Development of Car/Trailer Handling and Braking Standards. Vol. II: Technical Report for Phase I – Rear Wheel Drive Tow Cars
NTIS, November 1979

DISCLAIMER

DOT HS-805 327

DEVELOPMENT OF CAR/TRAILER HANDLING AND BRAKING STANDARDS

Volume II: Technical Report for Phase I—Rear Wheel Drive Tow Cars

Richard H. Klein
Henry T. Szostak

Systems Technology, Incorporated
13766 South Hawthorne Boulevard
Hawthorne, California 90205

Contract No. DOT HS-7-01720
Contract Amt. $245,544

November 1979
FINAL REPORT

This document is available to the U.S. public through the National Technical Information Service, Springfield, Virginia 22161

Prepared For
U.S. DEPARTMENT OF TRANSPORTATION
National Highway Traffic Safety Administration
Washington, D.C. 20590

PB80-196629

Development of Car/Trailer Handling and Braking Standards. Volume II: Technical Report for Phase I - Rear Wheel Drive Tow Cars

Systems Technology, Inc.
Hawthorne, CA

Prepared for

National Highway Traffic Safety Administration
Washington, DC

Nov 79

Technical Report Documentation Page

1. Report No.
DOT-HS0805 327

2. Government Accession No.

3. Recipient's Catalog No.
PB80 196629

4. Title and Subtitle
DEVELOPMENT OF CAR/TRAILER HANDLING AND BRAKING STANDARDS
Volume II: Technical Report for Phase 1 — Rear Wheel Drive Tow Cars

5. Report Date
November 1979

6. Performing Organization Code

7. Author(s)
Richard H. Klein and Henry T. Szostak

8. Performing Organization Report No.
TR-1114-1

9. Performing Organization Name and Address
Systems Technology, Inc.
13766 South Hawthorne Boulevard
Hawthorne, California 90250

10. Work Unit No. (TRAIS)

11. Contract or Grant No.
DOT-HS-7-01720

12. Sponsoring Agency Name and Address
Office of Passenger Vehicle Research
National Highway Traffic Safety Administration
Department of Transportation
Washington, D. C. 20590

13. Type of Report and Period Covered
Final Report
9/77-11/79

14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract
This report identifies suitable handling and braking performance requirements for rear-wheel drive passenger cars towing trailers. This includes straight line braking, trailer swing stability, tow car steady turn stability, and combined brake-in-turn stability. Over 2000 combination-vehicle tests with eight trailers and three tow cars (in over 250 different configurations), plus the results of previous trailer towing research were then used to validate and/or revise the criteria and procedures. For straight line braking performance a combination-vehicle deceleration criterion of 0.4 g is recommended. A tow car weight requirement based on trailer brake capability is presented to insure the combination vehicle will meet the deceleration requirement. For trailer swing stability the safety-related performance measure is damping ratio, or the equivalent cycles to one-half amplitude. A minimum damping ratio of 0.15 (or 3/4 cycles to 1/2 amplitude) is recommended. Test procedures and analysis curves are presented which can be used to determine damping ratios for each trailer. In this case a <u>minimum</u> hitch load criterion based on tow car weight is suggested to insure performance compliance. For tow car stability a tentative performance criterion of maintaining a positive tow car understeer gradient up to and including 0.3 g cornering is recommended. This requirement sets a limit on the <u>maximum</u> allowable hitch load for a given weight tow car, and is a function of load leveling torque applied by a weight distributing hitch. Due to the compatibility of these latter tow handling requirements, they have been integrated into a hitch load versus tow car weight graph unique to each trailer. Use of this graph by a user or manufacturer will define the proper hitch load range for a given tow car size and/or help specify the minimum weight tow car for a given trailer.

17. Key Words
trailer, trailer towing, vehicle dynamics, test procedures, handling, braking, combination vehicle, standards

18. Distribution Statement
Document is available to the public through the National Technical Information Service, Springfield, Virginia 22151

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified		

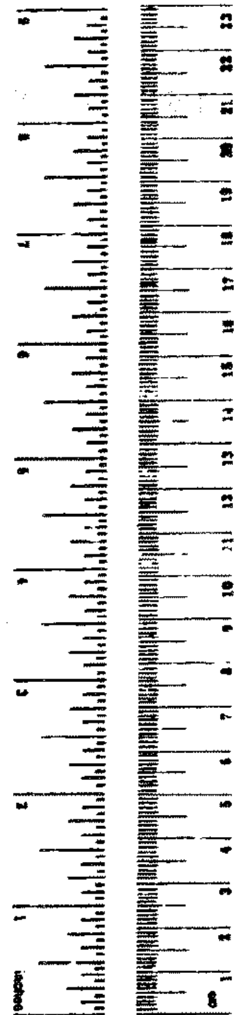
Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in ≡ 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

FOREWORD

This document comprises Volume II of a four volume technical report aimed at developing car/trailer handling and standards. A condensed executive summary of the program and key results is given in Volume I. This volume contains the main technical discussion, and summary test results of Phase I, for rear wheel drive tow cars. Volume III contains appendices providing raw data and other supportive material for the Phase I tests. Results of Phase II testing, using two front wheel drive cars, are presented in Volume IV. This latter phase represents a validation and revision of the requirements proposed in this volume.

The research program was accomplished by Systems Technology, Inc., Hawthorne, California, for the Office of Passenger Vehicle Research of the National Highway Traffic Safety Administration, under Contract DOT-HS-7-01720. The Contract Technical Manager was Dr. J. Kanianthra, and the STI Project Engineer was Mr. R. Klein. The STI Technical Director was Mr. I. Ashkenas.

Significant contributions made by STI staff members include Mr. H. Szostak for test direction and data analysis, Mr. L. Ingersoll for vehicle instrumentation and maintenance, Mr. T. Walsh for test driving, and Mr. G. Teper for development of automated data reduction techniques

Special acknowledgment is given for the fine cooperation and assistance extended to this program by the following organizations and individuals:

- Mr. J. Abromavage, U-Haul International; Chairman of Trailer Hitch Subcommittee of the SAE On-Highway Recreational Vehicle Committee
- Mr. R. Madison, Consultant for Recreational Vehicle Industry Association
- Mr. R. Wilkinson, Fleetwood Enterprises
- Mr. J. Shumway, Prowler Industries
- Mr. R. Franke, Coachman, Industries
- Mr. J. Carr, Shasta Trailers
- Messrs. C. Keck and S. Kulp, Holiday Rambler
- Messrs. D. Swanson and L. Huetsch, Atwood Mobil Products
- Mr. R. Chirakos, Dexter Axle Co.
- Mr. L. Caldwell, Eaz-Lift
- Mr. E. Kuma, Cal-Camp Industries

TABLE OF CONTENTS

Preceding page blank

TABLE OF CONTENTS (Continued)

LIST OF FIGURES

LIST OF FIGURES (Continued)

LIST OF FIGURES (Continued)

LIST OF FIGURES (Concluded)

LIST OF TABLES

SECTION I

INTRODUCTION

This report describes the development of braking and handling performance criteria and compliance formats that can be used to develop the foundation for passenger car/trailer safety standards. The need for this work stems from the considerably higher accident rate for vehicles pulling trailers than for passenger cars alone (Ref. 1). Critical parameters in car/trailer combinations (as opposed to passenger cars alone) are frequently determined by the driving public, via "rules of thumb," often with little regard to the few recommended practices available. As a result, accidents involving car/trailer vehicles can be caused by loss of control during straight-ahead and sub-limit (normal) driving, as well as during accident avoidance and limit-of-performance conditions. Accordingly, a need exists for a basic, uniform, performance-related handling criterion to improve product safety. The criterion must be directly relevant to dominant physical parameters and not legislate minor design details which might stifle competition. The related tests and measures must be simple and easily performed so as to not work a hardship on the smaller manufacturer. Finally, the criteria, tests, and measures must take into account the fact that the trailer manufacturer has no direct control over what the customer will do with the other two companion elements — the tow vehicle and hitch device — which have a profound interactive influence on the combined vehicle handling and safety.

In this regard, this program is a direct extension to prior work addressing the underlying problem of devising handling and braking tests and key performance parameters for automobile/trailer combinations. This prior work was accomplished by STI to develop "Handling Test Procedures for Passenger Cars Pulling Trailers" and to determine the "Effects of Weight Distributing Hitch Torque on Car-Trailer Directional Control and Braking" (Refs. 2 and 3, respectively) and by the University of Michigan, Highway Safety Research Institute (HSRI) to evaluate "Trailer Brake Performance" (Ref. 4). As an extension, this program takes full account of the analytical methods, test procedures, performance measures, and test apparatus used in accomplishing the preceding research. The major thrust of this effort, however, was to define the performance criteria, recommend compliance test procedures, and produce the foundation for a trailer handling/braking safety standard.

The approach taken in this program to accomplish the above task was threefold. First, preliminary analysis was performed to suggest the rule format and trends to be expected. This was documented in Ref. 5. Second, a full-scale test program was performed in which over 90 different hookup configurations were tested using eight trailers and three tow cars, as described in the next section. Primarily, only the four key test maneuvers recommended in Ref. 2 were used. These included straight line braking, step steer, pulse steer, and brake in turn, which are also described in Section II. The final data analysis effort used the results of the test phase to provide a reference for the analytical trends and an "in use" basis for performance criteria selection and location of final boundary lines.

Subsequent sections of this report are organized in parallel, according to the four key test procedures. That is, each section represents one test maneuver and stands alone in its treatment of analytical foundations, full-scale test results, development of tentative standards format, selection of performance criteria, and, finally, recommendations for a rule format and compliance test procedure.

Straight line braking performance is presented first. Modifications to the proposed rule format presented by HSRI in Ref. 4 are suggested and compared to the full-scale test data. Results show that a combination-vehicle deceleration criterion of 0.4 g can be met by all tested configurations. Based on this criterion a minimum tow car weight requirement as a function of trailer braking capability is derived and a <u>trailer-alone</u> brake test requirement similar to that proposed by the Canadian Standards Association (Ref. 6) is recommended.

Section IV describes trailer swing damping as evaluated by a "pulse steer" test procedure. This mode of trailer dynamic behavior is critical because large trailer oscillations can lead to tow car jackknife, trailer separation, and/or combination-vehicle (CV) rollover. For example, in a 1970 survey conducted by UCLA (Ref. 7) 39 percent of their trailer towing respondents (1331 people) had developed "snaking" problems with their trailers. Accident data from California, New Mexico, and Ontario, Canada (also tabulated in Ref. 7) showed that nearly all trailer towing accidents had developed snaking or weaving problems prior to the accident. These statistics point to the importance of trailer stability in car/trailer safety and lead to such questions as: how can trailer designs be improved? how can the proper range of trailer hitch load be more quantitatively determined? and how can trailers be tested to insure that stability is sufficient for a safe combination-vehicle configuration at highway speeds? Results of this program show that trailer damping can be improved by designs that stress low inertia, long wheelbases, heavy-duty tires, and large tow car to trailer weight ratios. By selecting a minimum trailer stability criterion (e.g., 0.15 damping ratio or 3/4 cycles to 1/2 amplitude was recommended in this program), it was possible to derive and validate a <u>minimum</u> hitch load boundary as a function of the tow car to trailer weight ratio.

Section V describes the changes in tow car steady turn stability (in terms of understeer gradient) due to increased hitch loads and lateral acceleration. This problem is much more difficult to identify from an accident data basis or from any standards for passenger cars alone (since there are none). However, it has been well documented that the above factors do cause large adverse changes in the basic tow car's static stability. In this program a performance criterion of "maintaining a positive tow car understeer gradient during cornering up to 0.3 g" was selected, and full-scale test results were used to adjust the analytically derived boundaries. This procedure produced <u>maximum</u> hitch load boundaries consistent with current trailering practice and those allowable by tow car manufacturers.

Combining the trailer damping and tow car stability criteria results in an integrated <u>allowable hitch load range</u> unique to each trailer and tow car weight, and thus would be useful to a trailer owner by helping select the proper hitch load for a given tow car size or to a trailer manufacturer by helping specify the minimum weight tow car for a given trailer model.

Section VI discusses results pertaining to the combined cornering and braking test procedure. Since all combination-vehicle configurations tested were able to meet (or exceed) the straight line deceleration performance criterion of 0.4 g while simultaneously not losing control during the steady cornering performance test at 0.3 g, no changes were recommended for the preceding integrated handling and braking formats.

The final section summarizes the recommendations presented in each of the individual sections.

SECTION II

TEST PROGRAM

In order to validate the proposed rule formats described in Ref. 4 and suggested in Ref. 5, a 6 month test program was undertaken. This section presents details of the test vehicles, combination-vehicle configurations, and test procedures germane to the full-scale test portion presented in this volume. Additional vehicle specifications, instrumentation details, complete run logs, and raw data are presented in Volume III.

A. TRAILER SELECTION

Characteristics considered in selecting the test trailers were type, weight, class, number and position of axles, brake type, and suspension design. The major overlap occurs between type and weight, since the other features generally correlate with weight. Although there are literally hundreds of trailer types, they can be divided nicely by their representation on the highway (Ref. 1) into the following four classes:

- Travel trailers
- Boat trailers
- Camper (or utility) trailers
- Horse trailers

Each can then fall into one of the four weight classes defined by the SAE Standard for trailer couplings and hitches (Ref. 8):

- Class I: $GVW \leq 2000$ lb
- Class II: $2000 < GVW \leq 3500$ lb
- Class III: $3500 < GVW \leq 5000$ lb
- Class IV: $5000 < GVW < 10,000$ lb

where Class I trailers are light duty, can be towed with small compact cars, and generally have hitch loads less than 200 lb. Class II are medium duty, such as travel trailers up to 18 ft or small power-boat trailers, and can be towed by a mid-sized car. Class III is generally the highest class towed by passenger cars, since the hitch loads can be as high as 750 lb. Class IV trailers such as large tandem axle travel trailers up to 31 ft might be towed by a large passenger car, although their hitch load may exceed those allowable (or recommended) by the automobile manufacturers.

In general, trailers over 4000 lb GAWR have tandem axles: boat trailers or utility trailers over 1500 lb have surge brakes; and Class II or above travel trailers have electric brakes. Currently, there are no regulations or formal, i.e., SAE, recommendations for load distributing hitches as a function of trailer weight; although they are generally used when hitch loads exceed 300 lb.

Combining all these factors led to the selection of the eight trailers described in Table 1. This includes four travel trailers (two Class III single axle, one Class III tandem axle, and one Class IV); one Class II single-axle boat trailer with surge brakes; two Class I camper and utility trailers with single axles and no brakes; and one Class IV horse trailer (since its short wheelbase, wide track, and brakes on only one axle make it an unusual configuration).

B. TOW VEHICLE SELECTION

Based on contract requirements, three tow cars, representing an intermediate, compact, and subcompact, were to be selected. To provide a range of design differences each was to be represented by a different major automobile manufacturer, i.e., GM, Ford, and Chrysler. All were to be 1976 model year or newer, so that they would comply with the passenger car braking standard FMVSS 105-75 (Ref. 9).

The final selection of tow vehicles is described in Table 2. These include a 1976 Chevrolet Monte Carlo, 1976 Plymouth Volare, and 1978 Ford Mustang II. The test weights used throughout the remainder of the report are [illegible], [illegible], and 3400 lb, respectively, and represent that given in Column 3 of Table 2, plus the additional 80 lb for the third class hitch head, load leveling bars, and instrumentation.

TABLE 1. TEST TRAILERS

TYPE	MANUFACTURER/ MODEL	OVERALL LENGTH	TEST WEIGHT / GVWR	AXLES	BRAKES	TIRES (PRESSURE)
Utility	U-Haul/AV	11 ft 6 in.	1500/2600	1	None	6:70 x 15 LT LRC (45 psi)
Camper	Starcraft Starmaster 6	15 ft 3 in.	1600/2090	1	None (Surge optional)	5:30 x 12 LRC (80 psi)
Small Travel	Shasta (1978)	17 ft 6 in.	3000, 3880/3888	1	Two 10 in. electric (Dexter)	8:55 x 15 LRC ST (50 psi)
	Traveleze (1967)	18 ft 7 in.	4000/ Unknown	1	Two 10 in. electric (Kelsey Hayes)	7:00 x 15 LT LRC (45 psi)
Medium Travel	Prowler/L	21 ft 8 in.	4000/5000	2	Four 10 in. electric (Dexter)	7:75 x 15 ST LRB (35 psi)
Large Travel	Holiday Rambler 127	28 ft 2 in.	6000/6800	2	Four 10 in. electric (Dexter)	7:00 x 15 LT LRC (45 psi)
Boat	American with 16 ft boat	19 ft 10 in.	3000/3100	1	Two 10 in. surge (Atwood)	H78-14 LRB (32 psi)
Horse	Stidham/B	10 ft 10 in.	4000, 5800/5960	2	Two 10 in. electric (Interstate) rear axle only	7:75-15 LRB (32 psi)

TABLE 2. TEST TOW CARS

VEHICLE TYPE (GVWR)	SIZE	CURB WEIGHT / TEST WEIGHT[a] (lb)	WHEELBASE (in.)	TIRES
1976 Chevrolet Monte Carlo (5513)	Intermediate	4140/4660	116	GR70-15 TPC 1007
1976 Plymouth Volare (4775)	Compact	3400/4030	112.5	D78-14
1978 Ford Mustang II (3861)	Subcompact	2715/3330	96	BR-78-13X

[a]Includes driver (180 lb), instrumentation (275 lb), hitch receptacle (95 lb) and one-half fuel.

Note: hitch head, hitch angle sensor, and load leveling bars add 80 lb to "test car alone weight" and are not included as part of trailer hitch load.

C. TEST PROCEDURES AND CONDITIONS

Four basic test maneuvers were used. These included straight line braking tests, handling tests (step steer and pulse steer), and a combined handling and braking test (brake in turn). Each is discussed below.

1. Straight Line Braking Test

Braking tests were tailored after FMVSS 105-75 (Ref. 9), HSRI (Ref. 4) and SAE Recommended Practice J134 (Ref. 10) for the tow car alone, trailer alone, and combination-vehicle, respectively. They were shortened, however, to include only the preburnish effectiveness, burnish, and second effectiveness tests. The sequence is given in Table 3, although this was not always followed exactly for all vehicles. Appendix D of Vol. III gives the exact sequences and raw data results for each vehicle.

Several points are worth noting in Table 3. First, a test speed of [illegible] mph was selected. This was done to tie in with the previous HSRI and

TABLE 3. GENERAL STRAIGHT LINE BRAKE TEST PROCEDURE

TEST	VEHICLE	SPEED (mph)	NUMBER OF STOPS	DECELERATION (ft/sec^2)	EXISTING REQUIREMENT	REMARKS
Preburnish Effectiveness	Tow car alone	40	12	Max	91 ft	6 below lockup, 6 above lockup. Brake temperature 150-200 °F. Also obtain deceleration at lower pedal pressures.
	(Intermediate only)	60	2	Max	216 ft	
Burnish	Tow car alone	40	200	12	—	1 mile between stops.
	Trailer alone	$40 \rightarrow 40\sqrt{1-\frac{W_t\text{-}HL}{W_{cv}}}$	200	$12\,\frac{W_t}{W_{cv}}$	—	1 mile between stops.
Second Effectiveness	Tow car alone	40	6	Max	91 ft	3 below lockup, 3 above lockup. Also at decreasing pedal pressures.
		60	2	Max	204 ft	Intermediate only.
	Trailer alone	$40 \rightarrow 40\sqrt{1-\frac{W_t\text{-}HL}{W_{cv}}}$	10	To be determined	—	Increasing brake voltage from 1 to 12 V
	Combination-vehicle	40	6 Minimum	Max	$15 < F_b < 120$ SAE J135	Trailer lockup allowed 3 below lockup } tow car 3 above lockup }

Note: $W_t - HL$ = Trailer weight without hitch load (static axle weight)
W_{cv} = Combination-vehicle total weight

STI work that used 40 mph as the test speed. Second, the trailer-alone and combination-vehicle (CV) procedure allowed trailer lockup. This was consistent with SAE J134 and that recommended by HSRI. Third, the CV tests were aimed at maximum performance. This was defined as "incipient" tow car wheel lockup; hence the fixed pressure brake actuator mechanism was set to provide at least three stops just below lockup and at least three stops at partial lockup.

2. Handling Tests

The step and pulse steer test procedures specified in Ref. 2 were used. These are described below.

a. Step Steer Test

A constant amplitude step steer was input and held for a minimum of 90 deg path change at constant speed. The steer angle input was adjusted to provide a 0.3 g turn at 30 mph for the combination-vehicle. This was usually between 60 and 90 deg steering wheel. The test was rerun at speeds between 10 and 50 mph in 5 mph increments to derive understeer gradient. When a combination-vehicle had a jackknife potential (i.e., high hitch load), additional 2.5 mph test speed increments were used in order to obtain data points in the transition range to jackknife. Both left- and right-hand turns were performed; however, due to data variability the right-hand turns were discontinued later in the test program.

An alternative test procedure tailored after SAE XJ266 (Ref. 11), "Passenger Car Steady State Directional Control Response Test Procedure," was also used. This constant radius test procedure required driving the vehicle around a 200 ft radius circle at increasing speed. Data were taken with the steering wheel position and throttle position fixed at a steady-state condition. The vehicle was then accelerated to the next speed at which data were taken. In general, this corresponded to 0.05 g lateral acceleration increments. Steer angle was plotted versus lateral acceleration to determine understeer gradient.

b. Pulse Steer Test

The vehicle was driven in a straight line at 55 mph and a fixed amplitude rapid pulse steering wheel was applied to excite the tow vehicle and trailer dynamic modes. Four replications were performed to provide a measure of the variance in damping ratio.

3. Combined Handling and Braking Test (Brake in Turn)

Constant brake level stops were initiated from 40 mph during a steady-state turn on a 355 ft radius circle. This provided 0.3 g lateral acceleration. Brake pedal pressure levels were increased on succeeding runs up to lockup of one tire on one axle of the tow car. In all cases the steering was held fixed during the deceleration interval. The test was also performed with full and with partial trailer brakes.

4. Additional Tests

Several other peripheral tests were performed to check analysis or data consistency. These included a trailer-alone damping test (external input applied at axle); tests to determine effects of speed, inertia, and lateral acceleration on trailer damping; and coast-down versions of step steer and constant radius circle tests to determine power effects. These are described in detail in Appendix B (Vol. III).

5. Test Conditions

All tow vehicles and trailers were new or put in "as new" condition with OEM brakes, tires, and adjustable air shocks. Each tow vehicle was also equipped with a Class III frame-mounted hitch and Kelsey-Hayes electric brake controller.

Tire inflation pressure of all vehicle tires was maintained at the manufacturers' recommended cold inflation pressure for the test loading condition. The vehicle was then driven at 40 mph for 15 mi to establish the "hot" inflation pressure. This inflation pressure was then maintained for all tests conducted under the given loading condition. In addition to maintaining inflation pressures, all new tires were "broken in" prior to effectiveness testing. For the tow vehicles, the burnish procedure was adequate for this purpose. For the trailers, several turns (both left and right) around the 200 ft radius circle were performed.

Trailer electric brakes were set up using an external resistor mounted in the tow car. Since no quantitative procedure was provided by the manufacturer for selecting the resistor value it was set up such as to provide a minimum of 10 volts at the trailer brakes with full controller application. Measured trailer brake voltages for each test are recorded in Appendix D of Volume III. In the tests of decreased trailer brake effectiveness the resistor value was increased accordingly.

D. TEST CONFIGURATION SUMMARY

In addition to the 24 potential combination-vehicles (3 cars × 8 trailers), several other variables were considered in order to develop meaningful handling and braking performance standards. These were the trailer weight, hitch load, load leveling torque, air shocks, and trailer brake authority. In general, each combination-vehicle was tested through a range of hitch loads. At the heavier hitch loads a minimum of two values of load leveling were then used. These corresponded to the current recommended practice of "+25 percent" (i.e., 25 percent of the hitch load is transferred to the tow vehicle front axle) and that recommended by STI in Ref. 3 of "only that necessary to relevel the combination vehicle after air shocks have been used to their fullest."

The resulting matrix of test configurations is given in Table 4. This table shows 92 different configurations tested in seven different maneuvers. The matrix is not full factorial, however, since the trailer damping test (pulse steer), for example, is only relevant with light hitch loads, and tow car stability and braking tests (step steer, straight line brake, and brake in turn) are only significant with heavy hitch loads. This selection process resulted in 250 total configuration/maneuvers requiring a total of over 2000 actual test runs. The actual run log summary is contained in Appendix C of Vol. III.

TABLE 4. FULL-SCALE TEST SUMMARY

TRAILER	TEST CONFIGURATION				TEST PROCEDURES																				
	WEIGHT	HITCH LOAD	LOAD LEVELING	AIR SHOCKS	SST			CRT			PS			SLB			BIT			BTA			OTHER		
					I	C	S	I	C	S	I	C	S	I	C	S	I	C	S	I	C	S	I	C	S
Utility	1500	0	N →	N →							x	x	x				x								
		2.5		Y									x												
		5		N Y Y							x	x	x				x								x
		7.5		Y									x												
		10		N Y Y	x		x	x		x	x	x	x	x	x	x	x		x						
		20	↓	Y	x			x			x			x			x						x		
Camper	1600	0	N →	N →							x	x	x												
		5		Y →	x						x	x	x	x			x								
		10			x			x			x	x	x	x		x	x		x						
		15			x			x			x			x			x								
		20	↓	↓	x			x			x			x			x								
18 ft Travel	3000	13	25	N	x			x			x			x			x			x					
			N	Y	x			x			x			x			x								
	3420	5	N	Y			x			x			x												
		7.5	N	Y			x						x												
	3880	5	N 20 31	N →							x	x	x										x		
		7.5	21										x												
		10	25 35 34		x			x			x	x	x	x		x	x								
		15	23 23	↓		x						x	x		x			x			x			x	
		20	-25	Y	x			x			x			x			x								
		20	25	N	x			x			x			x			x								
19 ft Travel	4000	5	N 0 N	Y N Y							x	x	x												
		10	N 10 0	Y N Y							x	x	x												
		15	25	N	x			x			x			x			x			x					
		15	-32	Y	x									x			x								
22 ft Travel	4000	0	N	N							x														
		5	N →	Y →		x			x		x	x	x												
		7.5	N →	Y →		x			x			x	x												
		10	N	Y	x			x			x			x			x								
		10	0 -29 0	N Y Y	x	x		x	x		x	x	x	x		x	x			x					
		10	25	N	x			x			x			x			x								
		12.5	-6	Y		x			x			x													
		15	-7	Y		x			x			x			x										
		15	17	N		x			x			x			x			x			x			x	
		17.5	4	Y		x			x			x													
		20	25	N	x			x			x			x			x								
27 ft Travel	6000	5	N 13	Y N							x	x											x		
		10	-15	Y	x																				
		10	25 15	N N	x	x		x			x	x		x			x			x					
		.5	-20	Y	x																				
		[illegible]	0	Y				x						x			x								
		[illegible]	25	N	x			x			x			x			x								
Boat	2275	26	28	N		x			x															x	
	3000	[illegible]	N →	N →			x			x	x	x	x												
		[illegible]	N	Y									x												
		5	N →	Y →			x			x	x	x	x												
		5	0	N			x			x			x												
		7.5	0	N									x												
		10	I N 32	Y Y N	x		x	x		x	x	x	x	x		x	x		x	x					
		15	N -14	Y Y	x						x			x	x										
		[illegible]	25 35 21	N N N	x		x	x		x	x	x	x	x	x										
Horse	5800	10	LL not	Y →	x			x			x									x					
		15	possible		x						x														
	[illegible]000	5	with									x	x												
		10	this			x						x	x		x			x			x				
	[illegible]000	10	trailer	↓							x														

Notes:

Hitch load – percent of trailer weight
Load leveling – percent of hitch load transferred to front axle
Air Shocks – Yes if used to level CV; No if not used
SST – step steer test (steering set for 0.3 g at 30 mph)
CRT – constant radius turn (400 ft diameter circle at 0.3 g at 30 mph)
PS – pulse steer (steer pulse at 55 mph)
SLB – straight line brake (maximum deceleration from 40 mph)
BIT – brake in turn (maximum deceleration from 40 mph at 0.3 g cornering)
BTA – trailer-alone brake capability test
Other – miscellaneous tests (calibrations, damping, inertia, braking, etc.)
I – intermediate; C – compact; S – subcompact

SECTION III

STRAIGHT LINE BRAKE PERFORMANCE

A significant problem in trailering safety is increased stopping distance due to increased total vehicle mass without proportional increase in braking effectiveness. At the present time there are no federal braking performance standards for recreational or utility trailers designed for towing by passenger cars or light trucks. There are, however, braking standards for the tow vehicles, i.e., FMVSS 105-75 (Ref. 9). In addition, even if there were a trailer brake standard, the many variables present in tow car/trailer hookups have sufficient influence such as to alter the expected "combination-vehicle" stopping distances. In other words, the total may not necessarily be equal to the sum of the parts. With this problem in mind, the basic objective of this portion of the program was to:

> "...conduct sufficient tests to identify appropriate braking requirement levels for combination-vehicles and trailers alone to form the basis of a federal standard."

The approach taken (in this program) to accomplish this objective was to investigate rational combination-vehicle stopping distance requirements based on trailer-alone deceleration capabilities. This is similar to that accomplished by HSRI in Ref. 4. Many combination-vehicles and hookup variables were then tested to develop practical limits for the requirements.

The following subsections present analytical models that can be used to estimate the influence of trailer and tow vehicle factors on CV braking performance. The full-scale results are then presented and compared to the proposed rule format of Ref. [illegible] and our analysis results. To aid in selecting braking criteria, existing recommended practices and test procedures are discussed. The end products are recommendations for meaningful trailer brake performance standards, test procedures, and hookup practices.

A. ANALYTICAL CONSIDERATIONS

The analysis presented in Ref. 4 assumed ideal conditions of tow car and trailer such that the combination-vehicle deceleration could be derived from the sum of the braking forces divided by the sum of the masses, i.e.,

$$a_{x_{cv}} = \frac{a_{x_c} W_c + a_{x_{ta}} W_t}{W_c + W_t} \tag{1}$$

where

$a_{x_{cv}}$	Combination-vehicle deceleration in g units
a_{x_c}	Tow car deceleration capability in g units
$a_{x_{ta}}$	Trailer-alone deceleration capability in g units
W_c	Tow car weight (lb)
W_t	Trailer weight including hitch load (lb)

In essence, the penalty for trailer braking performance poorer than the combination-vehicle is shifted to the tow car in terms of a weight differential. For example, if we assume a minimum tow car brake capability of 0.6 g (i.e., that compatible with FMVSS 105-75) and furthermore arbitrarily select a combination-vehicle deceleration criterion of 0.5 g, then the tow car to trailer weight requirement, as a function of trailer brake capability, would be as shown by the upper boundary line in Fig. 1. Weight ratios above the boundary would ideally provide deceleration exceeding 0.5 g, whereas weight ratios below the boundary would not meet the criterion. Note that unbraked trailers would require a tow car weight about five times the trailer weight if the tow car can only provide the minimum deceleration specified by FMVSS 105-75. If the tow car had greater deceleration capability, i.e., 0.67 g as assumed in Ref. 4, then the tow car weight boundary in Fig. 1 is reduced.

Due to the general nature of the format shown in Fig. 1, this approach is highly desirable as a standard. However, since it is generalized, it does not account for load transfer, tow car brake proportioning, load equalization, etc., that will vary from one hookup to another. To investigate these factors, the complete braking equations presented in Appendix D

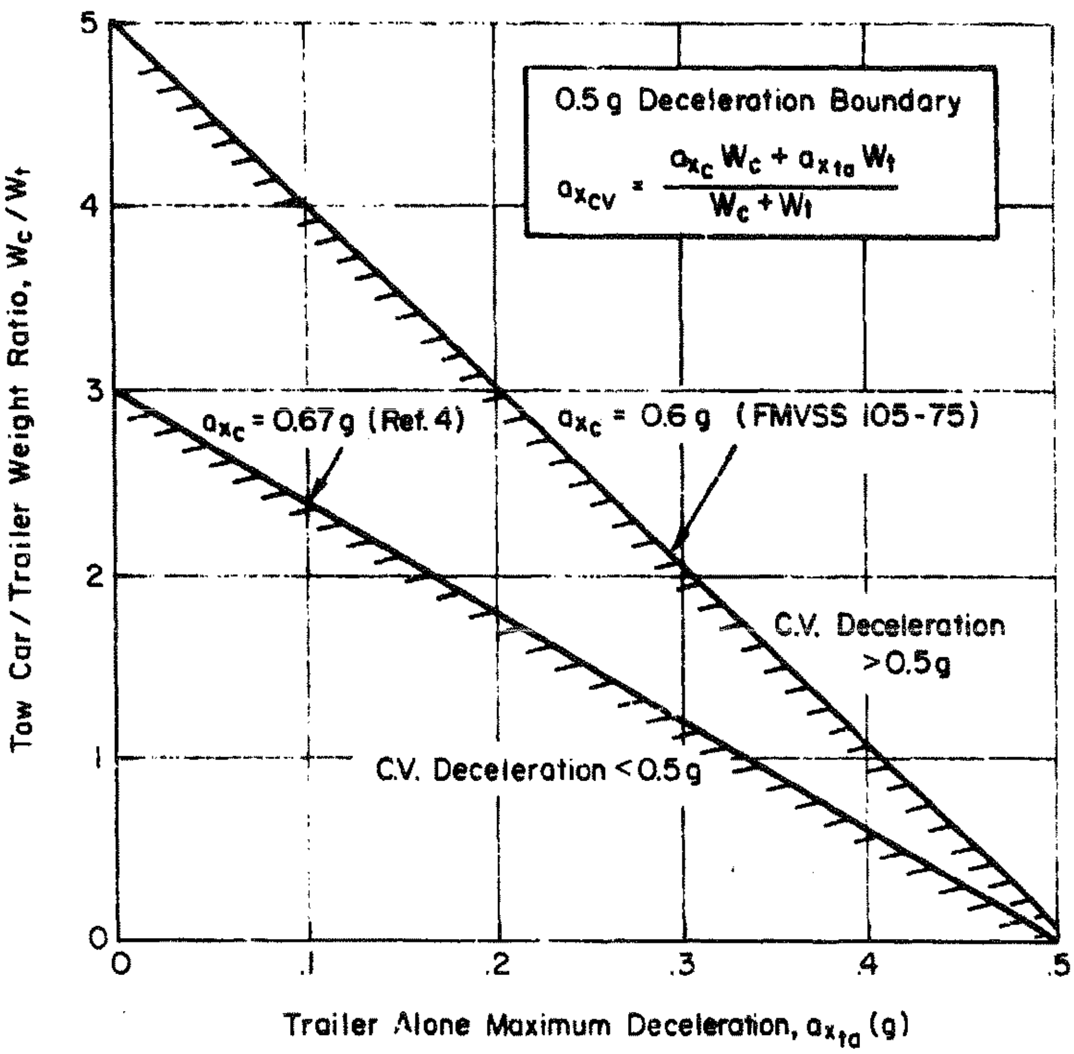


Figure 1. Braking Standard Format
Suggested in Ref. 4

of Vol. III were applied to a variety of actual trailer configurations. The results are shown in Fig. 2 for a tow car capable of decelerating at 0.6 g. This required a minimum friction coefficient, μ, of 0.625 for a tow car brake proportioning of 60 percent front, 40 percent rear. When compared to the simplified model of Ref. 4 (corrected for static hitch load transfer) it can be seen that both models are similar when the tow car is much heavier than the trailer, i.e., W_c/W_t is high. Obviously, at this weight ratio there is only a small influence of the trailer on the tow car. However, at the smaller weight ratios (allowable with active trailer brakes), the complete model requires significantly more trailer braking capability than the simplified version would predict. Viewed another way, the complete formulation shows a heavier tow car would be required to stop the CV at the specified deceleration rate. The main reason accounting for the different solutions is the large dynamic hitch load which reduces the tow car front tire brake forces.

With load leveling, the curves are shifted down (to lower tow car weight requirements) since the front axle maintains a larger vertical load. Tow car front wheel lockup is still the limiting condition.

One of the most significant factors in the proposed rule format of Fig. 2 is the selection of the combination-vehicle deceleration criterion. This is especially pertinent to unbraked trailers where any CV deceleration requirement will result in a maximum allowable trailer weight for each tow car GVWR. To show the sensitivity of tow car weight to CV deceleration requirements, the equations of Appendix D were simplified to that shown in Fig. 3. This assumes optimum tow car braking proportioning such as to meet a minimum 0.6 g deceleration, a 10 percent hitch load, and the dimensions given in Fig. 2. Two trailer weights (1500 and 3000 lb) are shown. From this figure it is clear that current unbraked trailers as heavy as 3000 lb being towed by nominal sized tow cars (i.e., 4000-5000 lb) will not be able to meet a 0.5 g CV deceleration criterion. In fact, even a 1500 lb trailer would require a tow car exceeding 5000 lb GVWR in order to meet a 0.5 g CV criterion.

The next step is to compare these analytical predictions with full-scale test results. This is accomplished in the next two subsections for

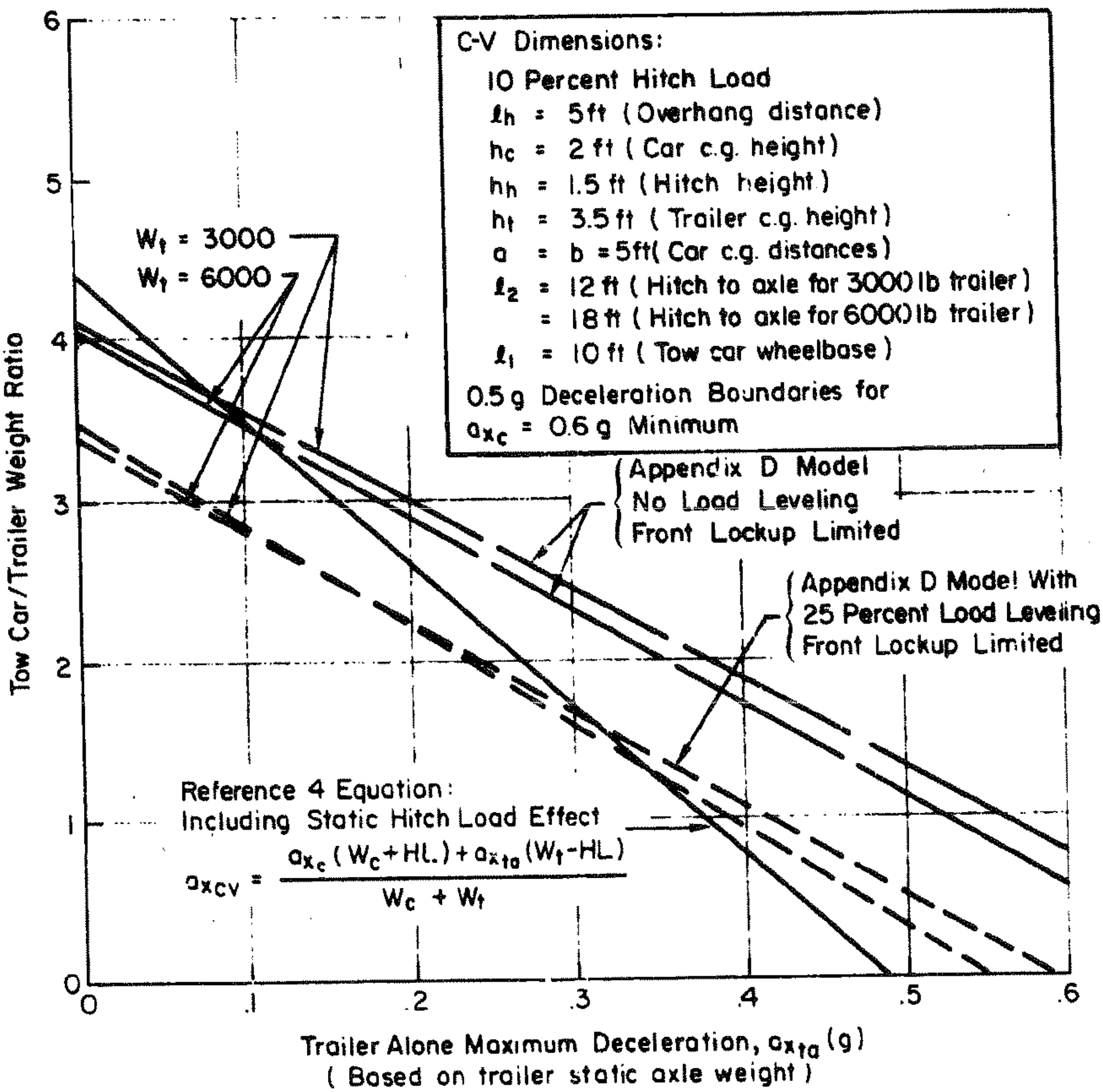


Figure 2. Comparison of Static and Dynamic Brake Performance Models at 0.5 g CV Deceleration

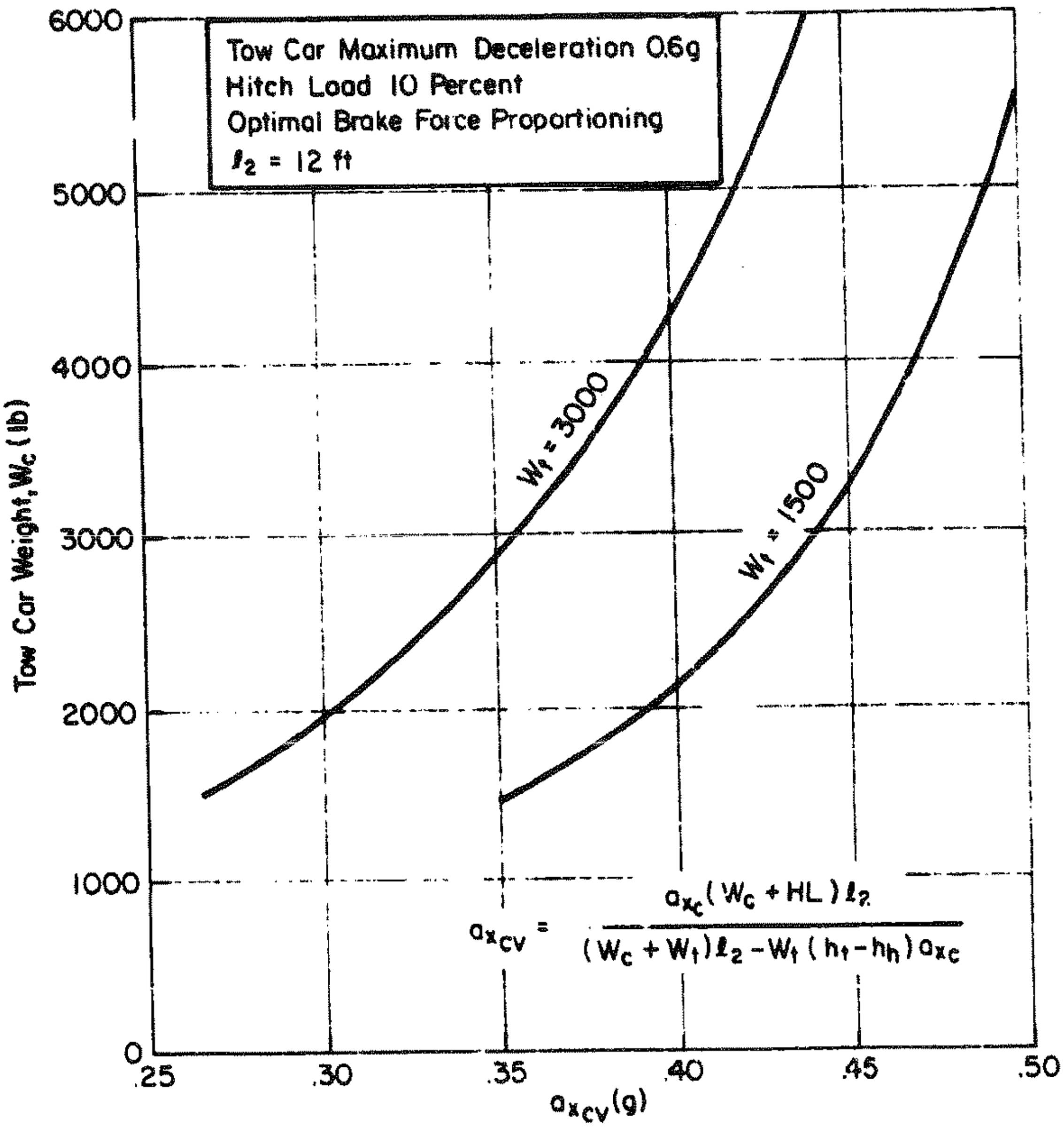


Figure 5. Deceleration Capability for 1500 and 3000 lb Nonbraked Trailers

three tow cars, eight trailers, and 22 combination-vehicles at various hitch loads and load leveling.

B. FULL-SCALE TEST RESULTS

Straight line brake tests from 40 mph were performed with tow vehicles alone, with combination-vehicles, and with trailers alone. As described in Section II, the test procedures followed the format of FMVSS 105-75 (Ref. 9), HSRI (Ref. 4), and SAE J134 (Ref. 10) for the tow car, trailer, and combination-vehicle, respectively. The following represents results for the second effectiveness tests.

1. Tow Car Alone

First, in Figs. 4-6 are the tow vehicle alone stopping distances (and decelerations) as a function of pedal force for each of the three tow cars. All three vehicles exceeded the requirements of FMVSS 105-75. In fact, the measured deceleration levels at the test weight were 0.7 to 0.75 g, as opposed to the 0.6 g required by FMVSS 105-75. Minimum-lockup pedal forces were about 50 lb for vehicles with power boost and about 100 lb for the "compact" without power boost. The "intermediate" exhibited rear lockup when lightly loaded and front lockup when heavily loaded. Since trailer hitch loads will increase the front lockup tendency, longest CV stopping distances will occur at the heavy weight condition.

2. Trailer Alone

The next result pertains to the trailer-alone braking capability. The measured deceleration levels are given in Table 5 for the actual axle weight. The minimum deceleration refers to what would have been measured if the full gross axle weight rating (GAWR) had been used. It is clear that the brake axle manufacturers are designing for 0.4 to 0.45 g. Only the Atwood system had higher capability. Discussions with Dexter Axle Co. (Ref. 12) confirmed that their design axle weight rating is based on achieving 0.43 g deceleration. This value is also recommended by the Canadian Standards Association (Ref. 6).

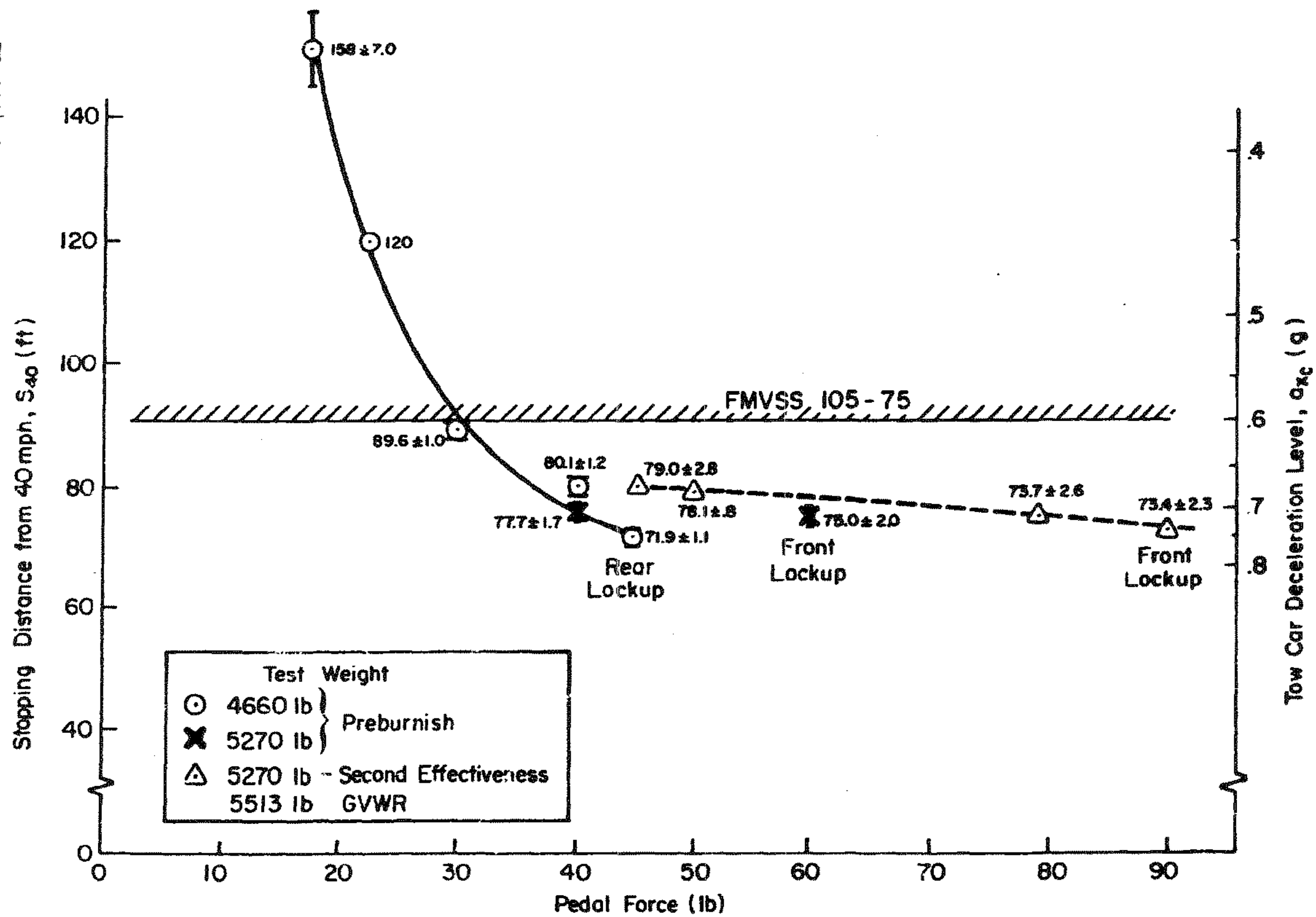


Figure 4. Straight Line Brake Stopping Distances from 40 mph for Intermediate

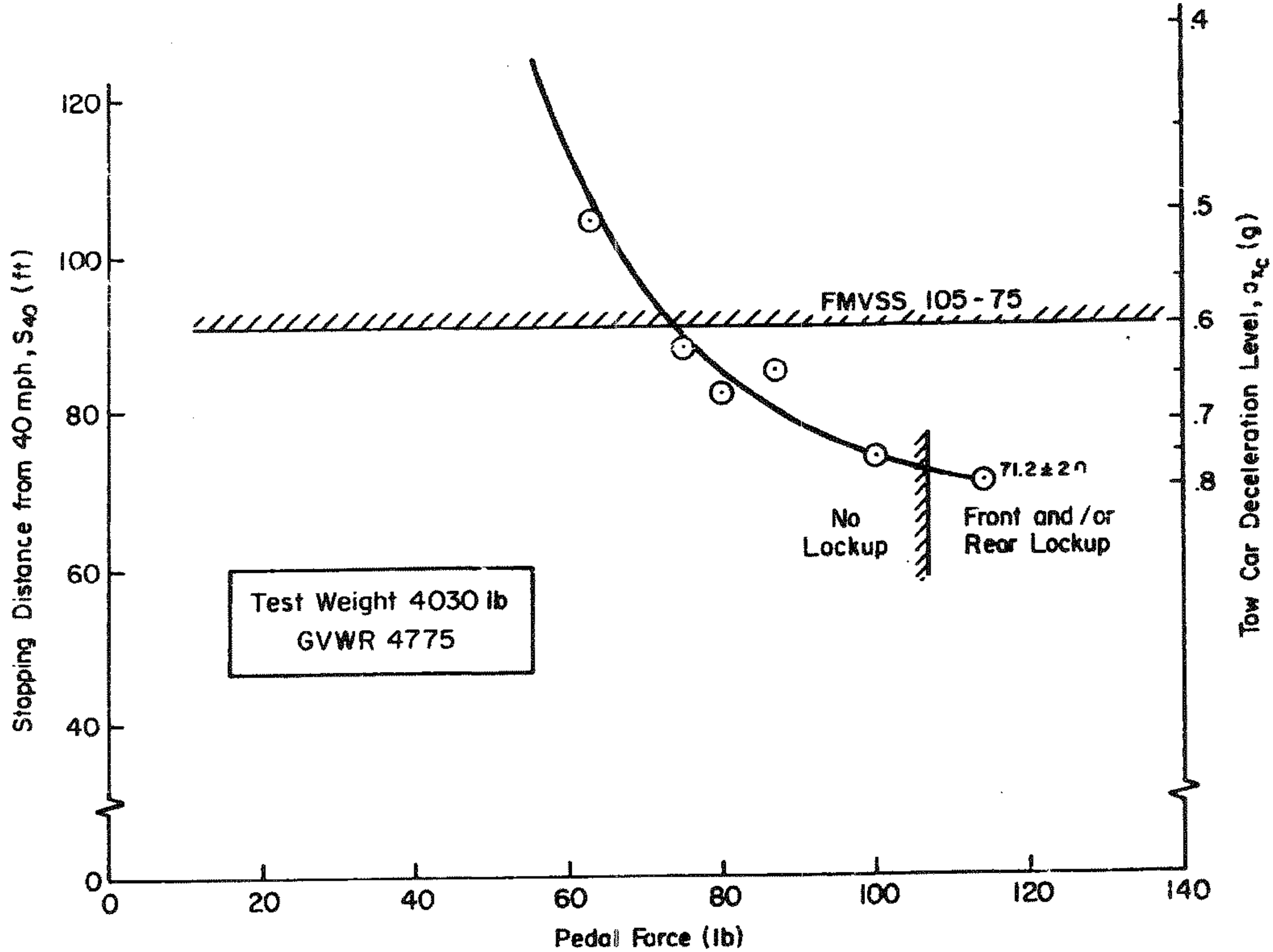


Figure 5. Straight Line Brake Stopping Distances from 40 mph for Compact

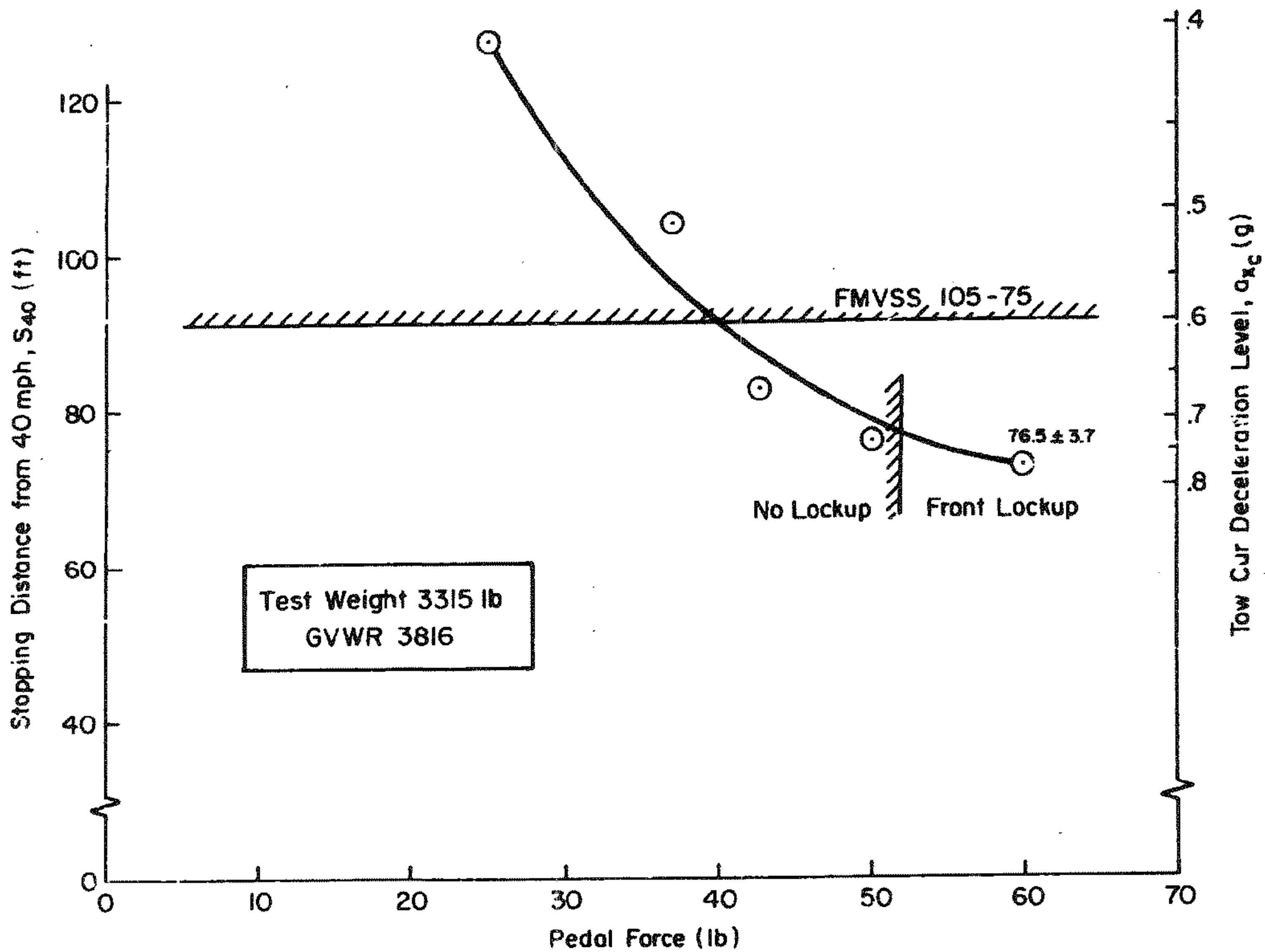


Figure 6. Straight Line Brake Stopping Distances from 40 mph for Subcompact

TABLE 5. TRAILER-ALONE DECELERATION LEVELS

TRAILER	TEST AXLE WEIGHT (lb)	MEASURED a_{xta} (g)	GAWR (lb)	MINIMUM a_{xta} (g)	BRAKE DESCRIPTION
Utility	1350	0	2600	0	None
Camper	1440	0	2090	0	Bendix (deactivated)
18 ft Travel	3490	0.44	3500	0.44	Dexter, two 10 in.
19 ft Travel	3480	0.47	3500	0.47	Kelsey-Hayes, two 10 in.
22 ft Travel	3610	0.67	5020	0.48	Dexter, four 10 in.
27 ft Travel	5660	0.43	6800	0.36	Dexter, four 10 in.
Horse	5535	0.42	5960	0.39	Interstate, two 10 in.
Boat	2645	0.70	3100	0.60	Atwood, two 10 in.

3. Combination Vehicle

Tables 6-8 present the combination-vehicle stopping distances as a function of hitch load, load leveling, and in some cases trailer weight. Appendix D (in Vol. III) contains the raw data. Lockup of one wheel on one side of the tow car defined the maximum braking applied even in trailer lockup occurred first. In general, most CVs were able to provide a 0.5 g deceleration level (i.e., 107 ft from 40 mph). The exceptions are highlighted in Tables 6-8 and discussed below.

a) Intermediate plus horse trailer. When loaded to 5800 lb (97 percent GVWR) or greater, maximum braking could not be applied without hitting the hitch head on the ground. The maximum weight limit may be significantly less. No load leveling is possible, and suspension linkage provides an unstable variation in hitch load as a function of hitch height, i.e., lower ball, higher hitch load. It would have to be recommended that the maximum load of this trailer be limited when being towed by a passenger car.

b) Intermediate plus large travel trailer at 6000 lb. The 10 percent hitch load plus 25 percent load leveling condition and the 15 percent hitch load with minimum leveling condition had CV decelerations of 0.47 g. This is probably due to not being able to lock up the trailer

TABLE 6. COMBINATION-VEHICLE STRAIGHT LINE STOPPING DISTANCE RESULTS FOR INTERMEDIATE SIZE TOW CAR[a]

HITCH LOAD (% W_T)	LOAD LEVELING (% HL TO FRONT AXLE) \ WEIGHT (lb)	TOW CAR ALONE	UTILITY	CAMPER	18 FT TRAVEL		19 FT TRAVEL	22 FT TRAVEL	27 FT TRAVEL	BOAT	HORSE
		[illegible]60[b]	1[illegible]00	1600	3000	4000	4000	4000	6000	3000	[illegible]800
[illegible]	None	71.9±1.1 / 6		100±2.1 / 8							Hitch scrapes
10	None		92.9±1.9 / 6	106±1.5 / 6		101±4.6 / 4		93.6±2.9 / 7		95.8±4.6 / 6	Hitch scrapes
	0							95.6±3.0 / 6			
	25							87.3±3.5 / 6	112±4.1 / 5		Not possible to set up
13-15	None			98.6±5.0 / 10	98.5±3.5 / 7					102±3.7 / 5	
	−30 → 0						(−32%) 91.8±.6 / 6		(0%) 114±4.0 / 6		
	25				86.7±2.5 / 7		94.4±1.9 / 5		105±2.9 / 11	106±2.6 / 6	
20	None		96.7±5.0 / 7	95.5±3.8 / 3							
	−25					98.7±4.5 / 6					
	25					101±4.4 / 8		90.3±2.9 / 7			
PEDAL FORCE (lb)		45	33-35	35-40	28-40	30-37	30-38	37-40	30-40	30-37	
LOCKUP TENDENCY		Both rear	None	None	BF_c, L_t	BF_c	B_t, BF_c	LF_t	BF_c	BF_c	R_t
NO LOCK DISTANCE		80.1±1.2 / [illegible]	[illegible]±1.9 / 6	10[illegible]±1.5 / 6	88.3±1.3 / 3	97.6±.6 / 4	108±2. / 2	92.3±1.8 / 5	104±2.4 / 6	97.7±[illegible].2 / 3	—
CONDITION (HL-LL)		13-25	—	—	13-25	20-25	15-25	10-NO	1[illegible]-25	10-NO	

[a]From 40 mph, mean ± standard deviation; number in lower right corners indicates number of runs in sample.

[b]3700 lb when acting as tow car (includes hitch hardware).

TABLE 7. COMBINATION-VEHICLE STRAIGHT LINE STOPPING DISTANCE RESULTS FOR COMPACT SIZE TOW CAR[a]

HITCH LOAD	LOAD LEVELING \ WEIGHT (lb)	TOW CAR ALONE	UTILITY	CAMPER	18 FT TRAVEL	19 FT TRAVEL	22 FT TRAVEL	27 FT TRAVEL	BOAT	HORSE
		4030 [b]	1500	1600	4000	4000	4000	6000	3000	4000
0	Car alone	71.2±2.0 / 4								
10	None									115±2.1 / 6
15	None		102±3.8 / 6							
	−14 → −7						84.7±2.0 / 6		88.8±5.0 / 6	
	7-1/2								92.4±5.2 / 5	
	17-23				91.8±1.0 / 4		83.3±3.2			
PEDAL FORCE			96		96		101		102	82
LOCKUP TENDENCY			BF_c		LF_c		LF_c, LF_t		BF_c, RF_t	—
NO LOCK DISTANCE			101 / 3		92 / 2		—		94.5 / 3	

[a]From 40 mph, mean ± standard deviation; number in lower right corners indicates number of runs in sample.

[b]4120 lb when acting as tow car (includes hitch hardware).

TABLE 8. COMBINATION-VEHICLE STRAIGHT LINE STOPPING DISTANCE RESULTS FOR SUBCOMPACT SIZE TOW CAR[a]

HITCH LOAD	% LOAD LEVELING \ WEIGHT (lb)	TOW CAR ALONE	UTILITY	CAMPER	18 FT TRAVEL	22 FT TRAVEL	BOAT
		3315[b]	1500	1600	4000	4000	3000
0	Car alone	76.5±3.7 / 6					
10	None		104±3.4 / 8	110±4.0 / 10			
	0					90.8±3.2 / 6	
	32-36				108±3.8 / 5		109±4 / 10
PEDAL FORCE (lb)		53	61	54	66	62	61
LOCKUP TENDENCY		LF_c	LF_c	LF_c	RF_c	RF_c, BF_t	RF_c
NO LOCK DISTANCE		78.4 / 3	103 / 3	108 / 7	106 / 2	89.6 / 1	107 / 6

[a]From 40 mph, mean ± standard deviation; number in lower right corners indicates number of runs in sample.

[b]3400 lb when acting as tow car (includes hitch and accessories).

brakes. Only the 15 percent hitch load with 25 percent load leveling exceeded 0.5 g deceleration. This trailer weight was too heavy for the two lighter tow cars.

c) Compact plus horse trailer at 4000 lb. Early lockup of both trailer wheels (on rear axle) caused increased hitch load, which caused early front lockup of the tow car. Note the pedal pressure was only 80 percent of that used for braking with the other trailers.

d) Subcompact plus 18 ft travel trailer. No trailer brake lockup could be obtained with this trailer; thus it was not possible to develop maximum trailer braking force.

e) Subcompact plus unbraked camper trailer. The combined weight of tow car plus trailer cannot provide 0.5 g CV deceleration even when the tow car provides 0.7 g. It appears that for this tow car the minimum tow car to trailer weight ratio for 0.5 g CV deceleration would be [illegible].

f) Subcompact plus boat trailer. Operation of a surge brake system is a function of the horizontal hitch force developed between the tow car and trailer. Effects of surge brake gain and actuation time delay cause this CV to have less than 0.5 g deceleration capability. Further analysis of the surge brake is presented in the next subsection.

In addition, Tables 6-8 provide the following general results:

a) Changes in stopping distance appear slightly improved with load leveling, although the results are not totally consistent.

b) Pedal forces were less than the 120 lb requirement of SAE J135.

c) No-lockup stopping distances were not significantly different from those allowing a tow car lockup.

d) Load leveling appears to reduce the effectiveness of the surge brake; however, the differences are not statistically different. Specifically, the four applicable configurations showed the following stopping distances from 40 mph:

	No Leveling	With Leveling
Intermediate + 3000 lb boat trailer at 15% hitch load	102 ± 3.68	106 ± 2.56
Compact + 3000 lb boat trailer at 15% hitch load	88.8 ± 5.03	92.4 ± 5.21

These stopping distance figures were derived from a minimum of five repeat "maximum performance" runs (see Appendix D). A standard t' statistical test performance on these data cannot accept the hypothesis that the two means are different at the 0.05 level. Additional tests would be required to show statistical differences. In any case the stopping distances noted above are less than previously seen with other surge brake actuators where load leveling totally bound up the surge mechanism (Refs. 2 and 4).

C. COMPARISON WITH TENTATIVE STANDARDS FORMAT

The first subsection described an approach for selecting a tow car to trailer weight ratio as a function of trailer-alone braking capability such as to guarantee a minimum combination-vehicle deceleration level. The previously described results allow such a comparison to be made.

The tow car to trailer weight ratios for the 22 CVs tested in this program are listed in Table 9. They do not include hitch load transfer. The trailer-alone deceleration levels have previously been listed in Table 5. Combining these two tables produces the predictions shown in Fig. 7. Since trailer axle loads at the test condition were less than the GAWR, the minimum trailer alone deceleration levels are given by the dashed lines.

For reference, two 0.5 g combination-vehicle deceleration criteria lines are overplotted in Fig. 7. These were derived in Figs. 1 and 2 and reflect the simplified and complete models, respectively, for a tow car capable of 0.6 g deceleration. Note that in the region of 0.4 g there is very little difference between the two models. For example, with either model, four

TABLE 9. TOW CAR TO TRAILER WEIGHT RATIOS, W_c/W_t

TRAILER / TOW CAR	UTILITY	CAMPER	18 FT TRAVEL	19 FT TRAVEL	22 FT TRAVEL	27 FT TRAVEL	HORSE		BOAT
							LIGHT	HEAVY	
Intermediate	3.17	2.97	1.22	1.19	1.19	0.79	NT	0.82	1.58
Compact	2.74	2.56	1.06	1.02	1.02	NT	1.02	NT	1.37
Subcompact	2.27	2.12	0.88	0.85	0.85	NT	0.85	NT	1.13

NT = Not tested.

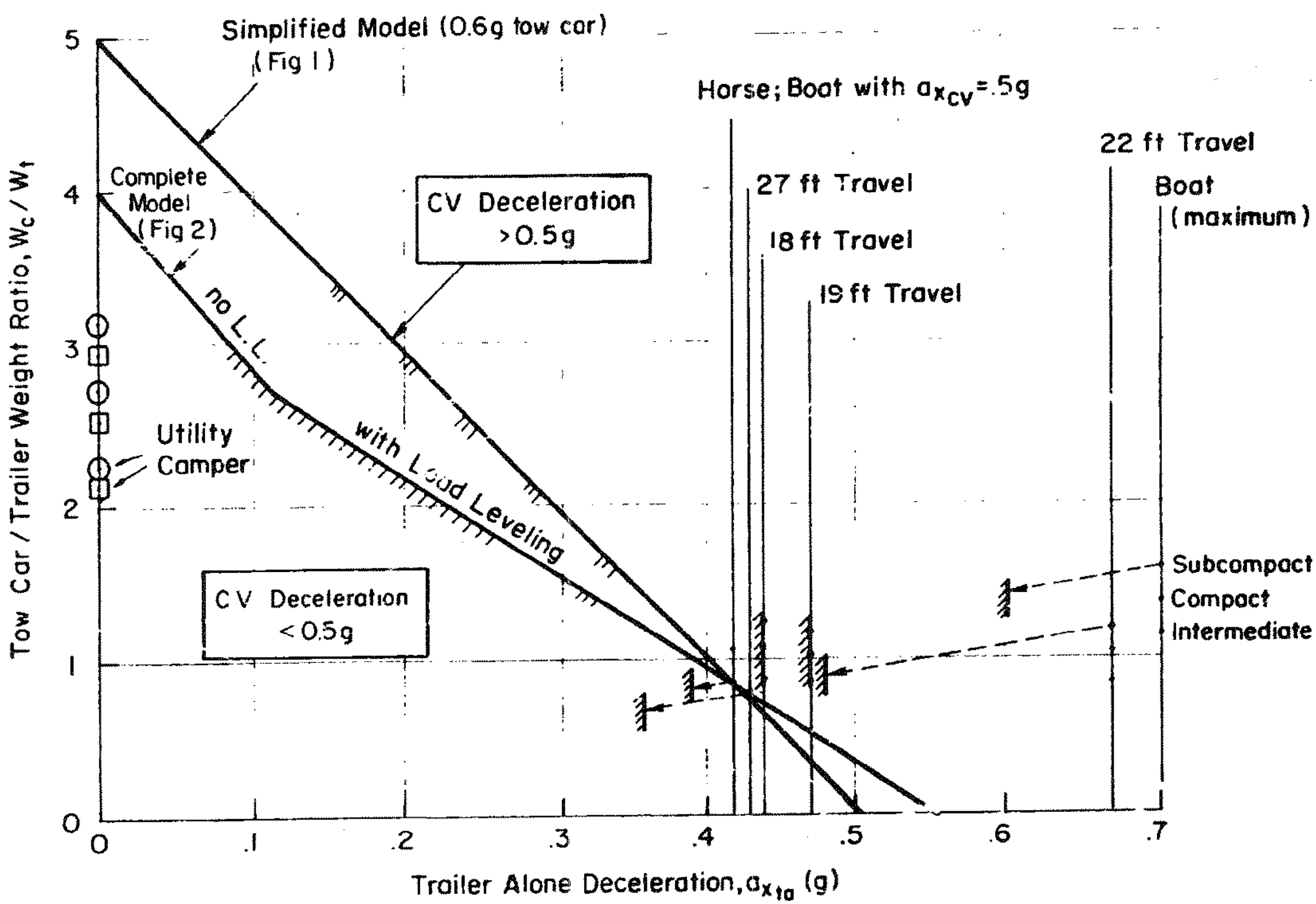


Figure 7. Comparison of Predicted Test Results with Hypothetical 0.5 g CV Straight Line Brake Requirement

trailers should be able to provide a combination-vehicle deceleration in excess of 0.5 g, i.e., they fall above the criterion line. For two trailers (the 27 ft travel trailer and horse trailer at test weight) the CV deceleration should be close to 0.5 g with the intermediate-sized tow car. At full GAWR or with lighter tow cars, neither should be able to provide 0.5 g. Both unbraked trailers should also fall below the criterion boundary.

1. Electric Brake Trailers

In terms of actual performance, measured results are first compared for the trailers with electric brakes. As expected, both the horse and 27 ft travel trailers recorded combination-vehicle deceleration less than 0.5 g. The subcompact plus 18 ft travel trailer was expected to barely exceed 0.5 g, and in fact was very close (108 ft recorded versus 107 ft required). The 19 and 22 ft trailers were predicted to exceed 0.5 g, and did so.

2. Surge Brake Trailers

The next comparison applies to surge brake trailers. Since the braking force in this braking system is developed through a horizontal hitch force (trailer pushing on the tow car), the effect of tow car size is more significant than with electric brake trailers. Thus, in this case the minimum tow-car/trailer weight ratio will be affected by the amount of brake force developed per pound of horizontal hitch force. This ratio is called the surge brake gain, G.

In order to use the format shown in Fig. 7 the "effective" trailer-alone deceleration level for a surge brake system must be determined. It is not realistic to simply apply increasing horizontal hitch forces and measure trailer-alone deceleration; the actual applied hitch forces during deceleration may be significantly less. (This is not true for electric brake trailers where full trailer brake voltage can theoretically be developed at any tow car brake level.) Using the simplified model, this "effective" trailer-alone deceleration level can be found when the surge brake gain, G, is known. The equation is:

$$a_{x_{ta}} = \frac{a_{x_{cv}}[G/(1+G)]}{1 - \%HL/100} \qquad (2)$$

Using the 0.5 g criterion illustrated in Fig. 7, a surge brake gain of 3.0 (derived in Appendix B for the boat trailer), and a 10 percent hitch load, the "effective" trailer-alone deceleration for our test trailer was actually only 0.42 g. In other words, the boat trailer — as a combination-vehicle — cannot apply the deceleration level previously shown in Table 5 (and Fig. 7) when the combination-vehicle is only decelerating at 0.5 g. It really can only apply 0.42 g deceleration to the combination. Referring to Fig. 7 now, this deceleration level is equivalent to the horse trailer and thus would require a tow car exceeding 0.9 times the trailer weight. Since this ratio was exceeded in the test program (minimum ratio was 1.13), the boat trailer combination-vehicles should have exceeded 0.5 g deceleration. When viewed in this context it is now apparent why the subcompact/boat-trailer configuration was very close to 0.5 g (109 ft measured versus 107 ft required).

It should also be mentioned that if the surge gain were increased then the trailer-alone deceleration would also be higher. In the limit, trailer-alone deceleration approaches the CV deceleration; and thus the trailer could be stopped at 0.5 g by any weight tow car. In terms of the complete model, however, Fig. 7 shows that the trailer-alone deceleration must exceed 0.55 g in order to meet a 0.5 g deceleration requirement with any weight tow car.

The effect of surge brake gain can possibly be understood more clearly with the help of Fig. 8. This figure plots the combination-vehicle deceleration level as a function of tow car to trailer weight ratio. Various surge gains are applied to a minimum deceleration tow car (i.e., 0.6 g). For comparison with Fig. 7 we can select a 0.5 g CV deceleration criterion and determine the weight ratio at any surge brake gain. For example, with no brakes (G = 0) the weight ratio is 5, the same as the left boundary in Fig. 7. A gain of at least 4 is required if the trailer weight is equal to the tow car weight. Viewed another way, a surge brake trailer of

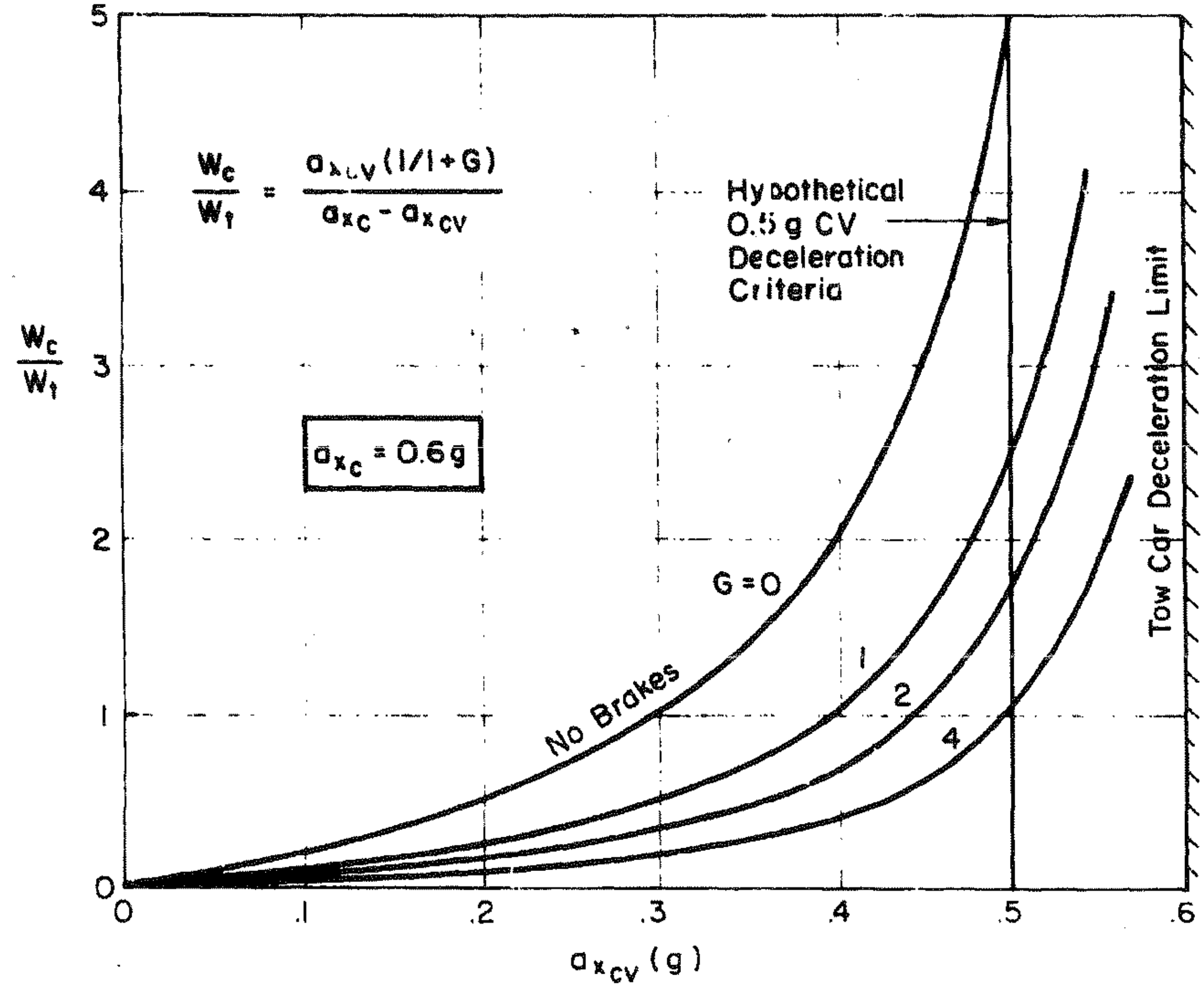


Figure 8. Simplified Analysis of Surge Brake Gain Effect on Combination-Vehicle Deceleration

weight, $Wt_{surge\ brake}$, can be shown to have the stopping capability equivalent to an unbraked trailer of weight, $Wt_{no\ brakes}$, via the relationship:

$$Wt_{surge\ brakes} = (G + 1)Wt_{no\ brakes} \tag{3}$$

For the example configuration in this program the surge brake gain was 3; thus, a surge-brake-equipped trailer of 6000 lb could ideally be stopped in a distance equivalent to a 1500 lb unbraked trailer.

The effect of surge brake time delay is also significant. This parameter represents the physical time delay between tow car brake application and subsequent trailer brake actuation, and is due to the time required to build up horizontal hitch forces. Based on results obtained with one surge brake mechanism in this program, an average time delay of 0.30 sec was determined. Since this delay only acts on the trailer brakes, we would expect stopping distances from 40 mph to be 6 ft longer than those predicted from deceleration level only. This delay probably accounts for the differences between measured and predicted stopping distances for the boat trailer. However, as long as any braking standard is specified in terms of stopping distance, there would be no necessity to determine a brake actuation time delay. In effect, the trailer brakes would have to develop a higher average deceleration level to make up for the inherent time delay.

Breakout forces in a surge brake mechanism may also be significant. For example, when load leveling was applied to a 2222 lb camper trailer, in the Ref. 4 study, the surge brake was rendered essentially useless. Also, in previous STI tests reported in Ref. 2, high hitch loads caused binding of the surge mechanism such as to increase the breakout force beyond the maximum horizontal hitch force. In this case the brakes were again rendered nonoperational. Although these limit cases become obvious in maximum performance brake stops, smaller breakout force levels would not show up. On the other hand, it is necessary to provide some finite breakout level to avoid spurious brake applications on a rough road or to allow for backing up without applying trailer brakes. The breakout level should not be so high, though, as to keep the trailer brakes from operating during downhill towing.

In summary, surge brake trailers can be rated with the same rule format as electric brake trailers; however, it will be necessary to determine the surge brake gain in order to establish the "effective" trailer-alone deceleration level.

3. Unbraked Trailers

The final comparison applies to unbraked trailers. As indicated in Fig. 7, all 1500 and 1600 lb unbraked trailer combination-vehicles should not be able to meet a 0.5 criterion. However, since the two cars used in this program were able to achieve greater than 0.6 g deceleration, all but the smallest tow car plus largest unbraked trailer (i.e., subcompact/camper) exceeded 0.5 g. In fact, if a tow car deceleration of 0.71 g is used, the simple model will identically predict the measured results. Additional comparisons of the unbraked trailer data with the complete model are shown in Fig. 9. This analysis assumes tow car deceleration of 0.70 g. Although tow car brake balance measurements were not made, the stopping distance results shown in Fig. 9 indicate near-optimum tow car braking. It is clear from these data (and the other trailer data) that the biggest impact of any combination-vehicle deceleration standard would be in a limitation on the maximum allowable unbraked trailer weight for a given tow car weight. Consequently, the selection of the CV criterion must be carefully considered. This will be discussed in the next subsection.

D. BASIS FOR TRAILER-ALONE AND COMBINATION-VEHICLE DECELERATION CRITERION

Several references can be used as benchmarks for the selection of a CV deceleration, or stopping distance, requirement. These include: SAE recommended practice, FMVSS 105-75 for vehicles other than passenger cars, Federal Highway Administration Transportation Code, state trailer towing laws, and recommendations by the Canadian Standards Association. Each is discussed below.

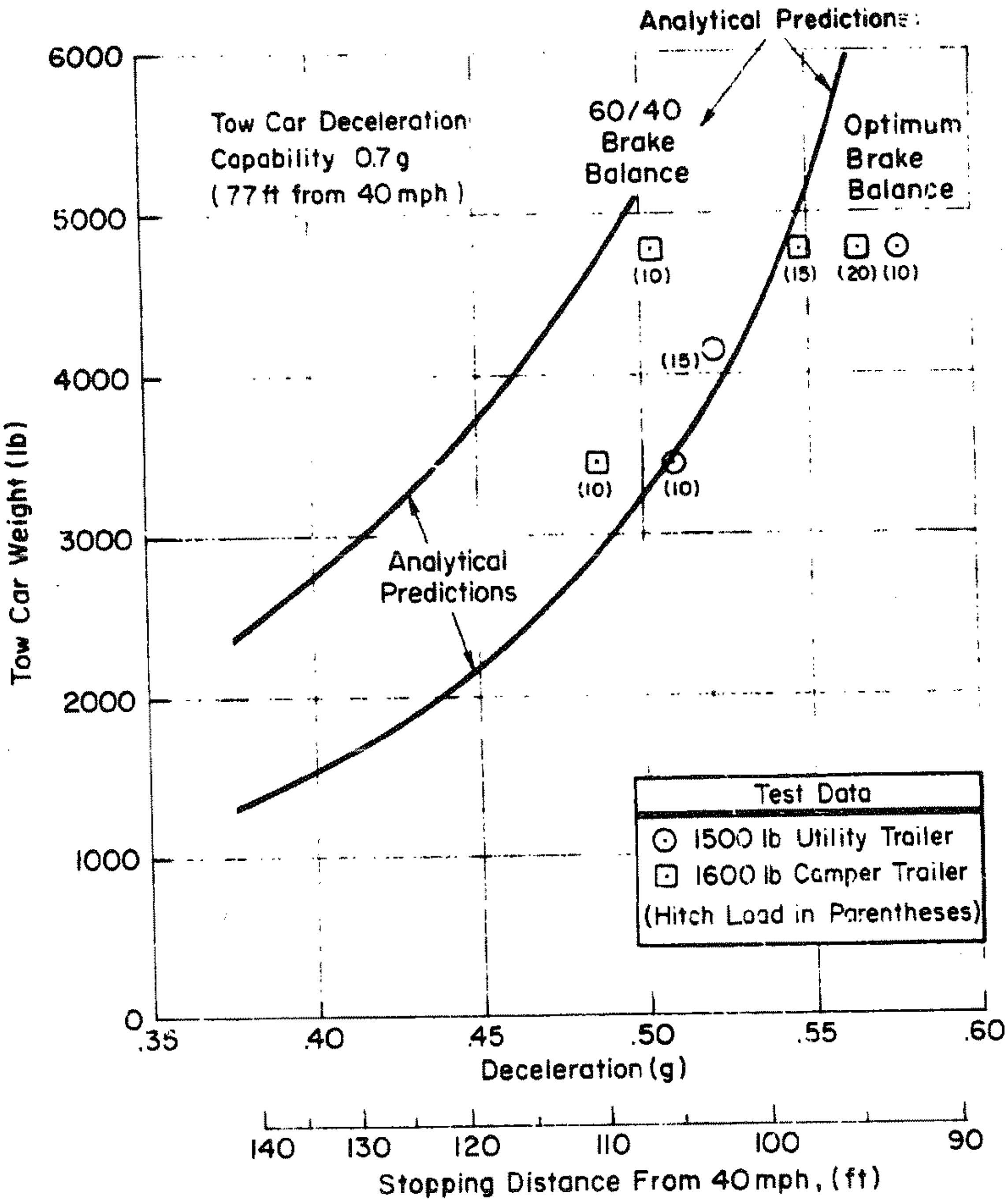


Figure 9. Analytical Versus Measured Stopping Distances for 1500/1600 lb Nonbraked Trailers

1. SAE Recommended Practice

SAE Recommended Practice J134 (Ref. 10) entitled "Brake System Road Test Code — Passenger Car and Light Duty Truck-Trailer Combinations" is intended for use by tow vehicle manufacturers to determine the maximum allowable unbraked trailer weight and by trailer manufacturers (although no trailer-alone test procedures are specified). The test procedures are patterned after FMVSS 105-75 in terms of effectiveness tests, burnish tests, fade tests, etc. The "Service Brake System Performance Requirements — Passenger Car-Trailer Combinations," SAE J135, specifies CV performance in terms of tow car pedal pressure necessary to provide 0.5 g deceleration. For reference, the test procedures and requirements of J134 and J135 are given in Table 10. Although much of the J134 test procedure is concerned with burnish, fade, and recovery, the essence of the effectiveness test requirements is that the combination-vehicle should be able to stop at 0.5 g deceleration from 30 and 60 mph without exceeding 100 or 120 lb pedal force, respectively. Only one test point is required to show compliance, and wheel lockup is allowed.

2. FMVSS 105-75

The second approach toward specification of a CV deceleration (or stopping distance) criterion is to place greater reliance on the existing requirements of FMVSS 105-75, as might be applied to a combination-vehicle. This document provides stopping distance requirements for vehicles other than passenger cars in a weight class less than 10,000 lb GVW, as well as those for ordinary passenger cars. The stopping distance requirements for these "other" vehicles (usually buses) allows approximately a 60 percent relaxation in performance, yet the vehicle itself can be up to 100 percent heavier. The exact figures for stopping distance from 40 mph are shown in Table 11. In terms of deceleration, the 144 ft specified in FMVSS 105-75 converts to a 0.37 g capability.

3. FHWA Transportation Code

Another existing requirement is the Federal Highway Administration Brake Performance Requirement 393.52 (Ref. 13) for combinations of property carrying vehicles. The deceleration requirement in this document is 0.435 g.

TABLE 10

BRAKE SYSTEM ROAD TEST CODE REQUIREMENTS — PASSENGER CAR/TRAILER COMBINATIONS

TEST NUMBER	TEST TYPE	NUMBER OF STOPS	TEST SPEED (mph)	DECELERATION	INITIAL BRAKE TEMPERATURE	COOLING	REQUIREMENT
1	Preburnish Check	10	30	10	As is	1 mi at 40 between stops to get IBT	$10 \le F_p \le 55$
2	Preburnish Effectiveness	a	30 60	16 max 16 max	200° before each stop	To get IBT	$15 < F_p < 100$ $15 < F_p < 120$
3	Burnish	200	40	12	250° or 1 mi at 40	To get IBT	
4	Adjust Brakes						
5	Second Effectiveness	Same as No. 2					
6	Fade Baseline	3	30	10	150° before each stop	—	Record value of F_p
7	Fade	10	60	15 or F_p 200 lb	150° before <u>first</u> stop	0.8 mi at 60, 1 mi at 40 prior to recovery	First 4 stops will have $F_p \le$ 120, 147, 173, 200
8	Recovery	12	30	10 or F_p 200 lb	Record	1 mi at 40	Min a_x = 5 ft/sec^2 (at max F_p 200 lb) for first 5 stops and F_p < 150 lb by sixth stop for a_x = 10 ft/sec^2
9	First Effectiveness Spot Check	2	60	15	200° before each stop	To get IBT	Record max F_p
10	Second Reburnish	35	40	12	250° or 1 mi at 40		
11	Second Fade and Recovery	Same as Nos. 7 and 8 but make 15 fade stops					First 8 stops will have $F_p \le$ 120, 137, 143, 15[illegible], 177, 177, 189, 200 lb
12	Second Effectiveness Spot Check	Same as No. 9					
13	Third Reburnish	Same as No. 10					
14	Final Effectiveness	Same as No. 2					
15	Inspect						

[a] Increasing pedal forces on each run until [illegible] g achieved.

Conditions of Test:

Ambient temperature [illegible] °F	Hitch load = 10 percent trailer weight
Note uncontrollable conditions	Trailer weight = gross
Tow vehicle weight = curb + [illegible] lb (including hitch load)	Record trailer brake inputs
	Dry pavement

TABLE 11

FMVSS 105-75 STOPPING DISTANCE REQUIREMENTS FROM 40 MPH

VEHICLE	GVW (lb)	S_{40} (ft)
Passenger cars	≤ 5000	91
	Lightly loaded	87
Other than passenger vehicles	≤ 10,000	144
	Lightly loaded	144

4. State Laws

Several states also have combination-vehicle braking requirements (e.g., Refs. 14 and 15). These requirements are listed in Table 12. Most other states simply require brakes on the trailer when trailer weight exceeds some specified value (e.g., 1500 lb in 9 states, 2000 lb in 5 states, 3000 lb in 25 states) or when the trailer weight exceeds 40 percent of the tow vehicle weight.

TABLE 12

STATE TRAILER TOWING BRAKE REQUIREMENTS

STATE	STOPPING DISTANCE FROM 20 MPH	DECELERATION (g)	REMARKS
California	40 ft	0.34	Towing vehicles < 10,000 lb
Massachusetts	30 ft	0.45	
Oregon	37 ft	0.36	Plus all trailers exceeding 35 ft

5. Canadian Standards Association

A most relevant trailer braking reference is the Canadian Standards Association's proposed standard for trailer-alone braking performance (Ref. 6). This document specifies 0.435 g deceleration from 30 mph and 0.35 g from 60 mph. The test procedure described in Table 13 follows the format of FMVSS 105-75, where the deceleration levels and test speeds are ratioed by the trailer to car weight to compensate for the unbraked mass of the tow car. These requirements are used by the Dexter Brake and Axle Division of Phillips Industries (Ref. 12).

Based on the CSA recommendation of 0.435 g for trailer alone and FMVSS 105-75 for 0.6 g for tow car alone, it is, ideally, possible to achieve a 0.5 g CV deceleration since trailer weights are generally never larger than 1.25 times the tow car weight ($W_c/W_t = 0.8$). This criterion would also be consistent with SAE J135 and the assumptions in Ref. 4. However, based on the test data presented previously, many current electric and surge brake trailers at full GVWR would not be able to meet the 0.5 g CV criterion shown in Fig. 7. Also, maximum unbraked trailer weight would then be limited to less than 1000 lb.

In summary, based on the test data obtained in this program, it appears that a 0.5 g combination-vehicle deceleration criterion is optimistic in terms of stopping distance (i.e., 107 ft from 40 mph). Ideally, all braked trailer test configurations should have passed, but in many cases they did not. Also, if two of the test trailers had been loaded to maximum GAWR, they could not, even ideally, exceed 0.5 g. Both 1500 lb unbraked trailer CVs would also not have been able to exceed 0.5 g if the tow cars had only the minimum braking capability specified by FMVSS 105-75.

The next subsection presents recommendations for stopping distance criteria and trailer braking test procedures.

E. RECOMMENDATIONS

Based on the analysis and test results presented herein, recommendations can be made for CV braking standards and test procedures. These are summarized below.

TABLE 13. CANADIAN STANDARDS ORGANIZATION PROPOSED STANDARD D313; TRAILER RUNNING GEAR (TRAILER-ALONE BRAKING)

TEST NUMBER	TEST TYPE	NUMBER OF STOPS	REFERENCE SPEED, V_R [a] (mph)	DECELERATION (g)	INITIAL BRAKE TEMPERATURE	COOLING	REQUIREMENT
1	Brake Warming	To get IBT	40-20	$12\ W_t/W_{cv}$	200-250 °C	—	—
2	Instrumentation Check	≤ 10	40-20	$12\ W_t/W_{cv}$	200-250 °C	—	—
3	Preburnish Effectiveness	6 6	30 60	Maximum	200-250 °C 200-250 °C		.435 g .35 g } Only 1 stop need meet requirement
	Burnish		$V_T = 40 \times \sqrt{1600 - 1200\frac{W_t}{W_{cv}}}$	$12\ W_t/W_{cv}$		Accelerate to 40 mph	
5	Adjust						
6	Second Effectiveness	Same as No. 3					
7	Fade Baseline	≤ 8 3	Determine pressure to obtain $a_x = 12\ W_t/W_{cv}$ 30	$12\ W_t/W_{cv}$	200-250 °C	—	Measure SD; average
8	Fade Snubs	10	$V_T = 40 \times \sqrt{1600 - 1200\frac{W_t}{W_{cv}}}$	Maximum	200-250 °C before <u>first</u> stop		
9	Recovery	5	30	Baseline pressure	As is	1.5 mi between stops	Stopping distance of fifth stop will be +30, −40 percent of baseline average
10	Reburnish	Same as No. 4 but only make 35 snubs					
11	Third Effectiveness	Same as No. 3					

Conditions of Test:
- Ambient temperature: 32-100 °F
- Trailer axle weight: GAWR, when attached to tow car, including load leveling
- Hitch load = manufacturer recommended
- Attitude = ±2 deg level
- Tow car in neutral for all tests
- Wind = 0
- Pavement = ±1 percent level PCC (or equivalent SN)
- Lane deviation < 1 ft either side of trailer

[a] Test speed = $V_R\sqrt{W_t/W_{cv}}$ where W_t = static trailer axle weight (when attached to tow car) and W_{cv} = total combination vehicle weight.

1. Tow Car Weight Format

The rule format suggested by HSRI (Ref. 4) in which each trailer would be specified in terms of a minimum weight tow car appears to be a viable standard under the following conditions:

- The combination-vehicle deceleration criterion should be reduced to 0.4 g. This would produce the following tow car weight requirement if a complete braking model is used:

$$W_c = W_t(2.1 - 3.5a_{x_{ta}}) \tag{4}$$

  where

  W_c = Tow car weight, GVWR

  W_t = Trailer weight, GVWR

  $a_{x_{ta}}$ = Trailer-alone deceleration capability (g)

  All trailers tested in this program would pass this criterion if it is additionally assumed that a tow car cannot "practically" tow a trailer weighing more than 1.25 times the passenger car weight. The practical problems are acceleration, hill climbing, overheating, hitch mounting, etc.

- Trailer-alone deceleration measurements should be based on static axle test weight and then ratioed up to GAWR if lockup cannot be obtained at full trailer brake voltage. For surge brake trailers, the "effective" trailer-alone deceleration level must consider the surge brake gain as discussed in Subsection C.

- Stopping distance must be the primary performance measure for the combination-vehicle. For example, an average deceleration criterion of 0.4 g would require a stopping distance of less than or equal to 134 ft from 40 mph.

2. Trailer Brake Capacity

An alternative to the above format would be to specify a maximum allowable braked weight capacity. For example, based on a 0.5 g CV criterion, it would be desirable to see no more than 1500 lb per each 10 in. brake. A more conservative number would be 1250 lb, as was the case for the 22 ft travel trailer. The horse trailer at 2900 lb/brake would be (and is) totally unacceptable; the 18 ft, 19 ft, and 27 ft travel trailers at 1700 lb/brake are marginal; and the boat trailer (with no load leveling) and 22 ft travel trailer at 1500 and 1250 lb/brake, respectively, would in fact exceed a 0.5 g deceleration criterion.

3. Test Procedures

As a result of the tests performed in this program, several recommendations in terms of test procedures were also developed. These are as follows:

- A trailer-alone brake test procedure combining key aspects of the procedures used by Dexter (Ref. 23), HSRI (Ref. 4), and STI (in this program) should become a standard. Based on problems with brake parts, this procedure should also include the complete tests of SAE J134. A recommended test procedure is given in Table 14. To minimize instrumentation, stopping distance can be used in place of average deceleration between two speeds.

- Surge brake gain must be determined with a separate procedure that applies known horizontal forces to the hitch. This can be accomplished statically by applying a known force and measuring the trailer brake pressure (see Appendix B). This gain is then multiplied by the trailer-alone brake force per unit brake pressure to determine pounds brake force per pound hitch force.

TABLE 14. TRAILER-ALONE BRAKE PERFORMANCE TEST PROCEDURE

TEST NUMBER	TEST TYPE	NUMBER OF STOPS	TEST SPEED[a] (mph)	DECELERATION[b]	INITIAL BRAKE TEMPERATURE	COOLING	REMARKS
	Instrumentation Check	10	40 → 30	9 V to brakes	—	—	—
2	Preburnish Effectiveness	6[c] at maximum performance	$40\sqrt{W_t/W_{cv}}$	2, 4, 6, 8, 10, 12 V to trailer brakes	200-250°F	—	Plot average deceleration vs. applied voltage level. Determine maximum trailer-alone deceleration capability a_{xta}
3	Burnish	200	40 → $\sqrt{1600-1200\frac{W_t-HL}{W_{cv}}}$	$12(W_t/W_{cv})$	200-250 or 1 mi between snubs	—	—
4	Second Effectiveness		Same as No. 2				$a_x \geq 0.435$ g at 12 V
5	Fade Baseline	≤8 3	30 30	$12(W_t/W_{cv})$ Voltage necessary	200-250	—	Determine voltage necessary.[b] Measure SD; average.
6	Fade Snubs	10	40 → $\sqrt{1600-1200\frac{W_t-HL}{W_{cv}}}$		200-250 before <u>first</u> stop	1.5 mi at 30 mph, then start deceleration	
7	Recovery	5	30	Baseline voltage		1.5 mi between	SD for last stop +30%, −40% average baseline
8	Reburnish		Same as No. 3 except 35 snubs				
9	Third Effectiveness		Same as No. 2				$a_x \geq .435$ g at 12 V

Condition of Test: Use trailer brakes only. Plot stopping distance vs. speed for effectiveness tests. Compute deceleration from $a_{xta} = (u^2/2SD)$.

[a] $W_t - HL$ = static axle weight, W_{cv} = combination-vehicle weight.

[b] For surge brake trailers, voltage increments replaced by trailer brake pressure increments, increasing to lockup.

[c] Only one stop necessary at lower voltage levels; at maximum, attempt to have three below and three above wheel lockup.

SECTION IV

TRAILER DAMPING PERFORMANCE

Trailer stability represents the second performance parameter for which a vehicle handling standard is required. This trailer mode is the pendulous swing oscillation of a trailer commonly seen on the highway. If trailer mass and inertia are small with respect to the tow vehicle this motion is more of a nuisance than a handling problem and, in fact, cannot become unstable if the hitch point does not move. However, as the trailer approaches or exceeds the tow vehicle in mass and inertia the forces and moments applied by the oscillating trailer to the hitch point (and hence to the automobile) become large enough to cause loss of control, trailer separation, and/or combination-vehicle rollover.

Due to the frequency and damping separation between the tow car and trailer modes, the trailer mode oscillation can be accurately modeled by a simple second-order system response. By analogy, the resulting performance measure used to assess trailer stability is the reduction in swing amplitude with successive oscillations. In vehicle dynamics terminology (Ref. 16) such oscillatory stability is measured by cycles to 1/2 amplitude, or (an equivalent) damping ratio, ζ. For example, time history plots of trailer swing at different damping ratios are shown in Fig. 10 for reference. When $\zeta = 0$ the oscillation is sustained (undamped), and at $\zeta = 0.5$ the oscillation ceases within 1 cycle. At $\zeta = 1.0$ there is no oscillation. If damping ratio becomes negative, the oscillation amplitude increases with time, and hence is unstable — a very undesirable condition. The most significant changes pertinent to trailer swing occur at low or negative damping ratios, i.e., from negative to 0.3, where the oscillations are perceptible to the driver and where safety implications arise.

In this program 23 combination-vehicle configurations were tested at various hitch loads and load levelings to determine the resulting trailer damping ratios. In addition, a methodology was developed for predicting trailer damping and the hitch load necessary to meet a tentative minimum damping criterion. The following two subsections present the analytical

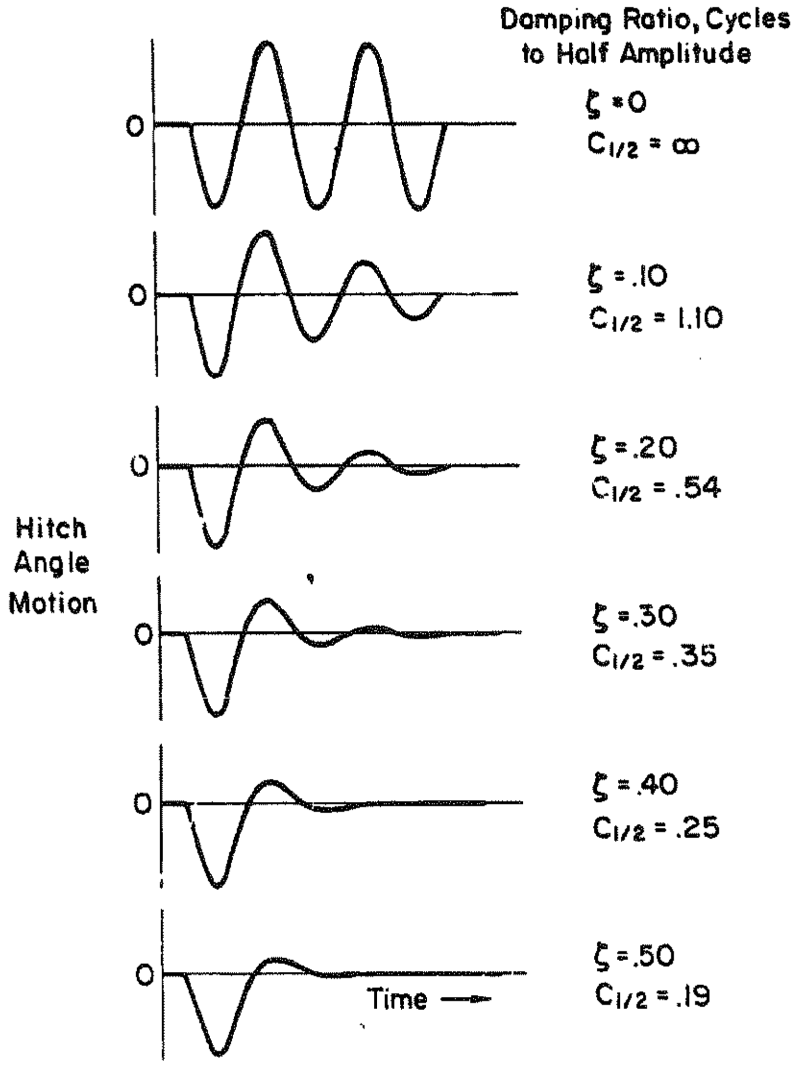


Figure 10. Second-Order System Indicial Response characteristics

background and supporting full-scale test results, respectively. Recommendations for a minimum allowable hitch load rule format, manufacturers' design guides, and test procedures are then presented.

A. ANALYTICAL CONSIDERATIONS

At first glance it might appear that the trailer swing mode would be one dynamic handling characteristic for which it would be easy to establish trailer test and performance requirements. Indeed under limited circumstances, such as a very high tow car to trailer weight ratio, it is. Unfortunately, the cases of concern are those in which trailer swing causes and interacts with the tow car response. Isolation of trailer damping performance is not so straightforward in this latter case. However, in this program we have developed and evaluated an approach for predicting the damping characteristics of these combination-vehicles.

The first step in this approach was to develop equations that separately treat both the trailer-alone and the combination-vehicle. The trailer-alone case represents a condition corresponding to a laterally immovable hitch point. In effect, this condition would occur if the trailer were being pulled by a tow car of infinite mass. In this case, the swing (or fish-tail) mode is a pendulous type oscillation that, once excited, will die out since no energy is added to the system. References 2 and 17 have shown that for this condition the trailer-alone oscillation frequency ($\omega_{\eta_{ta}}$) and damping ratio ($\zeta_{\eta_{ta}}$) may be calculated in terms of trailer parameters as follows:

$$\omega_{\eta_{ta}} = \sqrt{\frac{2Y_{\alpha_3}\ell_2}{I_{th}}} \tag{5}$$

$$\zeta_{\eta_{ta}} = \frac{\sqrt{\ell_2^3 Y_{\alpha_3}}}{U_o\sqrt{2I_{th}}} \tag{6}$$

where

Y_{α_3}	Cornering stiffness of trailer tires on one side (lb/rad)
ℓ_2	Hitch to trailer axle length (ft)
I_{th}	Trailer moment of inertia about hitch (slug-ft^2)
U_o	Forward speed (ft/sec)

Equation 6 illustrates the well-known fact that damping is most influenced by hitch-to-axle distance (the longer, the better) and speed (the slower, the better). In addition, stiffer trailer tires or smaller moment of inertia about the hitch also improve the damping. Note also in this "limit" case that the damping approaches zero as speed increases and <u>cannot</u> be negative (unstable).

The interaction of trailer damping with tow car weight was first investigated using a five-degree-of-freedom car/trailer model given in Ref. 3. The resulting damping ratio versus speed is shown in Fig. 11 for three tow car weights pulling a 3000 lb trailer. Note that if the tow car weighs 3000 lb (same as the trailer) we would expect an unstable trailer at 68 mph; a 6000 lb tow car would extend the speed for instability to 93 mph; and, as expected, a 600,000 lb tow car (academic, of course) never allows an unstable trailer. This is verified by the fact that at this vehicle weight the simulation result very nearly overlays the simple trailer-alone expression given by Eq. 6.

Since the interaction of the trailer and the tow car is dependent on the magnitude of the lateral hitch forces applied <u>by</u> the trailer to the tow car, it is useful to derive the hitch forces applied to an infinite weight tow car, i.e., a limit case wherein the hitch point cannot move laterally during trailer swing. For such conditions the peak lateral hitch forces

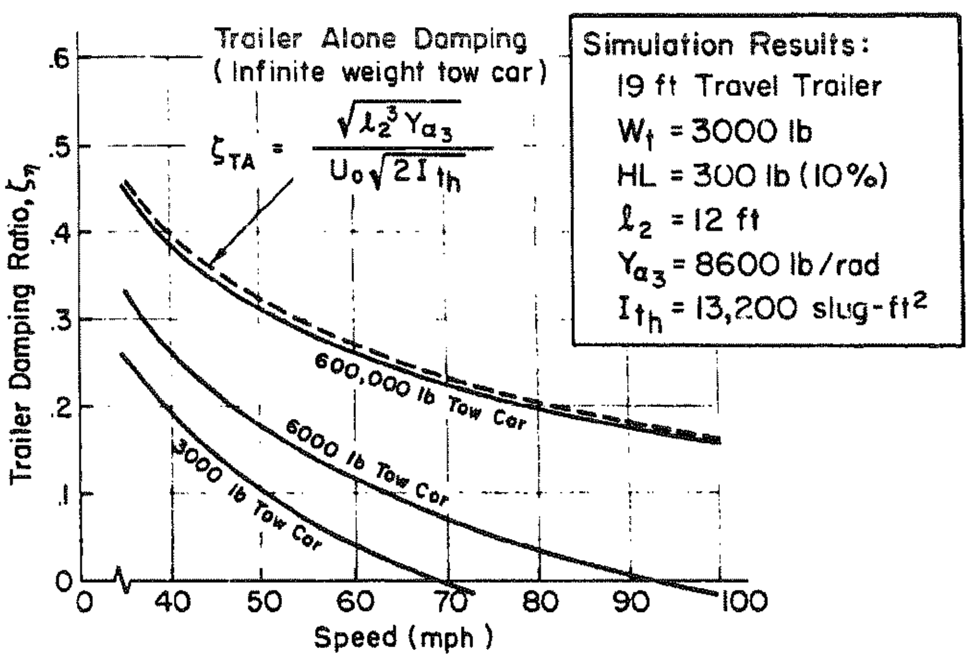


Figure 11. Effect of Speed on Trailer Damping

will occur at the peak swing amplitude of the trailer. Using the symbology from Fig. 12 we can then drive the lateral hitch forces, F_{y_h}, caused by the trailer lateral tire forces, F_{y_t}. Taking the summation of forces and moments yields:

$$\begin{aligned} \sum \text{Forces} &= F_{y_t} - F_{y_h} = m_t a_y \\ \sum \text{Moments} &= F_{y_t} h + F_{y_h} e = I_{t_o} \ddot{\eta} \end{aligned} \tag{7}$$

For this special case of an immovable hitch, the lateral acceleration (a_y) of the center of gravity is also equal to the distance e times the angular acceleration, i.e., $a_y = e\ddot{\eta}$. The forces and moment equations can then be solved for trailer acceleration:

$$\ddot{\eta} = \frac{F_{y_t} - F_{y_h}}{m_t e} = \frac{F_{y_t} h + F_{y_h} e}{I_{t_o}} \tag{8}$$

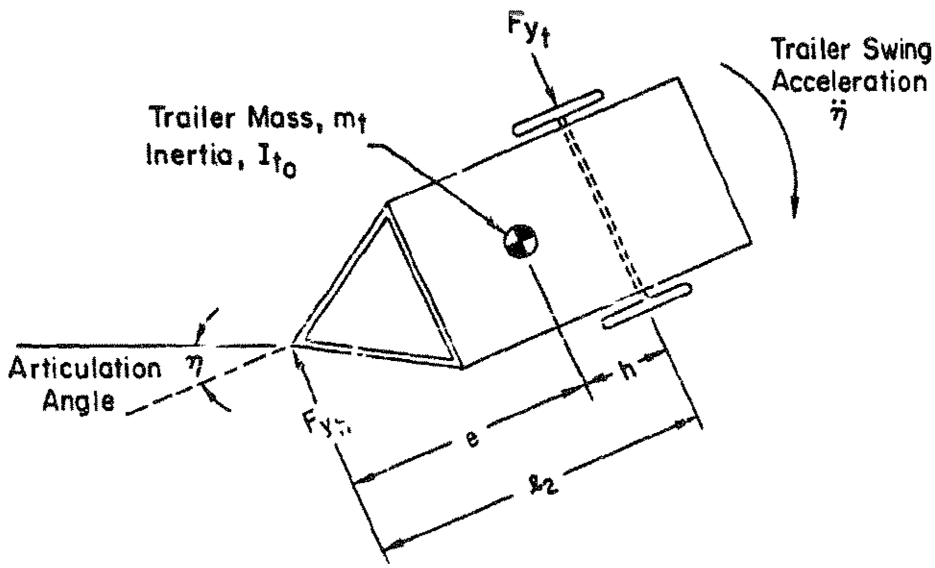


Figure 12. Trailer Inertia and Geometry Definitions

Rearranging produces the hitch force to tire side force ratio:

$$\frac{F_{y_h}}{F_{y_t}} = \frac{I_{t_o} - m_t eh}{I_{t_o} + m_t e^2} \tag{9}$$

Note that since h/ℓ_2 is the (vertical) hitch load, HL, divided by trailer weight, W_t, and $h + e = \ell_2$, then the second term in the numerator of Eq. 9 can be rewritten as:

$$\frac{HL\ell_2^2}{g}\left(1 - \frac{HL}{W_t}\right)$$

The denominator of Eq. 9 is simply the moment of inertia taken about the hitch point, I_{t_h}. In summary,

$$\frac{F_{y_{hitch}}}{F_{y_{tires}}} = \frac{I_{t_o} - m_t eh}{I_{t_o} + m_t e^2} = \frac{I_{t_o} - \frac{HL\ell_2^2}{g}\left(1 - \frac{HL}{W_t}\right)}{I_{t_h}} \tag{10}$$

where

I_{t_o}	Trailer moment of inertia about the c.g. (slug-ft^2)
I_{t_h}	Trailer moment of inertia about the hitch (slug-ft^2)
m_t	Trailer mass (W_t/g) (slugs)
e	Trailer hitch to c.g. length (ft)
h	Trailer c.g. to axle length (ft)
ℓ_2	Hitch to trailer axle length (e + h) (ft)
$(HL/W_t)100$	Percent trailer weight applied to hitch (i.e., "percent hitch load")

Note that when the numerator term is positive in sign the hitch force acts to produce a tow car yaw response which is in the same angular direction as the trailer angular acceleration, and hence will produce an amplifying factor in the trailer swing motions. On the other hand, by adjusting the percent hitch load the numerator can be made equal to zero, and hence there will be no lateral forces transmitted to the hitch. Physically, this c.g. location puts the center of percussion at the hitch point and hence any forces applied at the tire are not felt at the hitch. Since each term in Eq. 10 is a function of the trailer properties only, we have termed this ratio "hookup factor" (HUF). It can be controlled by the user in setting up the hitch load or by the trailer designer in setting lengths and inertias.

In addition, there must be a companion factor relating the tow car's sensitivity to lateral hitch forces. This will determine how much such forces contribute to the reduction in trailer damping. For the present we will denote this sensitivity as K_s, so that the overall combination-vehicle trailer damping ratio may be approximated by:

$$\zeta_{cv} \doteq \underbrace{\frac{\sqrt{\ell_2^3 Y_{\alpha_3}}}{U_o\sqrt{2I_{t_h}}}}_{\substack{\text{Trailer}\\ \text{Alone}}} - \underbrace{\left\{\frac{I_{t_o} - \frac{W_t \ell_2^2}{g}\left[\frac{\%HL}{100} - \left(\frac{\%HL}{100}\right)^2\right]}{I_{t_h}}\right\}}_{\substack{\text{"Hookup}\\ \text{Factor"}}} \underbrace{K_s}_{\substack{\text{Tow Car}\\ \text{Sensi-}\\ \text{tivity}}} \tag{11}$$

K_s is an empirical quantity based on the trailer/tow-car weight ratio, which must be derived from the full-scale tests presented in the next subsection.

From Eq. 11 it is clear that the trailer damping can be maximized when the trailer center of gravity is located to make the "hookup factor" equal to zero, or when the tow car is made infinitely heavy ($K_s = 0$). Analysis of "hookup factor" (HUF) as a function of hitch load (i.e., c.g. location) for representative trailer geometries indicates that a hitch load from 20 to 23 percent of the trailer weight will eliminate the tow car interaction, i.e., HUF = 0, for a wide range of trailer weights. Although this result is of interest, hitch loads cannot practically be this large without compromising tow car stability, i.e., loss of understeer. The real problem is to provide a sufficiently stable trailer at light hitch loads. For example, the hookup factor in Eq. 11 can be minimized by the trailer designer by increasing the wheelbase, ℓ_2, or by decreasing the moment of inertia, I_{t_0}. The user can improve the trailer damping by using a heavier tow car that restrains the hitch force interaction (thus making K_s smaller).

Since both hitch load and mass ratio significantly affect the trailer damping at a given speed, it is necessary to develop a tradeoff between hitch load and weight ratio to guarantee at least the minimum desired trailer damping for the combination-vehicle. This would then be in a form compatible with a requirement or recommendation to the user of a given trailer. A graph of this type is shown in Fig. 13 for a hypothetical 3000 lb trailer at two possible damping ratio criteria. For a given tow car weight, and corresponding W_c/W_t, this graph specifies the minimum hitch load that will meet the specified damping ratio criterion. With this trailer, for example, the hitch load must be at least 12 percent if the tow car and trailer are of equal weight and 0.2 damping is specified. Different trailers would have a different boundary location due to differences in wheelbase, inertia, tires, etc.

It should be mentioned at this point that increasing hitch load has a destabilizing effect on the tow car in terms of loss of understeer (Refs. 2 and 3). The tow vehicle directional control test discussed in the next section will put another criterion line on Fig. 13 such as to limit the maximum allowable hitch load as a function of two-car/trailer weight. That is, both the tow car and trailer directional (composite) handling requirements can be in the form of allowable hitch load versus weight ratio.

B. FULL-SCALE TEST RESULTS

Pulse steer tests were performed with the three tow cars and eight trailers described in Section II. This test procedure was developed in Ref. 2 to excite the trailer swing mode. Briefly, the procedure requires the combination-vehicle to be driven in a straight line at 55 mph with a 45-90 deg steering wheel "pulse" input applied as rapidly as possible. When recentered, the wheel is held tightly fixed to avoid steering force

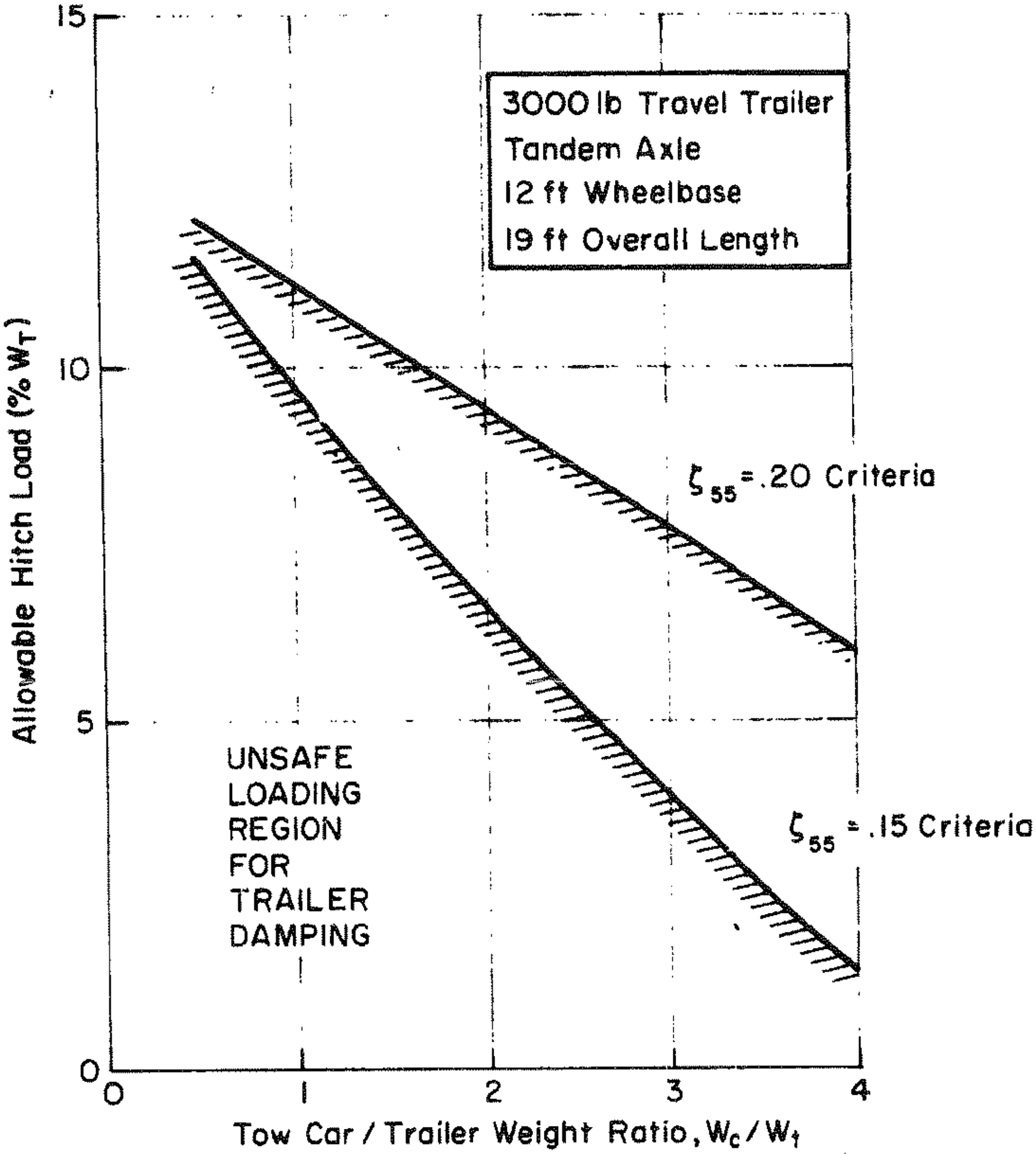


Figure 13. Potential Combination-Vehicle Trailer Damping Requirement

feedback from the trailer. The resulting trailer mode oscillation can be modeled by a damped second-order system response such as shown in Fig. 14.

The frequency and damping ratio of the resulting trailer articulation angle trace can be derived from the ratio of successive peak amplitudes, cycles to half amplitude, or other graphical means. In this program the articulation angle response was fitted (using a least-square technique) to that produced by an ideal second-order system. The identification interval begins 1 sec after the steering wheel angle is recentered and continues for several seconds. Time histories and trailer mode identification results for every test configuration are presented in Appendix E of Vol. III for reference.

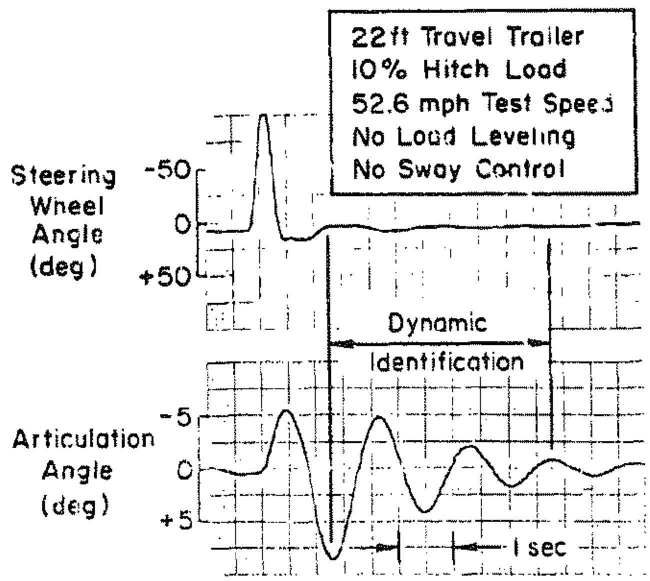


Figure 14. Example Steering Wheel and Trailer Time History for Pulse Steer Test Procedure

Two normalizations must be applied to the derived damping ratios in order to provide comparable results. These compensate for off-nominal speeds and for different load leveling values. For example, in many cases the actual speed at pulse application was not exactly 55 mph. In other cases the trailer was already unstable at a speed below 55 mph. Damping ratios at these off-nominal conditions were adjusted to the 55 mph reference speed using the simulation model of Ref. 3. These adjustments followed the curves shown in Fig. 15. One full-scale test configuration (also shown in Fig. 15) was run at several speeds to verify the damping versus speed slope.

The second adjustment was necessary to compensate for off-nominal values of load leveling. In general, the test plan was designed to cover three values of load leveling: none; minimum (as recommended in Ref. 3); and 25 percent (as recommended by hitch manufacturers). Minimum load leveling corresponded to using air shocks as much as possible and then

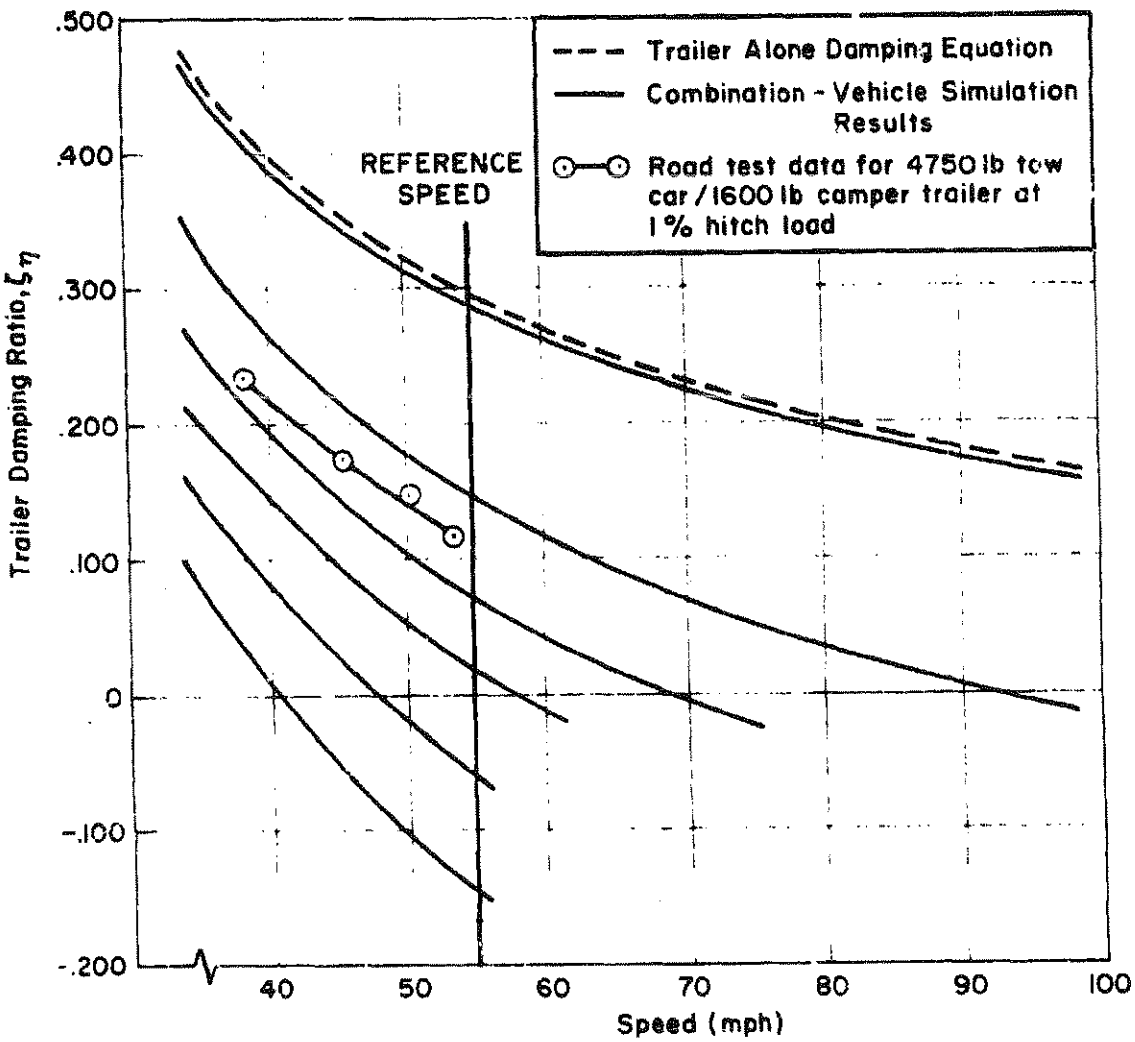


Figure 15. Trailer Damping Speed Adjustment Curves

applying only enough load leveling torque to relevel the combination-vehicle. Twenty-five percent load leveling corresponded to applying load leveling such as to transfer 25 percent of the hitch load to the tow car front axle. In most cases this also corresponded to releveling the combination-vehicle. However, since load leveling torque is controlled via changes in chain length, which is limited to minimum one-half link increments, the 25 percent desired could not always be achieved precisely. The actual percent load leveling values ranged from 15 percent to 36 percent for the nominal 25 percent setup.

Since it was determined in Ref. 3 that load leveling torque affects trailer damping of the combination-vehicle, it was necessary to apply a correction factor to normalize the unequal test conditions. To derive this correction factor it was necessary to convert the measured change in vertical tire loads to load leveling torque using the equations developed in Ref. 3. This procedure was accomplished for the five combination-vehicle configurations in which load leveling was varied. The results are shown in Fig. 16. From this plot the average change in damping ratio with load leveling torque was found to be 0.061 units per 1000 ft-lb. Since all combination-vehicle configurations were weighed and measured in the test program, the load leveling torque could be computed and the correction factor applied.

Final trailer damping data, corrected for speed and load leveling variations, are given in Table 15. This data table provides the basis for all subsequent discussions and comparisons. As noted earlier, the raw trailer damping data are presented in Appendix E of Vol. III for reference, or for application of additional analysis techniques.

The results presented in Table 15 can be plotted in various ways to illustrate the effects of the significant parameters. Primarily, these reflect the previously described effects, i.e., hitch load, trailer type, and tow car size, on combination-vehicle damping ratio. For example, Fig. 17 shows the damping ratio (at 55 mph) for each trailer at a constant hitch load of 5 percent. The order of trailer type presented along the abscissa generally reflects a decrease in damping when going from

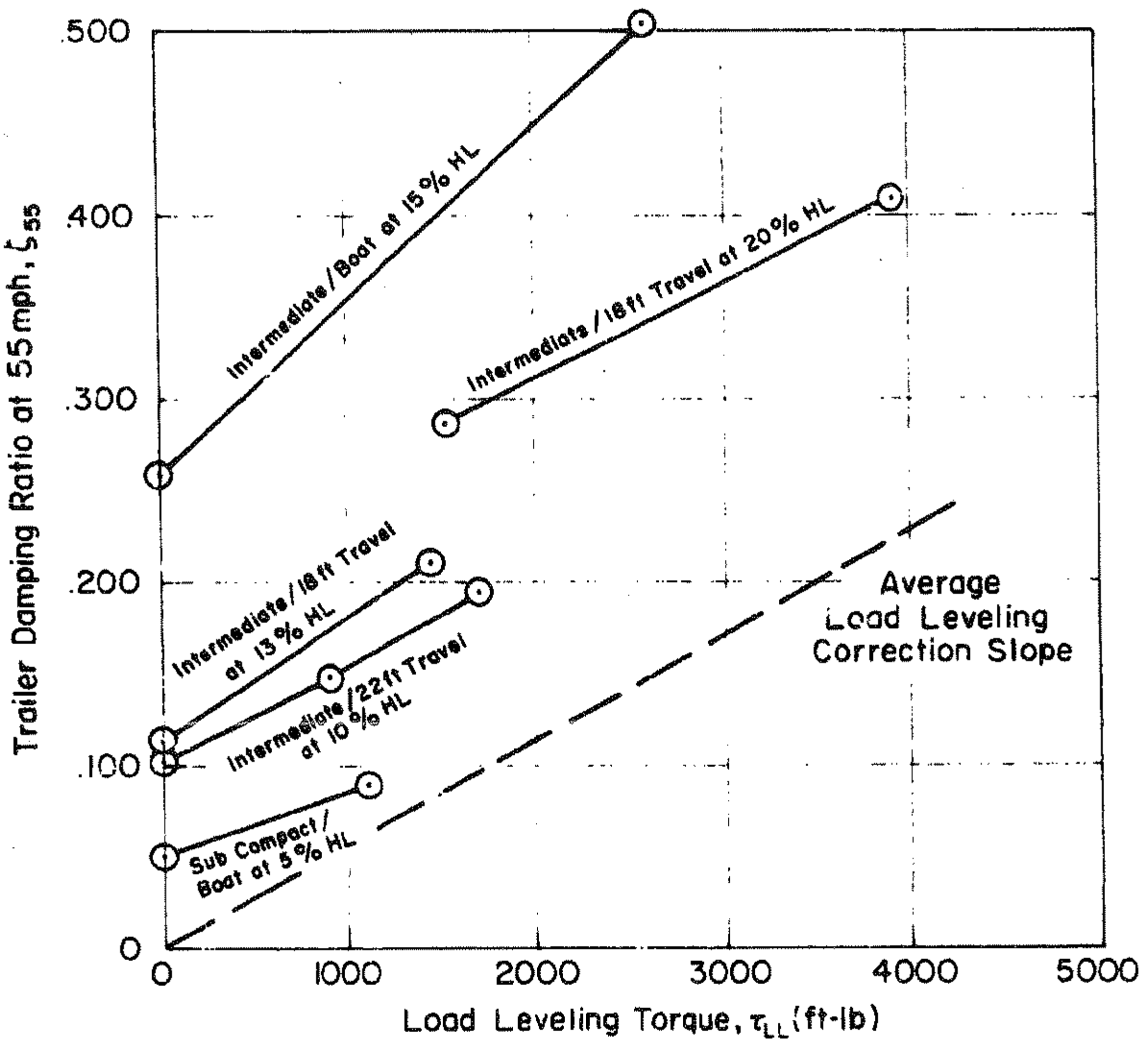


Figure 16. Effect of Load Leveling Torque on Trailer Damping

Reproduced from best available copy.

TABLE 15. TRAILER DAMPING RATIO AT 55 MPH

TRAILER	TEST WEIGHT (lb)	TOW CARS[a]	HITCH LOAD/LOAD LEVELING[b] 0	2-1/2	5 NONE	5 TEST	5 25%	7-1/2	10 NONE	10 TEST	10 25%	12-1/2	15 NONE	15 TEST	15 25%	17-1/2	20
Utility	1500	I	.39[illegible]		.135				.195								[illegible]
		C	[illegible]		.150				.250								
		SC	[illegible]	.390	.120			.165	.190								
Camper	1600	I	.1[illegible]5		.120				.190				.265				.28[illegible]
		C	[illegible]		.080				.160								
		SC	[illegible]		.040				.135								
18 ft Travel	3900	I			−.055		.006^c		−.020^c	.095	.104^c						.410 (25)
		C			−.120^c	−.040	−.163^c		.050^c	.190	.174^c		.185^c	.370	.365^c		
		SC			−.20^c	−.115	−.146^c	.010(7)	.040^c	.170	.144^c		.180^c	.370	.337^c		
	3420	SC			−.130			−.030								Too Heavy	
	3000	I										.111 .210 (25)					
19 ft Travel	4000	I			−.100		−.036^c		.030		.158^c		.058^c	.229	.250^c		
		C			−.20^c	−.150	−.135^c		−.060^c	.050	.066^c						
		SC			−.115		−.0[illegible]^c		−.020^c	.075	.[illegible]7^c					Too Heavy	
22 ft Travel	4000	I	−.055		.050		.116^c		.105	.195	.239^c						.510 (2[illegible])
		C			−.065		.000^c	.050	.086^c	.135	.216^c	.470 (−6)	.394^c	.510	.590^c	.710 (4)	
		SC			−.050		.007^c	−.020	.090^c	.180	.200^c					Too Heavy	
27 ft Travel	6000	I			−.080		.026^c		.000^c	.220	.211^c			.350	.303^c		
		C			−.190^c	−.100	−.090^c		.230^c	.410	.434^c					Too Heavy	
20 ft Boat	3000	I	.085		.130		.182^c		Data unreliable				.260	.504	.415^c		
		C	−.020		.030		.080^c		.100		.200^c		.230^c	.415	.380^c		
		SC	−.030	.035	.050	.090	.094^c	−.190(0)	.280^c	.390	.369^c		.135^c	.560	.[illegible]66^c		
Horse	5800	I							.113				.275				
	4000	C			−.080				.230								
		SC			−.130				.165							Too Heavy	

[a]Tow cars: I = Intermediate (4750 lb); C = Compact (4100 lb); SC = Subcompact (3400 lb).
[b]Hitch load (HL) = percent trailer weight; load leveling = percent HL transferred to tow car front axle. Values at nonstandard LL given in parentheses.
[c]Normalized point from measured data.

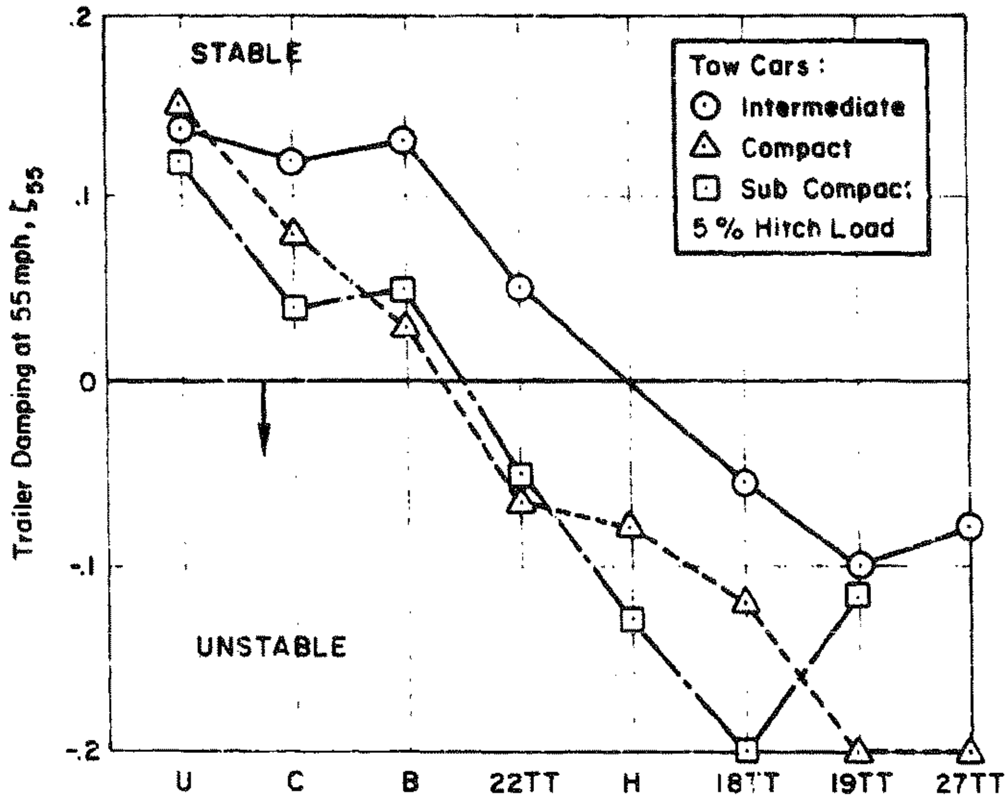


Figure 17. Trailer Damping at 55 mph for All Combination-Vehicles at 5 Percent Hitch Load

left to right, although the trends for each tow car are not necessarily the same. The main point to be derived from Fig. 17 is that trailer damping can easily vary from +0.15 (stable) to −0.20 (unstable) for this selection of trailers at 5 percent hitch load. Second, heavier tow cars appear to provide higher damping that the lighter tow cars.

If hitch load is increased, all trailers exhibit increased damping. Figure 18 shows the same ordering of trailers as in Fig. 17, and it can be seen that damping now ranges from +0.28 (stable) to only −0.05 (unstable), or an average positive damping increment of +0.15.

Due to the large effect of hitch load on trailer damping it is informative to graphically present this effect for each trailer individually.

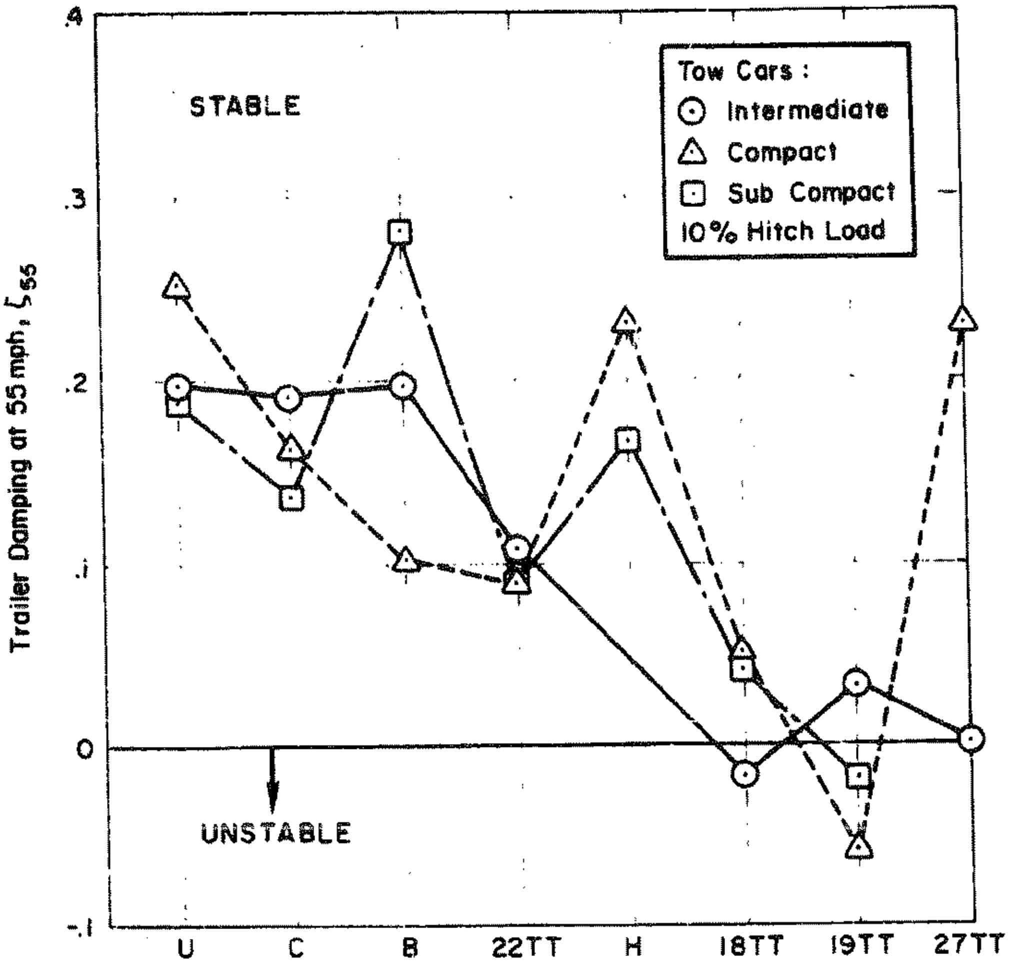


Figure 18. Trailer Damping at 55 mph for All Combination-Vehicles at 10 Percent Hitch Load

These plots are shown in Figs. 19-26 for the eight trailers, five with and without load leveling. Review of these figures shows the importance of hitch load and load leveling in providing a stable trailer mode. In particular, it is clear why manufacturers recommend hitch loads between 10 and 15 percent trailer weight.

Additional damping test results related to the trailer inertia, sway damper, non-zero lateral acceleration initial conditions, and steering free play are given in Table 16. The test results presented in Table 16 can be verbalized as follows:

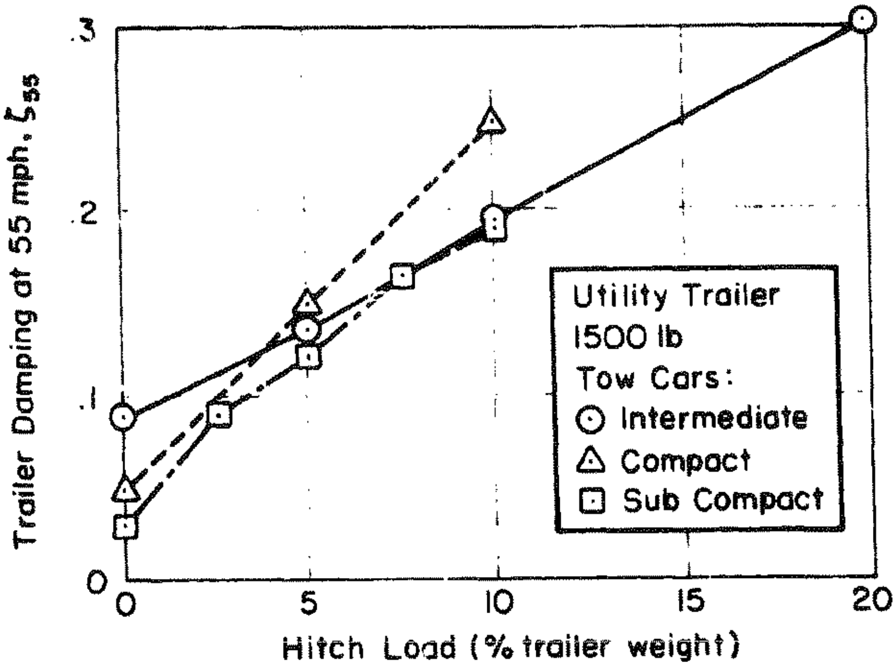


Figure 19. Trailer Damping as a Function of Hitch Load for 1500 lb Utility Trailer

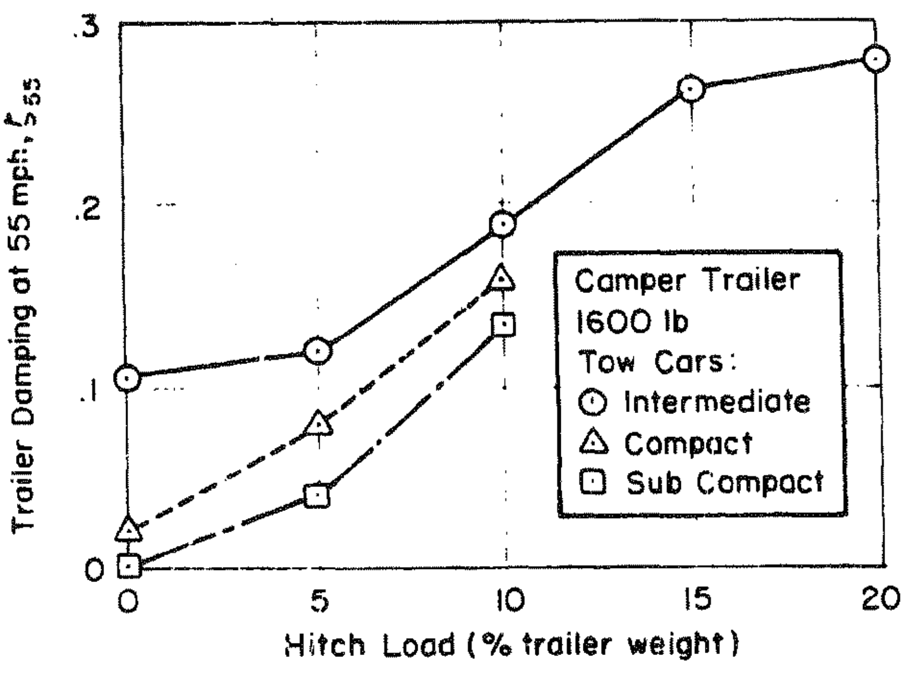


Figure 20. Trailer Damping as a Function of Hitch Load for 1600 lb Camper Trailer

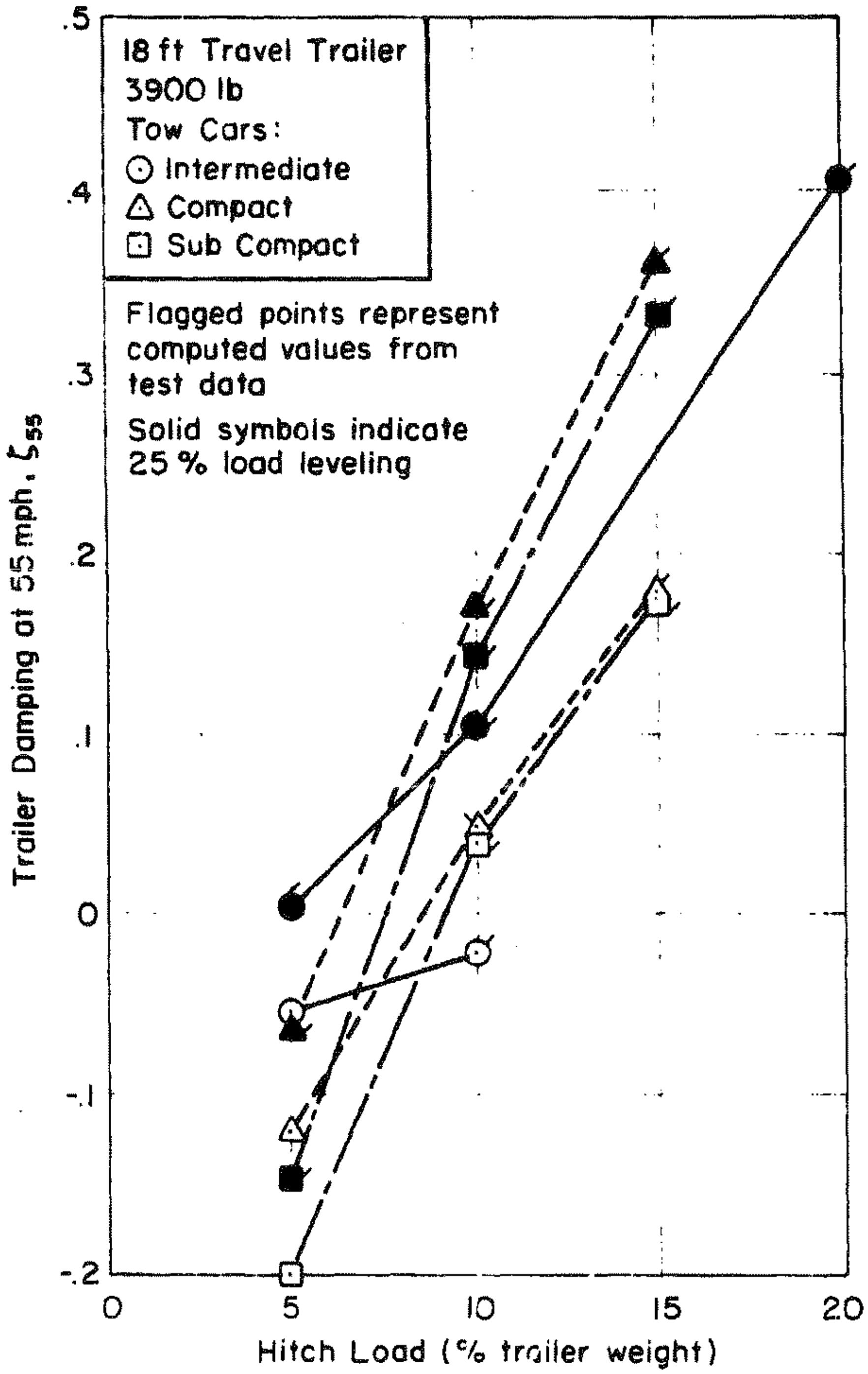


Figure 21. Trailer Damping as a Function of Hitch Load With and Without Load Leveling for 18 ft Travel Trailer

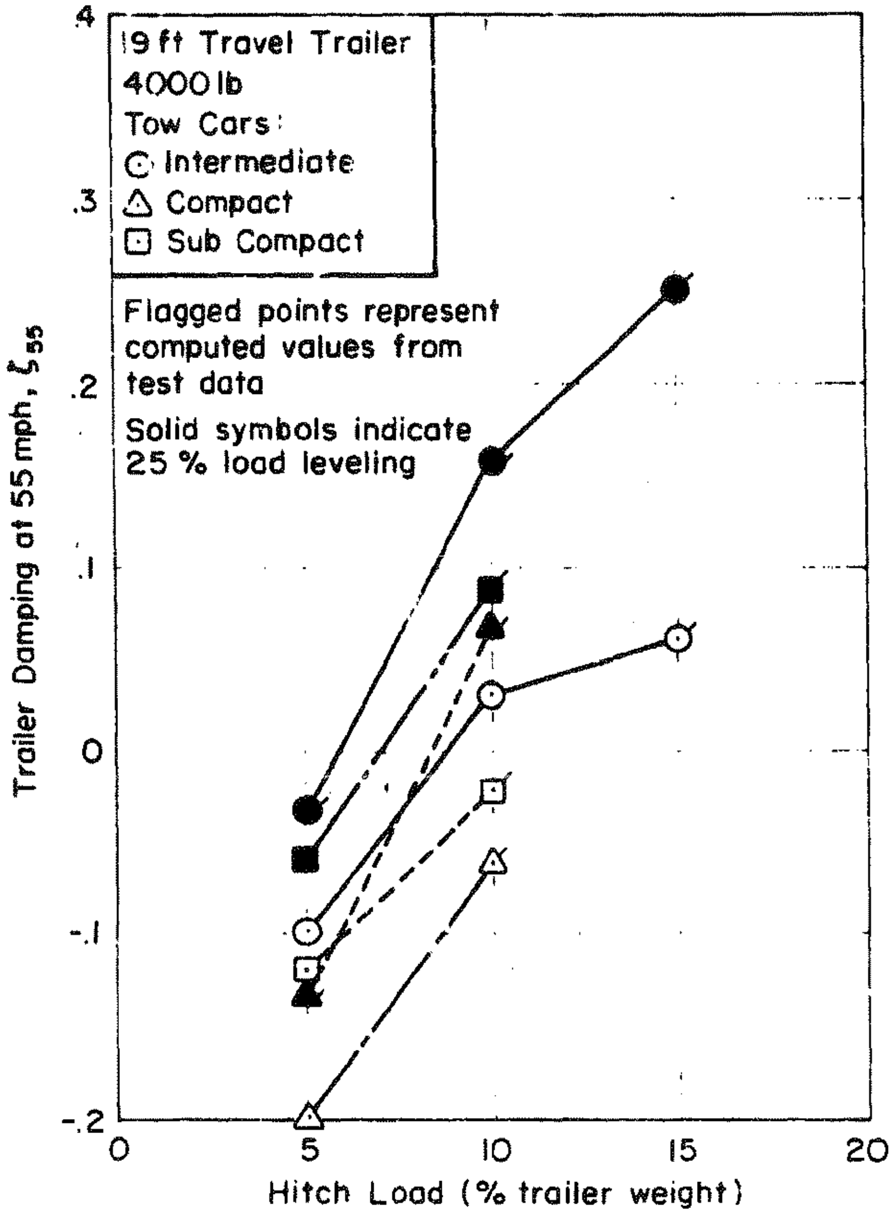


Figure ??. Trailer Damping as a Function of Hitch Load With and Without Load Leveling for 18 ft Travel Trailer

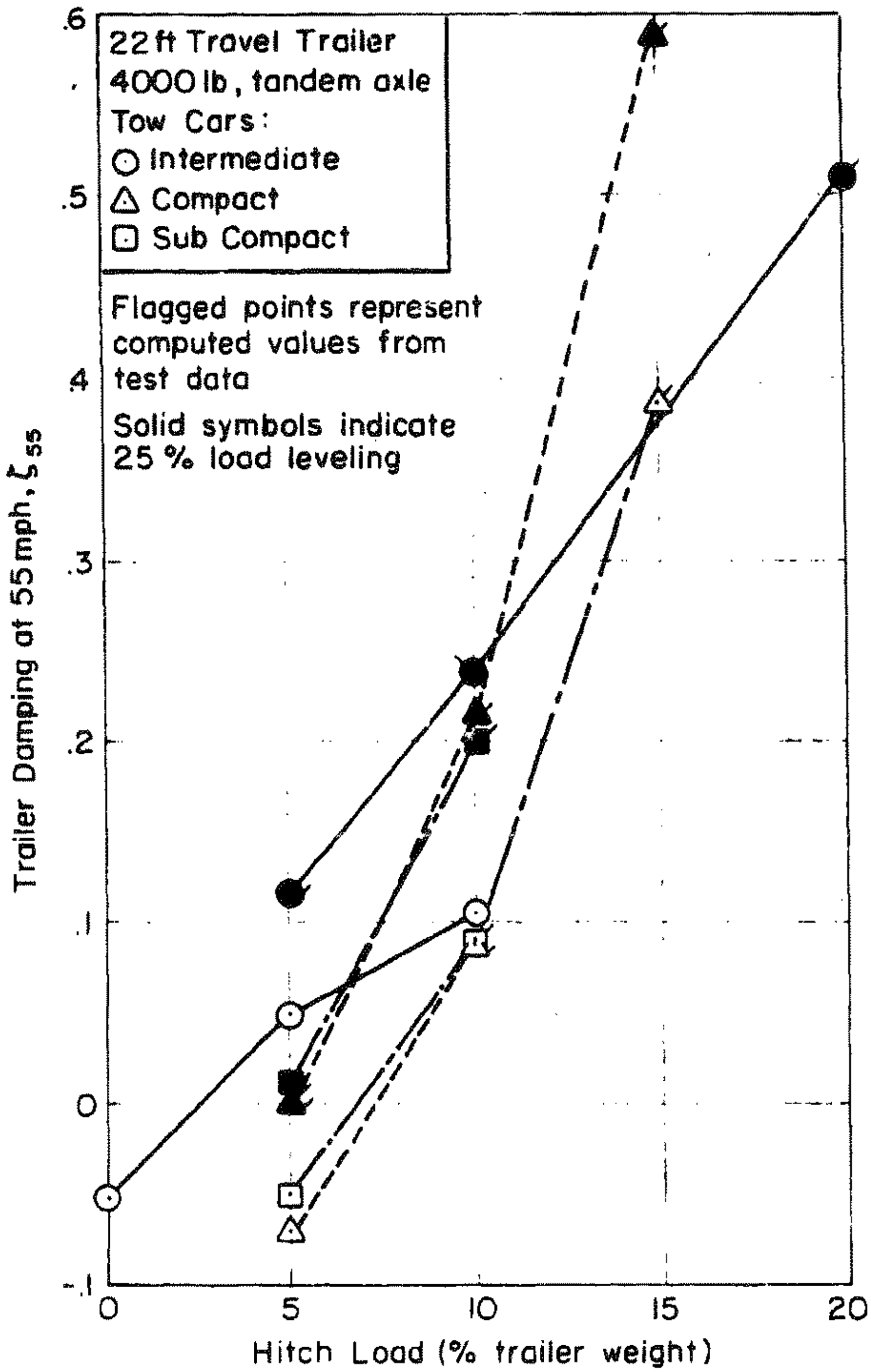


Figure 23. Trailer Damping as a Function of Hitch Load With and Without Load Leveling for 22 ft Travel Trailer

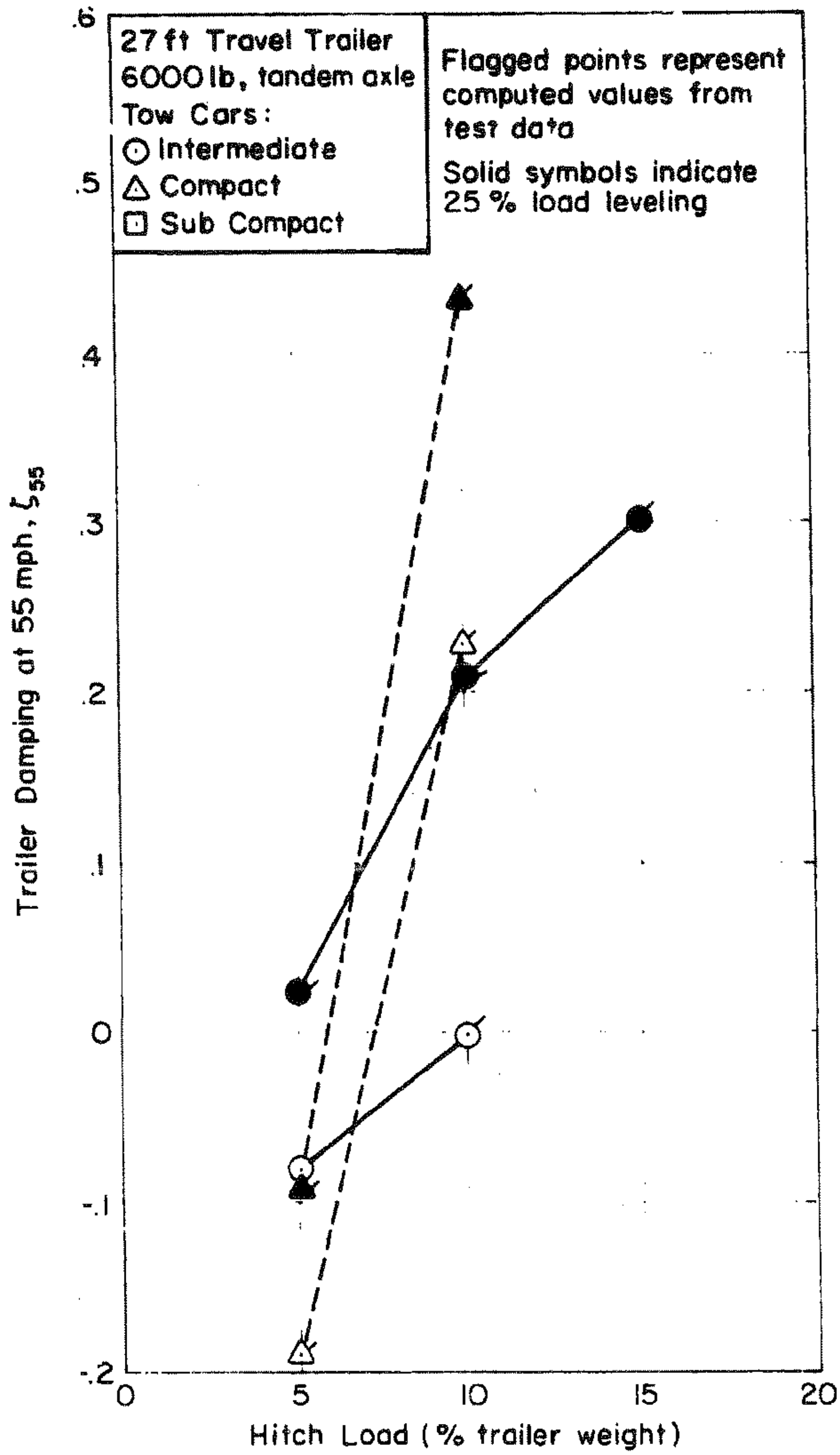


Figure 24. Trailer Damping as a Function of Hitch Load With and Without Load Leveling for 27 ft Travel Trailer

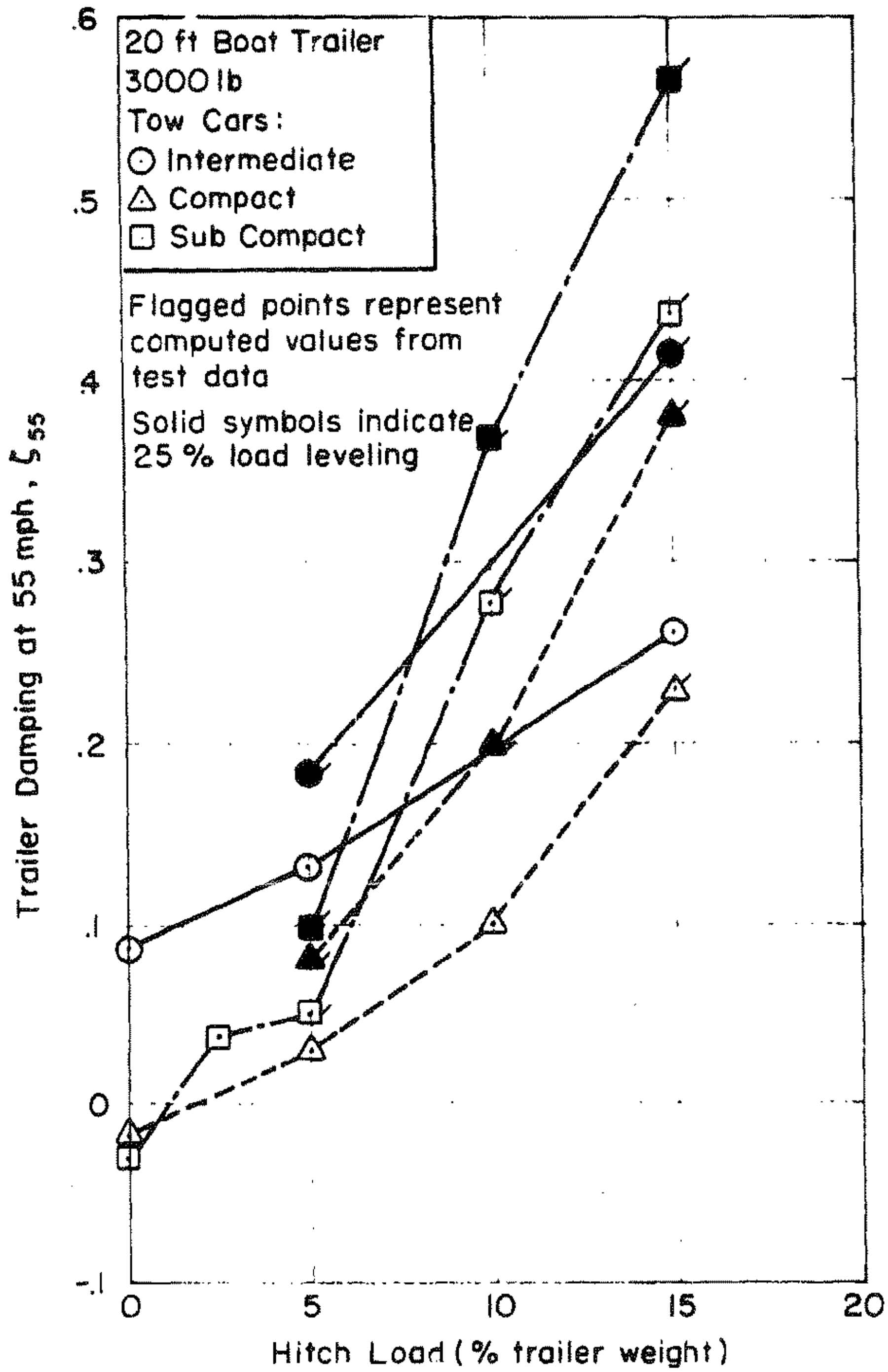


Figure 25. Trailer Damping as a Function of Hitch Load With and Without Load Leveling for 20 ft Boat Trailer

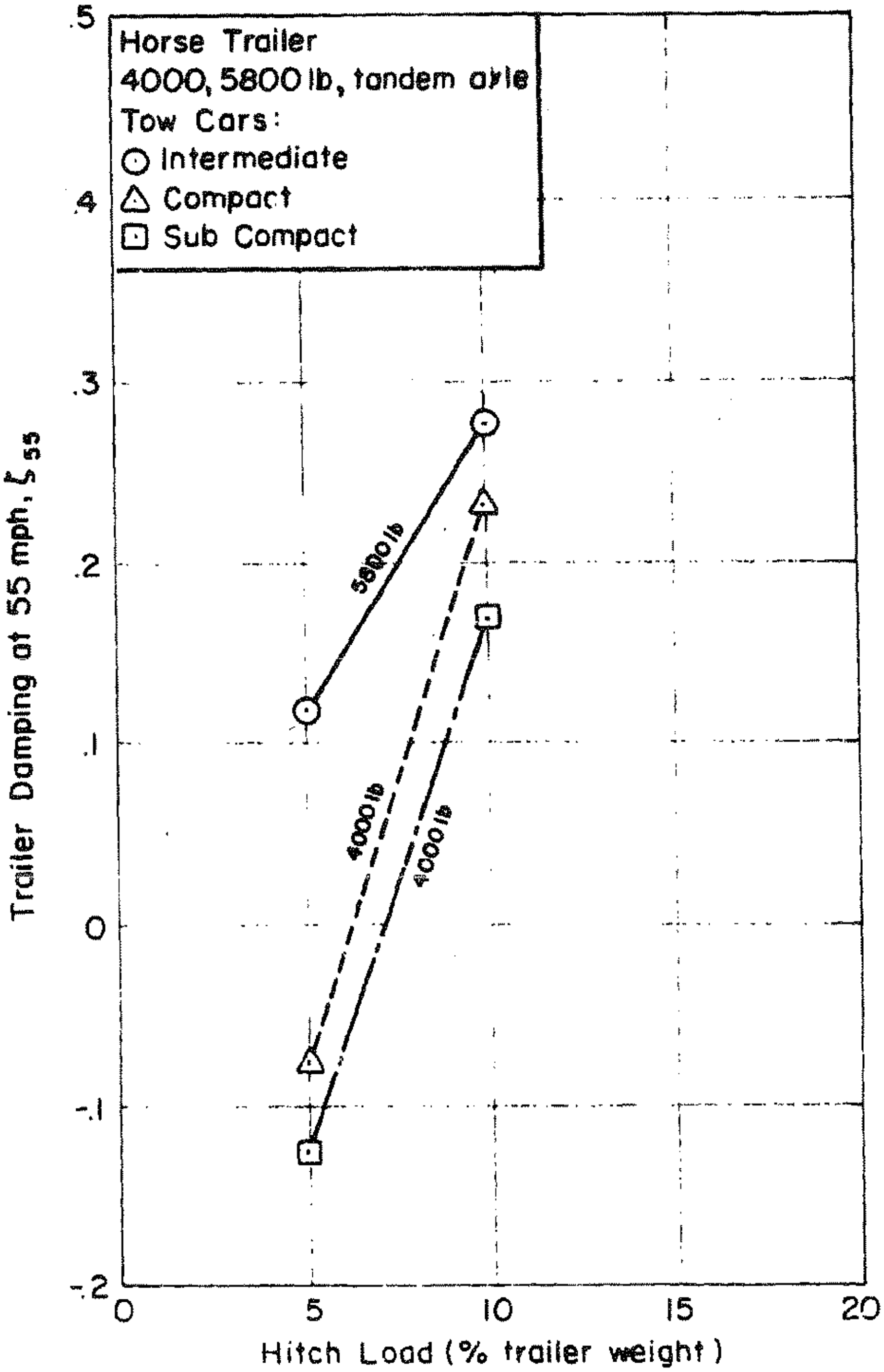


Figure 26. Trailer Damping as a Function of Hitch Load for Horse Trailer

TABLE 16. ADDITIONAL TRAILER DAMPING TESTS

TRAILER	TOW CAR	HITCH LOAD (%)	LOAD LEVELING (%)	TEST PURPOSE	MEASURED TRAILER DAMPING RATIO/ TEST SPEED (mph)	TRAILER DAMPING RATIO CORRECTED TO 55 MPH, ζ_{55}
Utility	Inter-mediate	20	None (air shocks)	Trailer-alone damping, ζ_{ta}	.345/58	.298
	Sub-compact	5	None (air shocks)	Inertia change at constant weight	.120/55	.120 with I_{to} = 321 slug-ft^2
					.060/55	.060 with I_{to} = 504 slug-ft^2
Camper	Inter-mediate	1	None	Speed variation	.232/38 .174/46 .149/50 .116/53	.110
22 ft Travel	Inter-mediate	10	None (air shocks)	Non-zero lateral acceleration (constant radius turn test)	.290/35 at $a_y \doteq 0$	.11 at $a_y \doteq 0$
					.00/35 at $a_y \doteq .42$ g	−.200 at $a_y \doteq .42$ g
18 ft Travel	Inter-mediate	5	None	Non-zero lateral acceleration	.00/50 at $a_y \doteq 0$	−.04 at $a_y \doteq 0$
					.00/38 at $a_y \doteq .48$ g	−.16 at $a_y = .48$ g
	Sub-compact	15	24	Steering fixed versus steering free	.370/55	.370 Fixed steering
					−.040/55	−.040 Free steering
27 ft Travel	Inter-mediate	5	None (air shocks)	Friction sway damper	.00/46	−.040 No sway damper
					.140/50	.105 Friction damper
		5	None (air shocks)	Dynamic braking sway damper	Speed limited by automatic brake application at 45 mph	Not possible to achieve 55 mph

1) Trailer-alone damping measured in full scale by setting the trailer c.g. at the center of percussion and applying a side force at the tire (discussed in Appendix E, Vol. III) checks very closely with the analytical expression given by Eq. 6.

2) Increasing trailer moment of inertia about the c.g. (separating load front and rear) decreases trailer damping in the predicted way.

3) There is a strong decrease in damping with increasing speed. Again, this effect was predicted by simplified analysis (Fig. 15).

4) Higher cornering (lateral acceleration) levels significantly reduce trailer damping. For example, at 35 mph the 22 ft travel trailer at 10 percent hitch load had damping of 0.29 at low lateral acceleration. This was reduced to zero damping at 35 mph at a lateral acceleration of 0.4 g. This effect is attributable to the loss of cornering stiffness of the trailer tires at high slip angles. This produces a loss in damping according to the trailer-alone damping equation, Eq. 6.

5) Steering free play can have a significant effect on reducing the trailer damping. For example, steering-fixed versus steering-free reduced damping from 0.37 to −0.04 at 55 mph. This effect has strong implications for tow cars with excessive steering free play and/or for drivers who allow the steering wheel force feedback to move the wheel. This in turn amplifies the trailer swing. Locking the wheel fixed is the best procedure.

6) Friction sway dampers can significantly improve trailer damping. At 55 mph the 27 ft travel trailer at 5 percent hitch load was unstable ($\zeta = -0.08$) without the sway damper and stable ($\zeta = 0.11$) with the damper. The electric brake type of damper acted primarily as a speed control device by limiting speed to that for zero damping.

The next subsection compares the reduced data in Table 15 with the analytical predictions given previously in order to develop a minimum hitch load boundary that will insure meeting specific trailer damping criteria.

C. COMPARISON WITH ANALYTICAL PREDICTIONS

The results presented in the previous subsection were designed to show the trends in trailer damping as a function of hitch load and tow car size. In general, these results followed predictions, although there were many cases in which the damping did not show the anticipated change with tow car size, especially for the larger trailers at heavy hitch loads being towed by the smaller cars. Also, the effect of load leveling was not included in the analytical considerations. Therefore, the purpose of data analysis at this point was to compare predicted results (as would be available from a trailer manufacturer) with measured results to see if our approach to a trailer damping standard is sufficiently accurate for practical use.

The first step in this analysis was to derive the trailer geometric, inertia, and tire properties necessary to solve Eq. 11. For example, lengths and weights were available from measurements made during the test program (see Vol. III, Appendix A); tire properties were derived from Refs. 18 and 19; and the moments of inertia were derived using the procedure discussed in Subsection E. The "hookup factor" can then be derived for various hitch loads and adjusted for roll steer if the trailer has a tandem axle. This tandem axle effect amounts to a reduction of 0.033 units in hookup factor from that derived without roll steer (see Vol. III, Appendix A). For reference, these parameters are given in Table 17.

The only unknown quantity remaining in the damping equation is the tow car sensitivity, K_s, which must be determined empirically. This was done by selecting a K_s that minimized the error between the predicted and the measured values of trailer damping and was a function of tow car and trailer parameters. Two physically justifiable quantities were fitted:

Simple trailer to tow car weight ratio	More complex yaw moment relationship
$C_1 \dfrac{W_t}{W_c}$	$C_2 \dfrac{(\ell_2 + \ell_h)}{\ell_1} \dfrac{Y_{\alpha_3}}{Y_{\alpha_2}}$

where

ℓ_1	Tow car wheelbase
ℓ_2	Hitch to trailer axle distance
ℓ_h	Tow car rear axle to hitch distance
Y_{α_2}	Tow car rear tire cornering stiffness
Y_{α_3}	Trailer tire cornering stiffness

TABLE 17. TRAILER PARAMETERS

TRAILER	TEST WEIGHT (lb)	ℓ_2 [a] (in.)	Y_{α_3} [b] (lb/deg)	0% HITCH LOAD I_{t_o} (slug-ft^2)	0% HITCH LOAD I_{t_h} (slug-ft^2)	0% HITCH LOAD HOOKUP FACTOR	0% HITCH LOAD ζ_{TA}	5% HITCH LOAD I_{t_o} (slug-ft^2)	5% HITCH LOAD I_{t_h} (slug-ft^2)	5% HITCH LOAD HOOKUP FACTOR	5% HITCH LOAD ζ_{TA}	10% HITCH LOAD I_{t_o} (slug-ft^2)	10% HITCH LOAD I_{t_h} (slug-ft^2)	10% HITCH LOAD HOOKUP FACTOR	10% HITCH LOAD ζ_{TA}
Utility	1500	92	121	382	3,127	0.122	0.280	378	2,848	0.087	0.293	394	2,614	0.056	0.306
Camper	1600	124.5	96	872	6,232	0.140	0.276	1,002	5,828	0.129	0.288	874	5,199	0.075	0.305
18 ft Travel	3900	125	218	Not Tested				2,889	17,089	0.126	0.292	2,889	15,640	0.094	0.305
19 ft Travel	4000	142	175	Not Tested				2,986	18,686	0.116	0.265	2,986	17,086	0.083	0.277
22 ft Travel	4000	[illegible]	250 (Two tires)	4,456	27,656	0.128[c]	0.313	4,310	25,310	0.094[c]	0.337	4,310	23,110	0.063[c]	0.353
27 ft Travel	6000	203	327 (Two tires)	Not Tested				9,940	57,940	0.103[c]	0.348	9,000	52,000	0.056[c]	0.367
Boat	3000	163	147	2,888	24,588	0.118	0.306	3,255	22,855	0.098	0.320	3,255	20,77[illegible]	0.063	0.336
Horse	4000	93	250 (Two tires)	Not Tested				1,081	7,821	0.093	0.258	1,081	7,096	0.058	0.271

[a] Hitch to trailer axle distance.

[b] Trailer tire cornering stiffness (tires on one side only).

[c] Adjusted for tandem axle roll steer.

The best fit was obtained with the simple relationship and $C_1 = 3.68$. This constant was determined by rearranging Eq. 11 and solving for the value of C_1 that produced the measured result, i.e.,

$$C_1 = \left[\frac{\zeta_{ta} - \zeta_{cv}}{HUF}\right] \frac{W_c}{W_t} \qquad (12)$$

In summary, the best fit analytical expression for trailer damping is given by:

$$\zeta_{cv} = \zeta_{ta} - (HUF)3.7\,\frac{W_t}{W_c} \qquad (13)$$

where trailer-alone damping ratio, ζ_{ta}, and the hookup factor, HUF, are functions of hitch load. Figures 27-34 illustrate the trailer damping ratios obtained with this analytically based expression and comparison with the measured data. Considering all trailers and hitch loads together, the mean damping ratio error was −0.011 (analytical lower than measured).

Possible sources of error in computing the damping ratio arise from the following:

1) Selection of trailer tire cornering stiffness, Y_{α_3}, for derivation of trailer-alone damping. Since same and model tire data was not solicited from manufacturers, an estimate was made from Refs. 18 and 19 for tires of a similar size.

2) Tow car trailer interaction when the natural frequency of the trailer mode becomes equal to or greater than the natural frequency of the tow car's directional mode. When this occurs (primarily with tow car to trailer weight ratios less than one at the higher hitch loads), the measured trailer damping ratio will be much higher than predicted. Although these high damping ratio conditions are not of interest when looking for minimum limits, they were included in the error analysis.

3) Measurements of roll steer gradients for tandem axle trailers. This affects the corrections to the hookup factor for these trailers.

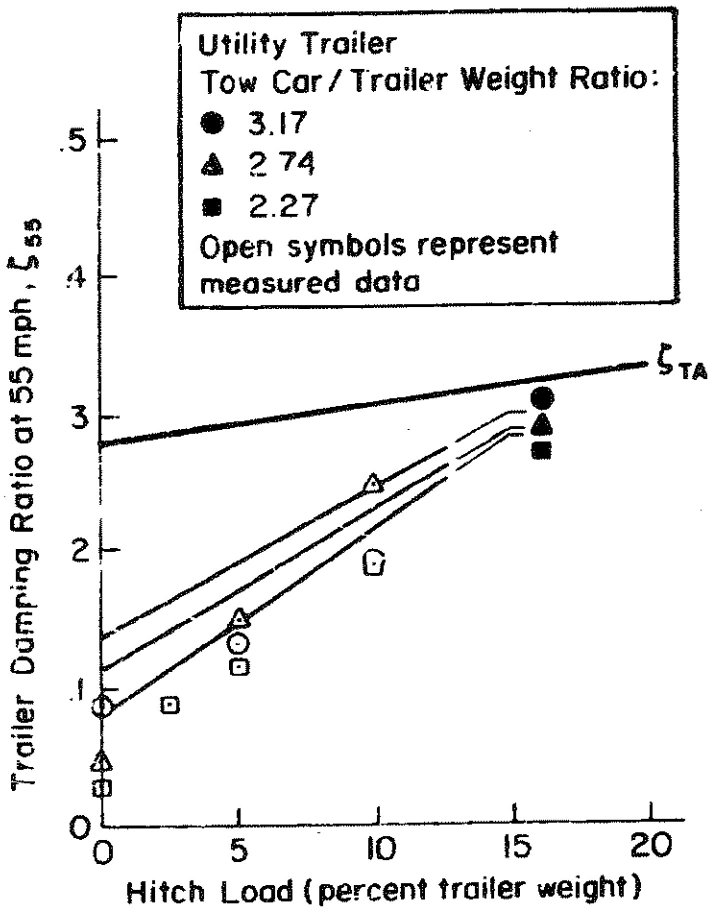


Figure 27. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for Utility Trailer

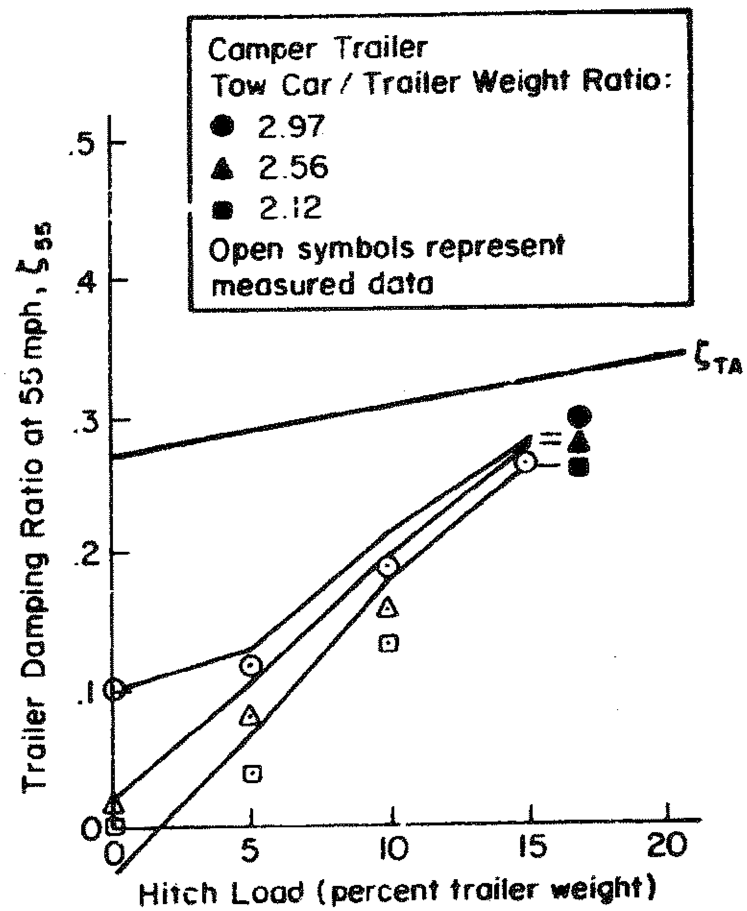


Figure 28. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for Camper Trailer

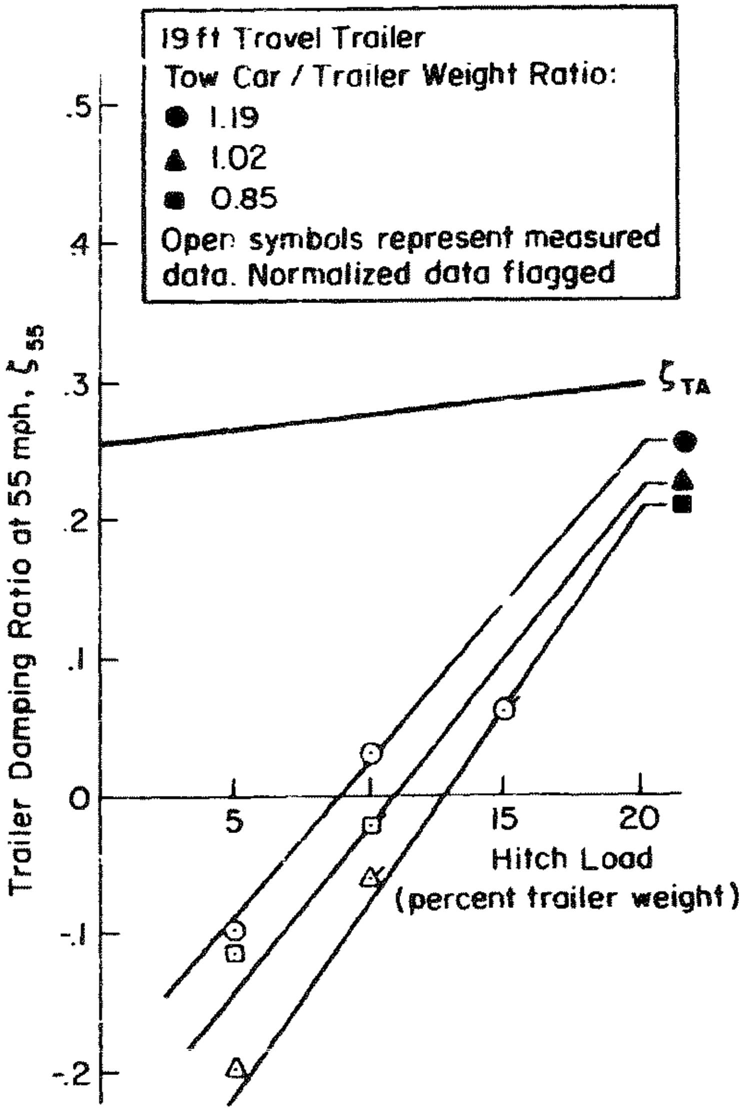


Figure 29. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for 18 ft Travel Trailer

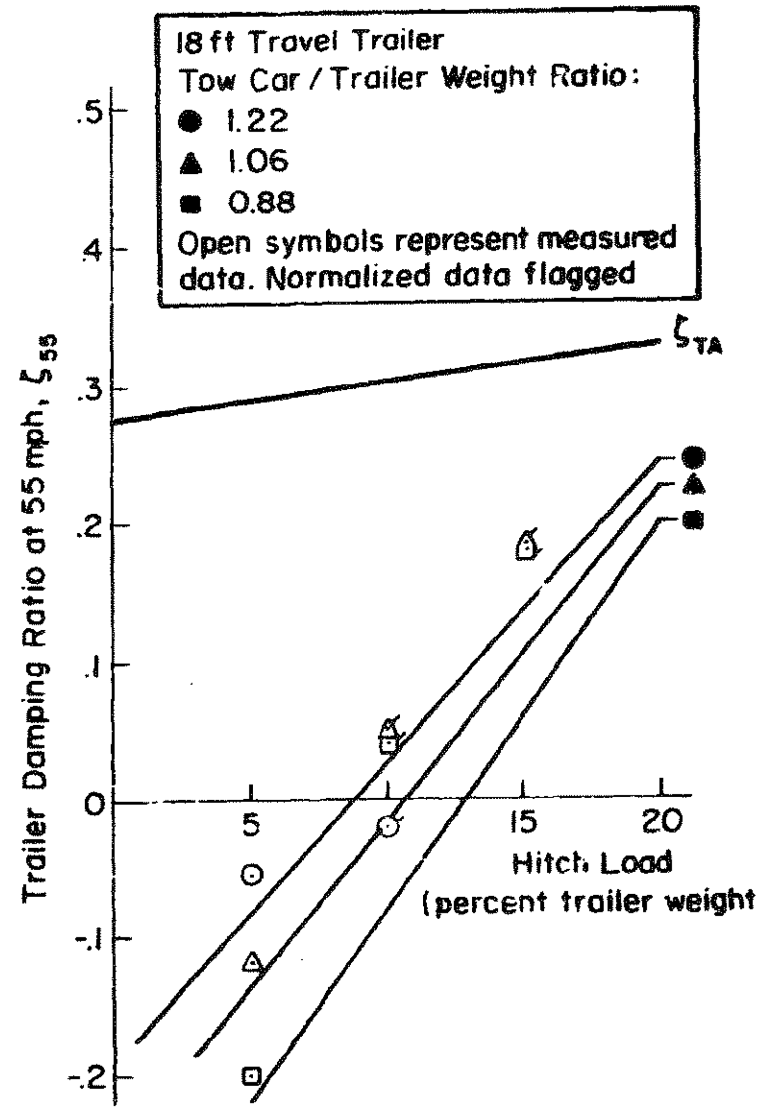


Figure 30. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for 19 ft Travel Trailer

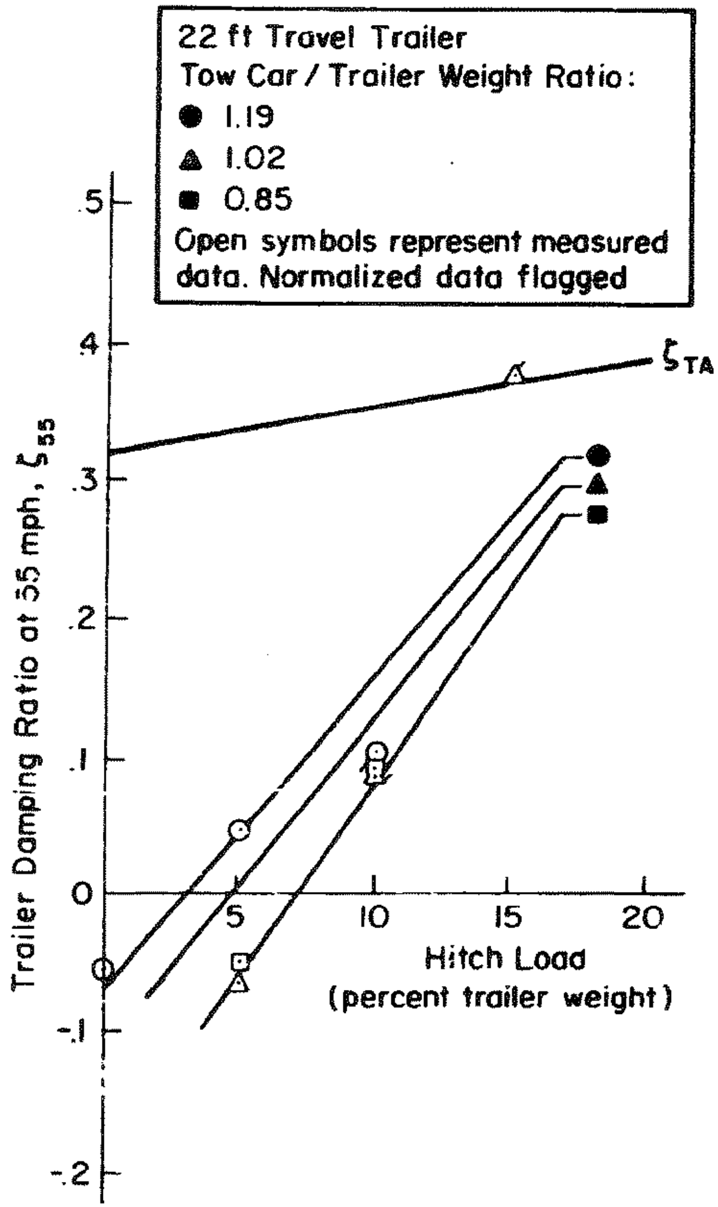


Figure 31. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for 22 ft Travel Trailer

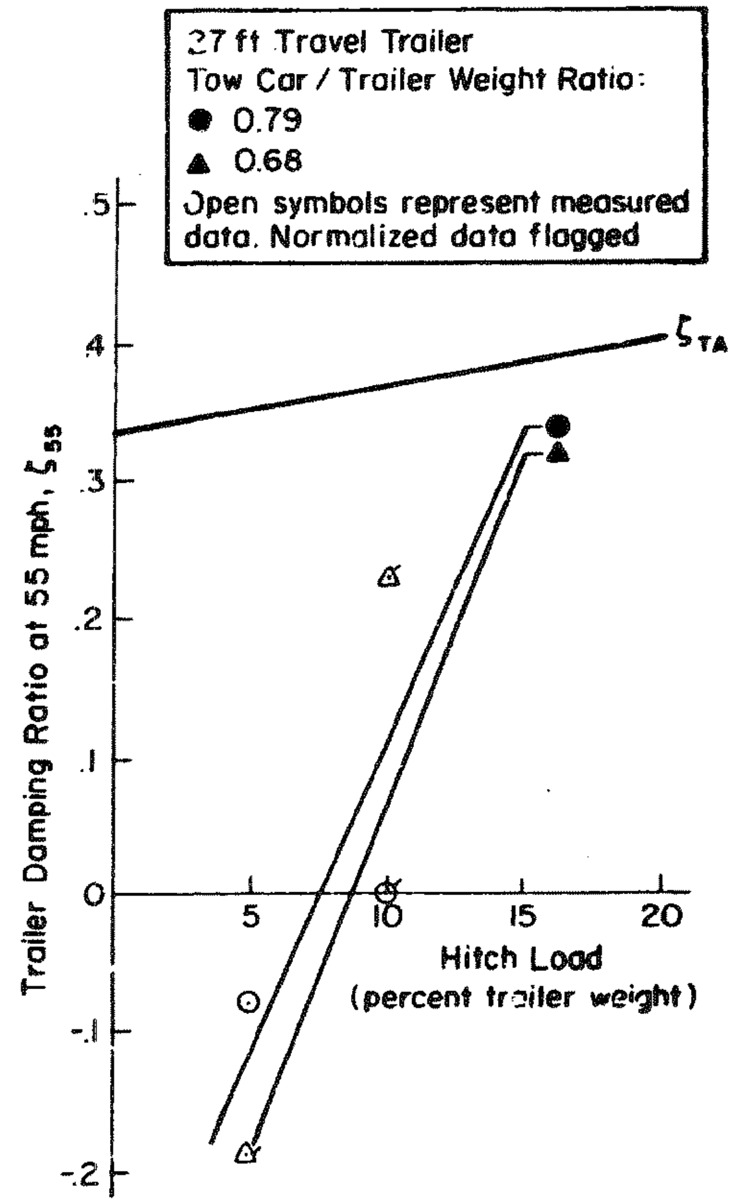


Figure 32. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for 27 ft Travel Trailer

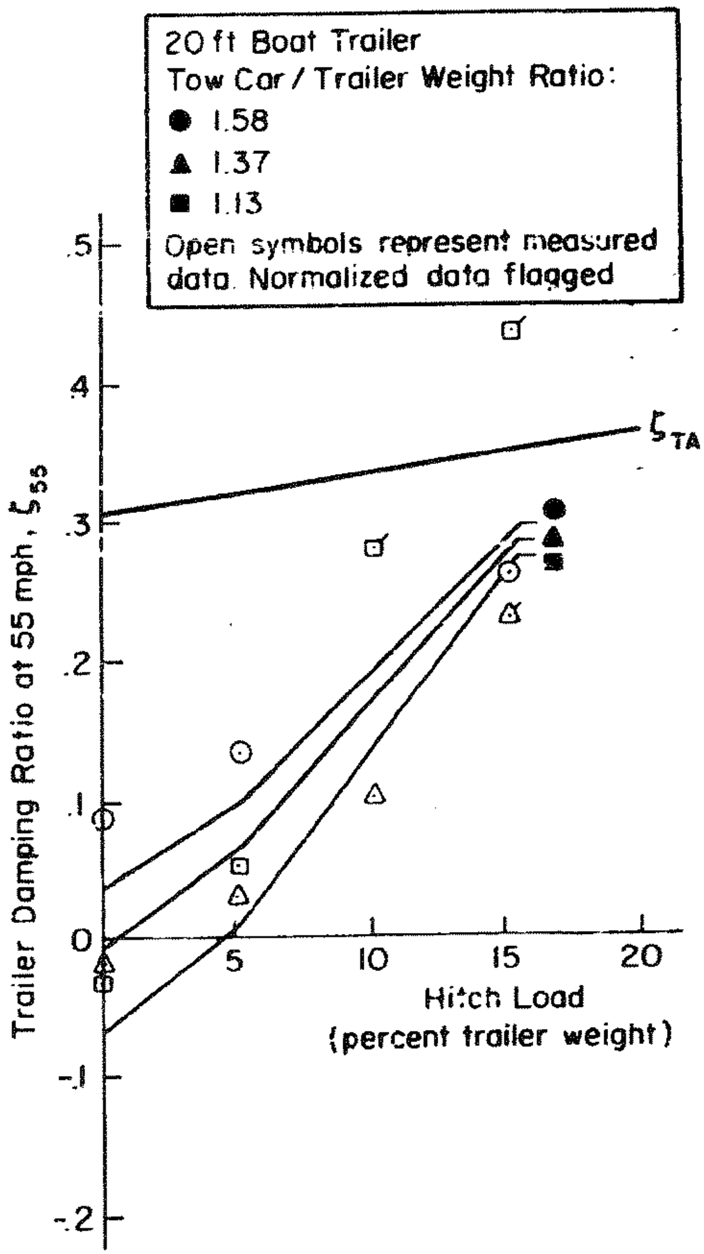


Figure 33. Comparison of Predicted and Measures Combination-Vehicle Trailer Damping for Boat Trailer

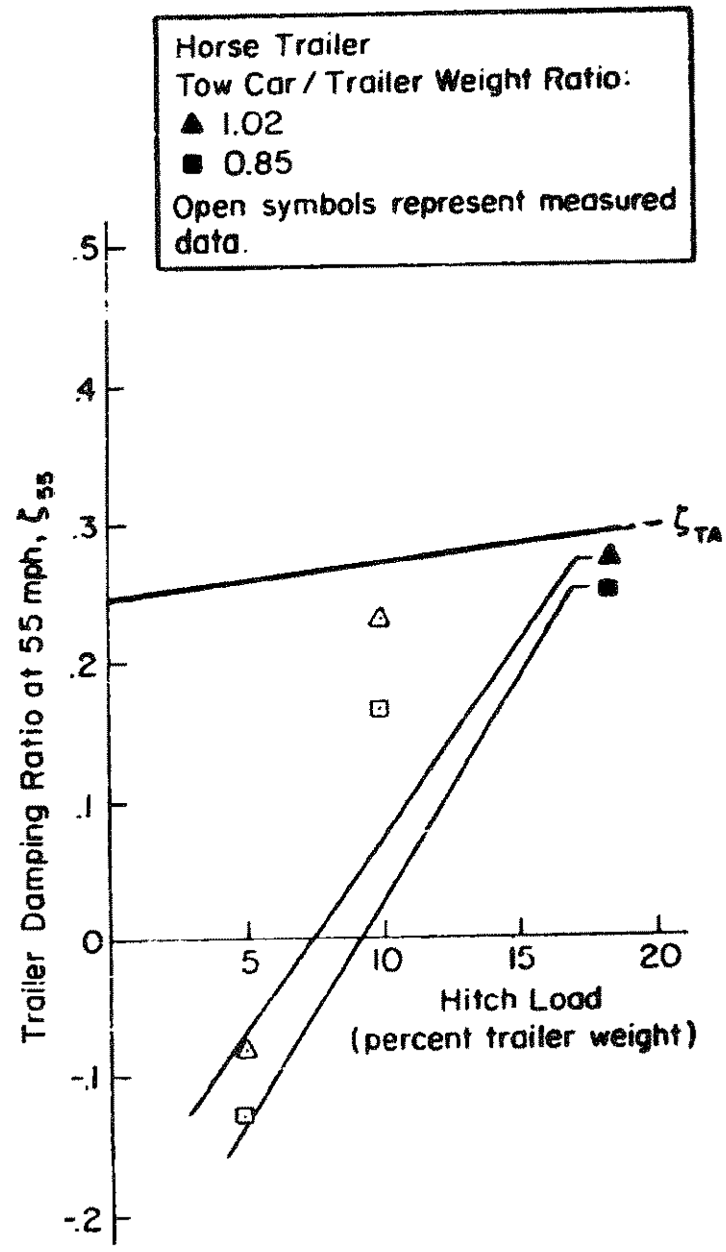


Figure 34. Comparison of Predicted and Measured Combination-Vehicle Trailer Damping for Horse Trailer

Although the predictions shown in Figs. 27-34 compare quite well to the "normalized" measured data, the results are not practical for large trailers where load leveling is used to redistribute the hitch load from the tow car tires and relevel the combination-vehicle. For example, for the configurations tested in this program, load leveling was necessary when hitch loads exceeded 300 lb. Up to this point, air shocks were capable of the releveling task. Consequently, for hitch loads greater than 10 percent of a 3000 lb trailer the tow car would probably be equipped with a load equalizer (Class III) hitch. This is also recommended in Ref. 20. Application of load leveling torque will then increase the damping from that predicted by Eq. 11. In fact, it was previously shown in Fig. 16 that load leveling improved damping approximately 0.06 units per 1000 ft-lb of applied torque. Prior results obtained in Ref. 3 showed a lesser average slope, i.e., 0.02 units per 1000 ft-lb.

The primary reasons why load leveling increases trailer damping are as follows:

1) Roll moment applied to the tow car which produces, via roll steer, a yaw response opposite to the trailer swing. The roll moment produced is a function of trailer articulation angle since, as shown in Fig. 35, the load leveling bar vertical forces become offset from the tow car centerline. This effect probably accounts for 2/3 of the total damping increase.

2) Friction forces applied at the load leveling bar pivots and at the hitch ball itself. For typical torques of 2000 ft-lb, calculations show that this effect is small, equivalent to a 20 lb increase in lateral force at the tires or about 2 percent of the total lateral forces acting at a 5 deg articulation angle.

3) Resisting forces applied by chain linkages angling forward and backward during trailer swing. Again this is a small effect, providing 10-20 lb equivalent side force at the tires for a 5 deg articulation angle and a chain length of 5 in.

4) Tire forces increased by increased vertical loads. Increased <u>trailer</u> tire cornering force, Y_{α_3}, provides the most "tire-related" influence to trailer damping. However, for a typical 25 percent load leveling condition the cornering force would be increased only about

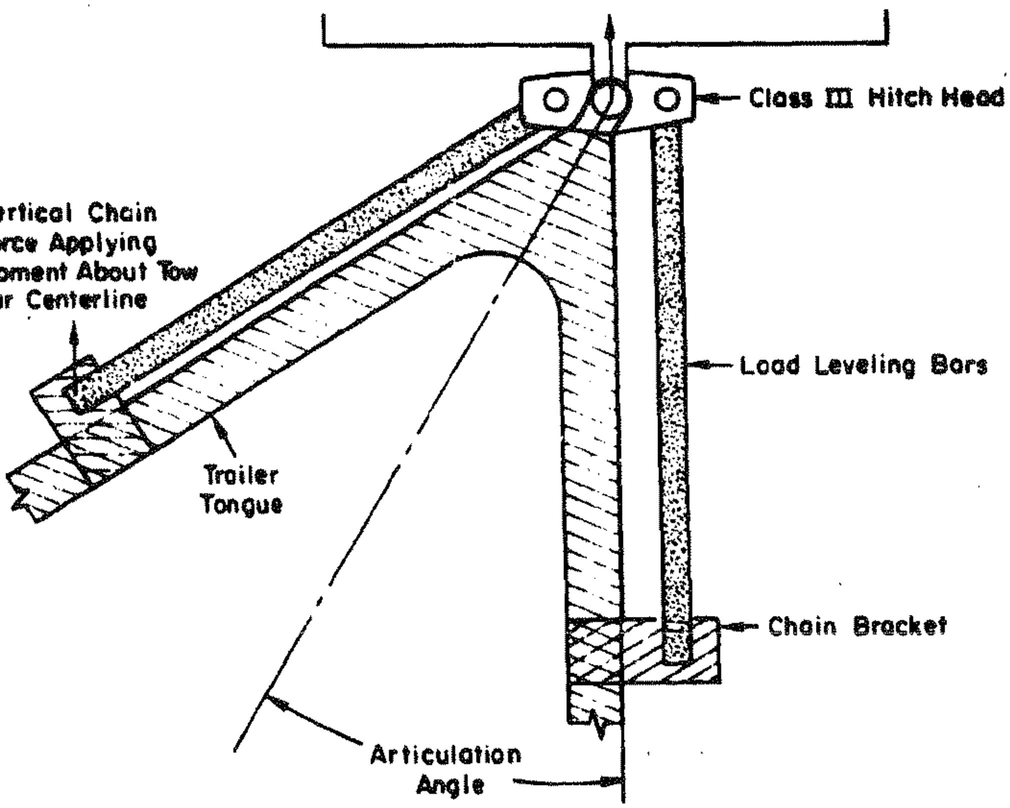


Figure 35. Roll Moment Applied by Load Leveling Bars
During Cornering or Trailer Swing

> 4 percent. Also, since this effect influences trailer-alone damping through a square root term (Eq. 6), the overall damping would be increased only about 2 percent due to trailer tire vertical load.

Due to the different types of Class III hitches available, and to the additional tow-car-related factors discussed above, it would be difficult for trailer manufacturers to accurately assess the effects of load leveling. Consequently, the specific effects of load leveling for each trailer probably would have to be tested with a representative set of tow cars to determine the combination-vehicle damping.

D. SELECTION OF TRAILER DAMPING CRITERION

Ideally, it would be desirable to use the analytical approach (Eq. 11) plus empirically determined constants to develop a <u>minimum</u> allowable hitch load for every trailer being towed by a passenger car. Since the analytical results presented in the previous subsection compare favorably with test data (especially at the lighter hitch loads), and approximate correction factors and simple test procedures have been established, we feel this approach can successfully be pursued.

However, the first question to be answered is how much stability (i.e., trailer damping) is necessary for safety. Obviously the trailer must be stable at highway speeds ($\zeta_{55} > 0$); however, the selection of a meaningful stability criterion represents the essence of any standard. For example, in Ref. 21 Korn stated that at 65 mph a given trailer should correct a sway condition within two cycles to be considered exhibiting satisfactory recovery characteristics. This "definition" cannot be translated directly into damping ratio, but from Fig. 10 it appears that a damping ratio of about 0.3 at 65 mph would be close to his criterion. Since 55 mph represents the current maximum legal speed, this damping can be extrapolated (using Fig. 15) to 0.35 at 55 mph. Trailer damping measurements given in Table 15 show this to be a very optimistic goal, except for very high hitch loads with load leveling. It can be seen from Table 17 that <u>trailer-alone</u> damping is generally not even this high.

A more realistic stability criterion would be a damping ratio of 0.15 at 55 mph. This translates to 3/4 cycle to damp to 1/2 amplitude and provides a stability margin to allow for the following situations:

1) <u>Overspeed margin.</u> Although different combination-vehicles will have somewhat different damping versus speed sensitivity than shown in Fig. 15, a 0.15 damping criterion at 55 mph will correspond, typically, to about a 90 mph critical speed, i.e., speed for trailer instability. Thus, the 0.15 damping criterion provides an overspeed margin of about 35 mph.

2) <u>Vehicle-in-use factors.</u> Reduced trailer tire cornering stiffness, steering free play, off-nominal inertia loadings, and non-zero lateral acceleration operating conditions have been shown to reduce the trailer damping. It is difficult

to justify specific damping margins to allow for these factors; however, current trailer recommendations specify hitch load be between 10 and 15 percent. Common use of these values of hitch loading can be verified by the survey of Ref. 22, in which 2675 trailers were weighed as they entered National Park campgrounds throughout the U.S. The average hitch load was 13 percent of the overall trailer weight. In this regard, results presented in Table 15 for 27 combination-vehicles at 10 percent hitch load had an average damping (at the tested conditions) of 0.18 ± 0.09. At 15 percent hitch load the average damping of 10 combination-vehicle configurations was 0.38 ± 0.12. Consequently, selection of a 0.15 damping criterion is well within current applications.

3) Downhill towing. Equations developed in Ref. 2 include horizontal force effects. The numerical example given in this reference can be used to compute a change in trailer damping of 0.004 units per 100 lb horizontal force. Thus, a 3000 lb trailer on a 6 percent downgrade would experience about a 10 percent reduction in damping at 55 mph if no braking were required. This reduction would be more significant if only tow car braking were used.

4) Tow car interaction. Trailer damping criteria higher than 0.15 would probably lead to the use of high hitch loads, the requirement for sway controls, or trailer redesign. As will be described in the next section, use of high hitch loads (the least costly approach) will degrade the tow vehicle yaw stability. On the other hand, simple calculations using Eq. 11 show that adding 2 ft to the tongue length would increase damping about 0.17 units.

For purposes of further illustrating the development of damping standards we will tentatively assume 0.15 represents a reasonable and justifiable criterion. The next question is how this criterion could be implemented and tested.

E. RECOMMENDATIONS

In the first subsection it was hypothesized that a minimum hitch load boundary based on tow-car/trailer weight ratio would be possible. The results and analysis presented in the subsequent two subsections showed that this approach was feasible, although the tow car size effect is not always consistent and the effects of load leveling will probably require individualized tests. However, differences in trailer damping due to trailer design and hitch load have been shown to be very significant, consequently, a minimum damping criterion and test methodology have been recommended. Based on test data from this program and calculations of vehicle-in-use factors, a damping criterion of 0.15 at 55 mph is suggested. From this a minimum hitch load boundary can be derived or experimentally determined, using the test procedures described below.

Derivation of the minimum hitch load boundary first requires determination of each trailer's "trailer-alone damping ratio" and "hookup factor" as defined in Eq. 11. These factors require measurement (probably by the trailer manufacturer) of yaw moment of inertia, geometry, and tire cornering stiffness. For example, yaw moment of inertia can readily be derived with a simple roller bearing turntable (such as used for wheel alignments), two coil springs, and the test setup shown in Fig. 36. The trailer wheels are held off the ground by placing the axle beam on a block, which is positioned on the roller bearing support turntable. A counterweight, W_x, is added to balance the trailer hitch load, i.e., $W_t h = W_x \ell_3$. At this condition the trailer floats freely on the turntable with no offset forces. Two coil springs are attached between the tongue and nearby ground anchor points. A prestretched preload provides a constant spring rate in both directions of travel (since no slack is permitted). The effective spring rate at the hitch (lateral force per foot of deflection) is measured by spring scales with the springs in this prestretched condition. The trailer is then gently oscillated (by hand) in the yaw direction (about ±3 in. amplitude), while maintaining the trailer in a level orientation. Elapsed time measurements are taken for several (e.g., 10) full oscillations and then repeated several times to insure consistency. Repeat run frequencies should be in close agreement (e.g., 3 percent) since damping is very small. The moment of inertia about the center of rotation can then be calculated from the equation:

$$I_1 = \frac{K_h \ell_2^2}{(2\pi f_n)^2} \tag{12}$$

where

K_h	Effective spring rate (lb/ft)
ℓ_2	Hitch to axle distance (ft)
f_n	Natural frequency (cycles/sec)

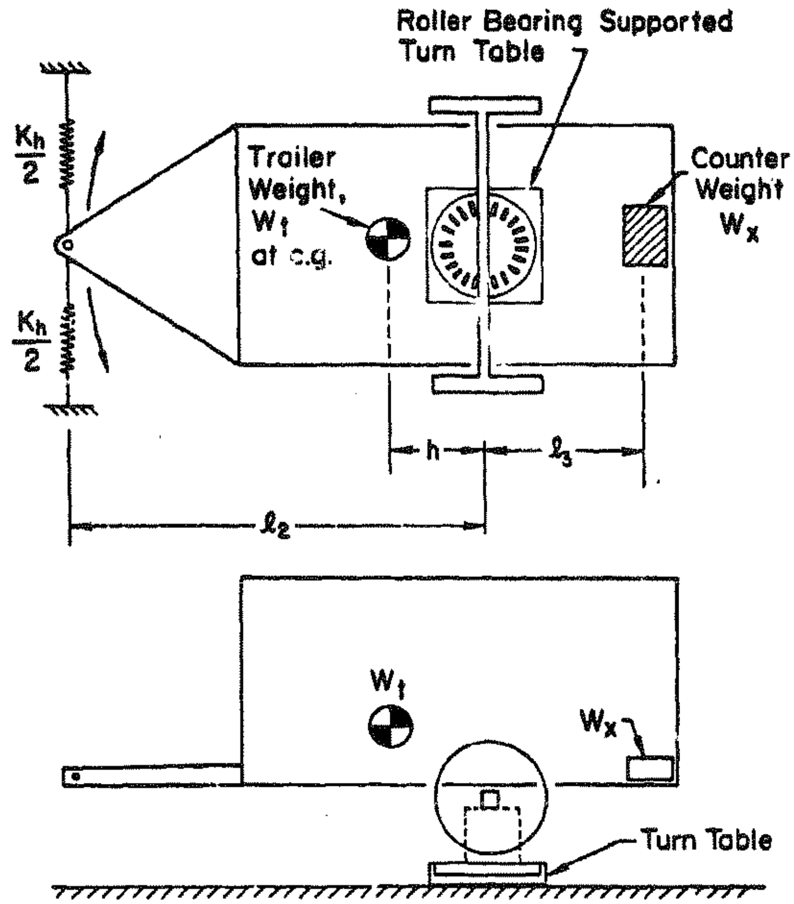


Figure 36. Test Setup for Trailer Moment of Inertia Measurement

This can be converted to the moment of inertia about the c.g., I_{to}, and about the hitch point, I_{th}, using the parallel axes equations:

$$I_{to} = I_1 - \frac{W_t h^2}{g} - \frac{W_x \ell_3^2}{g} \tag{13}$$

$$I_{th} = I_{to} + \frac{W_t}{g} (\ell_2 - h)^2 \tag{14}$$

The inertia values given in Table 17 were experimentally determined using this method.

Tire cornering stiffness, Y_{α_3}, is directly measurable with a tire test machine. This is usually accomplished by tire manufacturers, and the data would generally be available to the trailer manufacturer. If not, tire properties can be estimated by comparison to one of the tires already tested by Calspan and reported in Refs. 18 or 19.

Equation 11 is then used (with a tow car sensitivity constant of 3.7) to calculate, at various hitch loads, the tow car to trailer weight ratio that will produce a combination-vehicle damping ratio of 0.15 at 55 mph, i.e.,

$$\frac{W_c}{W_t} = \frac{3.7(HUF)}{\zeta_{ta} - 0.15} \tag{15}$$

Weight ratio versus hitch load plots have been prepared in Figs. 37-40 for the trailers weighing less than 4000 lb tested in this program. In these figures hitch loads above the boundary would ideally provide trailer damping ratios greater than 0.15. Using the utility trailer in Fig. 37 as one example, a minimum hitch load of 6 percent would be required for this trailer if towed by a 3000 lb car; this could be reduced to 0 percent if towed by a 5000 lb car. On the other hand, the 18 ft travel trailer shown in Fig. 40 would always require a hitch load greater than 13 percent, since tow cars greater than 6000 lb (i.e., 1.5 × trailer weight) are unrealistically large. However, hitch loads of this magnitude (i.e., 500 lb) would require load leveling, consequently the actual combination-vehicle would have to be tested experimentally to determine the minimum hitch load for acceptable trailer damping.

As described previously, the combination-vehicle trailer damping can be determined experimentally using a "pulse steer" test procedure. This procedure requires only a calibrated speedometer and an instrumentation sensor to measure trailer articulation angle or lateral acceleration of the trailer c.g. The combination-vehicle is driven in a straight line at [illegible] mph and a [illegible]-90 deg steering wheel "pulse" applied as rapidly as possible. When recentered, the wheel is held tightly fixed to avoid steering force feedback from the trailer. The resulting trailer damping coefficient

Tire cornering stiffness, Y_{α_3}, is directly measurable with a tire test machine. This is usually accomplished by tire manufacturers, and the data would generally be available to the trailer manufacturer. If not, tire properties can be estimated by comparison to one of the tires already tested by Calspan and reported in Refs. 18 or 19.

Equation 11 is then used (with a tow car sensitivity constant of 3.7) to calculate, at various hitch loads, the tow car to trailer weight ratio that will produce a combination-vehicle damping ratio of 0.15 at 55 mph, i.e.,

$$\frac{W_c}{W_t} = \frac{3.7(\text{HUF})}{\zeta_{ta} - 0.15} \tag{15}$$

Weight ratio versus hitch load plots have been prepared in Figs. 37-40 for the trailers weighing less than 4000 lb tested in this program. In these figures hitch loads above the boundary would ideally provide trailer damping ratios greater than 0.15. Using the utility trailer in Fig. 37 as one example, a minimum hitch load of 6 percent would be required for this trailer if towed by a 3000 lb car; this could be reduced to 0 percent if towed by a 5000 lb car. On the other hand, the 18 ft travel trailer shown in Fig. 40 would always require a hitch load greater than 13 percent, since tow cars greater than 6000 lb (i.e., 1.5 × trailer weight) are unrealistically large. However, hitch loads of this magnitude (i.e., 500 lb) would require load leveling, consequently the actual combination-vehicle would have to be tested experimentally to determine the minimum hitch load for acceptable trailer damping.

As described previously, the combination-vehicle trailer damping can be determined experimentally using a "pulse steer" test procedure. This procedure requires only a calibrated speedometer and an instrumentation sensor to measure trailer articulation angle or lateral acceleration of the trailer e.g. The combination-vehicle is driven in a straight line at 55 mph and a 45-90 deg steering wheel "pulse" applied as rapidly as possible. When recentered, the wheel is held tightly fixed to avoid steering force feedback from the trailer. The resulting trailer damping coefficient

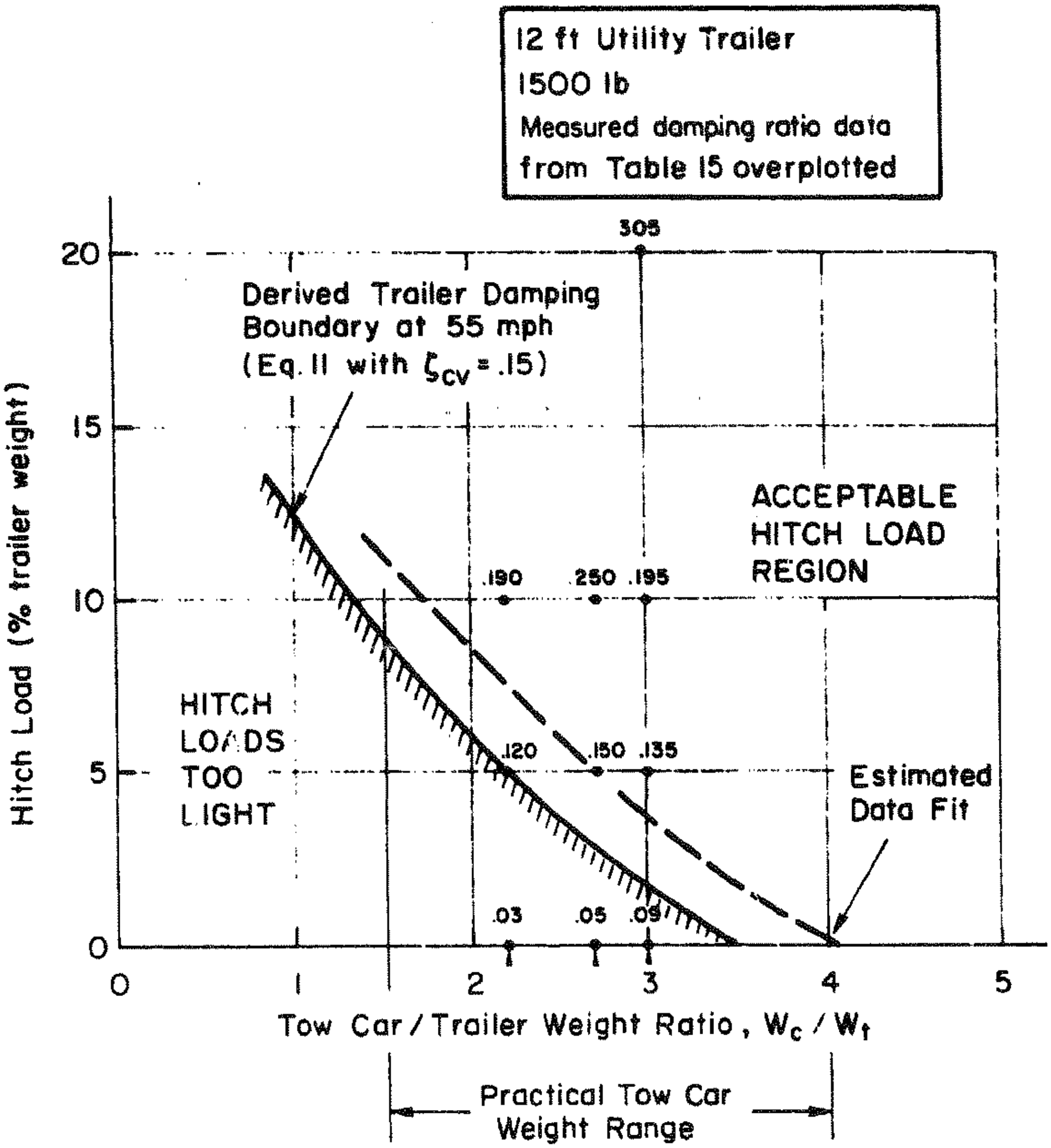


Figure 57. Comparison of Calculated Minimum Trailer Damping Boundary with Measured Data for Utility Trailer

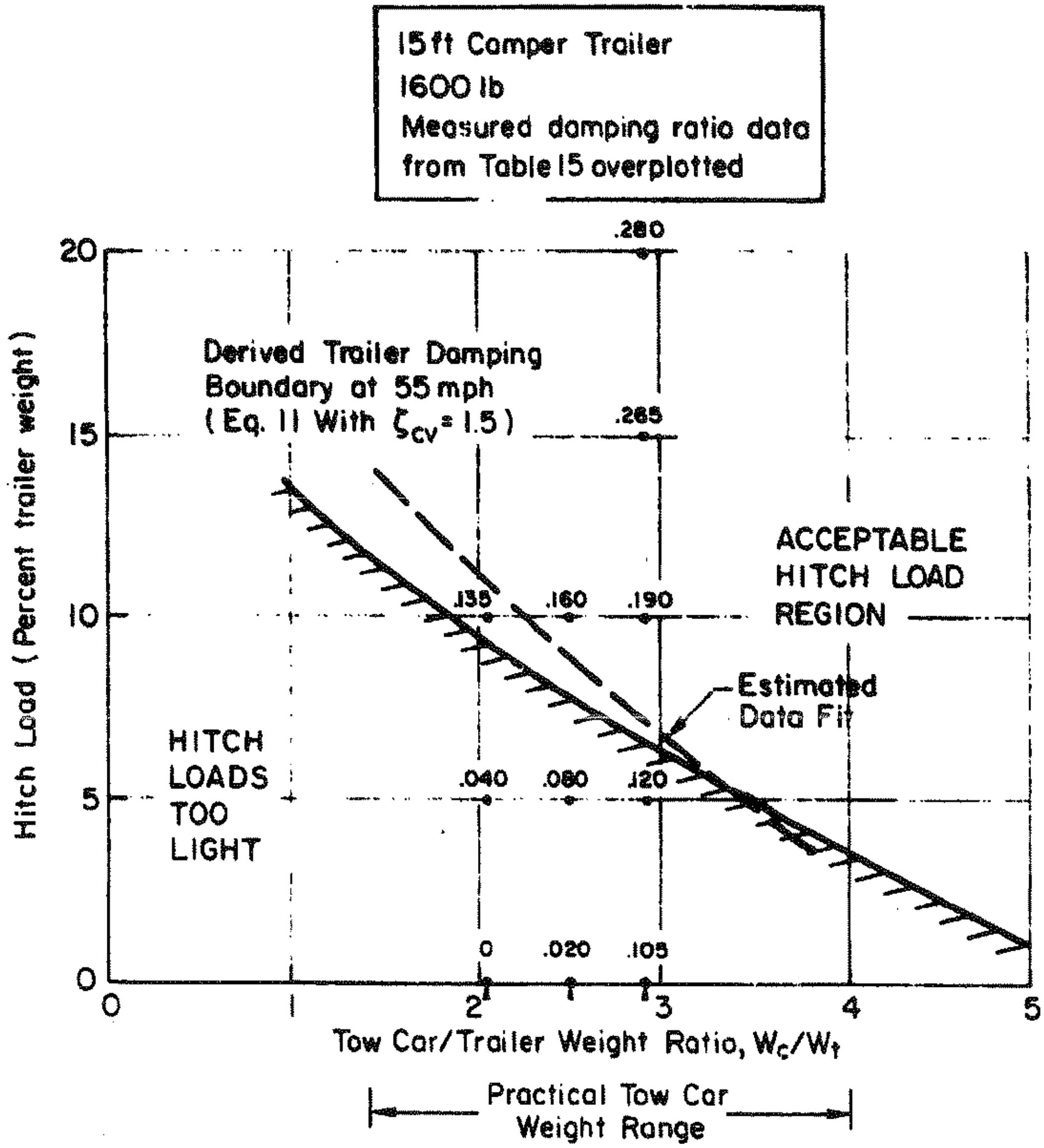


Figure 58. Comparison of Calculated Minimum Trailer Damping Boundary with Measured Data for Camper Trailer

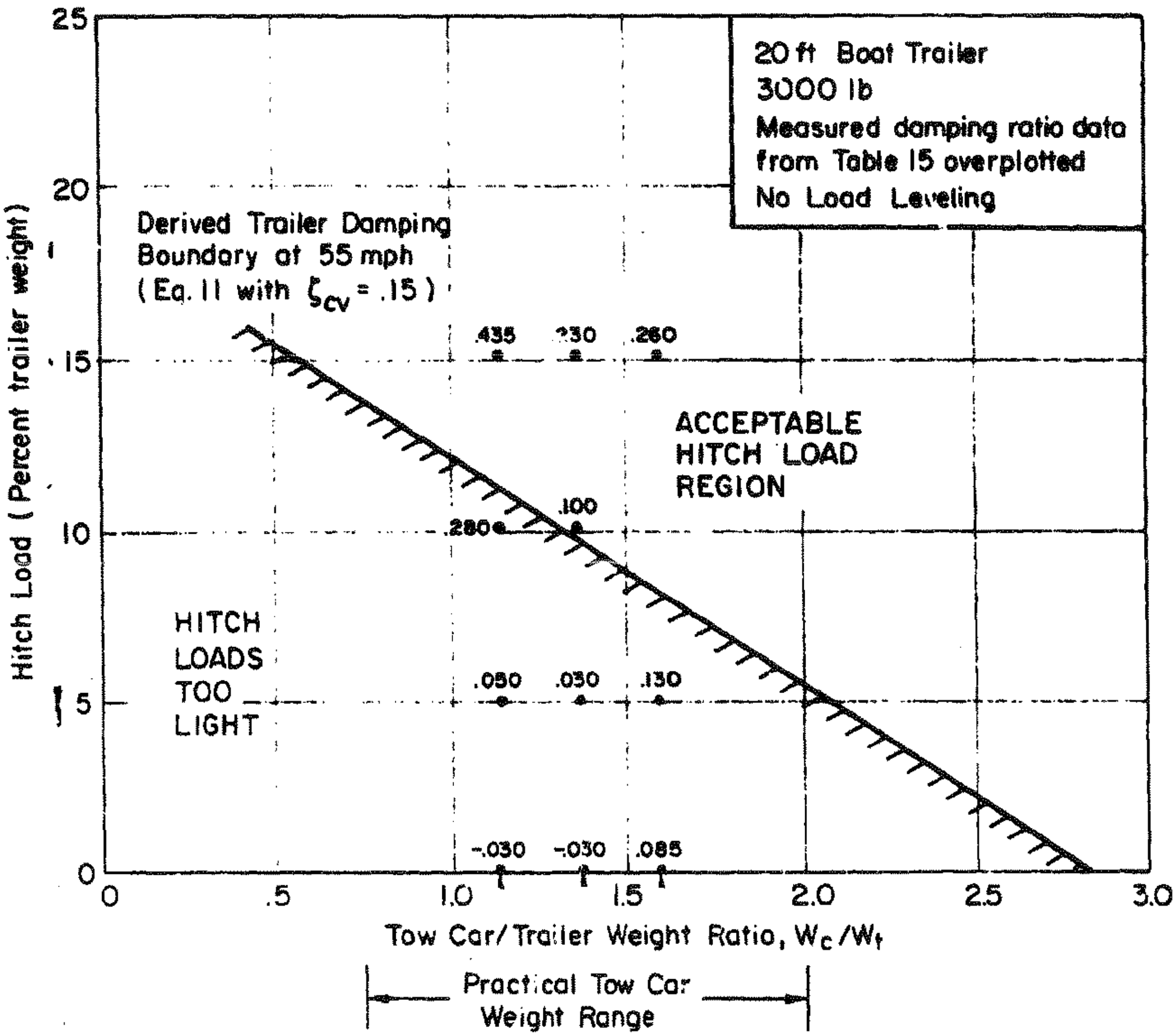


Figure 59. Comparison of Calculated Minimum Trailer Damping Boundary with Measured Data for Boat Trailer

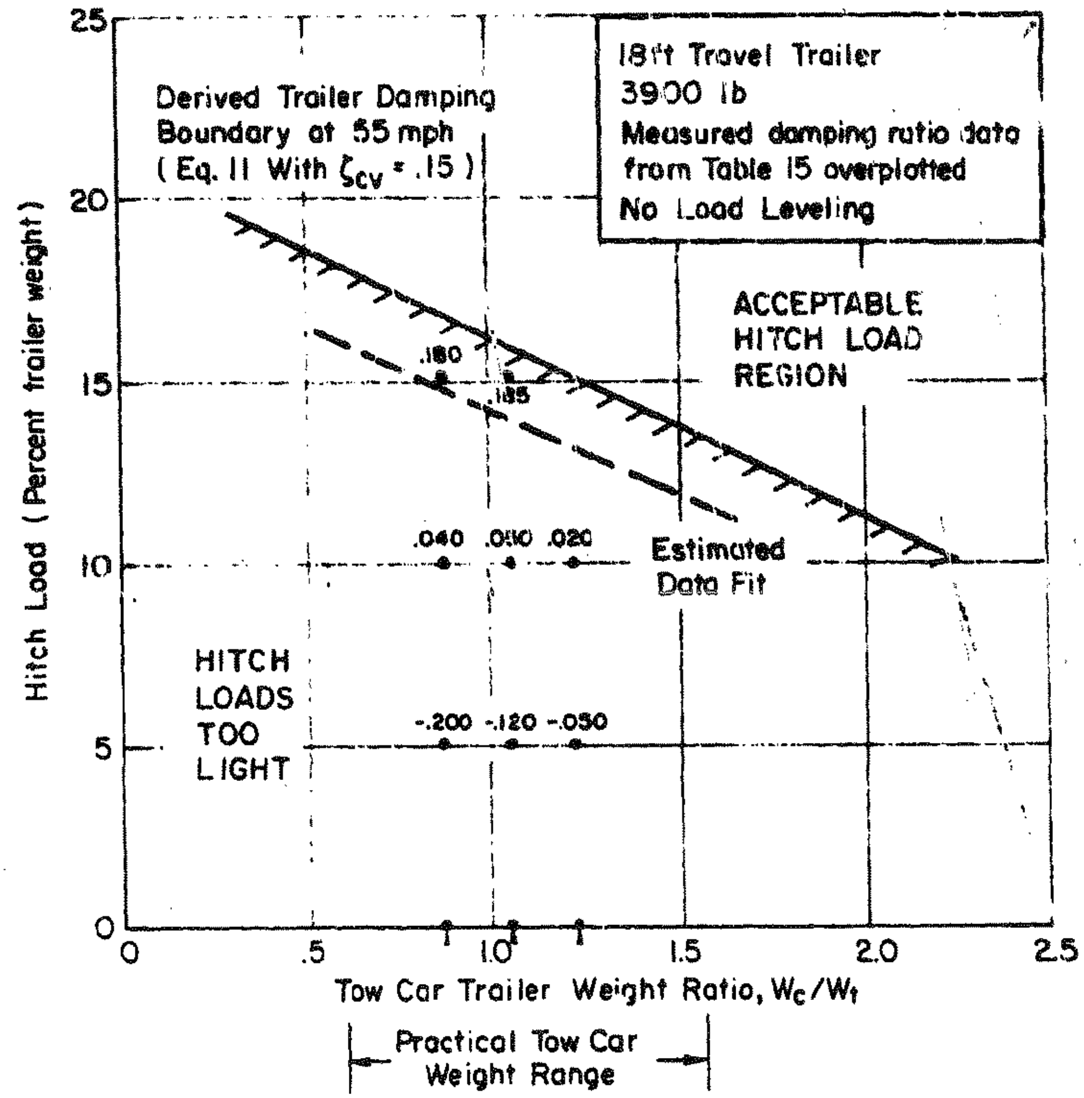


Figure 50. Comparison of Calculated Minimum Trailer Damping Boundary with Measured Data for 18 ft Travel Trailer

can be computed from the ratios of successive peaks as illustrated in Fig. 41. This procedure would be repeated at various hitch loads, and corresponding load leveling torques, to determine the minimum safe hitch load value.

Results of this procedure (Table 15 values) are overplotted on Figs. 37-40 in order to compare measured data with the predicted minimum

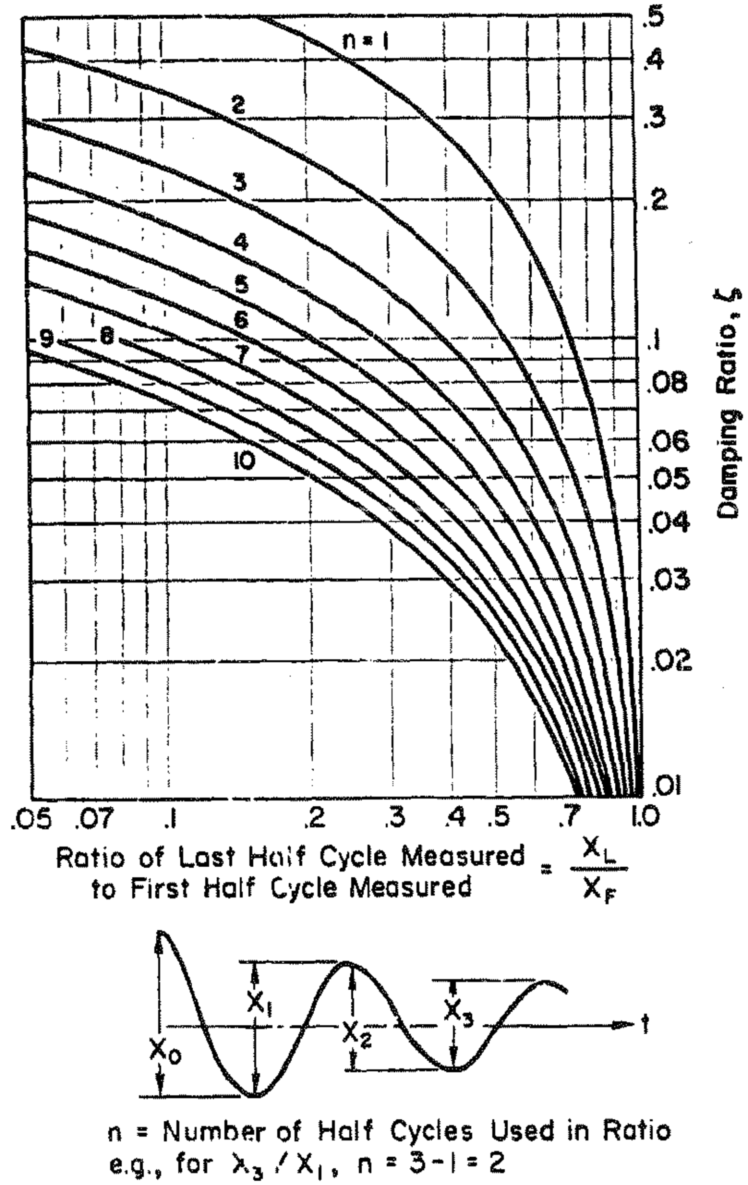


Figure 41. Derivation of Damping Ratio from Time History Response

damping boundary. As noted previously in deriving average errors, the comparison is quite favorable for lightweight trailers without load leveling. The boundary for heavier trailers is not practical (hitch loads too high) since load leveling must be used, and this in turn will increase the trailer damping. In effect, load leveling (or sway controls) will shift the boundary line to the left. Based on the test data the tow car size appears to be approximately doubled when load leveling is used, i.e., a 3000 lb tow car behaves as if it were a 6000 lb tow car.

The next section of the report describes the tow car understeer change due to hitch load which will place an additional (upper) boundary on the recommended hitch load plots.

SECTION V

TOW CAR STEADY-STATE TURN STABILITY

It has been shown in previous car/trailer handling studies (Refs. 2 and 3) that a sensitive, repeatable, and easily determined handling parameter for quantifying combined vehicle directional steady-state control and dynamic stability is the understeer/oversteer gradient, or stability factor, K. The importance of this parameter was also recognized in the Ref. 23 analytical study and in Ref. 24 where the first known attempt to establish a trailer towing performance safety standard was made. The latter employed maneuvers determining the limit influence of K rather than quantifying it. A companion stability factor for the trailer, K', was also developed and evaluated in the Ref. 2 program. In this section we have suggested an approach for using K to isolate requirements imposed on the tow vehicle by the trailer, compared full-scale results to tentative stability boundaries, and suggested recommendations for stability criteria and test procedures.

A. ANALYTICAL CONSIDERATIONS

Since the thrust of this entire section revolves around the effects of hitch load on the static stability (or understeer gradient) of the tow vehicle and trailer, a necessary starting point is to define the physical significance of tow car understeer.

Understeer gradient of the tow car is a fundamental steady-state directional control relationship between the tow vehicle yawing velocity (or side acceleration) and the steering wheel input. The "steady-state" phase can include constant lateral accelerations (other than zero) and constant fore and aft accelerations or decelerations <u>after</u> the initial transients have died out. When kinematic constraints are also considered, this relationship can be phrased in terms of the steer angle, δ_w, required to maintain a given fixed turn radius, R, i.e., as:

$$\delta_w = \frac{\ell_1}{R}\,(1 + KU_0^2) \quad ; \quad R = \frac{\ell_1}{\delta_w}\,(1 + KU_0^2) \tag{16}$$

Here the tow vehicle wheelbase, ℓ_1, divided by the radius of turn, R, is called the Ackermann steer angle, and the stability factor is a speed variation weighting. The Ackermann angle is the steer angle required when turn radius is independent of speed (i.e., for "neutral steer" when the stability factor

be oversteering, since less steer angle is needed than for neutral steer. When K is positive the vehicle exhibits the nominal understeering characteristic. In standard SAE notation (Ref. 16), stability factor is proportional to the understeer/oversteer gradient, expressed in deg/g, which is:

$$\kappa_{SAE}\ (\text{deg/g}) = 1845\ \ell_1 K\ (\text{sec}^2/\text{ft}^2) \tag{17}$$

when ℓ_1 is in feet.

It has been established in the full-scale testing of Refs. 2 and 3 that neutral or over-steer ($K \leq 0$) produces unsafe car/trailer directional control. As will be shown below, this can occur through changes in axle loading and tire side force properties which vary with hitch load, load leveling, braking, etc.

The stability factor for the automobile alone may also be expressed in vehicle parameters terms as:

$$K = \frac{W_c}{2g\ell_1^2}\left[\frac{b}{Y'_{\alpha_1}} - \frac{a}{Y'_{\alpha_2}}\right] \tag{18}$$

where the physical terms include the tow car weight (W_c), wheelbase (ℓ_1), front and rear center of gravity positions (a and b, respectively; $a + b = \ell_1$), and Y'_{α_1} and Y'_{α_2}, which are the <u>effective</u> tire side force coefficients at the front and rear axles, respectively. The effective side force coefficients combine the influence of tire, the geometrically designed-in understeer, and roll steer characteristics. If we were to separate these influences (and substitute the normal forces at each axle, $W_c b/\ell_1 = F_{z_1}$ and $W_c a/\ell_1 = F_{z_2}$), we obtain:

$$K = \frac{1}{2g\ell_1}\left(\frac{F_{z_1}}{Y_{\alpha_1}} - \frac{F_{z_2}}{Y_{\alpha_2}}\right) + k \qquad (19)$$

where Y_{α_1} and Y_{α_2} now represent tire characteristics only, and k represents the geometric understeer (Ref. 17). For front-engined domestic automobiles, $F_{z_1} > F_{z_2}$ (e.g, 55/45), and $Y_{\alpha_1} < Y_{\alpha_2}$ (front tires have generally slightly lower inflation pressure than the rear). Thus, the axle load/tire contribution to the stability factor is positive and the vehicle is basically understeering.

Now a hitch load is added. The stability factor for the combined vehicles can be similarly expressed as:

$$K = \frac{1}{2g\ell_1}\left[\frac{1}{Y_{\alpha_1}}\overbrace{\left(\frac{W_c b}{\ell_1} - \frac{W_t h \ell_h}{\ell_2 \ell_1}\right)}^{F_{z_1}} - \frac{1}{Y_{\alpha_2}}\overbrace{\left(\frac{W_c a}{\ell_1} + \frac{W_t h \ell_{1_h}}{\ell_2 \ell_1}\right)}^{F_{z_2}}\right] + k \qquad (20)$$

where the additional terms (defined in Fig. 42) are introduced by the hitch load applied to the tow vehicle at the attach point (ball). The parenthetic terms F_{z_1} and F_{z_2} are identically the normal loads at the front and rear axles assuming an unrestrained (freely pivoting) hitch. Note that the hitch load reduces the front axle normal load and increases the rear axle load. Thus, the presence of the trailer reduces the tow vehicle understeer and can cause the stability factor to become negative. This occurs more rapidly if the normal force on the rear axle approaches the tire maximum load rating, in which case Y_{α_2} decreases even more so. This can result in an oversteer (negative) contribution which equals or overpowers the geometric understeer component. In this instance K is negative at very low cornering levels and an unsafe condition exists.

Several factors concerning tow-vehicle-related requirements now emerge. First, one can calculate the hitch load which causes the stability factor

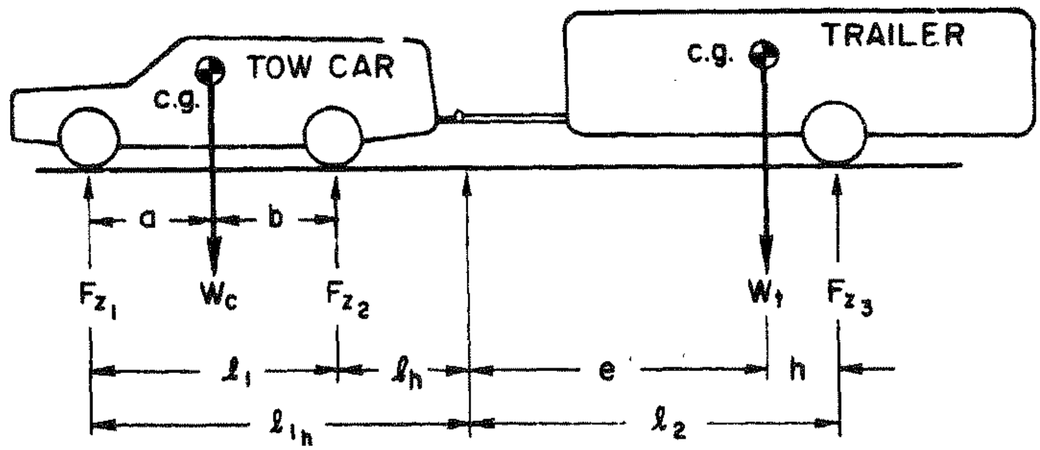


Figure 42. Tow Car/Trailer Geometry

to become zero at zero lateral acceleration. Further, using the nonlinear characteristics of tires at non-zero slip angles, the hitch load which causes the stability factor to become zero at higher values of lateral acceleration can also be calculated. For example, an analysis was made using several typical cars with standard-type tires. The methodology includes determining the static axle loads (including load leveling), calculating the load transfer (from inside to outside) during steady cornering at various lateral accelerations, determining the slip angles required from the tire characteristic curves at the computed normal load, and finally deriving the effective front and rear tire cornering stiffness at the various lateral accelerations. These tire coefficients are then used in Eq. 20 to compute the stability factor at each value of lateral acceleration. The results of this procedure are shown in Fig. 43 for an intermediate-sized tow car plus 22 ft travel trailer at 20 percent hitch load and 25 percent load leveling. Note that at 0.24 g lateral acceleration the tow car will become neutral steering. At lighter values of hitch load (or less load leveling), the whole curve would be shifted upward, and hence the point of neutral steering would move to higher lateral accelerations.

Using the approach described above we can specify, for example, the maximum allowable hitch load for a stability factor criterion of $K \geq 0$ up to 0.3 g. Assuming this criterion we can further plot the hitch load (or

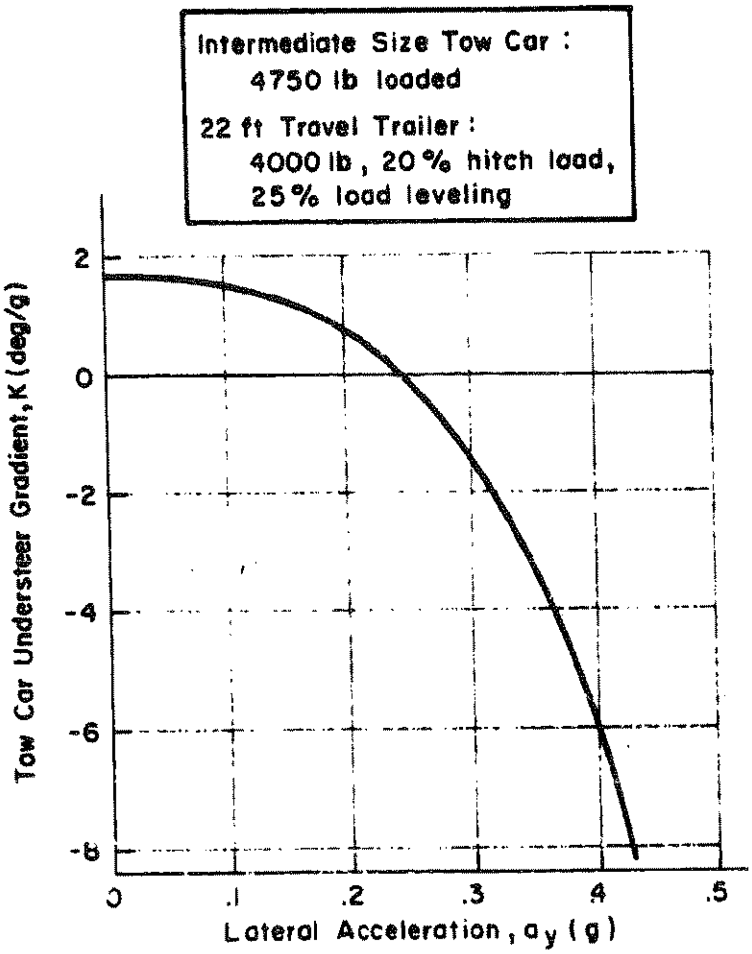


Figure 43. Calculated Variation in Tow Car Understeer as a Function of Lateral Acceleration

percent hitch load) as a function of tow car weight (or weight ratio). The use of load leveling alters the boundary in Fig. 44, since the rear axle vertical load is decreased. This results in lower tire cornering coefficient and thus a higher rear tire slip angle for a given level of lateral acceleration. This shifts the tow vehicle stability criterion line to the right, thus reducing the hitch load for a given tow car weight.

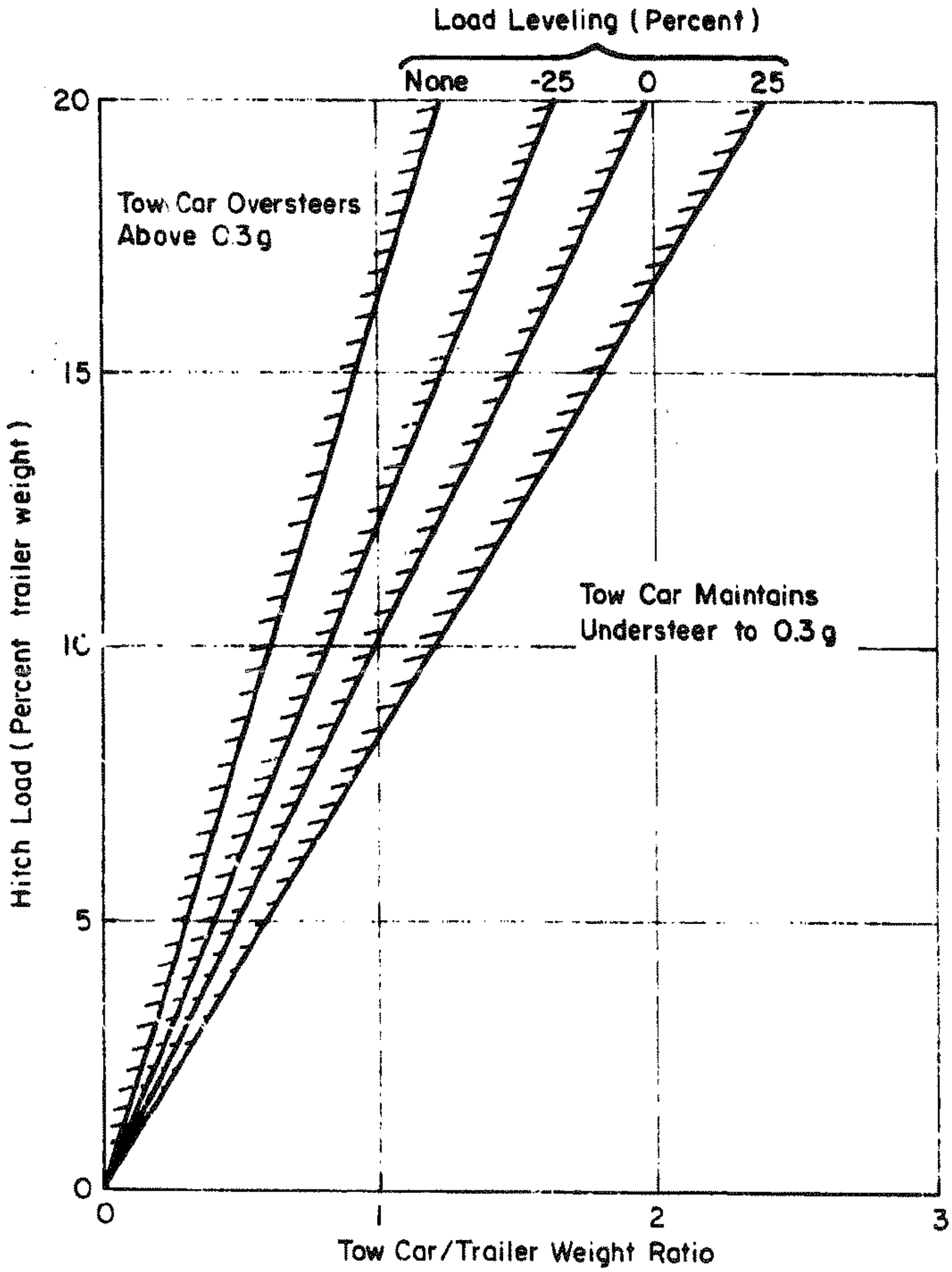


Figure 41. Analytically Derived Loading Boundary Necessary to Maintain Tow Car Understeer at 0.3 g Cornering

Full-scale results are presented next and compared to the theoretical stability boundaries of Fig. 44. Based on these comparisons and on current practices, a revised stability boundary is selected and test procedures are recommended.

B. FULL-SCALE TEST RESULTS

Over 70 understeer gradient tests were performed with the three tow cars and eight trailers. This included both the constant radius circle test procedures recommended by SAE (Ref. 11) and the step steer procedure previously recommended by STI in Ref. 2. Both procedures were generally performed in both the left- and right-hand directions. Data plots for each of the key configurations (in terms of steer angle versus lateral acceleration and/or turn radius versus speed squared) are presented in Vol. III, Appendix F.

The performance measures derived from these data were understeer gradient at zero lateral acceleration (K_o), lateral acceleration for neutral steer (a_{y_o}), and lateral acceleration for incipient jackknife ($a_{y_{JK}}$). Through the use of an electronic anti-jackknife brake mechanism this latter condition was actually inhibited from occurring. However, trailer articulation angles exceeding 15 deg (which triggered the anti-jackknife circuit) were defined as incipient jackknife conditions.

A summary of these performance measures for each combination-vehicle configuration is presented in Table 18. We will discuss the understeer results first, in this subsection, and then utilize the lateral acceleration for neutral steer results in the following subsection to empirically check the hitch load boundary previously illustrated in Fig. 44.

Figures 45a and 45b present the tow car understeer changes versus hitch load for each of the three tow cars. Left- and right-hand cornering data were averaged; however, due to power effects (rear tire torque) the understeer gradient in right-hand turns was always less than that in left-hand turns. For example, the average difference was 0.7 deg/g for 15 configurations in the constant radius circle test procedure. In any case, the results were as anticipated, i.e., both increasing hitch load and load leveling reduced tow car understeer. Viewed another way, the

TABLE 18. TOW CAR UNDERSTEER TEST RESULTS

TOW CAR	TRAILER	HITCH LOAD (lb)	LOAD LEVEL-ING	$K_{a_y=0}$ [a] (deg/g)	$a_{y_{K=0}}$ [b] (g)	$a_{y_{JK}}$ [c] (g)	REMARKS
Inter-mediate	None	—	—	5.0	>.5	—	Constant radius circle procedure
	Utility 1500 lb	150 300	N N	4.6 4.0	>.5 >.5	— —	
	Camper 1600 lb	160 256 336	N N N	4.7 4.8 4.3	>.5 >.5 >.5	-- — —	
	18 ft Travel 3000 lb	390 390	N +25	4.4 3.4	>.5 .35	— —	
	18 ft Travel 3900 lb	390 780 780	+25 −25 +25	2.9 1.8 1.4	.38 .35 .21	— >.42 >.38	
	19 ft Travel 4000 lb	600 600	−32 +25	1.6 0.9	>.33 >.26	— —	Step steer procedure
	19 ft Travel 4000 lb	400 400 400 800	N 0 +25 +25	4.3 3.3 4.3 0.9	>.5 .38 .28 .26	— — — —	Constant radius circle test
	27 ft Travel 6000 lb	600 600 900 900	−15 +25 −20 +25	2.4 0.8 −0.5 −1.1	.27 .16 0 0	— >.32 >.30 >.47	Left-hand turns only, step steer procedure
	Boat 3000 lb	400 450 450	N N +25	3.3 4.6 2.1	>.5 >.29 .48	— — —	Left-hand turns only, step steer procedure
	Horse 5800 lb	580 840	N N	2.8 2.1	>.33 >.35	>.5 —	Step steer procedure

[a] $K_{a_y=0}$: Average understeer gradient determined from left- and right-hand turns; slope taken from 0 to 0.2 g; in all cases $K_{RHT} < K_{LHT}$.

[b] $a_{y_{K=0}}$: Lateral acceleration for neutral steer right-hand turns; if prefaced by >, value given represents maximum tested condition.

[c] $a_{y_{JK}}$: Lateral acceleration at last data point exhibiting oversteer tendency ($K < 0$).

(continued on following page)

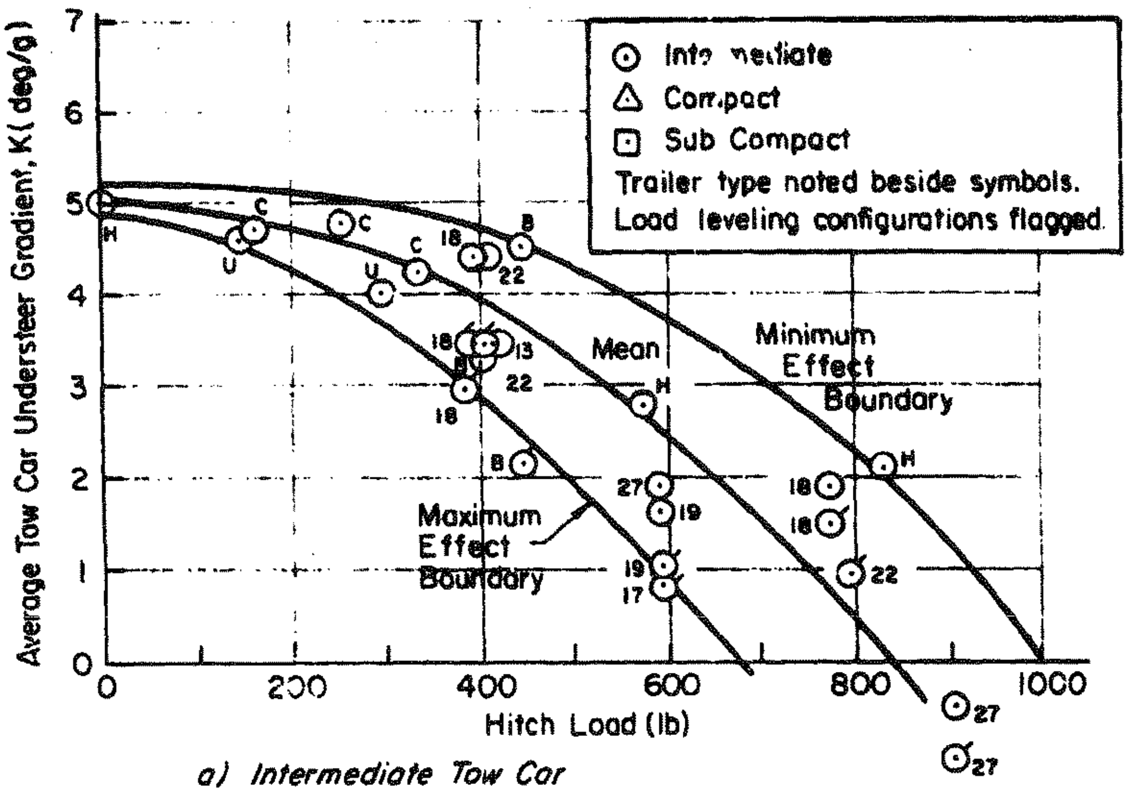
TABLE 18. (CONCLUDED)

TOW CAR	TRAILER	HITCH LOAD (lb)	LOAD LEVEL-ING	$K_{a_y=0}$ [a] (deg/g)	$a_{y_{K=0}}$ [b] (g)	$a_{y_{JK}}$ [c] (g)	REMARKS
Compact	None	—	—	6.2	>.5	—	Left-hand turns only, step steer procedure
	18 ft Travel 4000 lb	585	N	4.2	>.5		
	22 ft Travel 4000 lb	200	N	4.2	>.5	—	
		300	N	3.8	>.5	—	
		400	−29	1.5	>.5	—	
		500	− 6	0.4	.25	>.5	
		600	− 7	0	0	>.5	
		600	+17	0	0	>.45	
		700	+ 4	0	0	>.5	
	27 ft Travel 6000 lb	600	+15	−0.7	0	>.38	
	Boat 2275 lb	600	+28	1.8	.40	>.55	
	Horse 4000 lb	400	N	4.9	>.5	—	
Sub-compact	None	—	—	4.1	>.5	—	Left-hand turns only, step steer procedure
	Utility 1500 lb	150	N	3.5	>.5	—	
	18 ft Travel 3420 lb	170	N	2.7	>.5	—	
		257	N	2.1	>.5	>.5	
	Boat 3000 lb	0	N	4.0	>.5	—	
		150	N	2.9	>.5	—	
		150	0	2.1	.5	>.48	
		300	+32	1.4	.3	>.41	
		450	+21	0	.17	>.35	

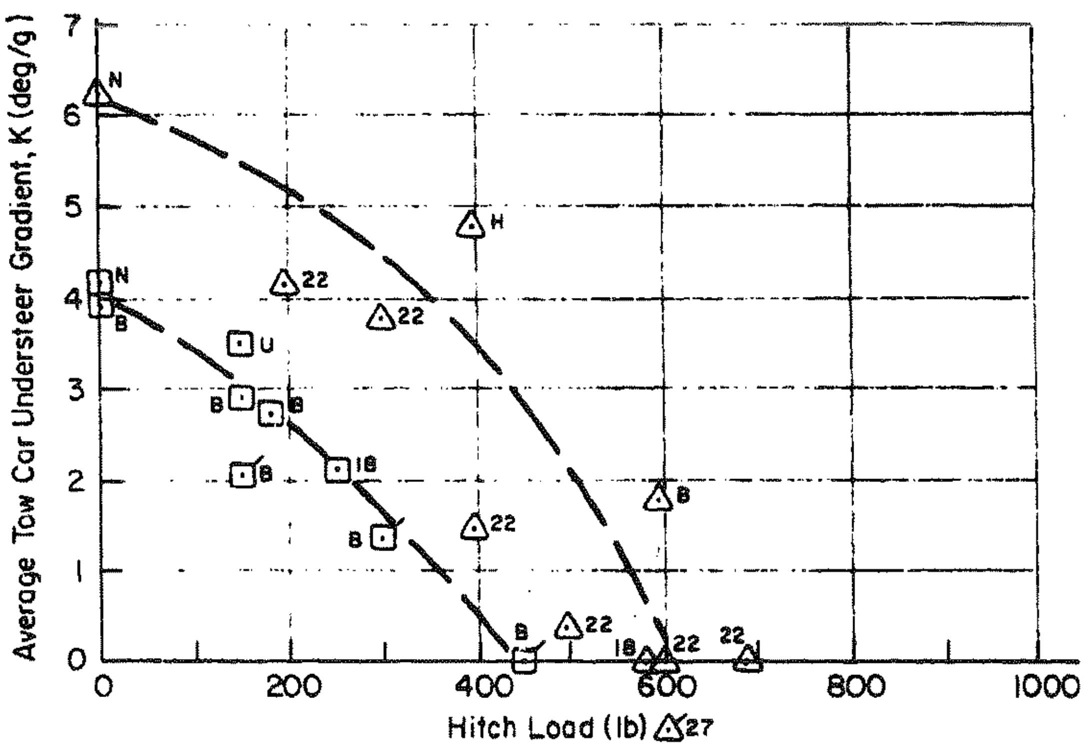
[a] $K_{a_y=0}$: Average understeer gradient determined from left- and right-hand turns; slope taken from 0 to 0.2 g; in all cases $K_{RHT} < K_{LHT}$.

[b] $a_{y_{K=0}}$: Lateral acceleration for neutral steer right-hand turns; if prefaced by >, value given represents maximum tested condition.

[c] $a_{y_{JK}}$: Lateral acceleration at last data point exhibiting oversteer tendency ($K < 0$).



a) Intermediate Tow Car



b) Compact and Sub Compact Tow Cars

Figure 45. Effect of Hitch Load on Tow Car Understeer at $a_y = 0$

reduction in understeer due to hitch load is related to the larger effective mass acting laterally on the tow car rear axle during cornering. In this regard, it is possible to represent the hitch load as a redistribution of vehicle front to rear mass ratio. This has been done in the conversion scale of Fig. 46 for the intermediate size tow car. From Fig. 45a it appears that this tow car transitions to oversteer ($K < 0$) at about 800 lb hitch load. In terms of mass distributions this implies that <u>any</u> loading producing more than 63 percent at the rear axle will result in oversteer at low cornering levels.

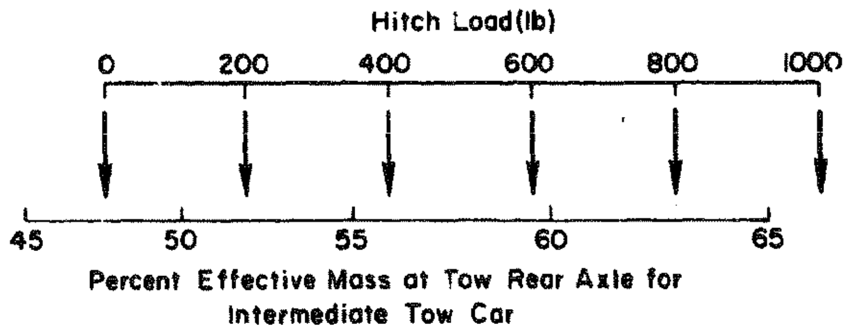


Figure 46. Hitch Load Effect on Tow Car Weight Distribution

The smaller tow cars allow even less hitch load to produce neutral steer, i.e., the compact allows about 600 lb and the subcompact can allow only about 400 lb. These levels convert to 61 and 59 percent rear axle load distributions, respectively. It should be mentioned that the hitch load values noted above are typically higher than those recommended by the manufacturers. For example, GM, Ford, and Chrysler recommend 600 lb maximum for their new full-sized cars, and Ford recommends a maximum of only 100 lb for its subcompacts. Although these lower values are probably based on other considerations (cooling, structure, etc.), following the manufacturers' recommendations would provide an understeer safety margin for increased cornering levels (e.g., recall Fig. 43). Hence, if a margin for higher lateral cornering is desired, then some positive value of understeer at very low cornering should be required.

The second most relevant factor which causes a reduction in understeer gradient is the amount of load leveling. Recall that this is due to the tire's lateral force generating capability, which is proportional to its normal load; consequently, load leveling, which <u>reduces</u> rear tire load, reduces the understeer. As listed in Table 19, five different trailers yielding seven different configurations were tested at the nominal "+25 percent hitch load transfer to the front axle" versus a "minimal" load leveling. Minimum leveling was defined as that required to relevel the CV after air lift shocks are used to their maximum. It turned out that the air shocks alone could relevel approximately 300 lb hitch load. For these seven cases listed in Table 19 the average increase in rear axle vertical load with "minimum" leveling was 446 lb <u>more</u> than with "+25 percent" load leveling. This added load was still within the tire's maximum rated load and improved the understeer gradient about 1.1 deg/g.

As stated previously, the results presented in Figs. 45 and Table 19 are not unexpected. However, as a matter of practical use in the development

TABLE 19

AVERAGE CHANGE IN UNDERSTEER GRADIENT AS A FUNCTION OF LOAD LEVELING FOR INTERMEDIATE TOW CAR

TRAILER		LOAD LEVELING		HITCH LOAD	REAR AXLE LOAD CHANGE DUE TO LOAD LEVELING	UNDERSTEER CHANGE DUE TO LOAD LEVELING
TYPE	WEIGHT (%)	MINIMUM (%)	NOMINAL (%)	(lb)	(lb)	(deg/g)
18 ft Travel	3000 3900	None −25	+25 +25	390 780	−540 −560	−1.0 −0.4
19 ft Travel	4000	−32	+25	600	−515	−0.6
22 ft Travel	4000	None	+25	400	−360	−1.0
27 ft Travel	6000 6000	−15 −20	+25 +25	600 900	−425 −465	−1.6 −0.6
Boat	3000	None	+25	450	−458	−2.5

of a handling safety standard, we must utilize these results in a different way in order to validate the tentative rule format presented in the previous subsection. This can be done by determining the lateral acceleration at which the tow car understeer goes to zero. This value is determined from the point of zero slope on the steer angle versus lateral acceleration plot (or turn radius versus speed squared plot). These curves are presented in Vol. III, Appendix F. This performance measure can then be viewed as a handling boundary once a minimum lateral acceleration criterion is established.

The lateral acceleration at $K = 0$ has been summarized in Table 18 along with the estimated lateral acceleration for jackknife. Since not all tests achieved the goal of 0.5 g lateral acceleration, the highest value achieved is prefaced by a > sign. In other words, the value of lateral acceleration for neutral steer (or jackknife) was greater than that achieved in our test. It is footnoted that due to the left/right turn differential the lateral acceleration for neutral steer is given for the worst case, i.e., the right-hand turn. Left-hand turns could sustain a higher lateral acceleration before transitioning to neutral steer.

The $a_{y_{K=0}}$ results presented in Table 18 are graphically presented in Fig. 47 as a function of hitch load. As expected, they also follow the same trend with increasing hitch load and load leveling as did the understeer plots of Fig. 45. If we were to accept, say, 0.3 g as a minimum safe boundary, then the allowable hitch loads would be significantly lower than those previously derived for neutral steer at zero lateral acceleration. However, the data in Fig. 47 are too scattered to attempt an empirical fit. The next subsection replots these results in a form comparable to the analytically derived boundary line (for neutral steer at 0.3 g cornering) and hence provides some guidance in selecting a maximum hitch load boundary based on the tentative criterion.

C. COMPARISON WITH TENTATIVE STANDARDS FORMAT

The tentative standard format described in Subsection A was based on providing a hitch load condition that would insure positive tow car understeer up to some lateral acceleration level. The analysis was based,

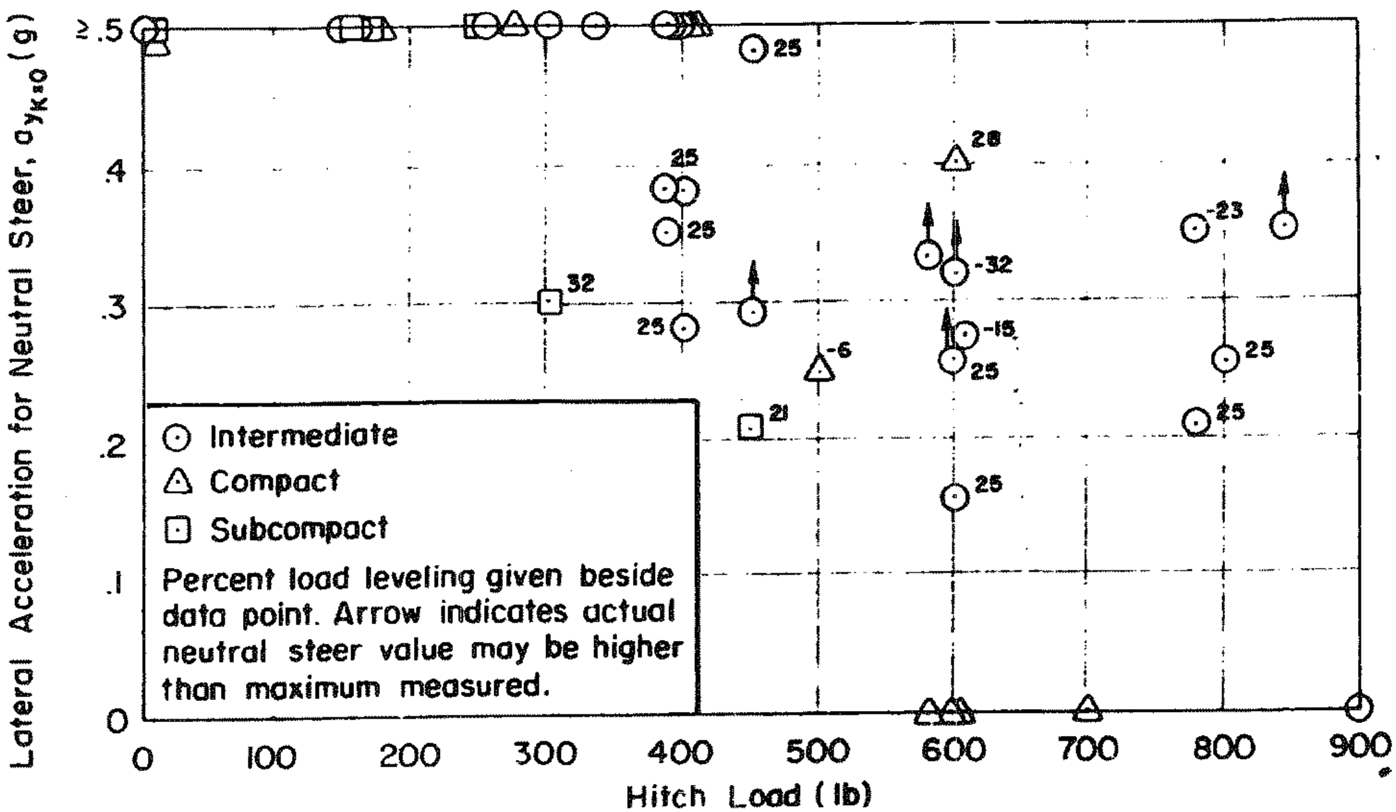


Figure 47. Effect of Hitch Load on Lateral Cornering Capability

somewhat arbitrarily, on 0.3 g, with the hope that this would also provide a cornering capacity of up to 0.5 g without jackknifing. In Figs. 48-50 the results previously presented in Table 18 are related to each hitch load/weight ratio condition in order to test the applicability of this format. Also noted in Figs. 48-50 are the load leveling percentages, since this variable also influences the location of the boundary lines. We will discuss each figure individually in order to see how the boundaries fit the data, or may be adjusted to fit the data.

Figure 48 presents the 25 combination-vehicle configurations towed by the intermediate-sized car. Close comparison of the boundaries versus conditions shows that in 20 out of 25 configurations the boundaries do properly separate the data. Of these 20 configurations, 13 were predicted to pass (i.e., were able to provide higher than 0.3 g before neutral steer) and 7 were predicted to fail (i.e., were unable to provide 0.3 g before neutral steer). In four cases (circled) the boundaries are too conservative. In other words, the CV was not expected to maintain understeer up to 0.3 g cornering, yet it did. In one case (dashed circle) the actual point of neutral steer was not reached at the maximum lateral acceleration tested of 0.26 g. Hence, we do not know if this configuration would have exceeded 0.3 g; although it does appear that the applicable boundary (+25 percent LL) is again too conservative.

In terms of practical application of the loading boundaries we can illustrate an example case using the intermediate tow car attempting to pull the 27 ft travel trailer. Note that none of the data points in Fig. 48 for this trailer passed the tentative criterion. In other words, at hitch loads of 10 percent or greater with recommended load leveling the CV cannot maintain understeer up to 0.3 g Hitch loads higher than 10 percent, without load leveling, should be able to pass; however, the rear tire load capability would be exceeded and the CV would be in a very tail-low attitude. Even with minimum load leveling a hitch load higher than 10 percent on this tow car would not result in maintaining understeer up to 0.3 g cornering. On the other hand, it was shown in Section IV that hitch loads less than 8 percent will produce excessive trailer swing (ζ_{55} < 0.15). Recall from Section IV that at 5 percent hitch load this

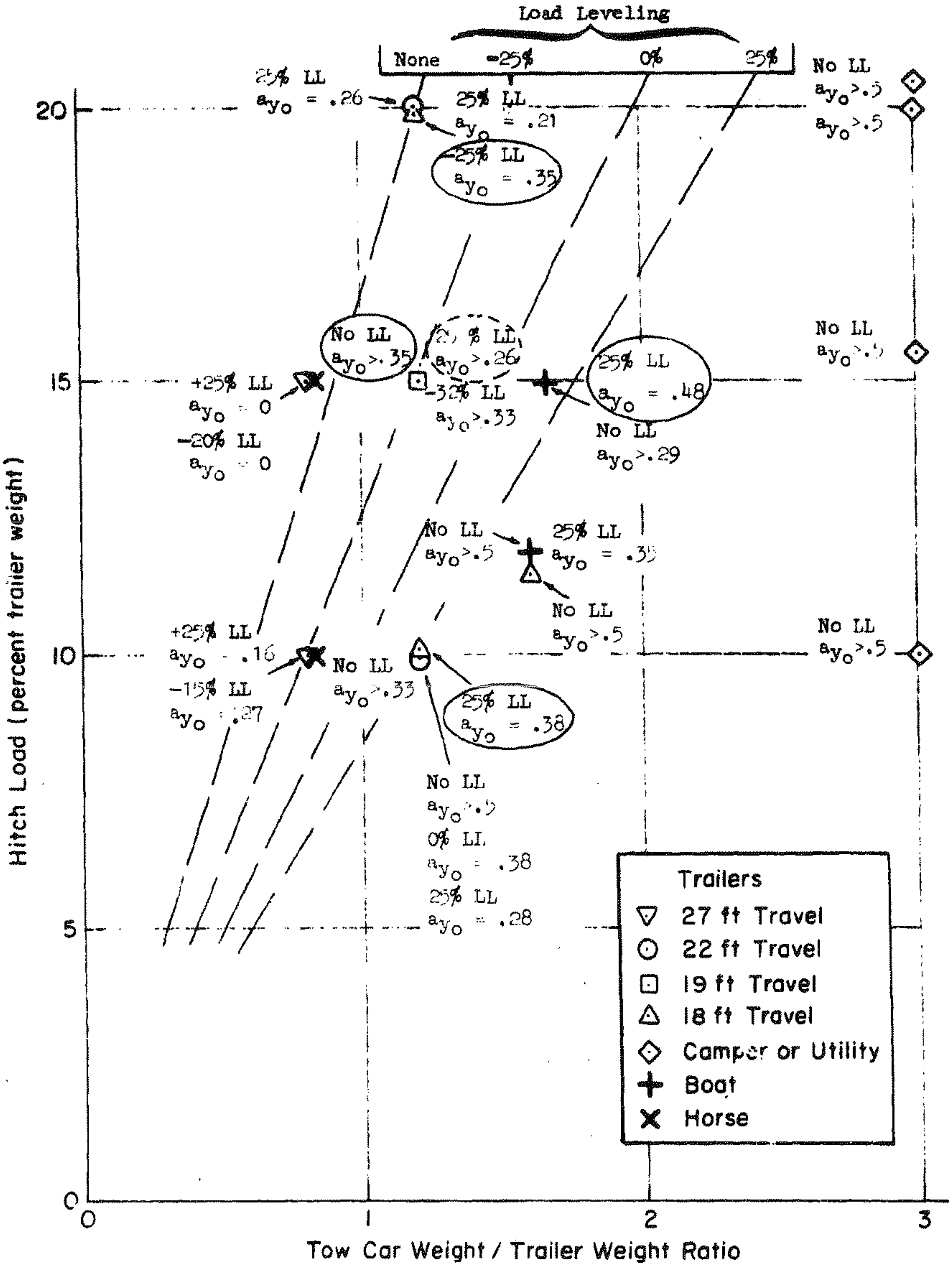


Figure 48. Comparison of Intermediate Tow Car Test Data with Allowable Hitch Load Boundaries

trailer was <u>unstable</u> above 45 mph when no automatic sway controls were used. Based on the above, it appears that for this tow car/trailer weight ratio (i.e., 0.78) the best hookup from a stability standpoint would be a hitch load between 8 and 10 percent with <u>no</u> load leveling. However, in practice, the 8-10 percent hitch load/no load leveling configuration would also not be acceptable, since the CV could not be releveled with air shocks alone. In summary, Fig. 48 tells us that this 6000 lb trailer should <u>not</u> be pulled with a 4700 lb tow car. Use of load leveling (necessary for releveling) at 8 percent hitch load would require a tow car/trailer weight ratio of 0.9, i.e., a tow car weight greater than 5400 lb. The use of automatic sway control would reduce this weight requirement, since lighter hitch loads would be possible without excessive trailer swing.

Figure 49 presents the eleven combination-vehicle configurations towed by the compact-size tow car. Again, comparison of the a_{y_0} data with the boundary shows that nine out of ten configurations are properly split by the boundary lines, i.e., four passed the criterion and five failed. One case was expected to fail but passed; again, the boundary was too conservative.

Figure 50 presents the eight combination-vehicle configurations towed by the subcompact-sized tow car. Comparison of the a_{y_0} data with the boundary shows that six configurations passed the criterion, one failed, and one was predicted to fail but passed — with the boundary again being too conservative.

Additional comparisons can be made by utilizing the data published in Refs. 2 and 3. Figure 51 overplots stability data (in terms of jackknife or no jackknife) for six trailers previously tested in these reports. Although the general "yes" or "no" jackknife data are not equivalent to the more quantitative lateral acceleration for neutral steer, $a_{y_{K=0}}$, the fact that the trailer jackknifed or not during a test run below 0.5 g provides a reasonable basis for comparison with the analytically derived boundary lines. In any case, the results shown in Fig. 51 for 29 CVs tested in Refs. 2 and 3 indicate that 16 configurations would be predicted to pass (were below the boundary lines) and <u>did not</u> jackknife, 11 were predicted to fail (were above the boundary line) and <u>did</u> jackknife, and

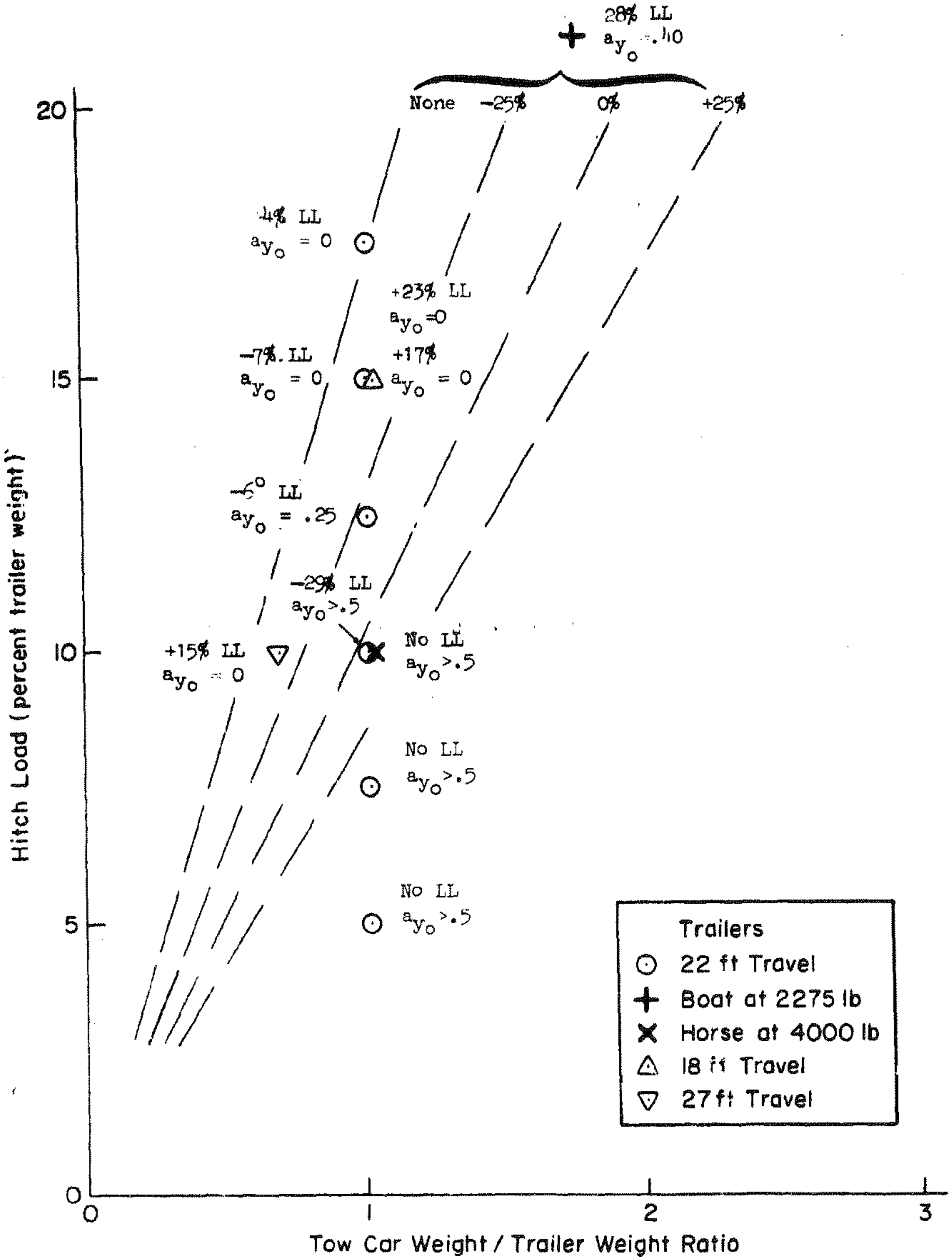


Figure 19. Comparison of Compact Tow Car Test Data with Allowable Hitch Load Boundaries

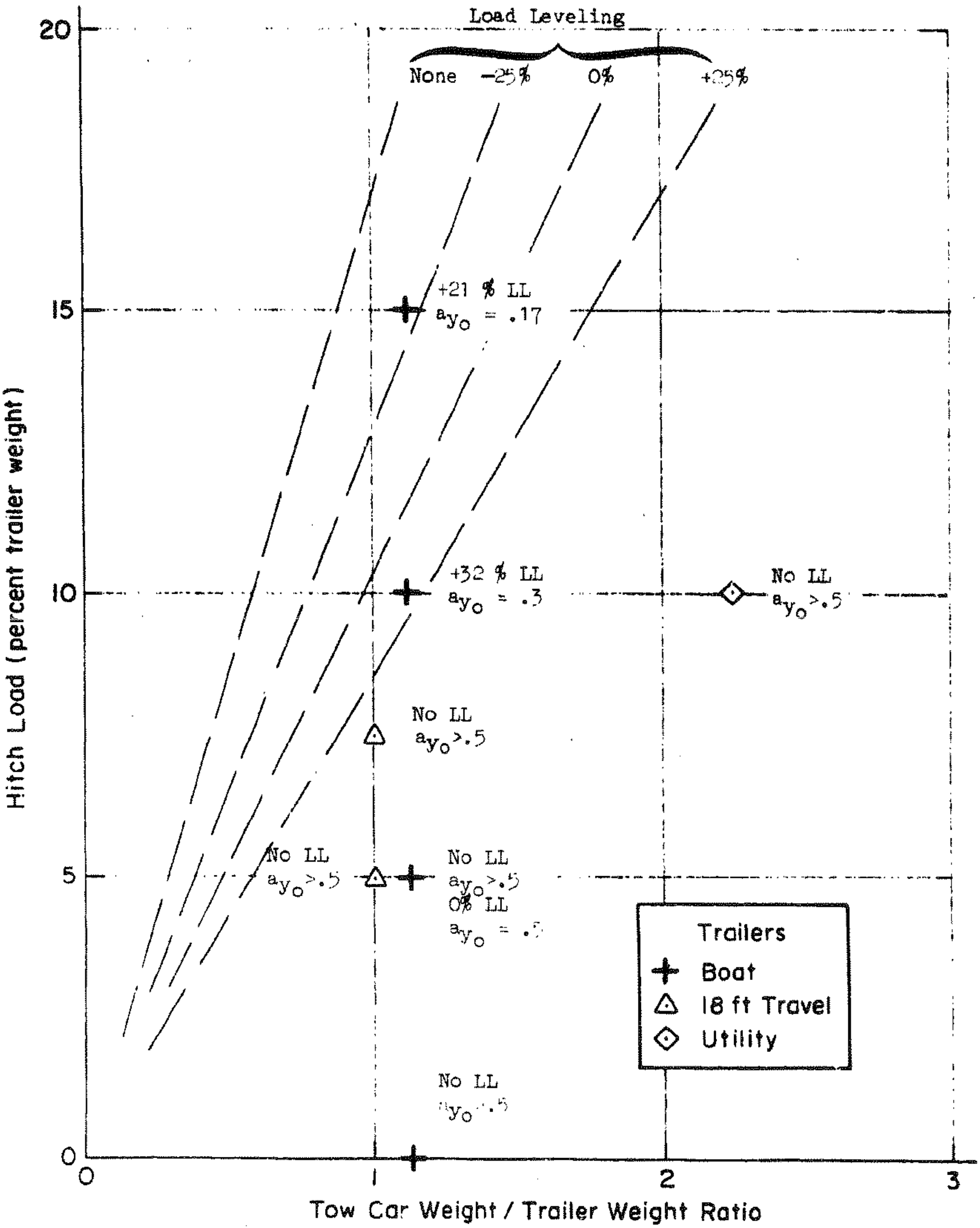
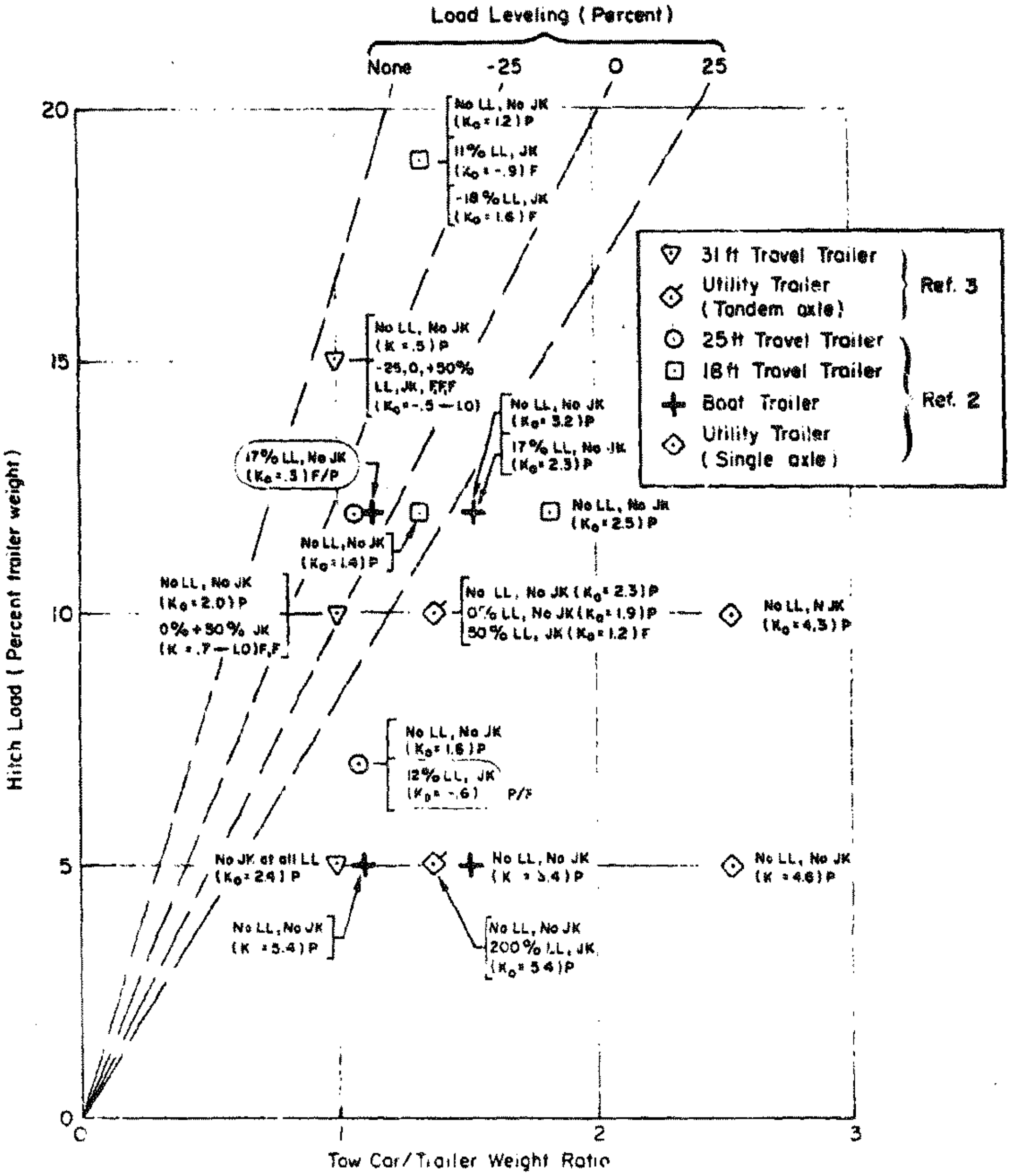


Figure 50. Comparison of Subcompact Tow Car Test Data with Allowable Hitch Load Boundaries



igure 31. Comparison of Previous Test Data with Allowable Hitch Load Boundaries

one configuration should have failed but passed. Only one additional configuration should have been stable (it fell below the boundary line) but actually <u>did</u> jackknife. This latter case represents an undesirable error since a 7 percent hitch load/12 percent load leveling condition was predicted to be safe but was <u>not</u>. There appears to be no reason why this 25 ft travel trailer should exhibit such an oversteer response at low g cornering ($K_o = -0.6$ deg/g).

In summary, the results of tests performed in this program (plus data taken previously) are very consistent with analytically derived maximum hitch load boundary lines. If anything, the boundary line may be slightly too conservative and should be rotated to the left. The next subsection describes a finalized criterion and justifies the selection of 0.3 g lateral acceleration as a reference condition.

D. SELECTION OF TOW CAR STABILITY CRITERION

Based on the comparisons presented in the previous subsection, a tow car stability criterion derived from maximum hitch load considerations appears to represent a valid rule format. The next question is up to what lateral acceleration should the tow car be able to maintain an understeering characteristic. A 0.3 g level was tentatively used as the test case, but no justification was given for this selection. This subsection provides some rationale for this criterion and then adjusts the hitch load boundary lines to better fit the empirical data.

In terms of the tow car alone, all passenger cars are designed to be understeering at all lateral acceleration levels. Depending on the tires, suspension, weight distribution, etc., this understeer may be a constant over the range of cornering g's (as was the case with the intermediate tow car) or may exhibit increased understeer at higher g cornering (as was the case for the compact and subcompact test cars). This latter characteristic results in a decreasing turn rate (or larger turn radius) as forward speed is increased. A requirement for this characteristic was specified for the Experimental Safety Vehicle (Ref. 25). However, when towing a trailer, it has been well established that the opposite occurs;

consequently, we must specify the conditions for the minimum acceptable understeer level.

The basis for this selection is the maximum lateral accelerations used by drivers in emergency maneuvering. Several researchers have measured this. For example, Ref. 26 indicates that nearly all of the 100 driver sample (in two cars) were unwilling to use more than 0.35 g in emergency obstacle avoidance. References 27 and 28 show a much wider range of peak lateral acceleration in studies using 48 drivers in 12 vehicle configurations. These data, summarized in Ref. 29, show a median peak value of about 0.4 g for evasive maneuvering. Considering that towing a trailer would make a driver even more conservative in maneuvering, it is not unreasonable to assume a lower limit value of, say, 0.3 g prior to the vehicle exhibiting an unstable behavior. On the other end of the spectrum we must expect at least 0.16 g, since highway designers (Ref. 30) base their maximum degree of roadway curvature on a side friction factor (i.e., lateral acceleration) of this level. Since there does not appear to be any additional justification data in this area, and the sources noted above are consistent with the tentative criterion, we will therefore assume the following:

> The maximum hitch load will be such as to insure the tow car (of any combination-vehicle configuration) to be able to maintain an understeer characteristic up to and including 0.3 g lateral acceleration.

Based on the results presented in the previous subsection, the above criterion can be met using a loading graph such as shown in Fig. 52. These modified boundaries are less conservative than those derived analytically and hence represent a better fit to the empirical data presented in Figs. 48 to 50. The boundary line for maximum load leveling (+25 percent) has been rotated to the left by 10 percent and a single "minimum" load leveling boundary has been placed midway between the "none" and the previous "+25 percent" load leveling boundaries. This single boundary replaces the previous 0 and −25 percent load leveling values in Fig. 44 and represents what would be necessary to relevel a combination-vehicle *if* air shocks were used

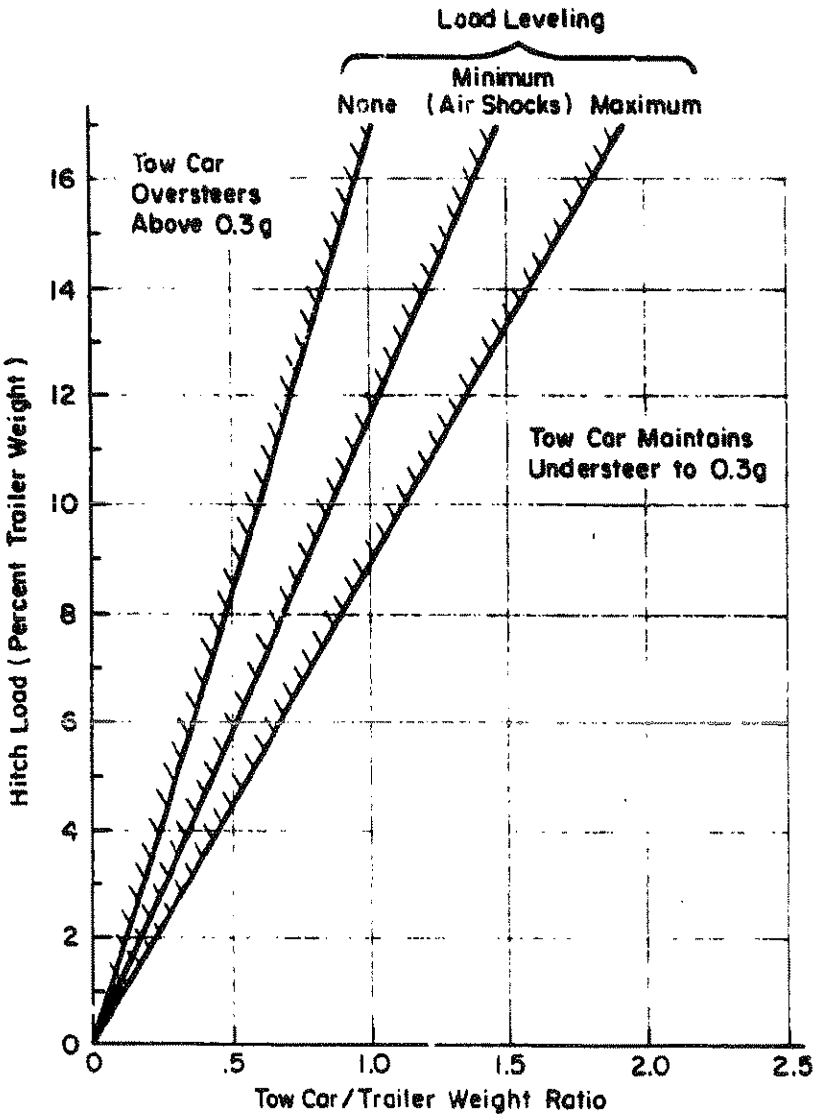


Figure 52. Proposed Maximum Allowable Hitch Load Boundary to Provide Understeering Tow Car up to 0.3 g Cornering

to relevel prior to applying any load leveling torque. This selection of boundaries is also more easily applied by a user, since it is almost impossible to determine the percentage hitch load transferred to the front axle of the tow car without very accurate scales under each axle. On the other hand, a "releveling" procedure using differential bumper height is well accepted in practice and easy to implement (refer to Ref. 3).

Combining the maximum hitch load boundary proposed in Fig. 52 with the previously determined minimum hitch load boundary (i.e., that necessary to insure acceptable trailer swing damping at 55 mph) results in an integrated tow car/trailer handling standard. This approach is recommended in the next subsection. Also presented in the next subsection are test procedures that can be used by manufacturers to evaluate the tow car stability boundary for their trailers.

E. RECOMMENDATIONS

Recommendations for two areas are discussed. The first pertains to a proposed integrated handling standard format which treats both the tow car and trailer stability characteristics. The second pertains to recommended test procedures for determination of passenger car stability when towing a trailer.

1. Integrated Handling Standard

Due to the success of the hitch load versus tow car/trailer weight ratio format used for both trailer swing damping (Section IV) and tow car steady-state turn stability, these tow boundaries can be integrated into one plot which specifies both a minimum and maximum hitch load. This is illustrated in Fig. 53 for four test trailers previously compared in Section IV, Figs. 37-40. In using this "allowable" hitch load plot, note first that each trailer has its own minimum hitch load boundary. Determination of this boundary is primarily a function of trailer wheelbase and moment of inertia. On the other hand, the maximum hitch load lines are the same for each trailer since this boundary is based on tow car stability. The allowable hitch load range falls between the two boundaries.

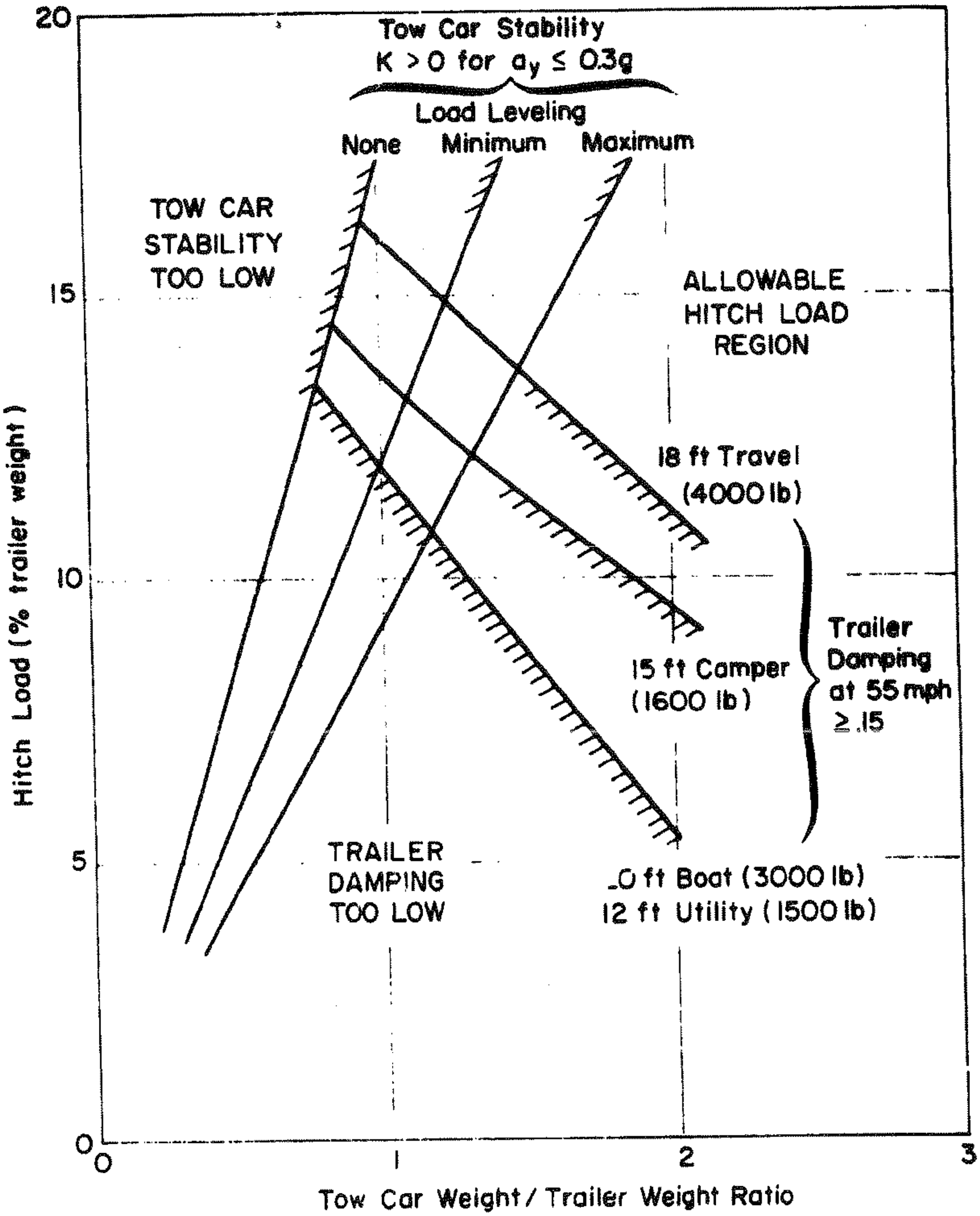


Figure [illegible]. Example Integrated Hitch Load Boundary Limits Based on Trailer Swing Damping and Tow Car Stability

For example, if the boat trailer were to be towed by a car weighing 1.5 times the trailer weight, then the allowable hitch load range would be from 8-1/2 percent to 17.5 percent. This latter hitch load value would require at least minimum load leveling.

The hitch load range for the 18 ft travel trailer is much more restrictive due to its lower swing damping. In fact, this 4000 lb trailer would probably require a tow car weighing at least 5000 lb, since a minimum hitch load with load leveling would be 15 percent (or 600 lb).

Since each trailer model will have a unique minimum hitch load boundary, the format of Fig. 53 could also be handled in absolute terms, i.e., the plot could be transformed to hitch load weight vs. tow car weight. This format is illustrated in Figs. 54a to 54h for each of the eight trailers tested in this program. The boundaries illustrated on each plot are based on the following:

a) Use of Fig. 52 for all trailers.

b) Use of analytically derived minimum hitch load boundary for each trailer based on Eq. 11 of Section IV (Figs. 36-40, for example). In essence, the analytically derived boundaries predict the general magnitude and slope of the required hitch load versus tow car weight.

c) Shifting the analytically derived minimum hitch load boundary up or down to match test data. This is especially important when load leveling is used.

Preparation of this type of plot is recommended as part of the trailer manufacturing test process and could even be included as part of the owner's manual.

Although the trailer examples presented in Fig. 54 give a good overall picture of the tow car/trailer tradeoffs, application by a user probably always will start with a specific tow car. In this case, the upper hitch load limit may be dictated by the tow car manufacturer due to limitations of power, cooling, structure, etc. Generally, the manufacturer's limit will occur <u>prior</u> to reaching the stability limit. For example, many subcompacts recommend hitch loads no more than 100 lb; whereas Fig. 54a would allow more than 300 lb. In short, manufacturers' maximum hitch load recommendations should <u>always</u> take precedence.

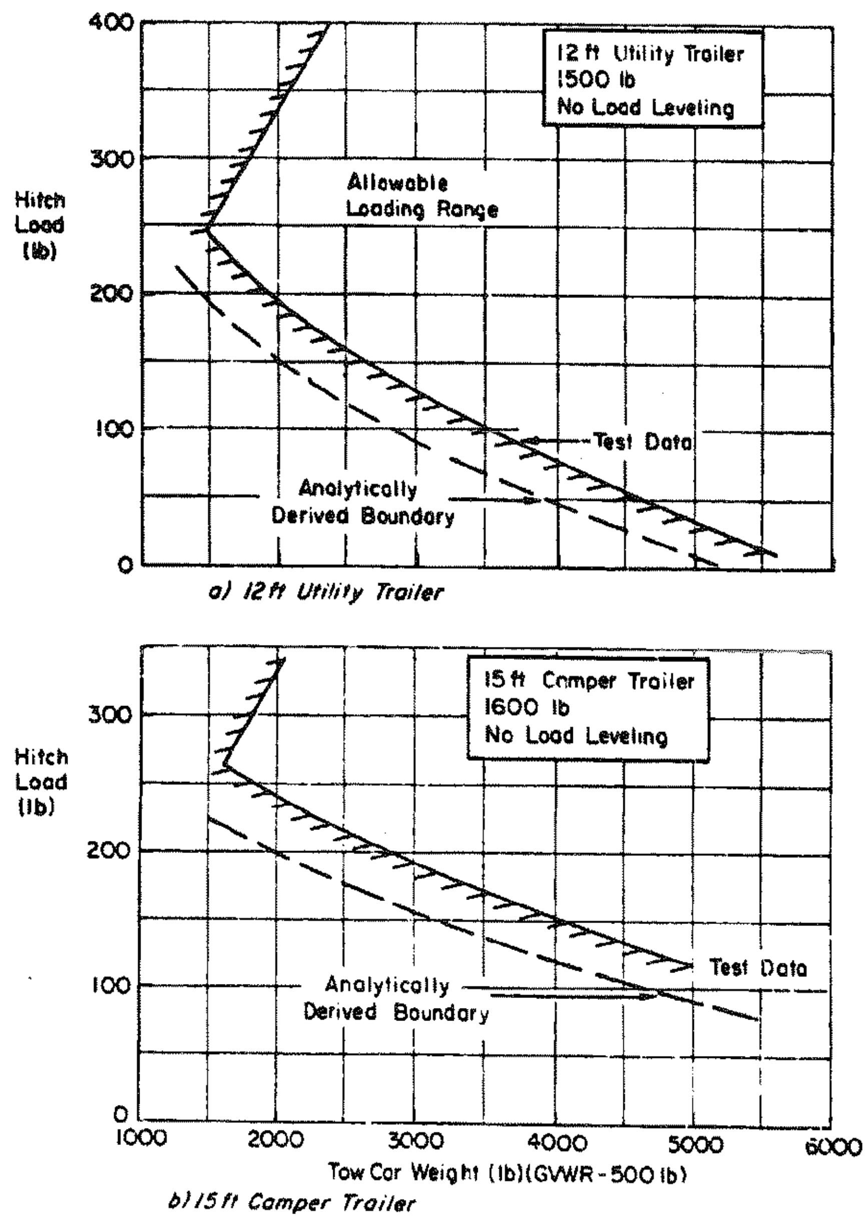
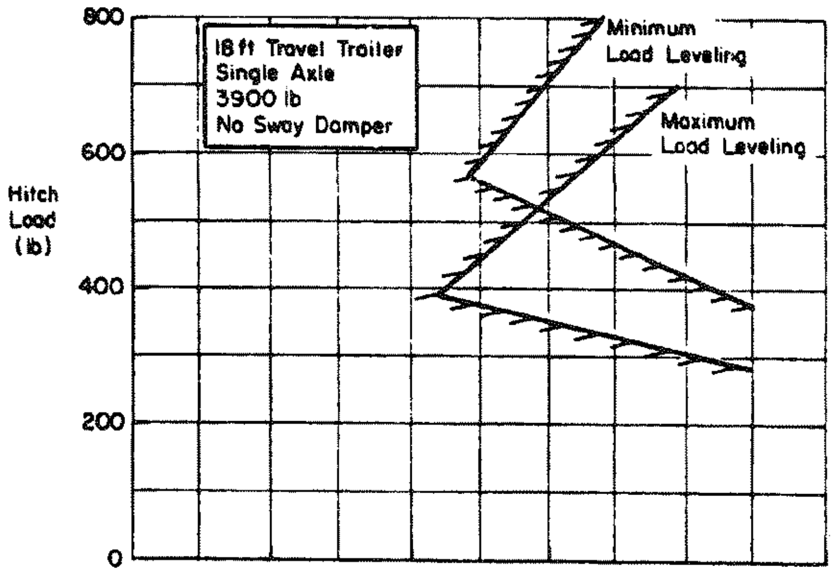
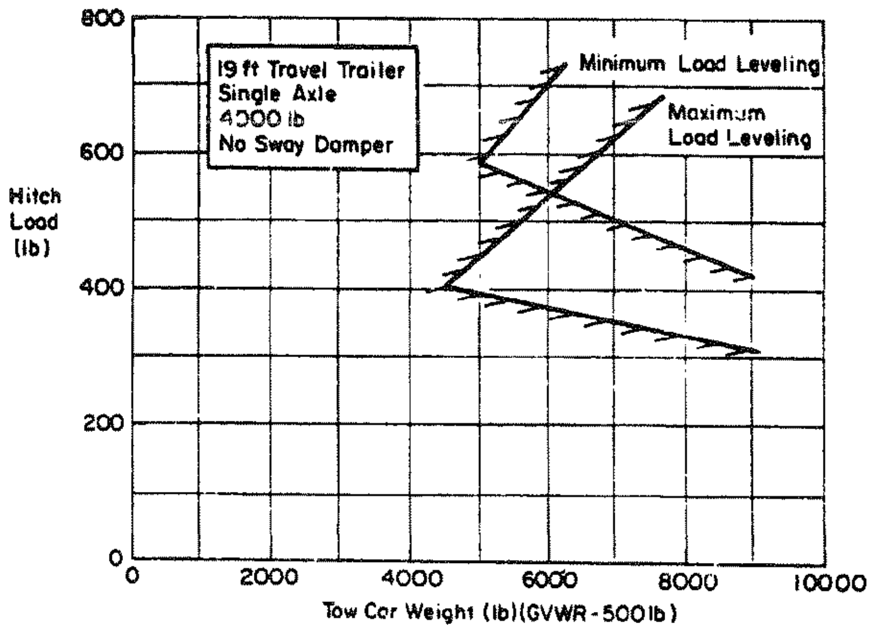


Figure 54. Recommended Integrated Trailer Handling Standard Examples

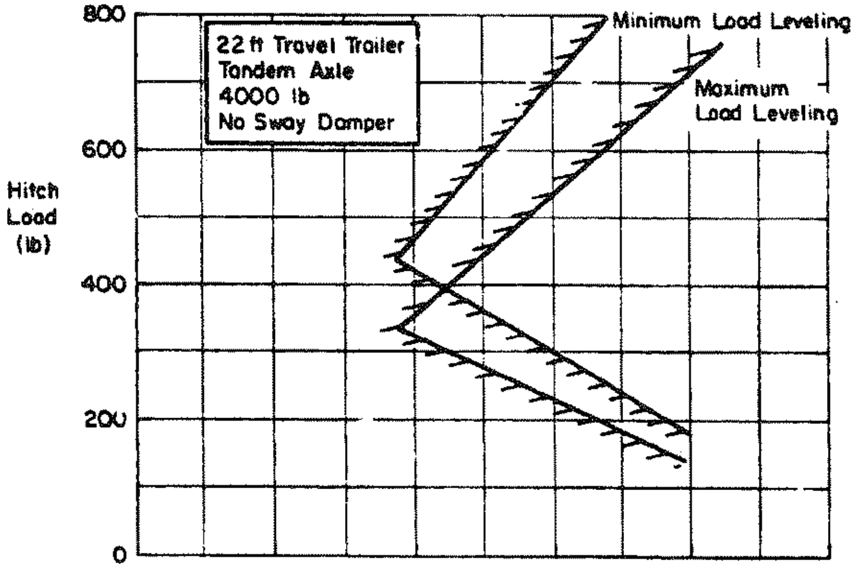


c) 18 ft Travel Trailer

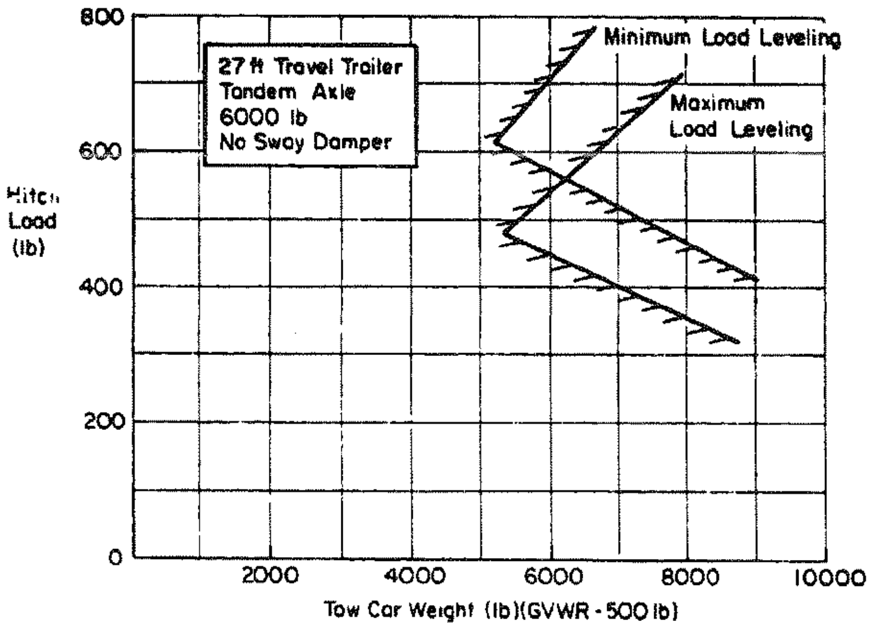


d) 19 ft Travel Trailer

Figure 54. (Continued)



e) 22 ft Travel Trailer



f) 27 ft Travel Trailer

Figure 54. (Continued)

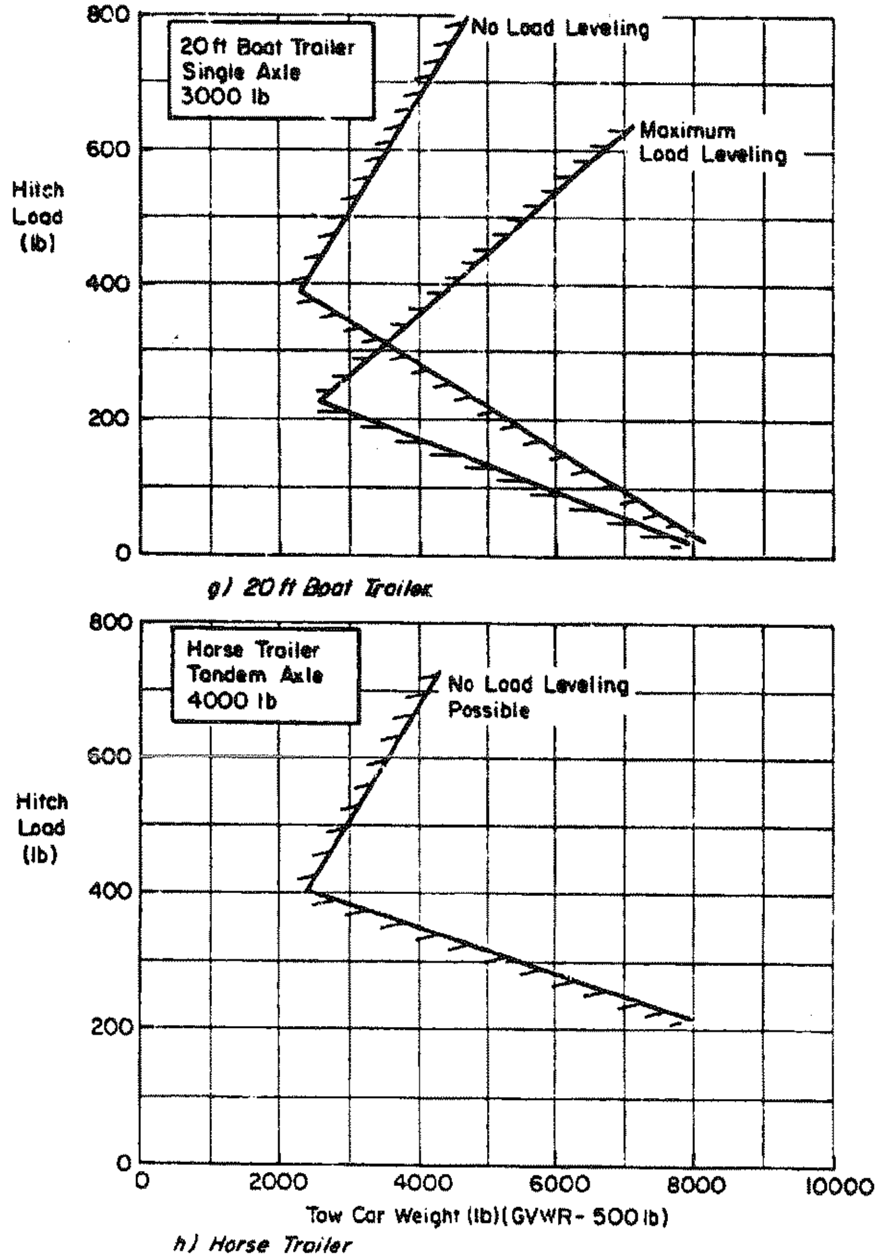


Figure 54. (Concluded)

2. Improved Test Procedure

Several procedural problems were uncovered (and solved) during this test program. These relate to determination of understeer gradient for a CV and effects of environmental factors. First, while the "step steer" test procedure recommended in Ref. 2 is a well-proven method for determining tow car understeer gradient, the constant radius circle test recommended by SAE (in Ref. 11) is more applicable for CVs that exhibit a non-linear variation in understeer gradient with lateral acceleration. In particular, this latter procedure provides a closer determination of the lateral acceleration level at which the tow vehicle becomes neutral steering.

Implementation of the constant radius method for trailer towing, however, requires a turn circle radius of 200 ft, as opposed to the 100 ft minimum recommended for passenger cars. This larger radius circle provides 0.3 g cornering at 30 mph and a maximum test speed of only 40 mph in order to exceed 0.5 g. The 40 mph speed requires 64 percent less power than the step steer procedure recommending a maximum speed of 50 mph. In this regard, a larger turn circle would not be acceptable, since it would not allow high enough lateral accelerations before running out of engine power (due to the concomitant higher forward speeds). Smaller radius circles would also not be acceptable, since they would not provide high enough forward speed to produce representative trailer articulation angles.

In addition, the constant radius method enables the change in trailer understeer gradient to more easily be determined, since trailer articulation angle is changing in a continuous manner. It should also be reiterated that due to engine torque effects right-hand data have a lower understeer gradient than left turns; consequently, although both left and right turns should be performed, the worst case (i.e., most critical) condition may be more meaningful.

Other procedural recommendations are as follows:

- Ambient wind has a significant effect on understeer measurements. For example, with the 27 ft travel trailer, a wind of 30 mph produced 40 percent changes in measured yaw rate data.

- Road surface should be smooth in order to minimize wheel bounce which complicates the determination of steer angle.

SECTION VI

COMBINED BRAKING AND CORNERING

The fourth combination-vehicle test procedure was aimed at uncovering tow car and/or trailer stability problems during a brake-in-turn maneuver. Ideally, if the automobile and trailer meet the individual and combined vehicle requirements developed previously, then there should be little or no response degradation during the combined maneuver. If a response degradation does occur, then the preceding requirements should be altered or new requirements developed.

Based on the above objectives, a brake-in-turn test procedure was accomplished using the steady-state 0.3 g turn at 40 mph procedure recommended in Ref. 2. A 355 ft constant radius circle was used. Brake pressures were increased on successive runs until wheel lockup (or jackknife) occurred. Results of these tests were compared with a preliminary stability requirement suggested by the results of Ref. 2.

It should be mentioned here at the outset that for <u>all</u> configurations no jackknife conditions occurred, and in <u>all</u> cases the CV exceeded 0.4 g deceleration. In only twelve limit deceleration cases was there a slight transient oversteer tendency; however, no loss of control tendency was observed.

A. PRELIMINARY REQUIREMENTS

A combined vehicle performance parameter was developed in Ref. 2 to relate initial tow vehicle directional rate of change (stability factor change) and deceleration level. Time duration of any adverse response was also a weighting factor. For example, Ref. 2 data suggested that yaw rate change per unit speed change, $\Delta r/\Delta U$, greater than $|0.3|$ (deg/sec)/mph sustained for 1 sec or longer <u>would</u> result in a car/trailer jackknife. It was felt that it would be almost impossible for the average driver, within one second, to recognize the onset of a jackknife condition and initiate corrective action (reduce braking and introduce countersteer). Therefore, the following safety boundary was recommended:

During a brake-in-turn at 0.3 g lateral acceleration the peak $\Delta r/\Delta U$ should be less than 0.30 (deg/sec)/mph and/or $\Delta r/\Delta U$ oversteer durations less than 1 sec for a 0.4 g deceleration from 40 mph.

In addition, it was hypothesized that it would be necessary to determine the point at which the ratio of longitudinal hitch forces to tow car weight produce a jackknife condition during a brake-in-turn maneuver. This occurs when the trailer braking is less than the tow car and hence would provide an additional constraint on the minimum trailer-alone brake capability.

B. FULL-SCALE TEST RESULTS

All test runs are identified in Table 20; however, only the maximum deceleration runs (or those exhibiting lockup) are of interest. Lower deceleration levels showed no unusual characteristics.

Results include maximum deceleration level, effective turn radius, maximum yaw rate change per unit speed change, and wheel lockup. Ideally, the turn radius would remain constant at 355 ft; however, any tightening of the turn (smaller radius) can be viewed as a transient oversteer tendency.

Only 12 out of 38 limit deceleration tests resulted in a transient oversteer tendency, and none of these cases resulted in a jackknife. Since none of the increased yaw rate transients exceeded 1 sec time duration, each of these runs falls below the previously suggested criterion and no conclusions can be drawn.

C. RECOMMENDATIONS

No additional recommendations can be added to the suggested braking standards provided in Sections III and IV and the integrated handling standard provided in Section V. It is suggested, however, that demonstration of a 0.4 g deceleration during 0.3 g cornering be maintained as part of the test procedure requirements.

TABLE 20. BRAKE-IN-TURN TEST RESULTS

TOW CAR	TRAILER	HITCH LOAD (Percent)	LOAD LEVELING (Percent)	$a_{x_{max}}$	$\Delta r/\Delta U$ $\left(\frac{deg/sec}{mph}\right)$	Δt (sec)	R_{min} (ft)	TOW CAR LOCKUP	RUN NUMBER
Inter-mediate	Utility 1500 lb No brakes	0	N	 .46	 0	Non-Limit	 355	 —	149, 151, 152 150
		5	N	 .41	 0	Non-Limit	 355	 —	142-146
		10	N	.52 .46 .57 .54	0 .48 0 0	 .43 	355 245 355 355	— RF — RF	99 100 101 102
		20	N	 .42		Non-Limit			127-133 134
	Camper 1600 lb No brakes	5	N	 .50 .46 .52	 0 0 0	Non-Limit	 355 355 355	 RF LF —	254, 258, 259 255 256 257
		10	N	 .44 .47	 .70 0	Non-Limit .80 	 230 355	 LF —	222, 223, 225-228 224 229
		15	N	 .48	 0	Non-Limit	 355	 —	205-211 204
		20	N	 [illegible] 43	 0 0	Non-Limit	 355 355	 LF —	180, 183-187 181 183
	18 ft Travel 3000 lb	13	N	 .45	 0	Non-Limit	 355	 —	448-452, 454 453
			+25	 .53	 0	Non-Limit	 355	 —	405-410, 412 411
	18 ft Travel 3900 lb	10	+25	 .47	 0	Non-Limit	 355	 —	534-538 539
		20	-25	 .40 .41 .39	 0 .69 0	Non-Limit .60 	 355 252 355	 — RF LF	470-473, 475, 476 474 477 478
			+25	 .47 .48 .44	 0 .27 .50	Non-Limit .30 .80	 355 238 200	 — — LF, LR	522-525, 529-53[illegible] 526 527 528
	[illegible] ft Travel [illegible] lb	[illegible]	[illegible]	 [illegible]	 0	Non-Limit	 355	 —	809-811, 813-816 812
			[illegible]	 [illegible]	 0	Non-Limit	 [illegible]		771-778, 780-782 779

(Continued on following page)

TABLE 20. (CONCLUDED)

TOW CAR	TRAILER	HITCH LOAD (Percent)	LOAD LEVELING (Percent)	$a_{x_{max}}$	$\Delta r/\Delta U$ $\frac{deg/sec}{mph}$	Δt (sec)	R_{min} (ft)	TOW CAR LOCKUP	RUN NUMBER
Inter-mediate	22 ft Travel 4000 lb	10	N			Non-Limit			329-331, 333, 334, 336, 337
				.46	0		355	—	332
				.40	.40	.70	247	LF	335
			0			Non-Limit			302-312, 314
				.63	.27	.80	236	—	300
				.52	0		355	—	301
				.46	.45	.90	194	LF	313
				.43	.47	.70	244	LF	315
			+25			Non-Limit			349-353
				.46	0		355	—	354
		20	+25			Non-Limit			369-371
				.47	0		355		372
	27 ft Travel 6000 lb	10	+25			Non-Limit			594-596, 598-601
				.48	0		355	—	597
		15				Non-Limit			669-676
				.44	0		355	—	668
						Non-Limit			635, 636, 638-645
				.46	0		355	—	637
	Boat 3000 lb Surge brake	10	N			Non-Limit			928-929
				.54	0		355	—	930
Compact	18 ft Travel 3900 lb	15	+25			Non-Limit			1300-1302, 1304-1307
				.61	0		355	--	1303
	22 ft Travel 4000 lb	15	+17			Non-Limit			1133-1136, 1138-1140
				.49	0		355	—	1137
	Horse 4000 lb	10	N			Non-Limit			1257-1262, 1264
				.41	0		355	—	1263
Sub-compact	Utility 1500 lb No brakes	10	N			Non-Limit			1637-1720
				.46	1.0	.40	355	LF	1715
	Camper 1700 lb No brakes	10	N			Non-Limit			1718-1720
				.46	1.0	.40	213	LF	1715
				.48	0		355	—	1716
				.42	1.2	.35	200	LF	1716
	Boat 5000 lb Surge brake	10	+30			Non-Limit			1587-1591, 1593-1595
				.46	1.2	.40	220	LF	1592
				.49	0		355	—	1596

SECTION VII

SUMMARY AND RECOMMENDATIONS

This program has provided the fourth step in the process of developing motor vehicle safety standards covering the handling and braking performance of passenger cars pulling trailers. Previous work has established meaningful test procedures, performance measures, and in one case a proposed rule format. This current step has proposed and evaluated a justifiable set of performance criteria and tested over 90 different combination-vehicle configurations against them. This section summarizes these criteria, suggests means for insuring compliance, recommends manufacturers' test procedures, and offers user guidelines.

A. HANDLING AND BRAKING PERFORMANCE CRITERIA

The following performance criteria were suggested for passenger car/ trailer combinations:

- All combination-vehicles shall be capable of stopping within 134 ft from 40 mph, i.e., average deceleration of 0.4 g.
- All trailers of a combination shall exhibit a minimum trailer swing damping ratio of 0.15 (i.e., 3/4 cycles to one-half amplitude) at 55 mph.
- All tow cars of a combination shall exhibit a positive understeer gradient up to and including 0.3 g cornering.
- All combination-vehicles shall demonstrate 0.4 g deceleration during 0.3 g cornering without incurring transient oversteer (increased yaw rate) longer than 1 sec duration.

B. MEANS FOR INSURING COMPLIANCE

Using the above criteria it was possible to derive tow car and trailer characteristics that would insure the combination-vehicle meets the requirements. For braking, these design or hookup features include:

- To insure 0.4 g combination-vehicle deceleration the tow car to trailer weight ratio shall be greater than or equal to:

$$\frac{W_c}{W_t} \geq 2.1 - 3.5\, a_{x_{ta}}$$

where $a_{x_{ta}}$ is the trailer-alone braking capability in g's. The greatest impact of this requirement would be on unbraked trailers, i.e., an unbraked trailer could not weigh more than 48 percent of the tow car. For braked trailers capable of providing 0.4 g deceleration (at GAWR), the trailer weight could, theoretically, be increased to 1.4 times the weight of the tow car.

- An alternate solution would specify a minimum trailer-alone deceleration capability of 0.44 g for Class II or larger trailers. This could easily be accomplished by not allowing more than 1500 lb to be supported by each 10 in. brake.

- Stopping distance should be the primary performance measure. This measure accounts for brake actuation time delays. Average deceleration can be computed from stopping distance.

- Surge brake gains should be greater than 2.0.

Trailer and tow car stability can be insured by specifying the following:

- Sufficient trailer damping (for each trailer model) can be provided by specifying a minimum allowable hitch for each tow car weight. This can be derived using the analytical/empirical equation:

$$\zeta_{cv} \doteq \underbrace{\frac{\sqrt{\ell_2^3 Y_{\alpha_3}}}{U_o \sqrt{2 I_{t_h}}}}_{\text{Trailer Alone}} - \underbrace{\left\{ \frac{I_{t_o} - \frac{W_t \ell_2^2}{g}\left[\frac{\%HL}{100} - \left(\frac{\%HL}{100}\right)^2\right]}{I_{t_h}} \right\}}_{\text{"Hookup Factor"}} \underbrace{C_1 \frac{W_t}{W_c}}_{\text{Tow Car Sensitivity}}$$

where

ζ_{cv} = Minimum damping ratio requirement, e.g., 0.15 at 55 mph

ℓ_2 = Trailer wheelbase (ft)

Y_{α_3} = Trailer tire cornering stiffness (lb/rad)

U_o = Forward speed (ft/sec)

g = Gravity (32.2 ft/sec^2)

I_{th} = Trailer moment of inertia about hitch (slug-ft^2)

I_{t_o} = Trailer moment of inertia about c.g. (slug-ft^2)

W_t = Trailer weight (lb)

W_c = Tow car weight (lb)

%HL = Percent trailer weight at hitch point

C_1 = 3.7 (empirically derived)

Given these data and a desired performance criterion value for the combination-vehicle damping ratio, ζ_{cv}, the minimum allowable hitch load for each weight tow car can be derived. This type of exercise was illustrated in Figs. 37-40. Full-scale "pulse steer" tests are recommended, especially for larger trailers that require load leveling, in order to check the final location of the minimum hitch load boundary line.

- Sufficient tow car stability can be provided by specifying a maximum allowable hitch load for each tow car weight and amount of load leveling. This relationship, illustrated in Fig. 52, was derived from an analytical basis and adjusted to match results of over 75 combination vehicle configurations. Due to other limitations, maximum hitch loads specified by the tow car manufacturer may be less than that required for stability; hence, the lower limit should take precedence.

- An integrated handling compliance plot can be derived for each trailer. Figures 54a through 54h (in Section V) provided examples of this format for the eight trailers tested in this program. Similar plots could be provided by trailer manufacturers as part of their owners' manuals. It should be noted how close the intersecting boundaries come to matching commonly accepted trailer practice, i.e., minimum tow car to trailer weight ratio must be about 1 or greater, and optimum hitch load should be about 10 percent. Increasing the tow vehicle size quickly opens up the allowable hitch load region. This consistency implies a proper selection of the performance criteria.

C. DESIGN TEST PROCEDURES FOR TRAILERS ALONE

Two procedures should be utilized by trailer manufacturers to determine trailer-alone brake capability and trailer-alone damping ratio.

1. Braking

Straight line braking procedures for the trailer-alone have been detailed in Table 14, which follows the format of SAE Recommended Practice J134 and the Canadian Standards Association proposed Standard D313. Key points of the procedure include:

- Lockup of one wheel on one axle is allowed.
- Brake stops are made from a test speed of $40\sqrt{W_t/W_{cv}}$ (mph) to account for the unbraked mass of the tow car.
- Average deceleration at the static axle weight, W_t, is calculated from stopping distance at 40 mph, SD_{40}. This must be ratioed to the gross axle weight rating, GAWR, if lockup cannot be obtained with trailer brakes.

$$a_{xt_a}\ (g's) = \frac{53.7}{SD40} \times \frac{W_t}{W_{GAWR}}$$

- Brake voltages are increased in increments up to maximum, at which point six incipient lockup runs are made.

For surge brake trailers a special apparatus is necessary to apply the brake pressure input. In this case incremental pressure changes replace the incremental voltage changes used for electric braked trailers. Also, the "surge brake gain," G, i.e., the amount of brake force generated per pound of horizontal hitch force, must be determined. Once this is known the "effective" trailer-alone brake capability for this type of trailer can be determined using the following equation.

$$a_{xta} = \frac{a_{x_{cv}}\left(\frac{G}{1+G}\right)}{1 - \frac{\%HL}{100}}$$

where $a_{x_{cv}}$ represents the combination-vehicle deceleration requirement.

It can also be shown that, theoretically, a surge brake trailer of weight W_t should have the stopping capability equivalent to an unbraked trailer of weight W_t', via the relationship:

$$W_t' = \frac{W_t}{G + 1}$$

2. Trailer Swing

The analytical/empirical expression derived in Section IV requires measurement of the trailer-alone moment of inertia as a function of hitch load. This is readily accomplished with a roller bearing turntable (such as used for wheel alignments), two coil springs, and a stopwatch. The trailer wheels are held off the ground with a block positioned on the roller bearing support turntable. A counterweight is added to the rear of the trailer to balance the hitch load such that the trailer floats freely on the turntable with no offset forces. The trailer is oscillated (by hand), and elapsed time measurements are taken to determine the frequency of oscillation. Knowing this (plus the coil spring rate and wheelbase), the moment of inertia can be calculated using the equations given in Section IV.

D. SUPPORTIVE RESULTS

Significant results reported in Sections III and IV of the report are summarized below.

1. Braking

- Several combination-vehicle configurations tested in this program were unable to exceed 0.5 g deceleration even though the tow cars exceeded the requirements of FMVSS 105-75 and the trailers were not loaded to full GVWR. These results very closely matched the analytical predictions using a generalized static braking model including load transfer, load leveling, and tow car brake proportioning.

- All combination-vehicle configurations tested in this program, except the horse trailer at full GAWR, would theoretically be able to pass a 0.4 g deceleration requirement.

- If loaded to full GVWR, the unbraked utility trailer would theoretically be unable to pass a 0.4 g deceleration requirement with the compact and subcompact tow cars. This result was not verified and should be studied further, since any combination-vehicle deceleration requirement will have the most impact on manufacturers of unbraked trailers.

- Trailer-alone deceleration levels for six braked trailers ranged from 0.36 g to 0.6 g when adjusted to GAWR. The current deceleration design goal for electric brake systems appears to be 0.435 g.

- All three tow cars exceeded the stopping distance requirements of FMVSS 105-75 from 40 mph during second effectiveness tests. Average deceleration for incipient lockup was 0.71 g.

- Suspension design of the horse trailer caused an undesirable increase in hitch load with increasing deceleration. Since this trailer does not allow for load leveling, braking tests could not be performed at full GAWR without scraping the hitch on the ground.

- Average actuation time delay of the surge brake system was 0.3 sec. No significant differences due to load leveling could be determined. Previously, as indicated in Refs. 2 and 4, load leveling rendered the surge mechanism inoperative.

- Combination-vehicle stopping distances appeared slightly improved with load leveling; however, the results were not totally consistent.

- Pedal forces in combination-vehicle braking tests were less than the 120 lb recommendation of SAE J135.

- Maximum performance combination-vehicle stopping distances with tow car lockup were not significantly different from those with no tow car lockup.

2. Trailer Swing

- For trailers with low hitch loads (where no load leveling was required) the trailer damping of the combination vehicle was closely predicted by an analytical/empirical relationship.

- There is a strong decrease in damping with increasing speed. This trend is biased up (to higher damping) with higher hitch loads and biased down (to lower damping) with lighter hitch loads.

- Longer trailer wheelbases (i.e., longer hitch to axle distances) are desirable in order to maintain high damping ratios at low hitch loads. In terms of sensitivity, increasing tongue length on an 18 ft travel trailer one foot increases damping 0.08 units. An ideal design goal is to have the wheelbase at least three times the radius of gyration (as measured about the center of gravity).

- Load leveling improves trailer damping. This effect is due, primarily, to the tow car roll steer geometry and hence is difficult to predict. However, empirical results showed an average increase in trailer damping of 0.06 units per 1000 ft-lb of applied load leveling torque. This is higher than previously measured in Ref. 3.

- A friction sway damper can significantly improve trailer damping. At 55 mph this type of damper was able to increase damping on a large travel trailer by 0.19 units. The electric brake type of damper acted primarily as a speed control device by limiting speed to that for zero damping.

- Increased trailer moment of inertia (separated load) reduces trailer damping in a predictable manner and hence is an undesirable condition.

- High cornering levels significantly reduce trailer damping. For example, damping ratio of the medium travel trailer was reduced 0.29 units when the lateral acceleration was increased to 0.4 g. This effect was attributable to the loss of cornering stiffness of the trailer tires at high slip angles, which in turn produces a loss in damping according to the trailer-alone damping equation.

- Steering free play can have a significant effect on reducing the trailer damping. For example, allowing steering to be free reduced trailer damping as much as 0.4 units at 55 mph as compared to that measured with steering held fixed. This effect has strong implications for tow cars with excessive steering free play and/or for drivers who allow the steering wheel force feedback to move the wheel. This, in turn, amplifies the trailer swing oscillation. Holding the wheel fixed is the safest procedure.

- A trailer-alone damping test procedure was successfully used with one test trailer to check analytically derived predictions. Results were sufficiently close as to not warrant formalization (or recommendation) of a separate trailer-alone damping test procedure.

3. Tow Car Stability

- For the intermediate tow car a hitch load of 800 lb produced a neutral steer response at low cornering (less than 0.3 g). If viewed as a mass distribution this implies that <u>any</u> loading resulting in a 37/63 front to rear weight distribution would likely result in oversteer at or above 0.3 g cornering.

- The compact and subcompact tow cars became neutral steering at approximately 600 and 400 lb hitch loads, respectively. These levels convert to 39/61 and 41/59 front/rear weight distributions, respectively.

- Load leveling 25 percent of the hitch load to the tow car front axle <u>reduced</u> understeer about 1.1 deg/g from that required to relevel the combination-vehicle with air shocks and minimum or no load leveling.

- Right-hand turn results showed understeer gradients 0.7 deg/g <u>less</u> than those for left-hand turns. This is attributable to rear axle torque effects during the turn at constant speed.

- Due to the general nonlinear variation in understeer gradient with lateral acceleration, a constant radius circle test procedure is further recommended as a combination-vehicle test procedure. This procedure provides a closer determination of the lateral acceleration at which the tow vehicle becomes neutral steering and provides a continuous readout of trailer articulation angle change with speed. This is necessary for trailer stability factor calculations. An optimum radius of 200 ft is recommended.

4. Brake-in-Turn

All combination-vehicle configurations could deceleration (from 40 mph) at or above 0.4 g during a 0.3 g turn without loss of control or transient yaw rate increase sustained for more than 1 sec. Consequently, no additional recommendations can be added to the suggested braking and handling standards previously presented.

E. USER GUIDELINES

Based on the full-scale results of this and previous NHTSA-sponsored trailer braking and handling programs several general recommendations for the user can be offered. These might be used to supplement public information documents such as Ref. 20.

- There is definitely an optimum hitch load for each tow car and trailer combination. Hitch loads too high, even with load leveling, will cause the tow vehicle to "dig in" during sudden turning maneuvers and sharpen the turn even further. The trailer will then tend to push the rear of the tow car into a jackknife position. Hitch loads too light lead to trailer swing. Heavier tow cars reduce the effect of both problems. In general, the tow car gross vehicle weight rating should exceed the trailer gross vehicle weight rating.

- Use of load leveling should be supplemented by the use of air shocks and heavy duty suspension. Do not exceed the manufacturer's maximum hitch load rating with or without these devices.

- Tire inflation pressure should be set for the maximum rated tire load. If recommended or allowable, it is desirable from a handling standpoint to set the front tires at a lower inflation pressure than the rear.

- Put heaviest load over the trailer axle to reduce the "barbell" effect of separated loads.

- If trailer swing occurs, hold the steering wheel fixed and let the combination-vehicle decelerate by itself or apply trailer brakes if available. Sway dampers are also useful in reducing trailer swing.

- Avoid sharp cornering at highway speeds. Lateral acceleration reduces trailer damping, tow car stability, and braking capability.

- Check tire and brake capacity of the trailer. Tire capacity is stamped on the tire by the manufacturer. Adequate brake capacity may be judged by multiplying the number of braked wheels by 1500 lb. For 10 in. brakes this value should exceed the trailer GVWR.

REFERENCES

1. Hutchinson, John W., et al., Recreational Vehicle Accident Investigation Study, DOT HS-801 523, Apr. 1975.

2. Johnston, Donald E., John W. Zellner, and Irving L. Ashkenas, Handling Test Procedures for Passenger Cars Pulling Trailers. Vol. II: Technical Report, DOT HS-801 936, June 1976.

3. Klein, R. H., and H. T. Szostak, Effects of Weight Distributing Hitch Torque on Car-Trailer Directional Control and Braking, DOT HS-803 248, Oct. 1977.

4. Bernard, J. E., et al., Trailer Brake Performance, Final Technical Report, DOT HS-802 101, Nov. 1976.

5. Klein, R. H., and H. T. Szostak, Plan of Work and Methodology, Car-Trailer Handling Standards Development, Systems Technology, Inc., WP-1114-1, Jan. 1978.

6. Trailer Running Gear, Canadian Standards Association, Proposed Standard D313, Sept. 1977.

7. Wojcik, C. K., and R. L. Mellinger, Pilot Study of Housetrailer and Truck Camper Safety, Phase No. 2, UCLA-ENG-7208, Dec. 1971.

8. Trailer Couplings and Hitches — Automotive Type, SAE J684e, Aug. 1974.

9. "Hydraulic Brake Systems (Effective January 1, 1976)," Federal Motor Vehicle Safety Standard 105-75, Federal Register, Vol. 41, No. 139, July 19, 1976, Paragraph 571.105-75.

10. Service Brake System Performance Requirements — Passenger Car-Trailer Combinations, SAE Recommended Practice J135, Mar. 1973; and Brake System Road Test Code — Passenger Car and Light Duty Truck-Trailer Combinations, SAE Recommended Practive J134b, Aug. 1975.

11. Passenger Car and Light Truck Steady State Directional Control Test Procedures, Proposed SAE Recommended Practice XJ266.

12. Personal communication from R. Chirakos, Chief Engineer, Dexter Axle Co.

13. Interstate Motor Carrier Noise Emission Compliance Regulations and Subchapter B — Federal Motor Carrier Safety Regulations. Chapter III - Federal Highway Administration, Department of Transportation. Paragraph 393.52 — Brake Performance.

14. "Safety in Tow — Depend on EAZ-LIFT; State-by-State Towing Laws and Travel Aids," EAZ-LIFT, Sun Valley, California

15. California Motor Vehicle Code — Division 12, Section 26301.5.

16. Vehicle Dynamics Terminology, SAE J670d, July 1975.

17. Bundorf, R. Thomas, "Directional Control Dynamics of Automobile-Trailer Combinations, SAE Paper 670099, Jan. 1967.

18. Roland, R. D., and D. T. Kunkel, The Influence of Tire Properties on Passenger Vehicle Handling. Vol. V: Measured Tire Performance Data, DOT HS-801 319, Jan. 1975.

19. Schuring D. J., Tire Parameter Determination. Vol. II: Technical Report; Part 1, Test Methodology, DOT HS-802 087, Nov. 1976.

20. "Travel and Camper Trailer Safety," available from General Services/ Distribution, NHTSA, Washington, D. C., DOT HS-802 586, Nov. 1977.

21. Korn, Wayne, "Travel Trailer Design and Construction — How It Relates to Towability," Travel Trailer Towing, SAE SP-259, 1964, p. 11.

22. Ludtke, Norman, Survey of Suspension Systems on Travel Trailers, Vol. I, DOT HS-800 979, Dec. 1973.

23. Collins, R. L., and J. P. Wong, "Stability of Car Trailer Systems with Special Regard to Trailer Design," J. Dynamic Systems, Measurement and Control, Vol. 96, No. 2, June 1974.

24. "Trailer Towing Performance — Passenger Cars and Multipurpose Passenger Vehicles, Federal Register, Vol. 35, No. 157, Para. 575.108, Thursday, Aug. 13, 1970.

25. DiLorenzo, F. A., "ESV Dynamic Performance," SAE Paper 730080, Jan. 1973.

26. Rice, R. S., et al., "Automobile Driver Characteristics and Capabilities — The Man-Off-the-Street" SAE Paper 760777, presented at the Automobile Engineering Meeting, Detroit, Oct. 1976.

27. Hayes, G. G., et al., Determination of Motor Vehicle Characteristics Affecting Driver Handling Performance. Vol. I: Technical Report. Vol. II: Appendices, DOT HS-801 421 and 422, Mar. 1975.

28. McRuer, D. T., R. H. Klein, et al., Automobile Controllability — Driver/Vehicle Response for Steering Control. Vol. I: Summary Report. Vol. II: Supporting Experimental Results, DOT HS-801 407 and 406, Feb. 1975.

29. Weir, D. H. and R. J. DiMarco, "Correlation and Evaluation of Driver/ Vehicle Directional Handling Data," SAE Paper 780010, Feb. 1978.

30. Geometric Design Standards for Highways Other Than Freeways, 1969, Amer. Assn. of State Highway Officials, 1969.