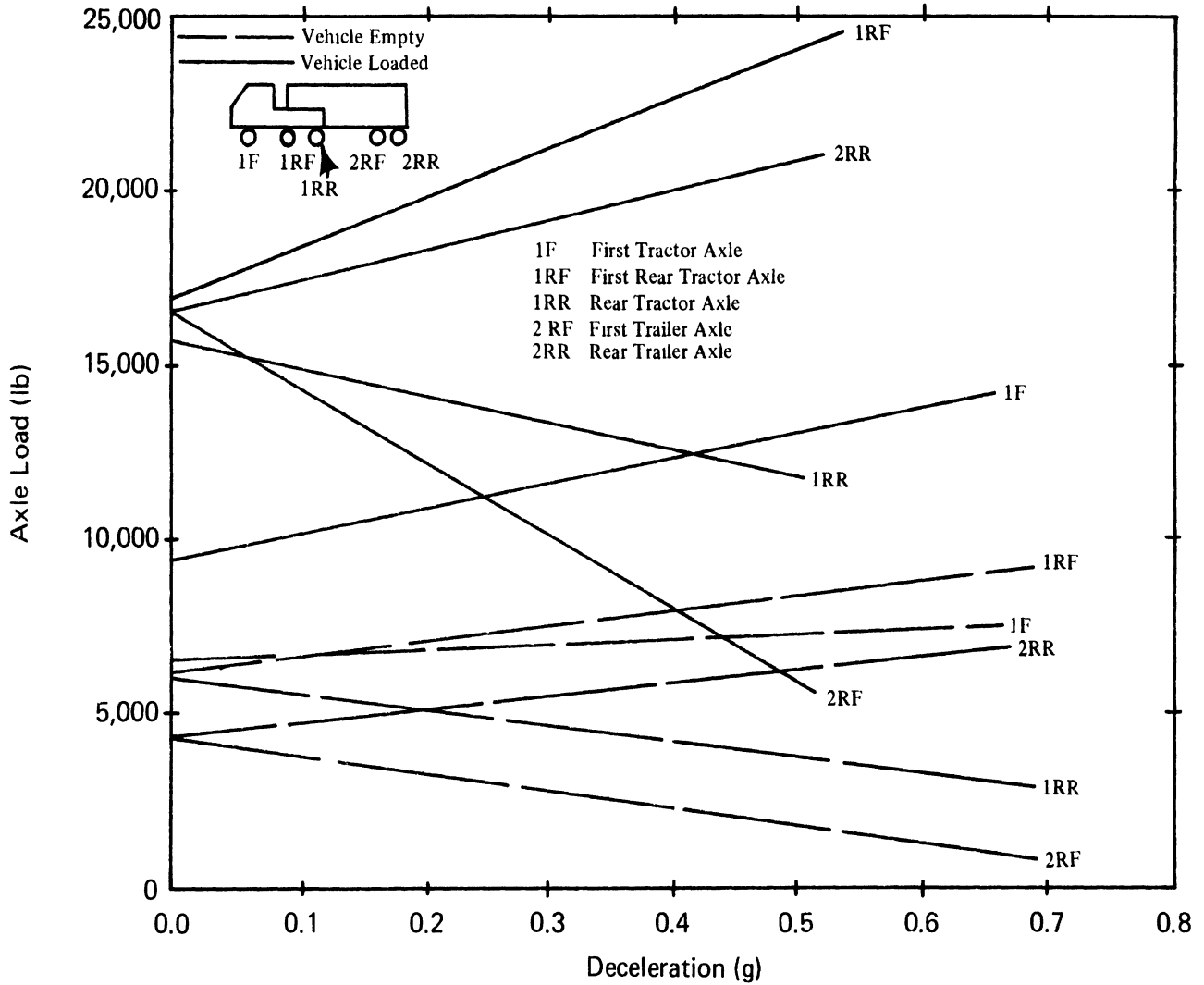FIGURE 117. DYNAMIC AXLE LOADS FOR TRACTOR-TRAILER 2-S2

FIGURE 118. DYNAMIC AXLE LOADS FOR TRACTOR-TRAILER 3-S2

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to axle load, if improvment in braking performance of articulated vehicles is to be achieved.

4.3.3 EFFECT OF THE VARIATION LOAD C.G. HEIGHT ON BRAKING PERFORMANCE OF THE LOADED TRACTOR. An analysis was made to determine the influence of load C.G. height on the performance of a loaded tractor, and to compare the performance of a tractor-trailer combination to that of a tractor carrying a load configured to simulate that which the tractor carries when pulling a trailer. Vehicle and brake system design data from the tractor and trailer of Vehicle 12 were used in the analysis.

The dynamic load diagram for the loaded combination, showing the relationship of total axle load and the horizontal and vertical kingpin reactions at the fifth wheel to the vehicle deceleration, is given in Fig. 119. Note that although the load on the front axle increases with deceleration, and the total load on the trailer axles decreases, the total load on the tractor rear axles remains fairly constant. This is accounted for by the fact that despite the increase in load on the tractor rear wheels from the increase in the vertical component of the kingpin reaction at the fifth wheel, load is transferred off this set of wheels due to the moment of the force from the horizontal kingpin reaction (overrunning force). Any attempt to configure a load for the tractor so that it is loaded "just as if it were pulling a trailer" is made difficult because of these load transfer characteristics peculiar to articulated vehicles.

The dynamic load diagram for the loaded tractor maintaining the horizontal position of the C.G. at the fifth wheel location is given in Fig. 120 for various load C.G. heights. (The load configuration assumed was roughly equivalent to that shown in the loading diagram for the White tractor, Appendix A). Note that, under these conditions, it is impossible to achieve front and rear axle loading curves equivalent to those obtained for the loaded tractor-trailer combination. Loads obtained at a deceleration of 0.5 g are summarized in Table 36.

TABLE 36

COMPARISON OF AXLE LOADS FOR LOADED TRACTOR AND
TRACTOR-TRAILER COMBINATION AT 0.5 g DECELERATION

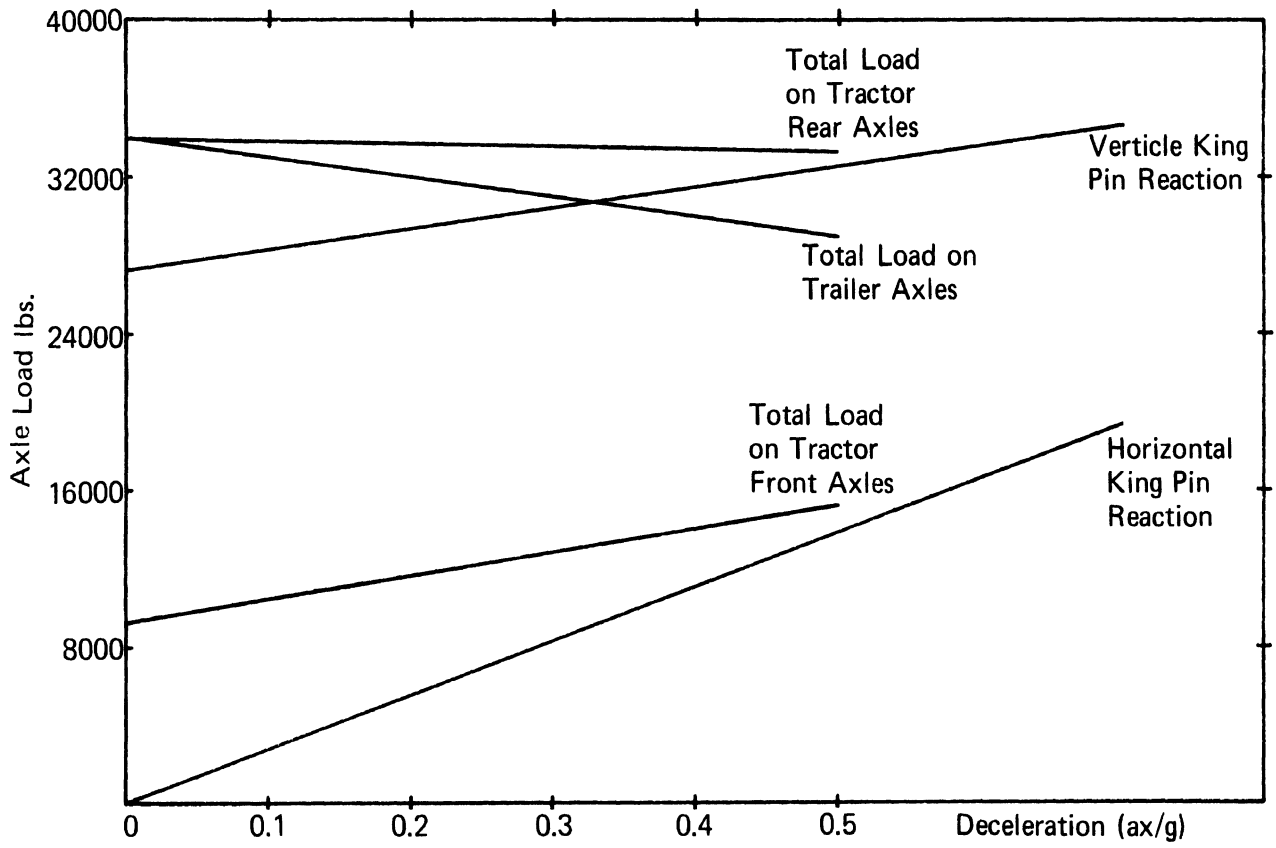|	Load on Tractor Axles Loaded Combination, lb	Load on Tractor Axles Loaded Tractor, lb		
	Fifth Wheel Height	CG Height		
Axle	48 in.	46 in.	70 in.	88 in.
Front	15,190	14,190	16,190	17,690
Rear	33,220	28,885	26,385	25,385

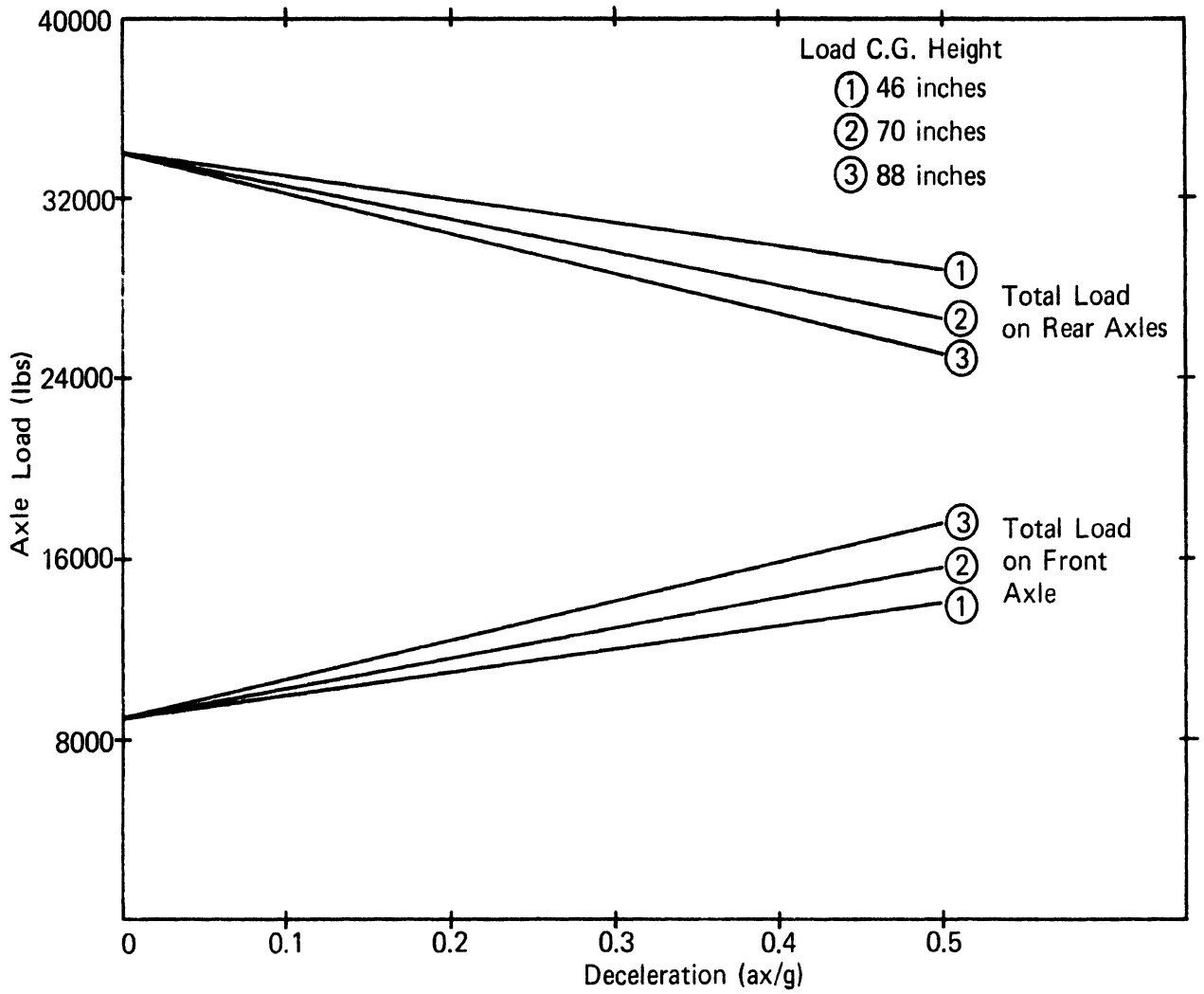FIGURE 119. DYNAMIC LOAD DIAGRAM, VEHICLE 12, TRACTOR-TRAILER

FIGURE 120. DYNAMIC LOAD DIAGRAM, VEHICLE 12, TRACTOR ONLY

The problem is further complicated by interaxle load transfer on the tandem axles, leading to premature locking of wheels on one of the axles. Taking interaxle load transfer into account, the dynamic load diagram for Vehicle 12 is shown in Fig. 121. Note that although the total load on the trailer axles decreases with deceleration and the total load on the tractor rear axles remains almost constant, interaxle load transfer causes severe unloading of the leading trailer axle and the rearmost tractor axle. The resulting brake force capabilities of each axle are shown in Fig. 122.

4.3.4 BRAKING PERFORMANCE WITH DUAL CIRCUIT AIR BRAKE SYSTEMS. Several possible dual circuit or split braking systems have been proposed for commercial vehicle braking systems (301,306,307). The purpose of these systems is to prevent total brake failure in the event of failure of a single system component. Three typical systems are evaluated in this section: a combination spring brake-service brake system,* a "horizontal split" system using dual actuator wedge brakes (stopmaster or twinplex), and a completely redundant system using dual diaphragm brake actuators with S-cam brakes.

The combination spring brake-service brake system was analyzed using the vehicle and brake system data from Vehicle 6, the Ford F-7000 two-axle tractor in combination with the Trailmobile 35-ft van trailer. The system utilizes two circuits, the primary circuit consisting of the tractor rear axle service brakes, and the secondary circuit consisting of the tractor front brakes and rear axle spring brakes. In a normal service application, the spring brakes are not utilized. However, in the event of primary system failure, the spring brakes become operable and can be modulated in conjunction with the front brakes. In the event of a secondary system failure, the tractor rear service brakes remain operable. In either case, normal operation of the trailer brakes is maintained. In making this analysis, using Equation 4-5, fade effects are ignored and it is assumed that the spring brakes on the tractor rear axle have a K-factor of 0.45. Under normal operation, Vehicle 6 has a brake force distribution (see Section 4.3.2) of $\phi_{1F}:\phi_{1R}:\phi_{2R} = 0.11:0.44:0.45$. In a failure mode in which the primary system is utilized, the distribution becomes 0.0:0.45:0.55 and the maximum deceleration capability is reduced by 11.5%. In a failure mode in which the secondary system is utilized, the distribution becomes 0.12:0.35:0.53 and the maximum deceleration capability is reduced by only 6.2%. Since the ideal distribution for this vehicle is 0.20: 0.42:0.38, it is obvious that the trailer wheels would be overbraked under either failure condition.

For the analysis of the "horizontal split" system, the vehicle and brake parameters for Vehicle 8, the Diamond Reo tandem-axle tractor and the Fruehauf 40-ft tandem-axle van trailer, are employed. The Diamond Reo tractor was equipped with dual actuator wedge brakes, having a brake factor of 5.0 if both actuators are operational, and a brake factor of 3.0 if just one actuator is operational. If a dual air piping circuit scheme is employed, wherein the primary circuit serves one actuator on each tractor brake, and the secondary

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*The dual circuit system is patterned after that developed for the Ford L-series trucks and tractors described in references (139) and (140).
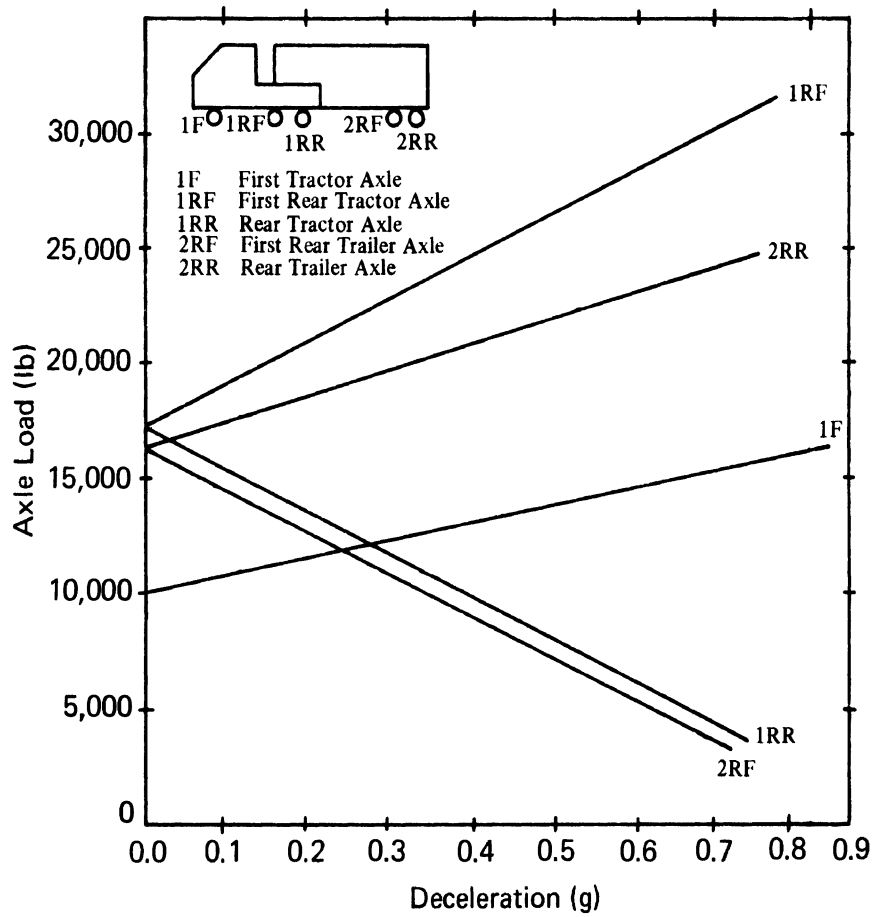
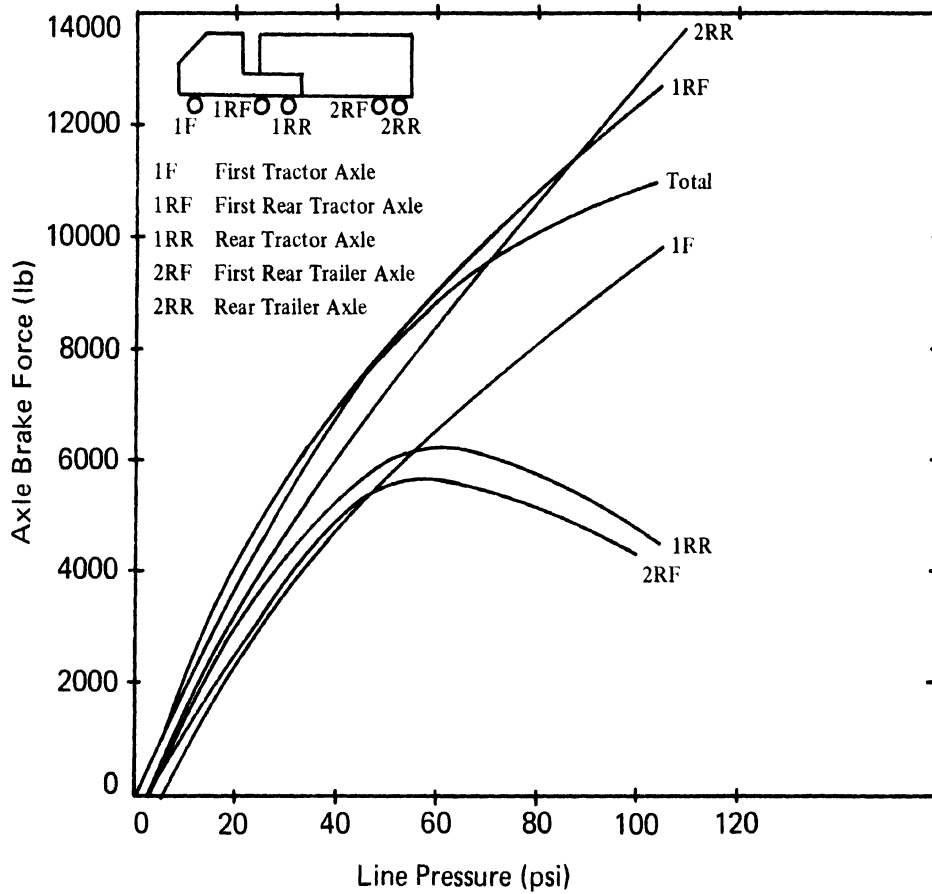FIGURE 121. DYNAMIC AXLE LOADS, VEHICLE 12, TRACTOR-TRAILER

FIGURE 122. BRAKE FORCES, VEHICLE 12, TRACTOR-TRAILER

circuit serves the other acutator on each brake, failure of either circuit would reduce the tractor brake effectiveness by 40%. It is assumed that in the failure of either circuit, trailer braking is maintained. The brake force distribution for this vehicle, originally 0.0:0.47:0.53, would be changed to 0.0:0.35:0.65 under failure conditions, and the total deceleration capability would be reduced by about 20%. The original brake distribution led to over-braking of the trailer axles, and failure under this scheme would only serve to accentuate this condition.

Dual diaphragm actuators for use with S-cam brakes provide for a completely redundant system in which the decrease in brake effectiveness under partial system failure is practically negligible. The actuator employed is identical to the conventional type except that it contains an extra diaphragm with sep-arate air supply lines. Under normal operation, the extra diaphragm is in-active, and the air pressure in the seperate lines is zero. If failure occurs in the service brake system, the secondary system is activated, and air is supplied through the separate lines to the extra diaphragm. This system pro-vides for the least loss of brake system effectiveness under failure condi-tions, but at the disadvantage of requiring completely redundant piping and special brake chambers.

4.4 THERMAL ANALYSIS

This study has included efforts that have been directed towards developing techniques to predict brake thermal response. The analysis was hindered by several factors, namely: (a) the instrumentation used in the experimental program did not lend itself to an accurate correlation of calculations and experimental results, (b) lining material parameters such as thermal conduc-tivity, specific heat, and density, could not be measured and thus had to be estimated, (c) the location of the thermocouples installed in the brake lining was not known with sufficient accuracy, and (d) the convective heat transfer coefficient could neither be accurately computed nor accurately determined from experiment. Notwithstanding these difficulties, calculations were made to predict brake temperatures as functions of time.

Finite difference methods were employed to compute the temperatures as a function of time at various points in the lining, $T = f(x, t)$, that would en-sue in three of the test procedures, namely, the fade test, the recovery test, and the brake rating test. This procedure was dictated by the necessity to introduce the convective heat transfer coefficient, h, as a variable dependent upon vehicle speed. The thermal properties needed for the analysis were ob-tained from published data (216, 217, 225) and a digital computer program was written to facilitate the numerical work involved.

Figures 123, 124, and 125 illustrate the application of this program to predict the variation in temperature that is produced in the brake lining of the medium truck. Although these figures indicate that reasonably good agree-ment was obtained between theory and test for this particular vehicle, agree-ment between theory and test was less satisfactory for other vehicles, there being, in general, discrepancies of the order of 20%. It can be concluded from these results that in order to develop techniques for predicting brake
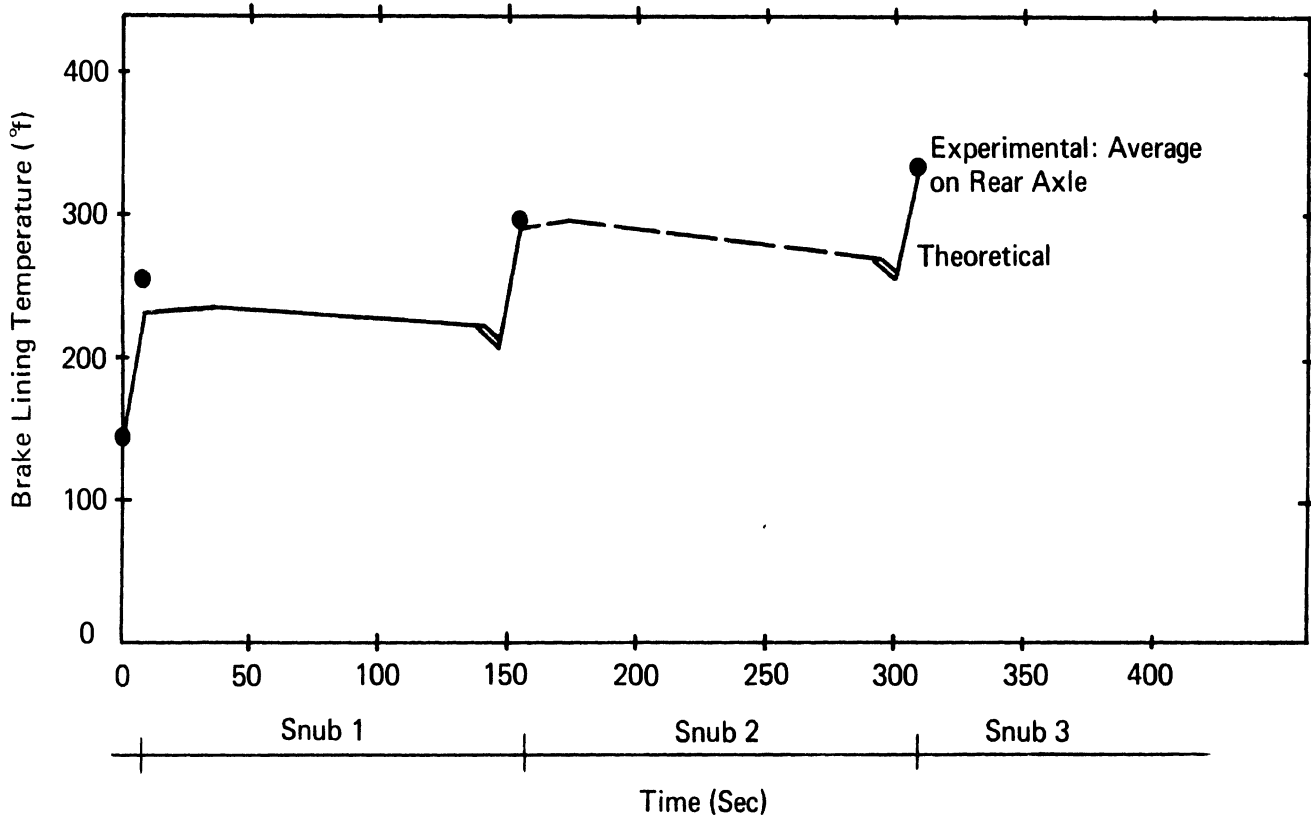
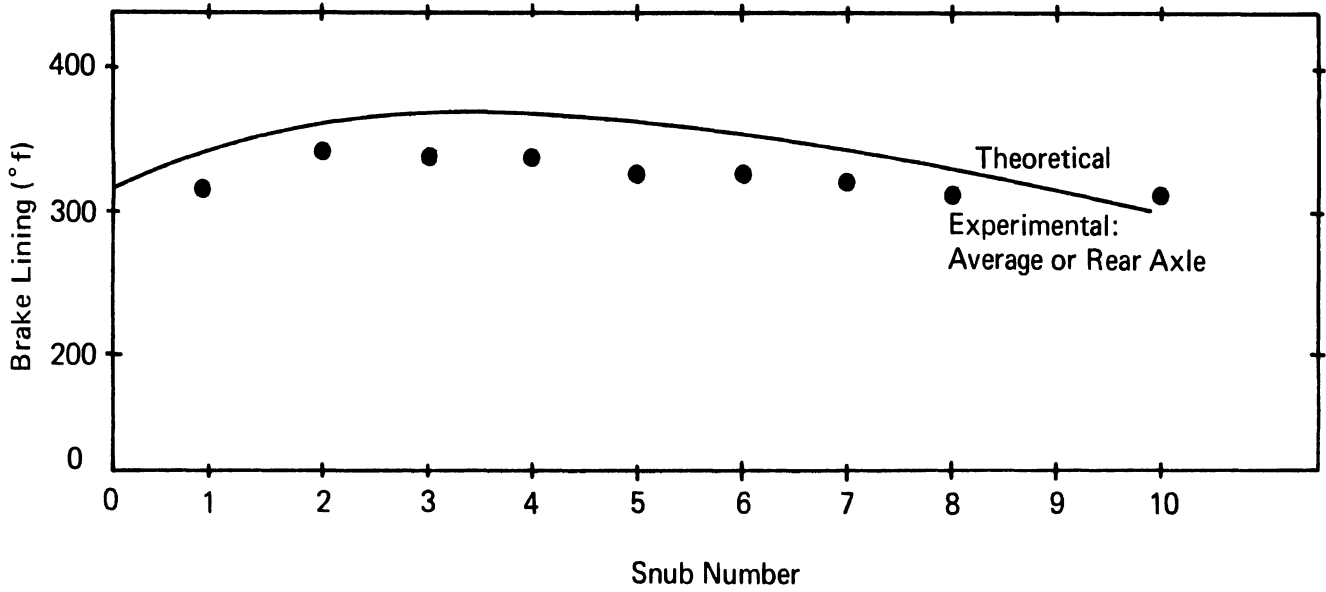FIGURE 123. BRAKE LINING TEMPERATURE, FADE TEST, MEDIUM TRUCK

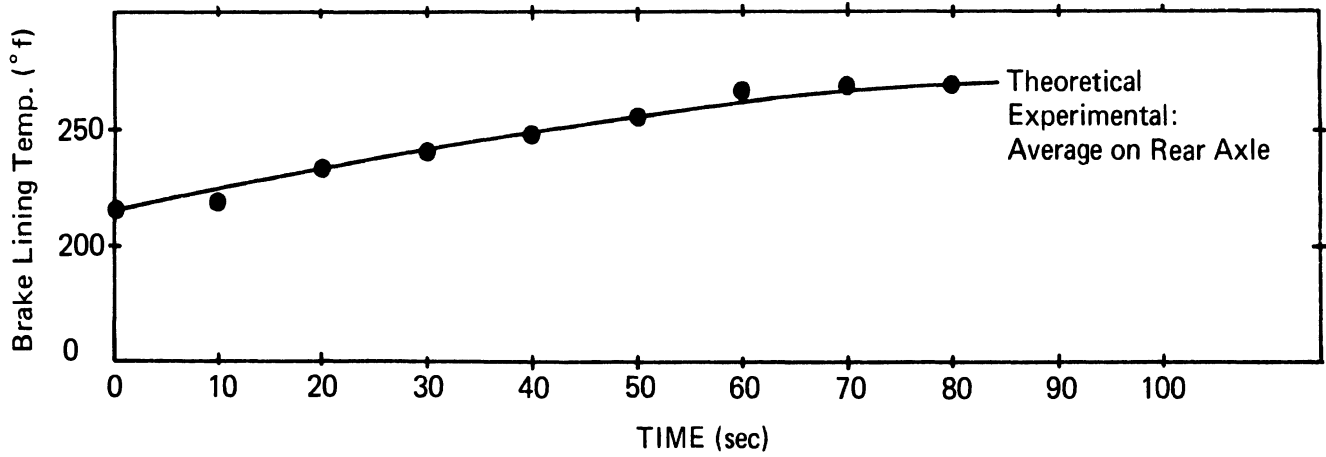FIGURE 124. BRAKE LINING TEMPERATURE, RECOVERY TEST, MEDIUM TRUCK

FIGURE 125. BRAKE LINING TEMPERATURE, BRAKE RATING TEST, MEDIUM TRUCK

thermal response based upon vehicle and brake system design data, further ana-
lytical work supported by carefully instrumented tests will be required.

4.5 VEHICLE DYNAMIC SIMULATION

Dynamic modelling and simulation were employed in the analytical phase
of this study to investigate the effectiveness of brake proportioning schemes
over a wide range of loadings and test surfaces, to assess the effect of vari-
ations in tractor and trailer brake response time on performance, and to evalu-
ate the braking performance of an integral truck equipped with a rear-wheel
antilock system. In addition, the results of another simulation effort, sup-
ported independently by HSRI to study the directional dynamics of tractor-
trailers in turning and braking maneuvers, are summarized.

4.5.1 VERTICAL PLANE MODEL. The dynamic simulation employed in this
study is based upon a model that represents either an integral 2-axle truck or
tractor, or a 3-axle tractor-trailer combination. Motions are constrained to
the plane of symmetry (vertical plane). Specifically, the wheels can bounce
and spin, chassis can heave, and pitch, and accelerate (decelerate) in straight
line motion. The braking system is modeled in a manner such that variable
time lags and delays in torque response can be introduced. Any desired brake
force distribution can be specified. Figure 126 depicts the essential features
of the model. A detailed description of this model is given in Appendix D.

4.5.2 BRAKE PROPORTIONING BASED ON STATIC AXLE LOADS. To determine the
improvement in performance that could be expected for a 3-axle tractor-trailer
combination (2-S1) using static load-sensitive proportioning valves, a typical*
2-axle tractor and 27-ft van trailer were used as a basis for the study. Brake
torque for each axle was calculated from typical design information. Table
37 gives the axle loads and brake torque distributions assumed for the study.
The tire-road interface was modeled using typical μ-slip curves with peak/
sliding coefficients of friction equal to 0.92/0.85 as shown in Fig. 127.

TABLE 37

AXLE LOADS AND BRAKE FORCE DISTRIBUTION

	Tractor Front Axle	Tractor Rear Axle	Trailer Axle	Total
Weight, Empty, lb	3,334	5,221	4,285	12,840
Weight, Loaded, lb	5,479	16,446	18,835	40,760
Fixed Brake Force Distribution, %	16	38	46	100
Proportioned Brake Force Distribution, Empty Vehicle, %	26	41	33	100

*The simulated combination closely resembled a Ford F-7000 tractor with
a Fruehauf Model FBB 27-ft van trailer.

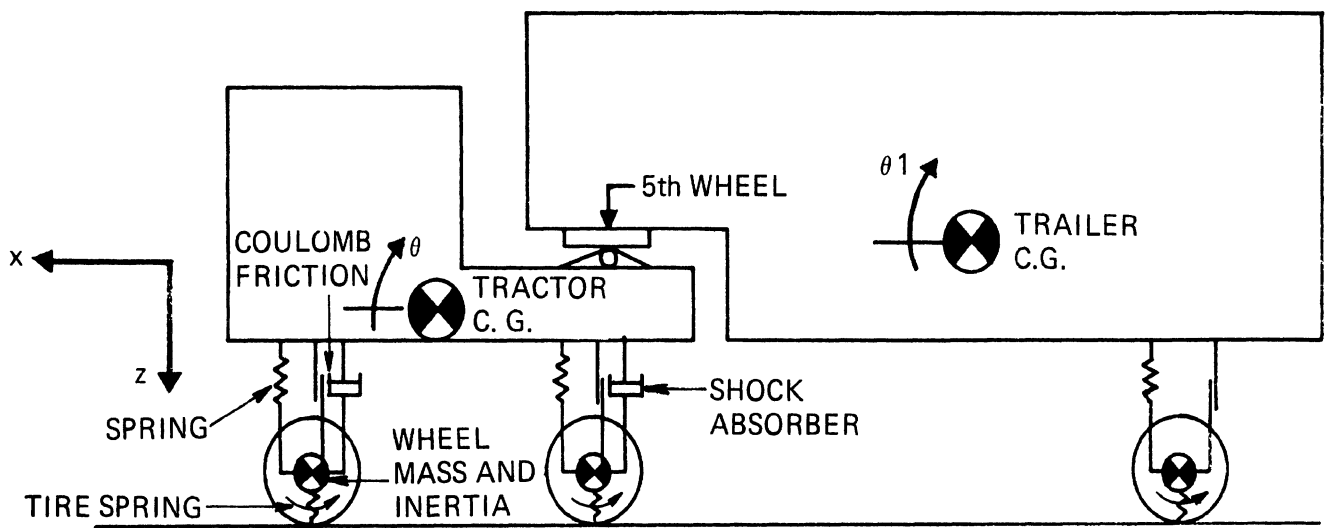FIGURE 126. ARTICULATED VEHICLE, VERTICAL PLANE MODEL
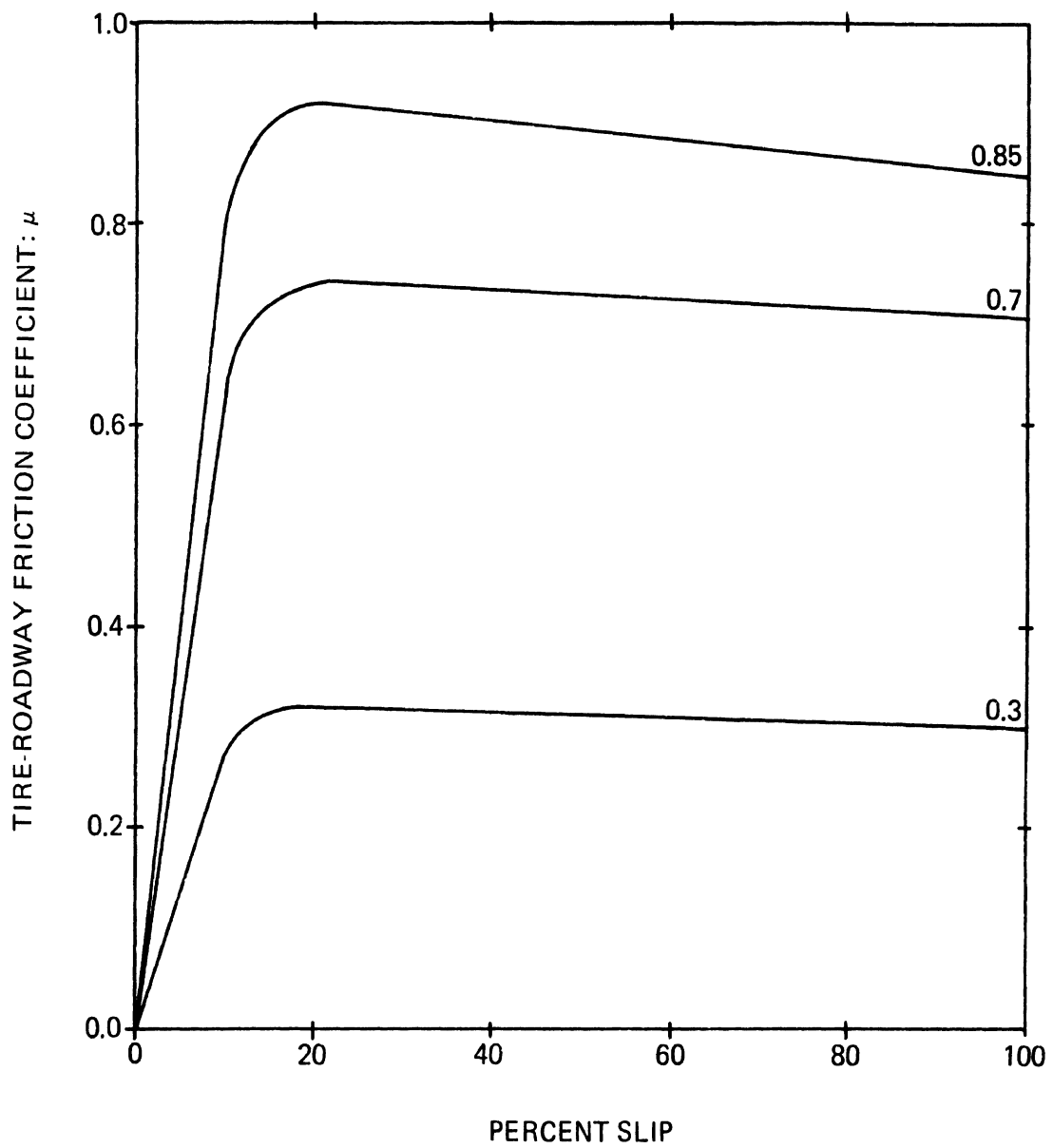
FIGURE 127. μ-SLIP-CHARACTERISTICS FOR SIMULATION OF TIRE-ROAD INTERFACE

In this preliminary investigation, no account was taken of the loss of brake effectiveness that is caused by fade during a single stop. The following findings were obtained:

(1) Without proportioning the loaded vehicle achieves a maximum deceleration of 16.7 ft/sec² before the trailer wheels lock up.

(2) The empty vehicle without proportioning achieves a maximum deceleration of 16.9 ft/sec² before the trailer wheels lock up.

(3) The empty vehicle with the brakes proportioned to the static loads on the axles (as shown in the last row of Table 37) can achieve a maximum deceleration of 23.8 ft/sec² without encountering wheel lock-up, an improvement in stopping capability of about 40%.

4.5.3 IDEAL PROPORTIONING. The objective of brake proportioning is to distribute braking forces among the various wheels of a vehicle in an ideal manner under a variety of loading and road surface conditions. Ideal brake force distribution is that which yields the maximum possible deceleration on any given road surface, implying, for a three-axle rig:

$$\frac{F_{x1}}{F_{z1}} - \frac{F_{x2}}{F_{z2}} = \frac{F_{x3}}{F_{z3}} = \frac{a_x}{g} \tag{4-17}$$

and

$$\frac{a_x}{g} = \mu \tag{4-18}$$

where

F_{x1}, F_{x2}, F_{x3} = the brake forces on the three axles of the vehicle

F_{z1}, F_{z2}, F_{z3} = the instantaneous normal loads on each axle

a_x = deceleration

g = gravity constant

μ = tire-raod interface coefficient

A second study was conducted to determine if a feasible proportioning scheme could be developed which would distribute the brake forces over a wide range of loadings and surface conditions such that Equations 4-17 and 4-18 would be satisfied to a maximum extent. The 2-S1 combination vehicle, namely, the Ford F-7000 tractor in combination with the Trailmobile 35-ft van trailer, was selected as the prototype for this study. Parameter data defining the vehicle and brake system were determined and introduced into the simulation. The effects of brake response time, fade, and pushout pressures were included. Fade effects were considered pressure-dependent and were modeled as was discussed previously. Preliminary runs were made to check that the simulation results matched test results obtained for the vehicle in the empty and loaded condition. The μ-slip characteristic used to represent the tires assumed peak/ sliding coefficients of friction equal to 0.76/0.7 in order to bring the simulated surface closer to that existing at BADC. Simulated performance and

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experimental data are compared in Fig. 128.

To increase braking efficiency to a maximum extent, a proportioning scheme should not only account for changes in static loading but changes in dynamic loading as well. Dynamic load- or deceleration-sensitive devices or sensors are rather difficult to employ to actuate brake proportioning valves because of noise and vibrational problems. However, dynamic axle loads and vehicle deceleration are dependent upon the applied pneumatic brakeline pressure (output of the treadle valve). Accordingly, a scheme was devised to utilize both static load and line pressure as proportioning variables. The characteristics of the valve simulated for the tractor are given in Fig. 129 and for the trailer in Fig. 130. To achieve best performance, it was necessary to increase the effectiveness of the existing brakes by a factor of 2.6 on the front axle of the tractor, by 1.3 on the rear axle of the tractor, and by 1.7 on the trailer axle.

Computer runs were made in a manner similar to the braking effectiveness test procedure. The vehicle was given an initial velocity of 60 mph and the brakes were applied at a given line pressure and held at this line pressure until the vehicle stopped. Line pressure was increased in finite steps until wheel lockup occurred. Figure 131 presents recorder traces that are obtained in a typical computer run.

Simulated braking runs were made for the baseline and ideally proportioned vehicles at all of the loading conditions given in Table 38 and two tire-road interface conditions: $\mu_{peak}/\mu_{sliding}$ = 0.76/0.70 and 0.325/0.3. Figures 132 and 133 show the maximum deceleration (wheels unlocked) achieved and the braking efficiency as computed using locked wheel friction as a reference value. We see that by means of this idealized proportioning scheme it is possible to achieve better than 90% braking efficiency for a tractor-trailer on the 0.7 and 0.3 surfaces, except for two cases on the dry surface, namely, the fully-loaded vehicle and the bobtail. The study also showed that changing the height of the center of gravity from 16 to 48 in. from the trailer deck had a negligible effect on braking performance.

4.5.4 BRAKE TIME RESPONSE. The importance of brake response time on the mean deceleration capability of a vehicle is discussed in detail in Section 2.6.1. Articulated vehicles, however, have a unique problem due to the difference in actuation time between the tractor brakes and trailer brakes. Although tests of the baseline vehicles indicated that this delay time is of the order 0.15 sec, it is conceivable that longer delay times could result due to unfavorable service conditions. For this reason, several computer runs were made to study the effect of an increasing delay time on the order of wheel lockup for an empty vehicle. The results are shown in Fig. 134, indicating 0.5 sec delay in application of the trailer brakes causes the tractor wheels to start to lockup until the trailer brakes finally apply. As delay times become longer, the tendency of the tractor rear wheels to lock becomes much greater, until at 1.8 sec the wheels lock completely. These results were obtained for an empty vehicle at a high coefficient surface, but the same results would be obtained for lower decelerations on a slippery surface.
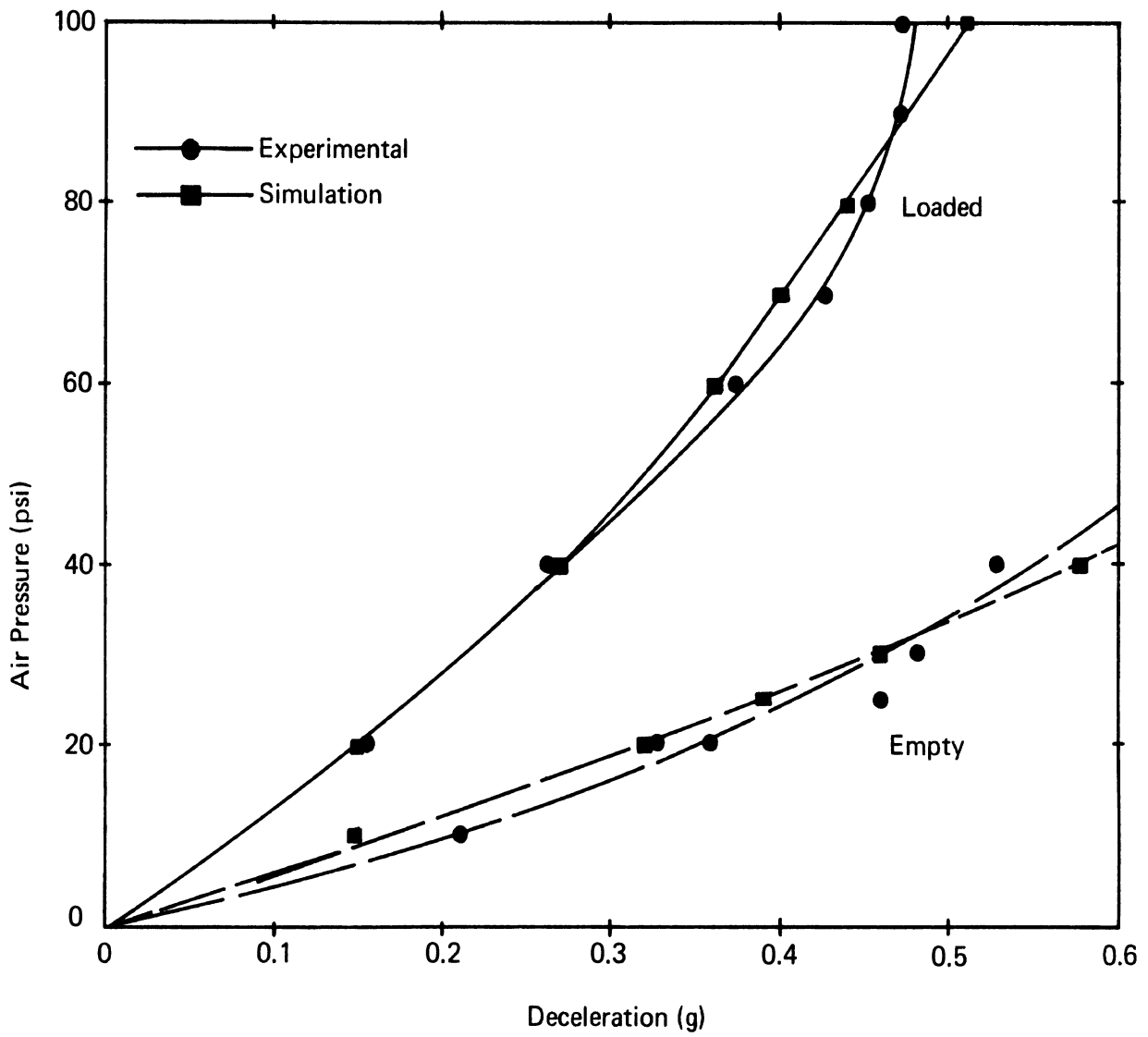
FIGURE 128. SIMULATED AND EXPERIMENTAL LINE PRESSURE-DECELERATION CHARACTER-
ISTICS
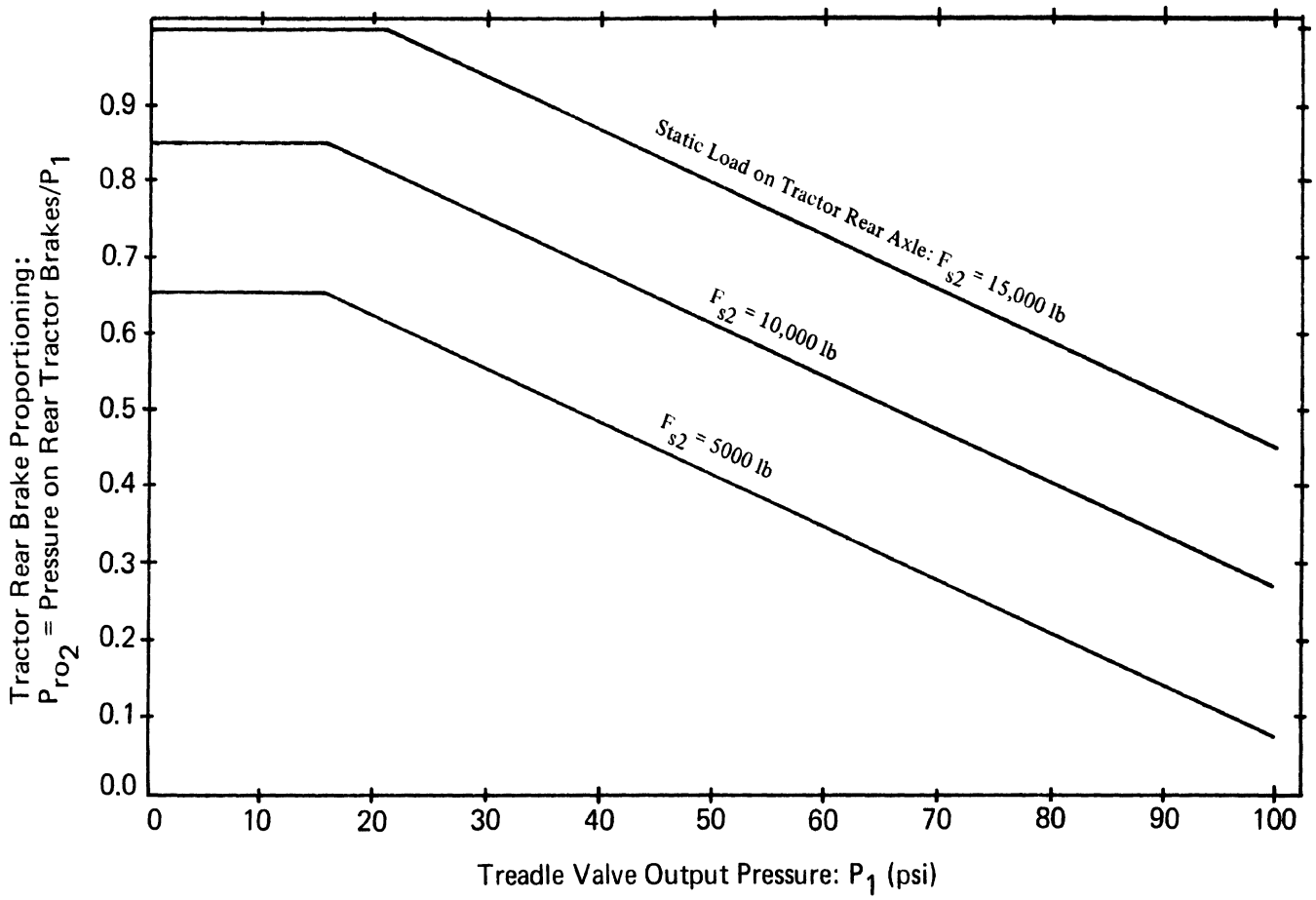
FIGURE 129. TRACTOR PROPORTIONING VALVE CHARACTERISTIC
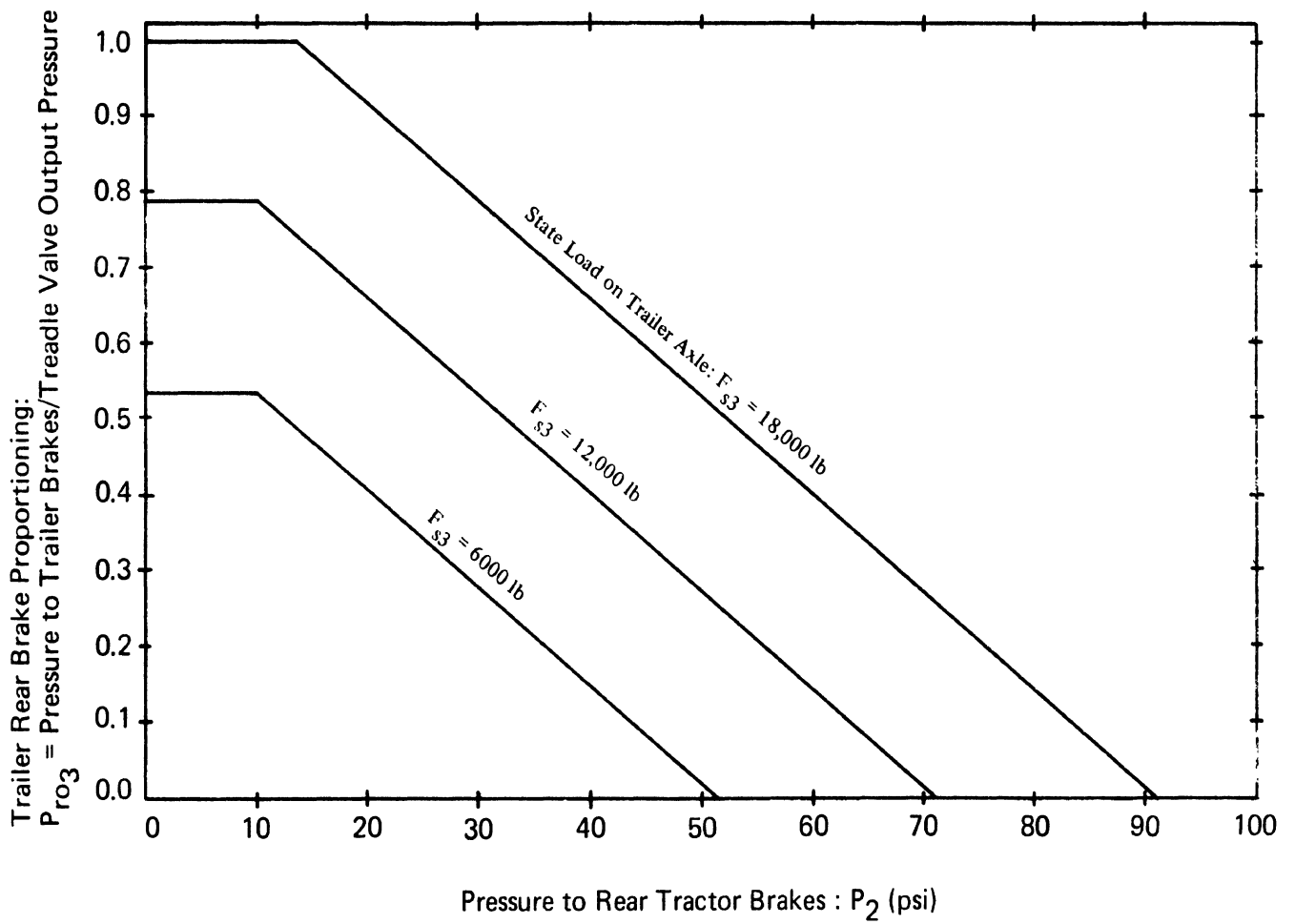
FIGURE 130. TRAILER PROPORTIONING VALVE CHARACTERISTIC

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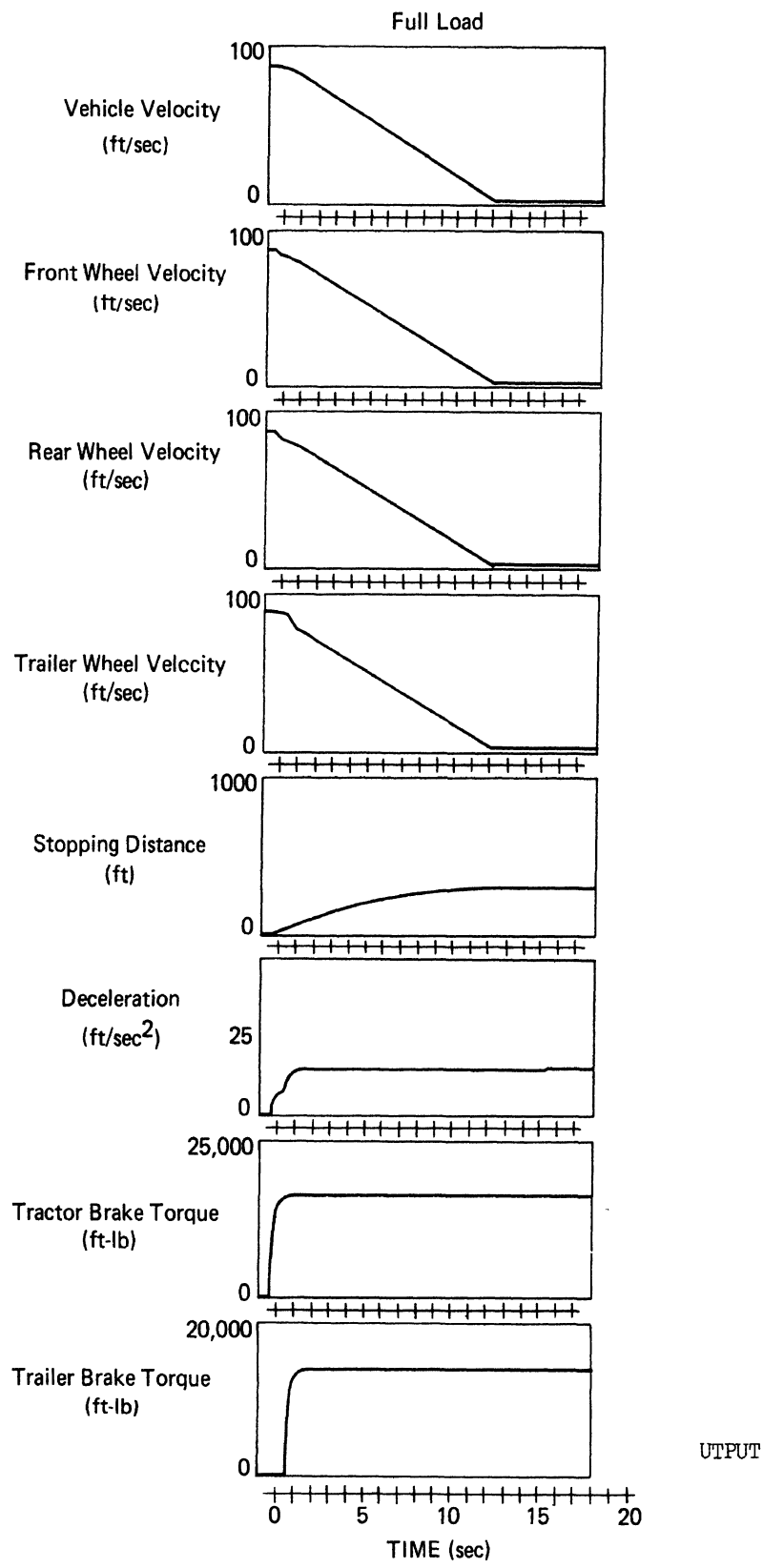
FIGURE 131. COMPUTER SIMULATION, SAMPLE OUTPUT

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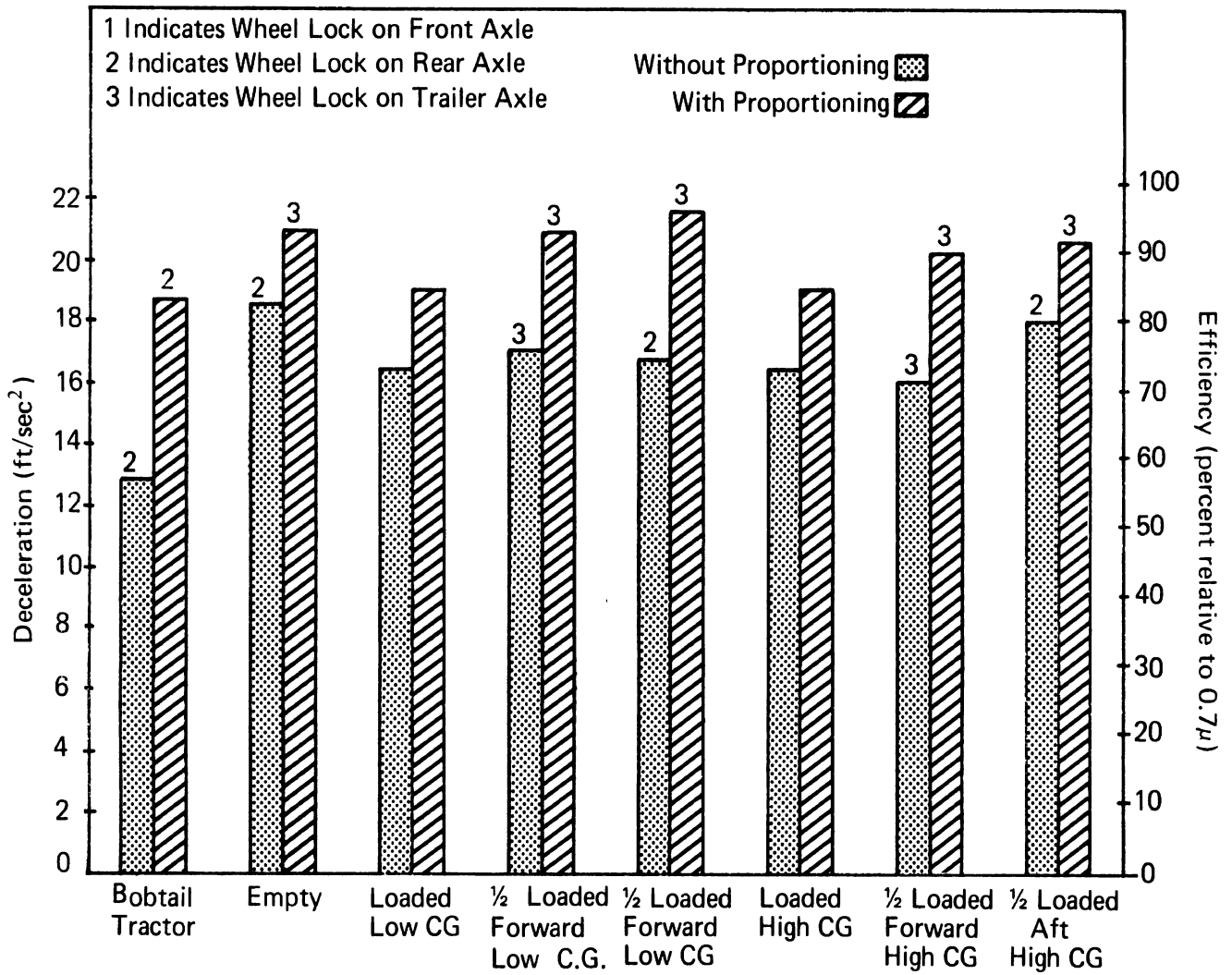
FIGURE 132. PERFORMANCE COMPARISON ON 0.7-μ SURFACE, WITH AND WITHOUT PROPOR-
TIONING

1 Indicates Wheel Lock On Front Axle
2 Indicates Wheel Lock on Rear Axle
3 Indicates Wheel Lock on Tractor Axle

Without Proportioning
With Proportioning

Deceleration (ft/sec^2)

Efficiency (percent relative to .3 μ)

10.6
10
9
8
7
6
5
4
3
2
1
0

110
100
90
80
70
60
50
40
30
20
10
0

Bobtail Tractor
Empty
Loaded Low CG
½ Loaded Forward Low CG
½ Loaded Aft Low CG
Loaded High CG
½ Loaded Forward High CG
½ Loaded Aft High CG
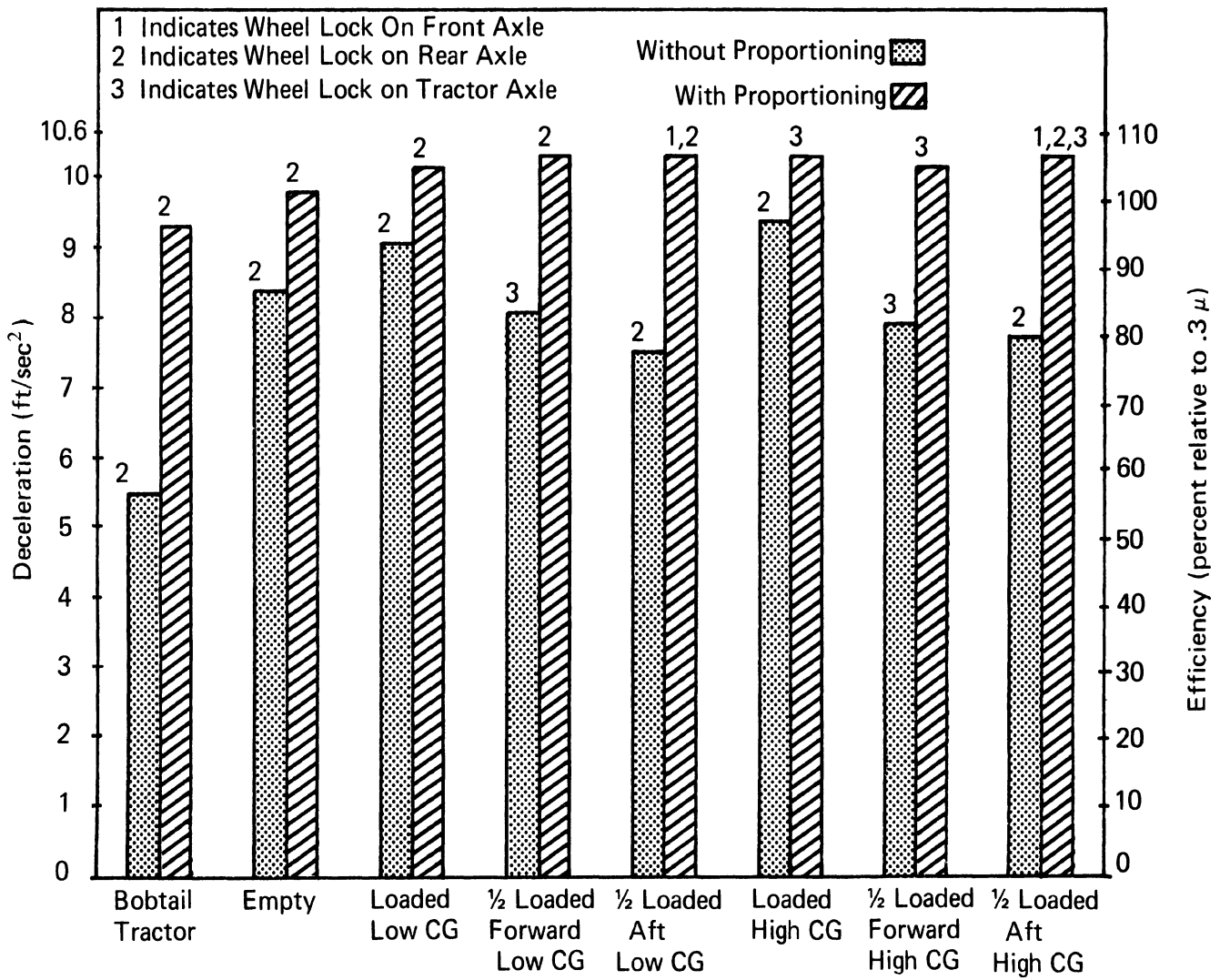
FIGURE 133. PERFORMANCE COMPARISON ON 0.3-μ SURFACE, WITH AND WITHOUT PROPORTIONING
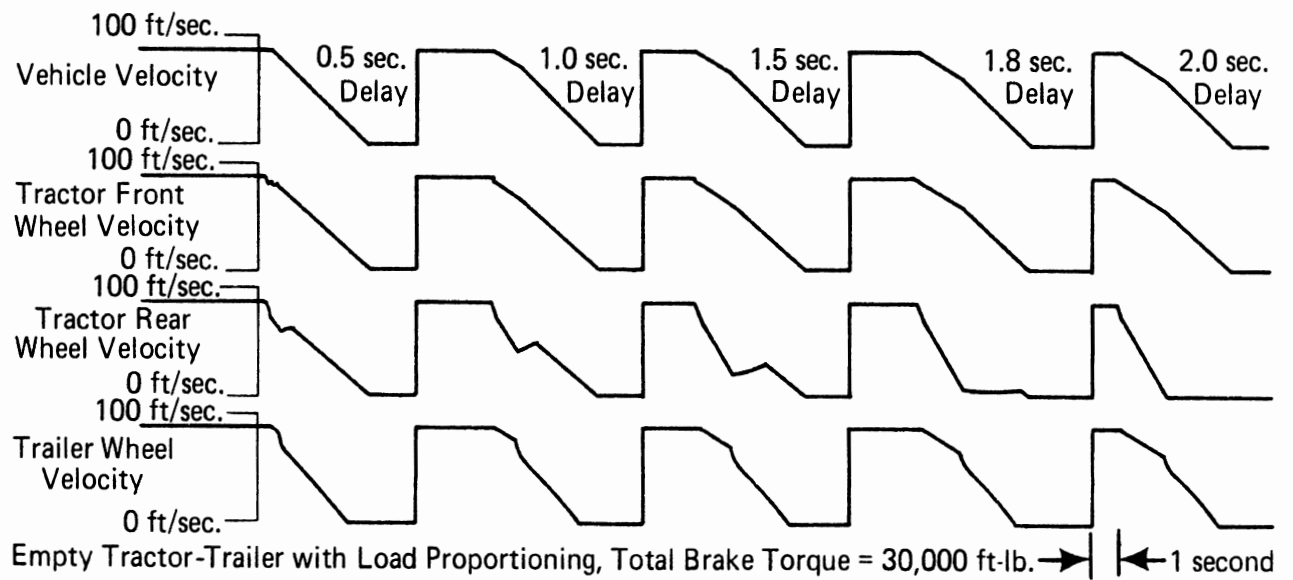
FIGURE 134. EFFECT OF TIME RESPONSE ON ORDER OF WHEEL LOCKUP

TABLE 38

LOADING CONFIGURATIONS AND AXLE LOADS USED IN SIMULATION OF IDEAL PROPORTIONING SYSTEM

Vehicle And Load Configuration	Load, lb	Load CG Height Above Trailer Deck, in.	Axle Loads			Total Vehicle Weight, lb
			Tractor Axles		Trailer Axles	
			Front	Rear		
Bobtail	none	-	2,794	3,341	-	6,135
Tractor-Trailer						
Empty	none	-	3,730	7,340	8,100	19,170
Low CG Full	23,150	16	5,460	16,480	20,070	42,320
Low CG Half Load, For'd	11,270	16	4,980	13,880	11,580	30,440
Low CG Half Load, Aft	11,290	16	4,310	10,380	15,770	30,460
High CG Full Load	23,150	48	3,730	16,480	20,070	42,320
High CG Half Load, For'd	11,270	48	5,460	13,880	11,580	30,440
High CG Half Load, Aft	11,290	48	4,980	10,380	15,770	30,460

4.5.5 INTEGRAL TRUCK EQUIPPED WITH WHEEL ANTILOCK SYSTEM. The purpose of this study was to determine the response of an integral truck equipped with a rear wheel antilock system to braking and steering inputs on a variety of test surfaces. Vehicle and brake system data for the loaded Chevrolet C-30 were employed in a horizontal plane simulation of a two-axle vehicle model* to which an antilock system, based on that described by Madison and Riordan (96), was added. The block diagram of the simulation model is given in Fig. 135.

Twelve simulation runs were made, as listed in Table 39, six of which involved straight ahead braking, and six of which involved combined steering and braking. In every case enough brake torque was applied to lock up all four wheels. The greatest gains in performance were made on the low coefficient surface, wherein stopping distance was consistently decreased by use of the antilock system. Figure 136 depicts the trajectories and resulting yaw angle for the braking and steering maneuver, with one run being made without braking to define the curve. The curves show that although steering was lost due to lockup of the front wheels, the yaw angle was considerably less when the antilock system was operational. The results of this study agree in general with the results of the vehicle tests, but point up the desirability of using an antilock system on the front as well as the rear wheels.

4.5.6 DIRECTIONAL DYNAMICS OF TRACTOR-TRAILERS IN COMBINED BRAKING AND STEERING MANEUVERS. A summary of the work described by Leucht (326) is given in this section. The computer simulation was based upon a mathematical model which describes the longitudinal, lateral, and yawing motions of a three-axle tractor-semitrailer on a flat road surface. Idealized steering and braking inputs were applied to the simulation to determine the influence of (1) design parameters, (2) operating conditions, and (3) the environment on the directional behavior of the vehicle under study.

The response to step steering was used to evaluate the influence of design parameters on transient response and steady turning behavior as measured by response times and lateral acceleration gain,** respectively. Decreasing the forward velocity, moving the fifth-wheel forward, increasing the wheelbase of the tractor, decreasing the load on the semitrailer, and moving the load to the rear of the trailer decreased the lateral acceleration gain and the response times. Increasing the friction at the fifth wheel decreased the lateral acceleration gain and increased the response time of the vehicle significantly.

The effects of brake-system parameter changes on directional response were evaluated by braking the vehicle from a steady turn with a fixed steer angle. Increased braking led to a sharper turn and an increased lateral response as measured by the yawing velocity, lateral acceleration, and sideslip angle of the tractor. The variation in loading condition was identified as a major cause of undesirable directional response with a fixed brake-torque

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*This model is described in detail in reference (325).

**Lateral acceleration gain is defined as the ratio of the steady-state lateral acceleration to the steer-angle input.
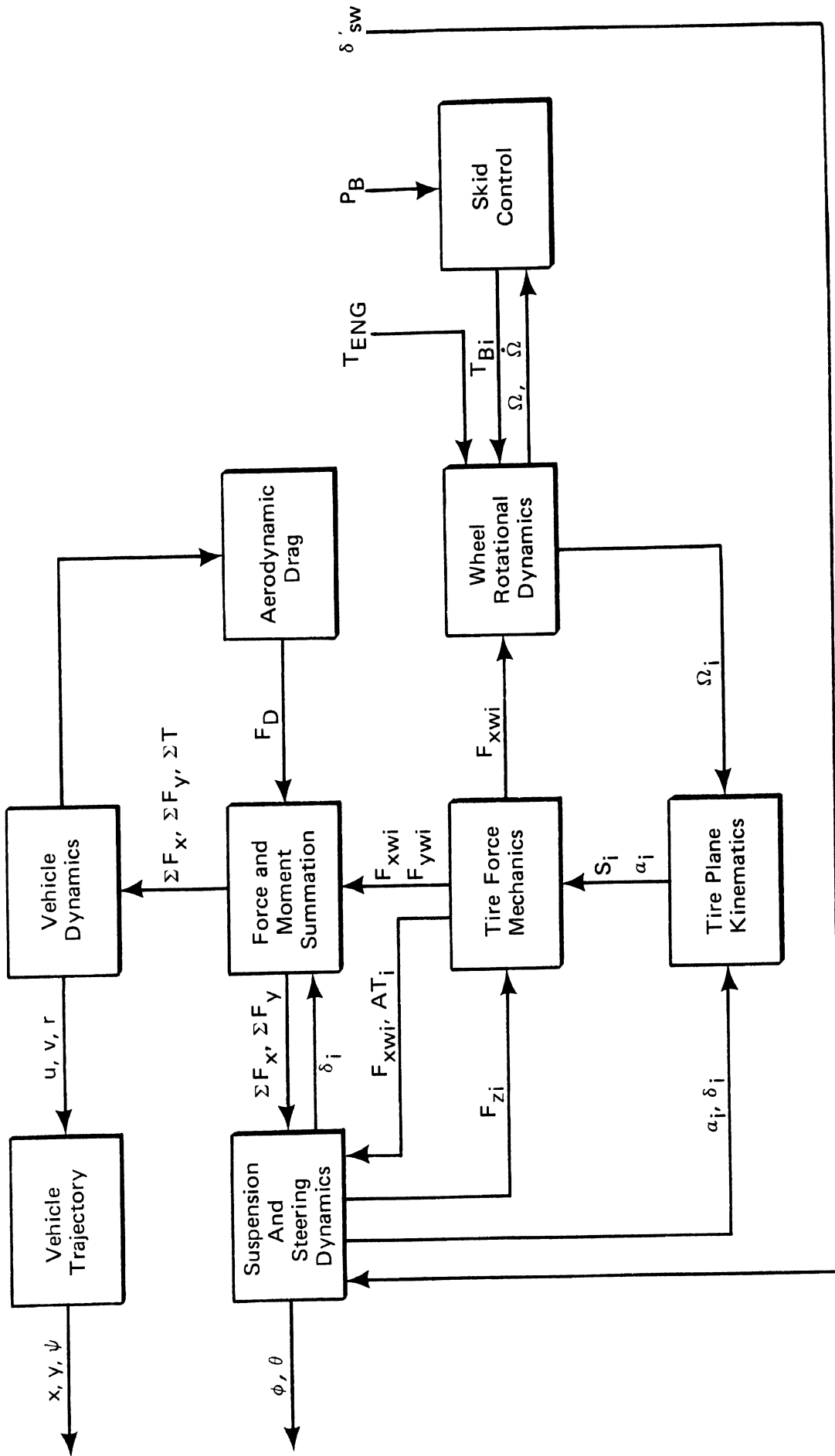
FIGURE 135. BLOCK DIAGRAM OF THE HORIZONTAL PLANE SIMULATION MODEL

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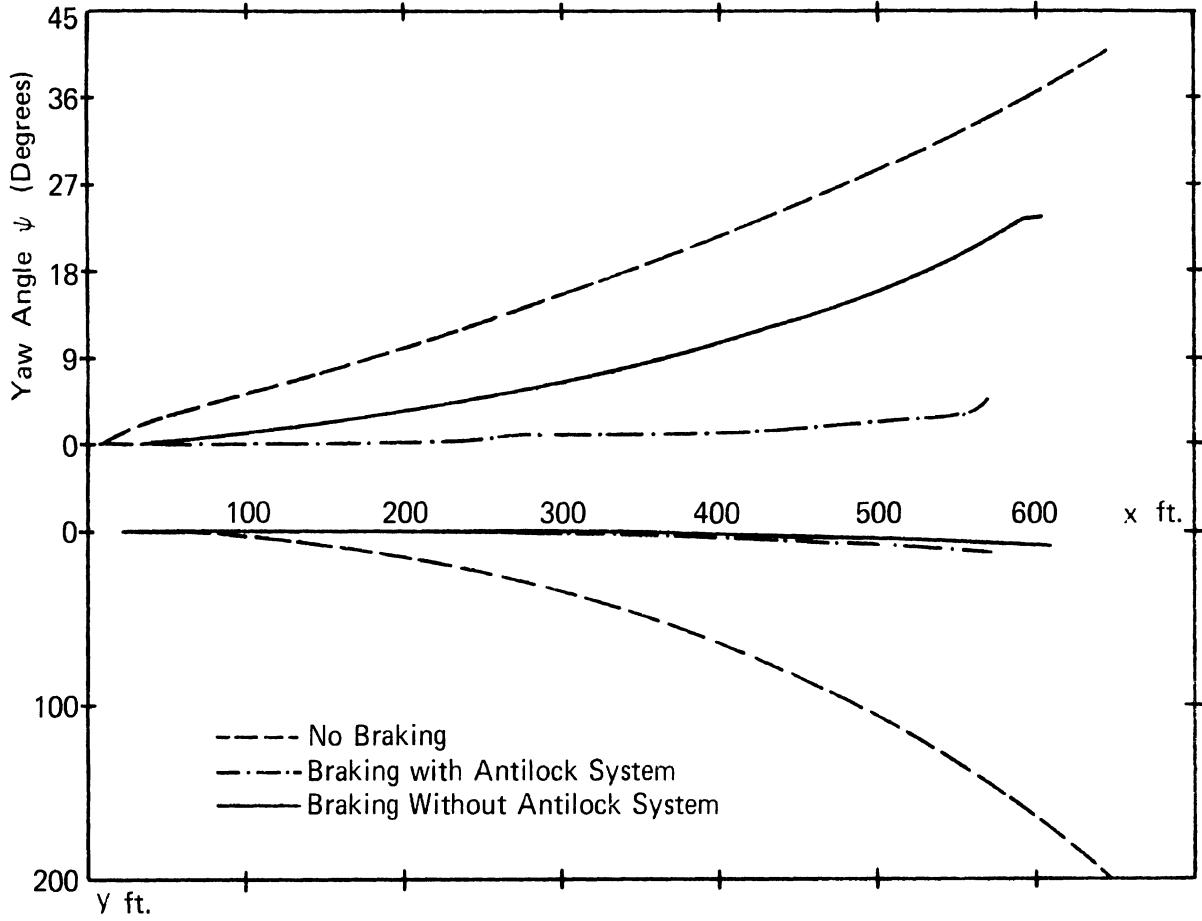
FIGURE 136. VEHICLE TRAJECTORY AND YAW ANGLE FOR TYPICAL SIMULATION RUN

TABLE 39

PERFORMANCE OF LIGHT TRUCK EQUIPPED WITH REAR WHEEL ANTILOCK SYSTEM

Run	V_o, mph	Front Wheel Angle, deg	μ_o	Average Decel, ft/sec^2	Stopping Distance, ft	Final Yaw Angle, deg	Antilock Operational
1	60	0	0.8	21.9	180	0	yes
2	60	0	0.8	20.5	180	0	no
3	60	0	0.4	11.8	310	0	yes
4	60	0	0.4	11.3	340	0	no
5	30	0	0.2	6.4	158	0	yes
6	30	0	0.2	6.2	165	0	no
7	60	5	0.8	21.9	180	4.5	yes
8	60	5	0.8	21.0	185	2.7	no
9	60	5	0.4	10.9	323	3.6	yes
10	60	5	0.4	9.7	335	14.4	no
11	60	5	0.2	5.1	573	4.5	yes
12	60	5	0.2	4.2	607	23.4	no

distribution. Brake-system design characteristics used to improve the system behavior were the brake-torque distribution, load-sensitive brake-torque control, and brake application times. Longitudinal and lateral brake-torque variations from a baseline distribution had the overall effect of decreasing the performance of the vehicle in a combined steering and braking maneuver without any wheels locking.

The effect of the forward velocity and friction at the tire-road interface on combined steering and braking was also studied by braking the vehicle in a steady-state turn. An increase in the forward velocity of the tractor increases the gain of the system resulting in a more severe lateral response being exhibited at higher velocities. The vehicle developed (1) a slower lateral response to braking, (2) more severe stable responses, and (3) more serious unstable responses on a surface with a lower coefficient of friction.

The ability of changes in basic design parameters to mitigate against producing an undesirable lateral response while braking in a steady turn was also examined. Increasing the wheelbase of the tractor and the amount of friction at the fifth-wheel were most effective in reducing the tendency for jackknifing in a combined steering and braking maneuver.

5. PERFORMANCE EVALUATION

In order to evaluate the findings obtained in the experimental and analytical phases of this study, it appeared desirable to categorize braking performance into a number of distinct facets or qualities.

The mechanics of the braking process suggest that there are at least five facets of braking performance deserving of consideration for commercial vehicles. These facets (or measures of braking performance) shall be referred to as effectiveness, thermal capacity, controllability, efficiency, and response, and are defined below.

5.1 PERFORMANCE MEASURES

Brake effectiveness is a measure of the gain of a brake in terms of torque output per unit input of line pressure at the brake chamber. When the braking level is lowered such that the tires are not being forced to operate near their adhesion or friction limit, the total braking force acting on a vehicle is linearly related to the total torque being generated by all of the brakes. Under these conditions, deceleration per unit value of brake line pressure serves as an overall measure of the braking effectiveness of the vehicle. For pneumatic systems characterized by a limit value of line pressure, a finite value of effectiveness for a given brake means that there is an upper limit to the brake torque that can be generated. If this maximum torque is insufficient to produce wheel lock, during the braking process, the maximum value of wheels-unlocked deceleration that can be achieved by the vehicle is degraded.

The thermal efficiency of a brake can be characterized by the ability of the brake to absorb heat generated in a single stop and to conduct or convect away heat generated in a series of stops. The combination of this thermal efficiency (which is mostly a characteristic of the brake drum) with the lining properties determines the fade resistance of the brake. With the instrumentation used in the tests, thermal efficiency could not be measured directly, and so thermal capacity or resistance to fade is measured in terms of the level of braking effectiveness that can be maintained during a series of rapidly repeated snubs (as specified in the fade test procedure for this program) or the number of snubs which can be accomplished in a given time interval (SAE J880), or the decrease in tow bar force in a towing test (as specified in this program for the brake rating test). The fade test as conducted was not successful for the large commercial vehicles tested in this program because brake fade could not be induced. However, if the requirements for the initial and final snub velocities are changed to correspond with those of SAE J786, that is, 40 mph to 20 mph instead of 60 or 50 mph to 10 mph, fade can be induced and a measure of thermal capacity could be obtained. One such performance measure is the number of snubs that can be produced at a specified level of deceleration with the time to accelerate back to test speed between snubs not to exceed a specified period. From the point of view of safety, a thermal capacity test should be based upon the most severe duty cycle that a vehicle might en-

counter in general use.

Brake controllability is used here in the sense of the driver being able to modulate brake force under a wide variety of loading and road surface conditions to minimize stopping distance while preventing wheel lockup. Deceleration/pedal force gain is a performance measure that can be used to characterize brake controllability (13).

Braking efficiency is a measure of the ability of a vehicle to utilize the friction forces available at the tire-road interface. Strictly speaking, it is defined as the ratio of the maximum wheels-unlocked deceleration capability of the vehicle on a given surface to the peak tire-road friction coefficient of that surface. When braking efficiency is determined experimentally, the surface on which the vehicle is tested must be measured to determine the peak tire-road friction coefficient. At the present time, adequate means do not exist for making such a measurement with large truck tires. Therefore, as an alternative, it is proposed that braking efficiency be determined by calculating this measure, as was accomplished for the test vehicles in this program.

Brake response time is defined as the time required for a brake to reach a given level of effectiveness from the time that the brake control (pedal) is activated. Measurements of response time in an actual stop would therefore require torque sensors on each braked wheel. Consequently, a more common means for determining the response time of pneumatic brake systems is to measure the time from the instant of pedal application (resulting in a full, fast opening of the treadle valve) to the instant a given pressure level is reached in the brake chamber.

Measurements show that the response times typically exhibited by air-brake systems are sufficient to influence the braking performance of commercial vehicles, as measured either by average deceleration or stopping distance. Synchronization of brake timing is important for preventing instabilities in articulated vehicles. Brake-release time is significant when the driver is attempting to modulate the brake force to prevent wheel lockup. Although the importance of response time is generally recognized, a consensus has yet to be reached with respect to defining a standardized test practice for measuring brake-response and brake-release times. Consensus is lacking on (a) how to specify the actuation of the system, (b) what point shall be selected to serve as "zero time" and (c) what level or percentage of the steady pressure response should be selected to serve as the basis for computing the response or release time. The problem is further complicated by the nonlinear character of pneumatic control systems; namely, the response times increase if the pressure increments requested by the driver are decreased. Although brake response time as measured by the time to achieve a specified level of effectiveness (i.e., brake torque output) is not exactly equivalent to the time required to attain a specified pressure level in the brake chamber, the latter measurement does serve to characterize the dynamic properties of a given brake system and to indicate the degree to which individual brakes on a vehicle are synchronized with each other. It appears that brake response, irrespective of the

precise manner in which it is determined either on application or release, is a significant measure of commercial brake system performance.

The performance measures defined above are summarized in Table 40.

TABLE 40

PERFORMANCE MEASURES

Quality Measures	Performance Measure	Symbol	Technique
1. Effectiveness	Maximum deceleration (wheels unlocked)	a_x	Effectiveness test
2. Thermal capacity	Number of snubs achieved at a given deceleration	n	Fade test
3. Efficiency	Braking efficiency	$-a_x/\mu$	Calculation of efficiency and tire/road interface test
4. Controllability	Deceleration/pedal force gain	$-a_x/F$	Effectiveness test
5. Brake response	Application and release times (air brake systems only)	T_a, T_R	Static response time test

5.2 MAXIMUM ACHIEVABLE PERFORMANCE BASED ON TESTS AND ANALYSIS

The maximum braking performance that can be achieved by a vehicle on a given test surface is limited by five factors:

(1) The frictional forces available at the tire-road interface

(2) The effectiveness of the vehicle's brakes, that is, the maximum torque capacity of the brakes

(3) The braking efficiency of the vehicle, that is, how well the brake torque is balanced axle to axle such that the tire-road frictional forces are best utilized

(4) The ability of the driver to modulate the pedal force such that maximum deceleration and minimum stopping distance are achieved without loss of directional control and stability

(5) The time response of the brake system to an applied pedal force

Tire-road interface tests indicate that the friction coefficient of truck tires, both peak and sliding, is significantly less than that of passenger car tires. The effectiveness tests point up considerable variation in tire-road friction coefficient among the various types of truck tires. For example, the light truck was capable of locked-wheel decelerations of as high as

28 ft/sec^2, while the disk-braked truck was capable of only 22 ft/sec^2 with the wheels locked. Since tests with both the Michigan Department of State Highways skid trailer and the HSRI instrumented car indicated that the friction coefficient of the test track varied only slightly over the entire period of testing, the difference in locked-wheel deceleration must be attributed to difference in tire traction. The importance of this observation cannot be overemphasized since it is pointless to increase the effectiveness of brakes if frictional forces corresponding to the increased brake torque capabilities cannot be produced at the tire-road interface. The reader is therefore cautioned to judge the performance capabilities of the larger trucks and tractor-trailers tested in this program in light of these findings on truck tire traction.

The maximum wheels-unlocked deceleration achievable, as determined either from an effectiveness test or a minimum stopping distance test, is a measure of not only the torque capacity of the brakes, but also of how well the vehicle utilizes the friction available at the tire-road interface. The maximum deceleration obtained on the dry surface is summarized in Fig. 137 for all of the vehicles tested. With performance depicted in this fashion, it is immediately obvious that the vehicles equipped with advanced brake and brake-control systems did not exhibit braking performance that was significantly better than was achieved by the baseline vehicles. In fact it can be argued that a truck, bus, or tractor-trailer with brakes balanced for maximum braking performance can exceed the performance achieved on a dry surface by advanced brake control systems. For example, the light truck, all the buses, and the 2-S2 tractor-trailer combination, exhibited performance on the dry surfaces as good as Vehicle 12 and considerably better than Vehicle 14. The disk-braked truck, however, was outperformed only by the light truck, and this result may stem from a difference in tire traction. Figure 138 perhaps better depicts the improvement in performance that can be expected by use of advanced brake-control systems. Since wheel lockup is viewed as undesirable, Fig. 138 presents only the maximum decelerations obtained up to the point of wheel lockup. Considerable improvements in performance are achieved for Vehicle 12 through use of the adaptive braking system, especially in the tests conducted on the low-coefficient surface, and in the tests conducted with the empty vehicle on the dry surface. It should be noted that the effectiveness of the brakes on Vehicle 12 was insufficient to lock up all the wheels on the dry surface in the loaded condition.

Vehicle 14 did not perform as well as Vehicle 12, on either the dry surface (even after making allowance for the slightly lower tire-road friction coefficient of the surface upon which Vehicle 14 was tested) or the wet surface. The empty tractor (bobtail) of Vehicle 14 produced a maximum deceleration of 19 ft/sec^2 on the dry surface, while the combination, empty or loaded, produced 15 ft/sec^2. Although the antilock system did improve performance of the combination as measured by the maximum deceleration achievable without wheel locking, the improved performance was considerably less than that which could have been achieved by better utilization of the forces in the tire-road interface.
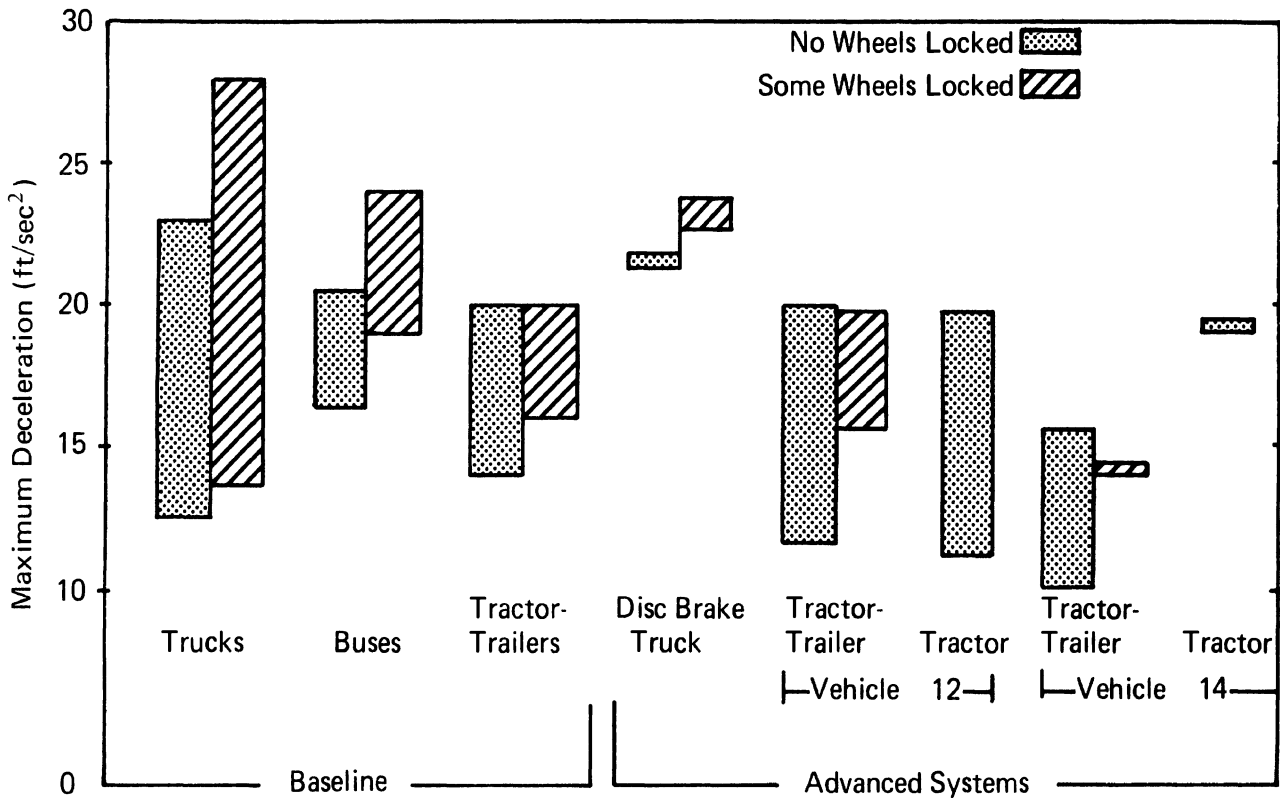
FIGURE 137. MAXIMUM DECELERATION PERFORMANCE RANGES, BASELINE AND ADVANCED
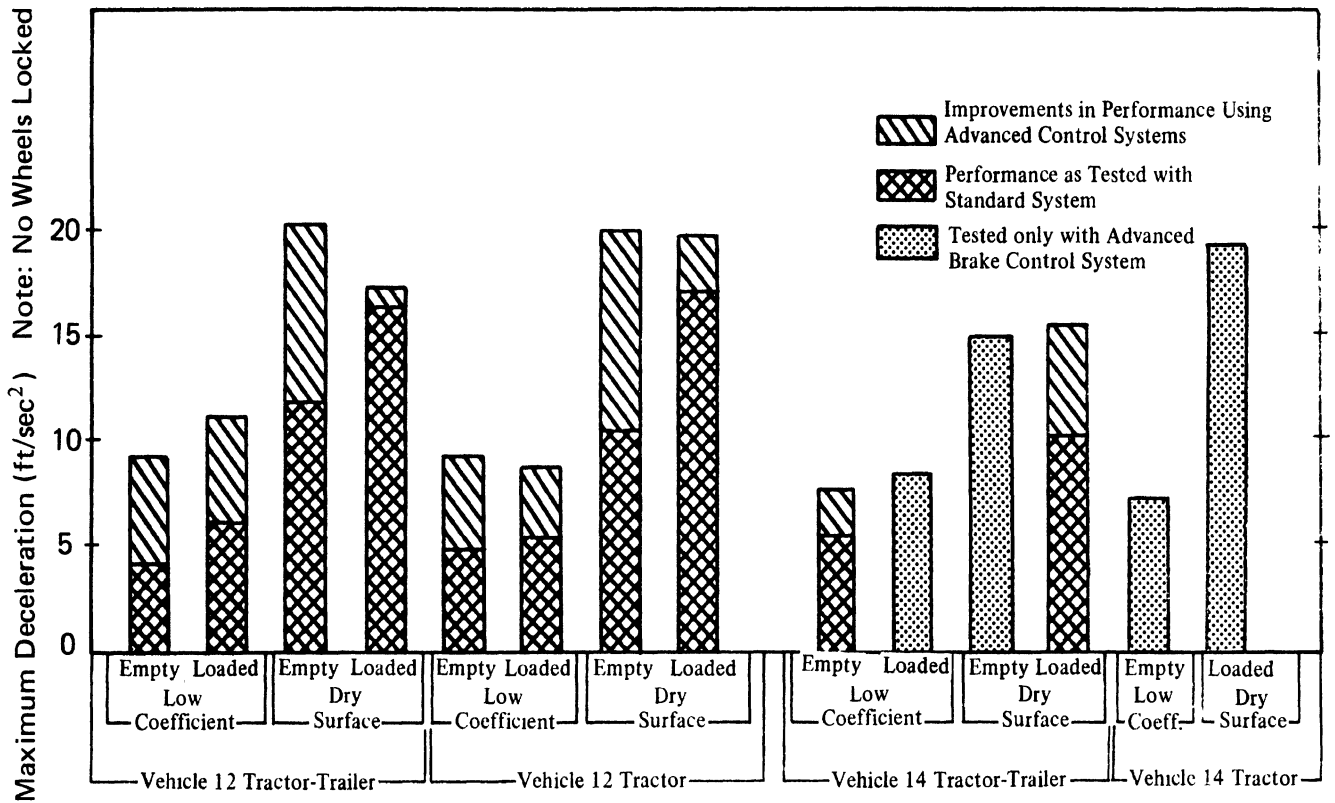SYSTEMS

FIGURE 138. IMPROVEMENT IN MAXIMUM DECELERATION PERFORMANCE WITH ADVANCED
SYSTEMS

Using the experimental data at hand, it is impossible to calculate exactly the braking efficiency of the tested vehicles, since the peak values of tire-road friction coefficient, upon which efficiency calculations must necessarily be based, have been determined in an approximate manner. However, using these approximate coefficients, the efficiency exhibited by each vehicle and system tested is tabulated in Table 41. Using efficiency as the criterion, the disk-braked truck showed best performance on the dry surface. The system which produced the best overall efficiency measure was the adaptive braking system installed on Vehicle 12. Next best was the antilock system installed on Vehicle 14. The smallest performance gains were achieved with the proportioning system. As pointed out earlier, minor adjustments in the proportioning valves, plus design steps that prevent premature lockup of the wheels on the leading axle of the trailer, would have markedly improved the performance obtained with the proportioning system. The simulation study described earlier has indicated the magnitude of performance improvements that can be expected from use of proportioning systems.

TABLE 41

OVERALL BRAKING EFFICIENCIES FOR ADVANCED SYSTEMS TESTED

Vehicle	System	Dry Surface		Low Coefficient	
		Empty	Loaded	Empty	Loaded
Disk Brake Truck	-	90	88	--	--
Vehicle 12 Tractor-Trailer	Standard	48	67	--	54
	Proportioning	47	67	--	--
	Adaptive	83	70	81	96
Vehicle 12 Tractor	Standard	46	70	43	47
	Proportioning	82	70	30	47
	Adaptive	82	80	80	75
Vehicle 14 Tractor-Trailer	Standard	--	45	46	--
	Antilock	66	68	66	72
Vehicle 14 Tractor	Antilock	86	--	63	--

If a vehicle is equipped with an adaptive or wheel antilock braking system, the driver does not have to modulate the pedal force to prevent wheel lock. However, for vehicles not so equipped, a recent study (13) at HSRI showed that the driver's ability in modulating the brake-pedal force to prevent wheel lockup is directly dependent upon the deceleration-pedal force characteristic of the vehicle. Of all the vehicles tested, the disk brake truck achieved the greatest maximum deceleration without lockup of wheels.

Figure 139 shows the deceleration-pedal force gain characteristics of the disk brake truck along with the recommended upper and lower limits derived from the HSRI study. It is interesting to note that those baseline vehicles in which the driver was able to modulate the pedal pressure such that a high deceleration was possible without wheel lockup all had deceleration-pedal force characteristics which were within the limits suggested by the HSRI study. These vehicles included the light truck, the three buses, and the 2-S2 tractor-trailer combination. It therefore appears reasonable to conclude that these limits may also serve as a guideline in selecting the deceleration-pedal force characteristics for buses, trucks, and tractor-trailers.

Measurements of brake response time show that the standard braking systems of Vehicles 12 and 14 had application/release times that were approximately the same or slightly longer than those of the baseline tractor-trailers. However, when using Syncron on Vehicle 12 and the antilock system on Vehicle 14, the response times of the trailer brakes were improved considerably. Results from the baseline vehicles, the standard systems of Vehicles 12 and 14, and the improvement of response using advanced systems are summarized in Table 42.

TABLE 42

SUMMARY OF BRAKE RESPONSE TIMES

| | Application/Release Time, sec | |
Vehicle	Tractor Rear Axle	Trailer Rear Axle
Average Baseline Vehicles	0.24/0.55	0.28/0.79
Vehicle 12 Standard System	0.27/0.30	0.40/0.70
Vehicle 12 with Syncron	0.27/0.30	0.30/0.40
Vehicle 14 Standard System	0.23/0.36	0.36/0.64
Vehicle 14 with Antilock	0.23/0.44	0.27/0.60

5.3 MEANS FOR IMPROVING PERFORMANCE

The above discussed findings, as derived both from analyses and test, indicate that three major steps will have to be taken to significantly upgrade the maximum braking performance of commercial vehicles.

First. The basic braking systems of the majority of these vehicles will have to be improved by use of more effective brakes, better brake balance, and faster system response on air braked vehicles.

Second. The traction characteristics of tires used on the majority of medium and heavy commercial vehicles will have to be improved so that the advantage of improved brake effectiveness can be fully utilized at the tire-road interface.

Third. Advanced brake control systems will have to be employed to allow rapid brake applications without instigating vehicle instability whether the
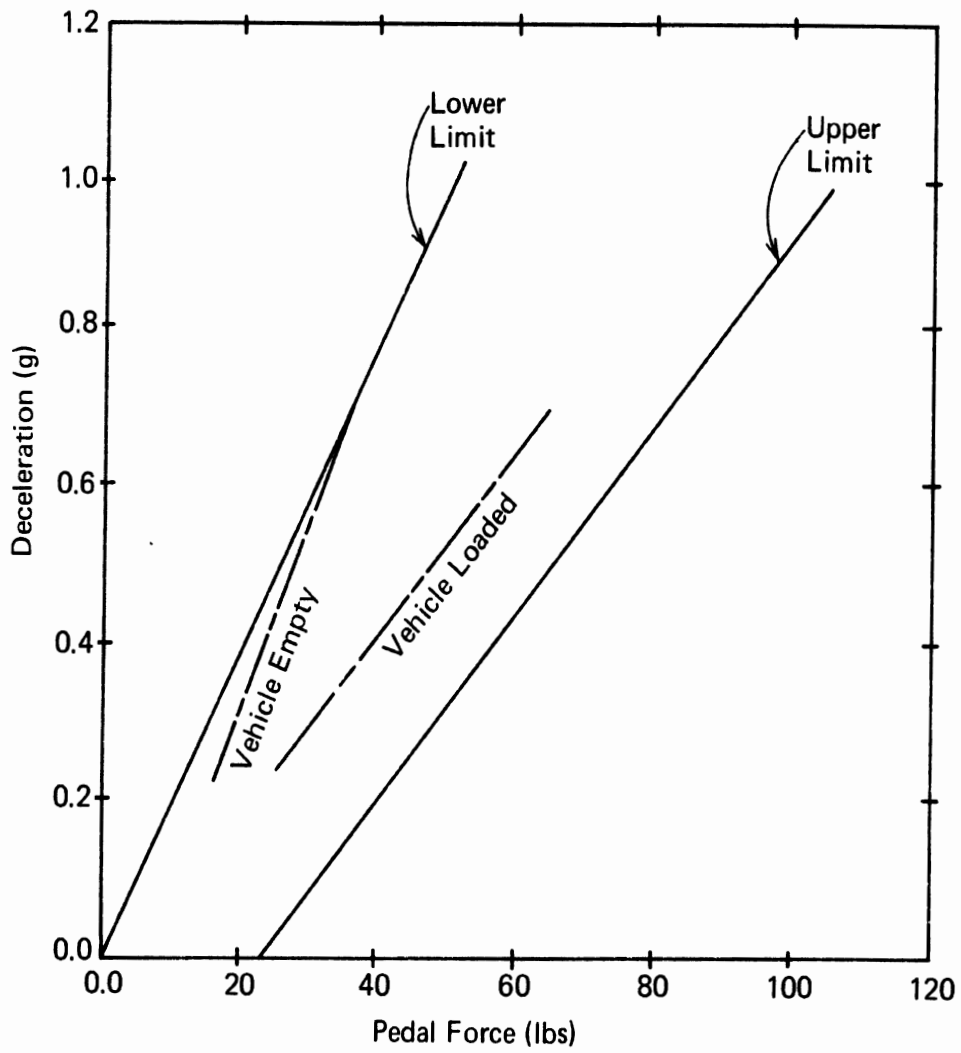
FIGURE 139. DECELERATION/PEDAL FORCE GAIN FOR DISK BRAKE TRUCK AND HSRI RECOM-
MENDED LIMITS

vehicle be loaded or empty, and operating on a dry or slippery surface.

A number of design alternatives exist for achieving these objectives:

(1) The effectiveness and fade resistance of the braking systems on medium and heavy trucks can be improved significantly by use of disk brakes.

(2) The effectiveness of the braking systems of tractors can be improved by use of large brakes on the front axle of tractors with tandem rear axles (a design configuration in which front brakes are generally absent) and by use of larger brakes on the front axle of two-axle tractors.

(3) The braking efficiency of many trucks and tractor-trailers can be improved by careful distribution of braking effort among the axles of the vehicle.

(4) The brake response time of air braked systems can be improved significantly through use of larger hoses, improved connectors and fittings, quick release valves, relay valves on tractors, and trailer brake synchronization.

(5) Braking performance can be improved significantly on trucks, buses, and tractor-trailers through use of the advanced brake control systems, which were evaluated by test and/or simulation in this program. These systems, ranked in order of potential for improving braking performance, are:

    (a) antilock system

    (b) dynamic load-sensitive proportioning system

    (c) static load-sensitive proportioning system

5.4 COST EFFECTIVENESS

Firm cost figures for the advanced systems tested are not available at the time of this writing. It is impossible, therefore, to make an accurate assessment of gain in performance per unit cost, for example, of antilock systems as opposed to proportioning systems.* The problem is further compounded by other factors. Prototypes and fleet test units usually cost more to produce than units in mass production. Maintenance required for advanced systems will increase vehicle operating costs. Increased weight due to larger brakes, stronger suspension systems and vehicle structures will increase initial vehicle costs and decrease payload capacity. Front brakes and power steering systems, if required on tandem axle trailers, will also increase costs. Cost savings, however, would result from use of advanced systems by decreased tire wear and less flat-spotting. With increased braking and control capabilities, drivers may be able to maintain faster schedules. The greatest potential cost advantage, however, is the decrease in accidents caused by skidding and subsequent loss of control.

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*Yarber has assessed relative system costs and performance gains for various antilock configurations in reference (327).

5.5 OTHER FACTORS

In considering any of the above design alternatives, one must bear in mind that the commercial vehicle has evolved over the years as a design compromise. The vehicle's structure, suspension, and brake system have been designed for a given level of average braking performance, with a capability of accepting a certain amount of overload in emergency situations. The pneumatic tires are part of this design compromise, since there isn't much point in designing high tractive capability into truck tires, at the expense of increased rolling resistance and higher wear rates, if that tractive capability cannot be matched by brake torque capabilities as constrained by brake size and brake design practice. The aim of this design compromise has been to produce vehicles which are safe and reliable within their performance range, and which are characterized by high payload/vehicle weight ratios and minimal operating and maintenance costs. To introduce a requirement for severely increased braking capability into the commercial fleet, as it has evolved, will necessarily require a reevaluation of the design of the entire system. The following points are suggested for serious consideration:

(1) More effective brakes will require stronger suspensions and stronger adjacent vehicle structures.

(2) Large brakes on the front axles of tractors could require new front axle and steering system designs, and, in many cases, the use of power steering.

(3) With increased deceleration capability, methods of cargo restraint will have to be reevaluated. On buses, passenger restraint systems may have to be utilized.

(4) The relatively high ratio of center of gravity height above roadway to truck width that is common to straight trucks, caused vehicle stability problems to be encountered at moderate decelerations in this test program. It is expected that the problem will be worse at higher decelerations. This problem may be alleviated by use of antilock systems. However, at this point in time, the problem is not clearly defined and requires much more study before a definite solution can be suggested.

(5) If proportioning and/or antilock systems are to be widely used, cognizance should be taken of maintainance and reliability problems associated with each system. Load-sensitive proportioning systems require mechanical, pneumatic, or other means of sensing changes in load. Due to wear, corrosion, and other degrading factors, the level of Coulomb friction in the suspension system may change, thus requiring periodic inspection and adjustment of the linkage. Since antilock systems for air-braked vehicles are still in the developmental stage, reliability problems with both mechanical and electronic components were encountered in the test program. It is mandatory that antilock systems have a high degree of reliability because of the human factors involved. The test program has pointed out that regardless of load or surface condition, the driver will make rapid, high-level brake

applications if he knows the antilock system is operational, whereas
he will be extremely sensitive to load and surface conditions when
applying the brakes without the antilock system in operation. Serious
stability problems are possible if the driver applies the brakes
rapidly thinking that the antilock system is operational where, in-
deed, it is not, due to a component failure.

6. RECOMMENDATIONS

6.1 RECOMMENDATIONS FOR A PERFORMANCE STANDARD

In making specific recommendations for a standard, careful consideration has been given to the necessity to upgrade commercial vehicle braking performance as quickly as possible to acceptable levels. Careful consideration has also been given to those points discussed in Section 5. Taking into account the system design problems which will result from increased performance requirements and the state of development of advanced systems, it is recommended that rules be promulgated which require upgrading the performance of trucks, buses, and tractor-trailers in three discrete steps, separated by appropriate periods of time.* As a first step, it is recommended that the rules require immediate action to upgrade braking performance to a level achievable by current design practice, that is, the best performance already demonstrated by baseline vehicles tested. For the second step, it is recommended that the rules require performance to be improved to the limit of the tire-road interface tractive capabilities of truck tires now available with due regard to realistic braking efficiencies. The second step may require use of load-sensitive proportioning systems on certain vehicles, and therefore sufficient lead time should be allowed for further development and testing of these devices. After an appropriate time interval to allow for development and testing of a reliable antilock system, the development of truck tires with better tractive characteristics, and the necessary design modifications of vehicle brake, suspension, and structural systems, it is recommended as a third step that performance equal to or approaching that of passenger cars be required along with use of an antilock system to insure vehicle stability over a wide range of vehicle loadings and road surface conditions. Summaries of the suggested performance requirements for each step are given as follows:

Step 1
- Maximum deceleration capability: 16 ft/sec^2**
- Minimum braking efficiency: 65% for surfaces having peak truck tire-road friction coefficients between 0.2 and 0.3
- Thermal capacity: Same as requirements of SAE J736 fade and recovery test except that 15 ft/sec^2 deceleration is required for fade snubs
- Deceleration/pedal force gain: HSRI recommendations as given in Fig. 139.
- Air brake response time: Application,0.25 sec (tractor), 0.35 sec (trailer); Release, 0.50 sec (tractor), 0.70 sec (trailer)
- Special systems required: None

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*Recommendations for a specific time frame or schedule to implement these steps cannot be made since information on lead times for introduction of design changes, development of new hardware and necessary manufacturing techniques is not generally available from vehicle, brake, and brake component manufacturers.

**It is presumed that this deceleration would be measured on a surface having a peak truck tire-road friction coefficient of at least 0.75.

Step 2
•Maximum deceleration capability: 20 ft/sec^2*
•Minimum braking efficiency: 75% for surfaces, as in Step 1
•Thermal capacity: Test upgraded to correspond with heaviest duty cycles experienced in class of service
•Air brake response time: Application, 0.25 sec (tractor), 0.30 sec (trailer); Release, 0.30 sec (tractor), 0.40 sec (trailer)
•Special systems required: Static load proportioning (if necessary)
Step 3
•Maximum deceleration capability: 24 ft/sec^2 with upgraded tires such that peak tire-road surface coefficient is at least 0.85 on a typical dry asphalt or concrete surface
•Minimum braking efficiency: 85% for surfaces having peak truck tire-road friction coefficient between 0.2 and 0.9
•Special systems required: antilock system
•Improved tires also required

6.2 COMPLIANCE WITH REQUIREMENTS

Test procedures similar to those used in the test program are recommended for determining brake effectiveness, fade resistance, deceleration/pedal force gain characteristics, and static response time of air-brake systems. If stopping distance tests are required, they should be made in conjunction with the effectiveness tests to ensure maximum unlocked wheel decelerations are achieved. Compliance with braking efficiency requirements can be made by requiring design calculations similar to those outlined in Section 4.2, and by validating the calculations by effectiveness tests on high coefficient and low coefficient surfaces.

Testing of tractor-trailer combinations presents a special challenge. Because of brake balance problems, a tractor which may perform very well with one trailer but perform poorly with another. Conversely, a trailer whose brakes were tested on a brake dynamometer and deemed adequate may perform well with one tractor but perform poorly with another. Also a tractor could perform well as a loaded straight truck, but poorly in combination with a trailer. For this reason it is recommended that tractors be certified to pull only those trailers with which it has been demonstrated by design calculation and test that performance of the combination vehicle is adequate.

6.3 FUTURE RESEARCH

As a result of the analytical work and vehicle testing accomplished in this program, certain problem areas have been pointed up which require careful study and research. Two of these problem areas require immediate attention:
(1) Truck tires: Test data are required to determine accurately the μ-slip characteristics of truck tires in order to assess the maximum potential braking performance possible with vehicles equipped with

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*It is presumed that this deceleration would be measured on a surface having a peak truck tire-road friction coefficient of at least 0.75.

these tires.

(2) Truck stability: Analytical work, accompanied by appropriate tests, is required to determine the causes of directional instabilities of integral trucks when subjected to severe braking maneuvers resulting in high decelerations. Study of combined braking/turning maneuvers to assess vehicle stability and limit maneuver capability should also be undertaken.

The thermal characteristics of commercial vehicle brakes also require further study. Analytical work accompanied by carefully instrumented tests should be performed to determine the factors influencing the thermal efficiency of the brake and the best means of measuring such efficiency. Research is also required to insure that any thermal capacity test specified should subject the brakes to duty cycles equivalent to the most intense cycle that the vehicle may encounter during its service life.

Finally, further study of the dynamics of articulated vehicles is necessary. An assessment of the state of knowledge relative to the longitudinal and lateral dynamics of articulated vehicles (328) indicates that two areas require investigation:

(1) The articulated vehicle presents unique control and stability problems, making demands on driver skill which should be evaluated in terms of potential for loss of control. A study of the vehicle over its full range of operating environmental and off-design configurations is required. By focusing particularly on "limit maneuver characteristics" (329), discrimination can be made among vehicle configurations on the basis of peak performance available for accident avoidance maneuvers.

(2) An accurate characterization of the traction mechanics of the tire operating under combined side and longitudinal slip is needed for accurate dynamic modeling and simulation of vehicle response in combined braking and turning maneuvers.