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Final Technical Report

IMPROVED PASSENGER CAR BRAKING PERFORMANCE

R.D. Ervin
J.D. Campbell
M. Sayers
H. Bunch

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16. Abstract

Test conditions were studied as candidates for extending the scope of the stopping distance requirements of FMVSS 105-75, "Hydraulic Brake Systems." conditions of interest included low and split friction surfaces as well as straight-line braking and braking-in-a-turn maneuvers. Two large test programs were conducted and various analytical efforts were applied to the examination of the candidate test conditions and methods. Throughout all of the study activities, only stopping distance performance was taken as the measure for evaluating the utility of the candidate conditions and methods.

It was concluded that only the low friction, straight-line braking condition constitutes a viable extension of the stopping requirements of 105-75. It was also found that stopping distances in a turn do not differ significantly from stopping distances measured in straight-line braking. Further, stopping distances on split friction surfaces do not appear generally useful as characterizations of vehicle safety quality.

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We wish to recognize the support of the staff of the Applied Physics Laboratory in the conduct of the vehicle simulation tasks.

Finally, we wish to note the special contribution of Ms. Jeannette Nafe for her organization and typing of this report.

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1.0 INTRODUCTION

This document constitutes the final report on Contract DOT-HS-6-01368 entitled "Improved Passenger Car Braking Performance." The project has been conducted by the Highway Safety Research Institute of The University of Michigan with support of the facilities of the Chrysler Corporation Proving Grounds and both the facilities and staff of the Bendix Automotive Proving Grounds.

The primary objective of this study has been to determine whether a basis exists for extending the stopping distance requirements of FMVSS 105-75 to cover conditions of low and split friction surfaces as well as braking in a turn. The current 105 standard, while nominally encompassing the general matter of the braking safety of hydraulically-braked vehicles, limits itself to requirements for straight-line stopping on a high friction (dry) surface. To the degree that assurance of adequate stopping performance on a dry surface does not also assure adequate stopping on other surface conditions, or while braking in a turn, the standard may be subject to revision.

Accordingly, this study was configured to apply both analytical and experimental techniques to the examination of differing surface and maneuvering conditions. The purpose of these examinations was fourfold:

1) To establish test procedures suitable for demonstrating representative vehicle stopping response under the subject conditions.

2) To provide an understanding of the mechanics of vehicle response under the braking conditions of interest.

3) To conduct full-scale tests so as to reveal the practical aspects associated with such candidate extensions to the federally-required method.

4) To evaluate the measured vehicle responses so as to determine if meaningful improvements in traffic safety would accrue from specification of performance under the candidate conditions.

The scope of the study was constrained at the outset to include only the stopping distance measure of vehicle braking response under low and split friction, and curved-path braking conditions. Thus the study was not to consider any of the directional response issues related to these conditions—although it was recognized that strong hypotheses do exist which connect the directional disturbances and "loss of control" results of braking to traffic safety.

Moreover, the confinement of interest to stopping distance measures, alone, serves to explain why recommendations are made herein for extending FMVSS 105-75 only to the inclusion of a straight-line, low-friction test condition. As will be shown, stopping distance performance on split friction surfaces or in curved paths is either conceptually unrelated to the broad interests of traffic safety or of negligible significance as an additional measure beyond that of straight braking on homogeneous surfaces.

The report is arranged to provide relatively brief discussion of each research task in the main body, with extensive appendices pertaining to detailed methods and results. An overview of project tasks is presented in Section 2.0, with discussion of individual tasks and presentation of results provided in Section 3.0. In Section 4.0, the possible extension of FMVSS 105-75 to cover the various candidate conditions is examined, leading to conclusions and recommendations in Sections 5.0 and 6.0, respectively.

2.0 OVERVIEW OF THE RESEARCH PLAN

The project consisted of four major tasks intended to lead toward conclusions relevant to the question of extending FMVSS 105-75. Since it was desired that test procedures be developed and employed to gather a representative set of braking data, it was first necessary to conduct an exercise to identify that limited set of passenger cars which would yield more or less representative braking performance. A proposed analytical approach toward this vehicle selection task was discarded in concern for the general inability to accurately predict differences in stopping distance among real vehicles—for want of parametric data describing brakes and tires in a comprehensive manner. Alternatively, then, a test program was executed, involving twelve passenger vehicles which had been manufactured since the effective date of FMVSS 105-75. These tests provided data which clearly discriminated among vehicles in terms of high and low friction braking and stopping in a curved path. Also, a general try-out of test methods was effected, permitting the identification of refinements which were to be implemented in the major test phase.

Subsequent to the initial test exercise, a quasi-static simulation effort was undertaken to clearly define the first-order mechanisms determining stopping distance performance under the conditions of interest. This effort established the relationships between the major vehicle parameters, the surface and maneuvering conditions, and the resulting constraints on minimum stopping distance. Specific inquiries made by way of the quasi-static simulation guided the selection of test conditions to be applied in the full-scale test series.

The major test effort involved conduct of an extensive matrix of tests on each of five selected passenger cars. The matrix contained 28 separate test sequences built around the first, second, and third effectiveness test formats of the 105 standard. The greatly expanded number of test conditions permitted both straight and right/left turning stops on low, high, and split friction surfaces, with the split friction condition being represented by both hi-right (that is, the high friction side of the split is situated on the right side of the vehicle) and hi-left orientations. Data taken in this test series clearly delineate the relative gain to be made if one were to specify stopping distance performance in a turn—in addition to the specification of straight-line stopping distance. Further, the data serve to put in focus the conceptual problems associated with the specification of split friction stopping performance.

The final task involved a large scale computerized analysis, part of which examined the sensitivity of test results to imprecision in the test condition variables. Together with a field survey of certain economic matters, the simulation effort was also applied in the examination of advanced braking system concepts. Advanced concepts were treated both in terms of their likely influence on performance capability and in terms of the costs likely to attend their introduction as production hardware.

Conclusions and recommendations were drawn with regard to the advisability of extending the stopping distance requirements of FMVSS 105-75. By way of implication, the general absence of recommendations to add more stopping distance requirements reveals that the directional or yaw disturbance aspects of braking on split friction surfaces and in a turn are seen as the more important safety issues associated with those braking conditions.

3.0 TECHNICAL DISCUSSION

In this section the methods employed, and results obtained, in conduct of the various elements of the research study will be presented. This discussion has been designed to provide a generalized treatment of the material while Appendices A through G have been prepared for detailed presentation of the results of the experimental and simulation efforts. Although findings deriving from gathered data are stated in the text of this section, definitive conclusions relative to candidate modifications of FMVSS 105-75 are presented in Sections 4.0, 5.0, and 6.0.

3.1 Survey Test Program

A test program was conducted on a sample of twelve vehicles in order to obtain a data set characterizing straight-line and curved-path braking performance of FMVSS 105-75-compliant passenger cars on low and high friction surfaces. Performance data on current braking systems for braking in a turn and on low friction surfaces is scarce. Thus the data obtained in this survey test series, along with data from more extensive tests of five vehicles (see Section 3.3) later in the project, provided the principle basis for determining whether augmentation of the 105-75 standard should be recommended.

The survey tests also provided a pilot exercise for refinement of the proposed test procedures to be applied in full-scale tests of five vehicles to follow. As originally planned, two of the twelve vehicles were to have been subjected to an additional set of braking tests on split coefficient surfaces. However, the test activity was terminated due to sub-freezing temperatures after only one vehicle was tested in the straight-line/split-coefficient condition. In the absence of the desired experimental data, the quasi-static simulation effort (see Section 3.2) was expanded to include split friction conditions.

3.1.1 Twelve-Vehicle Sample. In selecting the twelve vehicles, information was obtained from the MVMA specification sheets, Automotive News, and consumer journals regarding models available, sales volume, and brake system design. Vehicles were selected to provide representation from the four major American automobile manufacturers and to cover vehicle size ranging from sub-compact to full size. Generally, the vehicles selected were models exhibiting relatively high sales volume within the size/manufacture groupings, tending to make the sample representative of the highway population. A final constraint on sample selection was the availability of test vehicles from local rental agencies, dealers, or manufacturers, given the understanding that such vehicles would be used for braking tests.

All the test vehicles were manufactured after January 1, 1976 so that they should comply with the 105-75 standard. Except for the Plymouth Valiant and the VW Beetle, all vehicles in the 1976 production year were manufactured with disc front brakes. Thus, since drum front brakes are being phased out of design usage, they were not included in the sample. Most 1976 vehicles have disc front and drum rear brakes, but a few domestic and imported vehicles have four-wheel disc brakes, either as standard equipment or as an option. Since this appears to be a growing trend, one vehicle with four-wheel disc brakes was included in the sample. In selecting the sample, it was seen that most 1976 brake systems incorporate a proportioning valve between the master cylinder and the rear brake cylinders. The function of this valve is to reduce rear brake line pressure relative to the front at higher levels of master cylinder pressure. By use of such a function, the brake system designer avoids the well-known effects of employing a fixed proportioning system which is set to prevent rear-wheel lockup at high levels of deceleration (on high coefficient of friction surfaces such as specified by FMVSS 105-75). With such fixed proportioning systems, premature front-wheel lockup will occur on low coefficient of friction surfaces, producing longer stopping distances than could be obtained if the proportioning were closer to optimum for low deceleration stops. Several vehicle models were produced in 1976, however, which were of the fixed proportioning variety. Three of these were included in our twelve-car sample. Table 3.1 contains a list of the twelve vehicles which were tested, showing size classification and salient brake system features.

3.1.2 Test Site and Experimental Procedures. Braking tests using the twelve-vehicle sample were conducted on the skid traction facility at the Chrysler Corporation's Proving Grounds in Chelsea, Michigan. This facility consists of four adjacent lanes, twenty-eight feet wide and 1000 feet long with a long approach area from each direction. The composition and ASTM skid number of the four lanes in adjacent order are shown in Table 3.2. Wetting of the surface was accomplished with a multiple-head sprinkling system along one side of the test lanes which produced a reasonably homogeneous water depth by virtue of the uniform 1% grade across the lane. The lane width of twenty-eight feet was adequate to lay out curved paths for braking-in-a-turn experiments on the same surface area used for straight-line tests.

The HI-CO (high coefficient of friction) tests, straight and curved path, were conducted on the dry brushed concrete surface, SN_{40}(DRY) = 80, and the LO-CO (low coefficient of friction) tests were conducted on the wet jennite surface, SN_{40} = 30. Straight-line SP-CO (split coefficient of friction) tests were run with one vehicle on each of the three available surface junctions with the surface wet.

In addition to the skid numbers provided by Chrysler, the dry brushed concrete and the wet jennite surfaces were characterized by peak friction measurements. These measurements were made specifically to apply NHTSA's Braking Efficiency Technique [1], a method deriving a braking efficiency numeric from braking test data by comparing a vehicle's stopping distance with the computed ideal stopping distance of a hypothetical reference vehicle which makes optimum use of the available traction on the test surface. To obtain this numeric,

Table 3.1. Twelve-Car Sample Showing Size and Salient Brake System Features.

Size	Vehicle	Year	Manufacturer	Brakes Front/Rear	Proportioning Valve
Sub-compact	Chevette	1976	GM	Disc/Drum	No
	Pinto Wagon	1976	Ford	Disc/Drum	Yes
	Gremlin	1976	AMC	Disc/Drum	No
Compact	Nova	1976	GM	Disc/Drum	Yes
	Pacer	1976	AMC	Disc/Drum	No
	Volvo 244	1976	Volvo	Disc/Disc	Yes
Inter-mediate	Monte Carlo	1976	GM	Disc/Drum	Yes
	Fury	1977	Chrylser	Disc/Drum	Yes
	Torino	1976	Ford	Disc/Drum	Yes
Full Size	Buick LeSabre	1976	GM	Disc/Drum	Yes
	Ford LTD	1976	Ford	Disc/Drum	Yes
	Dodge Monaco	1977	Chrysler	Disc/Drum	Yes

Table 3.2. Test Surface Characteristics at Chrysler Proving Grounds.

Composition	SN_{40} Wet	SN_{40} Dry
Jennite	30	--
Epoxy-Coated Concrete	10	--
Brushed Concrete	50	80
Polished Concrete	15	--

peak friction measurements are made at two tire loads, representing the nominal front and rear tire loads on the reference vehicle, and at four velocities using the ASTM E-501 standard tire as a reference tire. The peak traction measurement data made with DOT's Surface Friction Dynamometer, or SFD, are shown in Figure 3.1. The curves drawn through the data points are the best least squares fit to the data points of curves with the general form $AV^2 + BV + C$ where V is velocity.

All HI-CO tests were run at an initial velocity of 60 mph and, in the curved-path case, on a curve with a radius of 801 feet, determined to produce an initial lateral acceleration of 0.3 g. For the LO-CO tests, the corresponding values were 40 mph initial velocity and a curved-path radius of 535 feet, producing an initial lateral acceleration of 0.2 g.

The performance measure derived from these tests was minimum stopping distance. Successive stops were made with increasing increments of constant pedal force until lockup occurred on either axle. Two additional stops were made at the constant pedal force giving minimum stopping distance with at most one wheel locked per axle, thus assuring vehicle controllability. Also, two driver "best effort" stops were made, permitting driver modulation of the brake pedal. Steering correction by the driver was permitted throughout the run. All tests were performed with the transmission in neutral per 105-75 test procedures. The test weight for each vehicle was curb weight plus driver, passenger, and instrumentation, or approximately the lightly-loaded vehicle condition used for the third effectiveness test in the 105-75 procedure.

In this survey test series, vehicles were tested essentially in the condition as received from the rental agencies. Each was inspected and serviced to assure that proper brake adjustment was attained, to determine that tires and brake linings were not excessively worn, and that the master cylinder, wheel cylinders, and brake lines were sound.

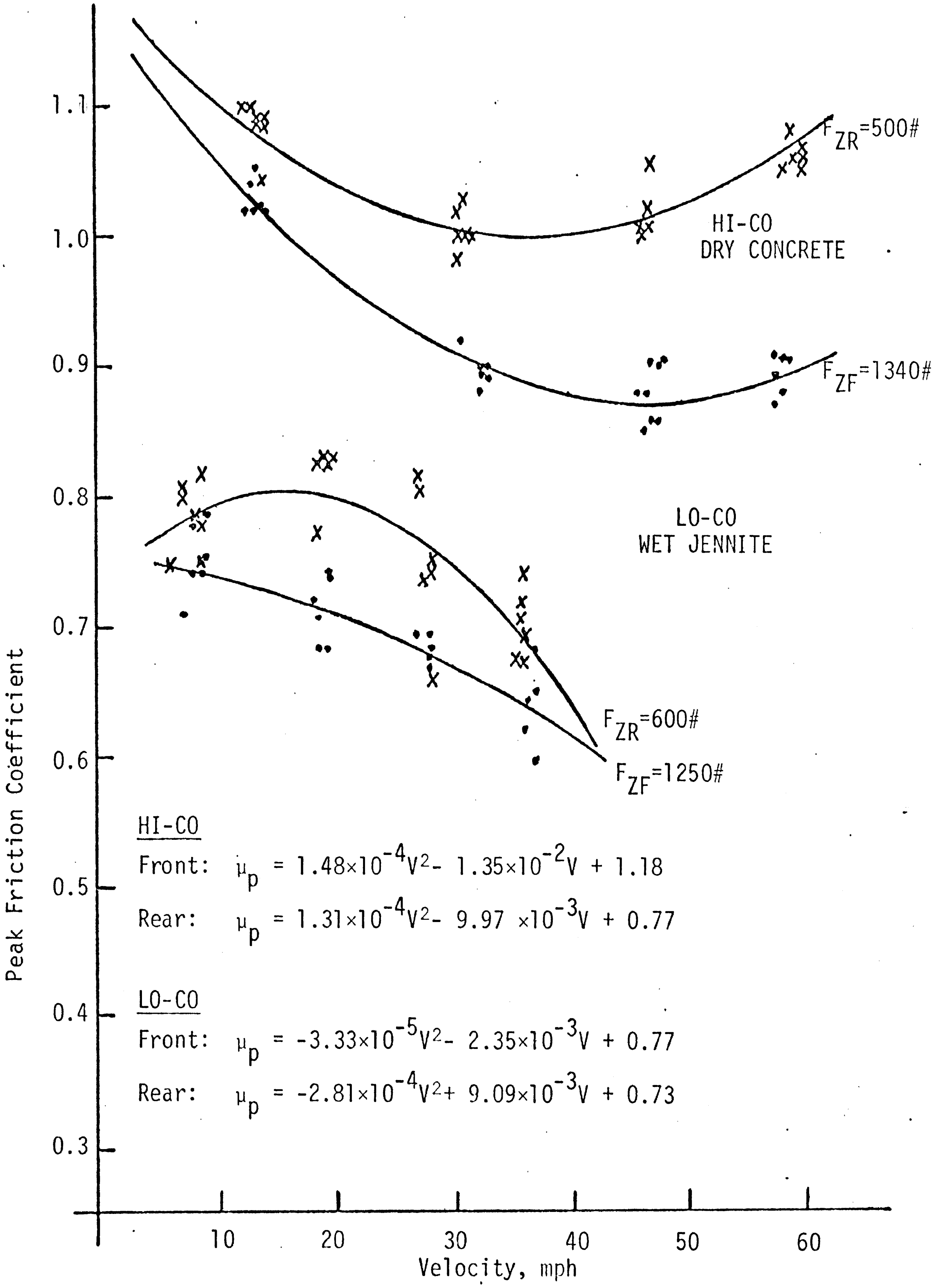


Figure 3.1. DOT Surface Friction Dynamometer peak friction measurements on dry concrete and wet jennite at Chrysler Proving Grounds, October 1976.

Each vehicle in turn was instrumented with a fifth wheel incorporating electronic circuitry for displaying initial velocity and stopping distance and a thermocouple was installed in the brake pad of the left front brake with temperature displayed to the driver to assure temperatures below 200°F before the start of each run. Wheel lockup was reported by spotters stationed on each side of the test lane.

The original approach toward objective application of the brake pedal, by which the driver applied the pedal manually while observing a meter display, steering, and assuring the desired initial velocity, proved to be a formidable driver task. Thus, a simple dead-weight pedal force application device was implemented which could be quickly and simply installed in any car. It consisted of a steel bar which clamped to the brake pedal and positioned a loading pan at the front edge of the driver's seat permitting the placement, before the test run, of one or more calibrated weights. When the driver released the hand-held pan, it dropped a short distance, rapidly applying a constant and highly repeatable pedal force. As shown in Figure 3.2, however, the dead-weight device produces an overshoot in the pressure transient at the output of the master cylinder. With manual brakes, the transient shows a large amplitude overshoot which damps to the final value within about 0.6 second— the over-pressure condition lasting for less than 0.2 second. The lag inherent in vacuum-boosted brakes, however, smooths the initial transient so that the final pressure level is reached in about 0.2 second. A second inertial effect was seen to derive from the moment about the brake pedal pivot which arises from the deceleration of the pedal application weight itself. As shown in Figure 3.2b, vehicle deceleration acting on the weight causes an increase in pedal force and thus brake line pressure over that obtained statically. Nevertheless, a quite constant pressure level is obtained throughout the braking period. These effects were judged to be less detrimental to the objective measure of stopping distance than the variations observed with direct driver control of pedal force. Thus the dead-weight pedal force applicator was used throughout the survey tests.

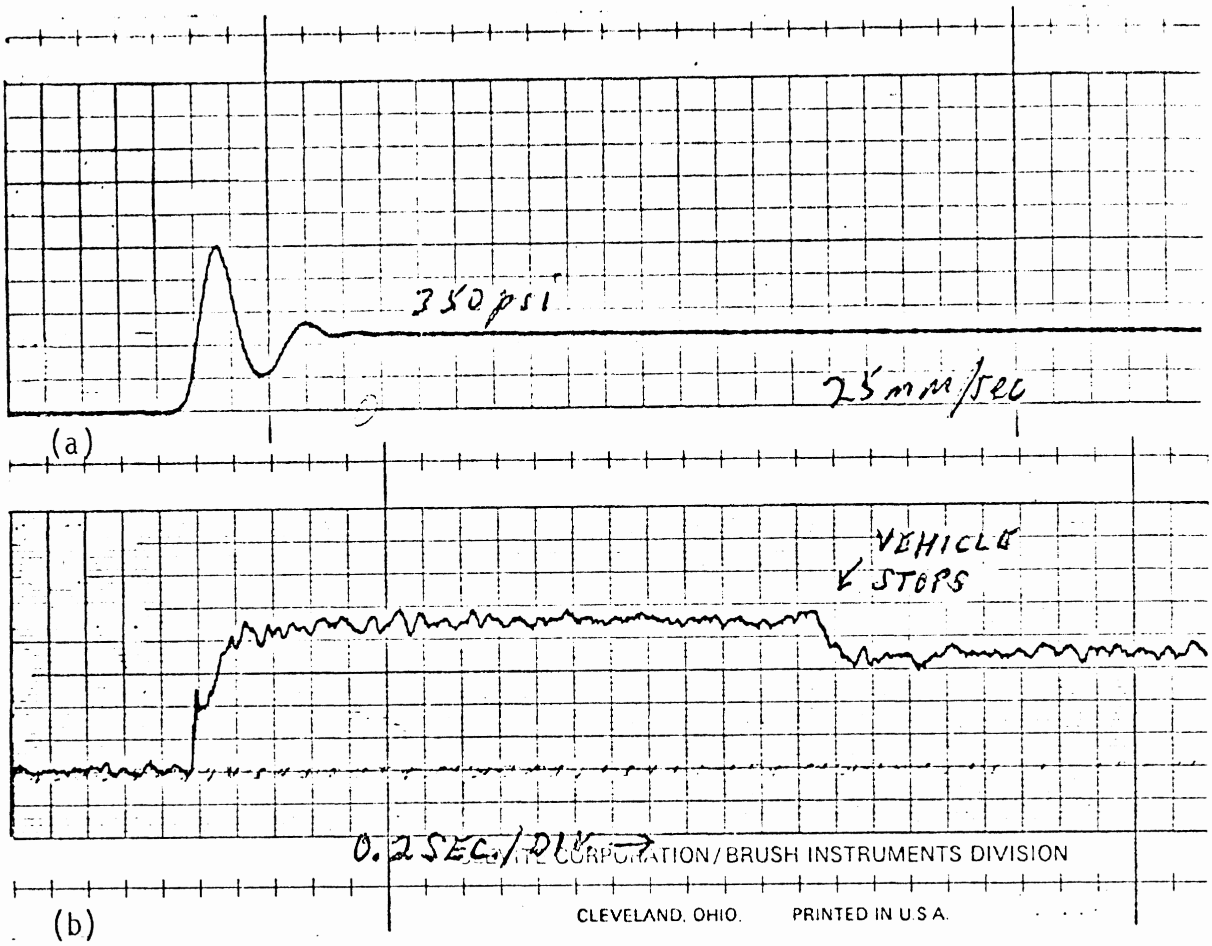


Figure 3.2. Brake line pressure transients resulting from application of the dead-weight brake application device. (a) Manual brakes, vehicle stationary. (b) Vacuum-boost power brakes, vehicle stopping.

3.1.3 Test Results on Twelve-Car Sample. A series of straight and curved-path (left and right turning) minimum stopping distance tests were made on each of the twelve vehicles on HI-CO (brushed concrete) and LO-CO (wet jennite) surfaces. Also, one vehicle was tested for straight-line braking performance on three differing split coefficient surfaces. In all tests, the vehicle was loaded to its curb weight plus about 400 pounds which included the driver, a passenger, and instrumentation.

Best stopping performances, defined by the shortest stopping distance out of three runs made at that constant pedal force which was determined to give optimum stopping without axle lockup, are plotted in Figures 3.3 and 3.4. Figure 3.3 is a data summary for the HI-CO tests which were conducted with an initial velocity of 60 mph and, in the case of braking in a turn, with an initial lateral acceleration of 0.3 g. Figure 3.4 presents a data summary for the LO-CO tests which were conducted with an initial velocity of 40 mph and an initial lateral acceleration of 0.2 g in the turn. The HI-CO straight-line braking test is equivalent to the third effectiveness test (lightly loaded vehicle) of the current FMVSS 105-75 standard. The requirement of the standard, for this condition, is a maximum stopping distance of 194 feet. Detailed results of these tests can be found in Appendix A, while the following discussion pertains both to those individual performance data and to the rank order data shown in Table 3.3.

In the HI-CO straight-line test (Figure 3.3), all cars except an AMC Pacer stopped in a distance less than the 194-foot requirement of 105-75. The best performing vehicle, a Ford Torino, stopped in 152 feet, followed closely by a Chevrolet Monte Carlo at 154 feet. The subcompact Ford Pinto station wagon ranked third with a straight-line stop of 157 feet. The average straight-line stopping distance of the small cars was 13 feet (8.0%) longer than the average for the large cars. The three vehicles without proportioning valves ranked 8th (Chevette), 11th (Gremlin), and 12th (Pacer).

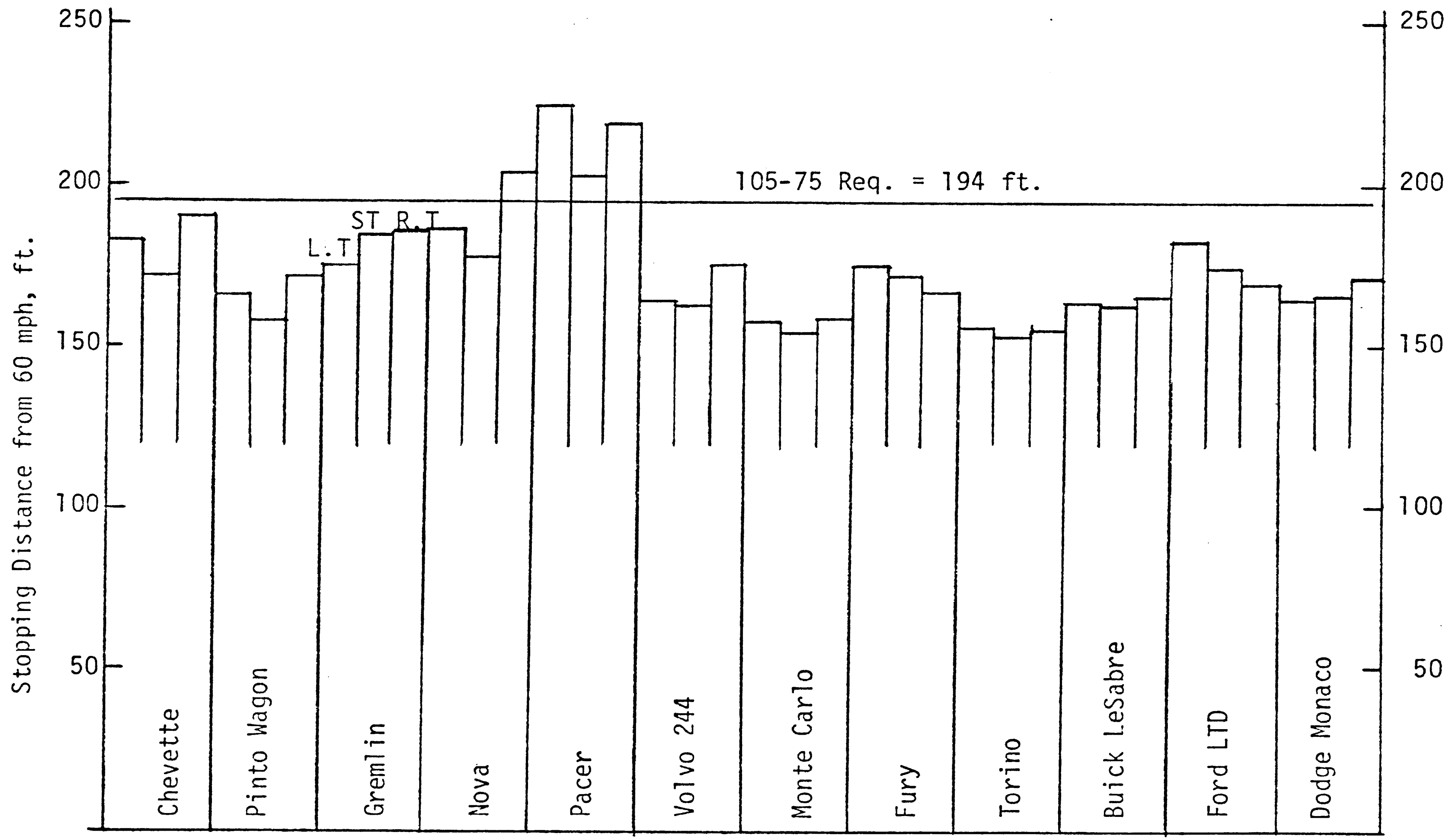


Figure 3.3. Minimum stopping distances for straight, left turning, and right turning stops from 60 mph on dry brushed concrete. Initial lateral acceleration, 0.3 g. Turn radius 801 feet.

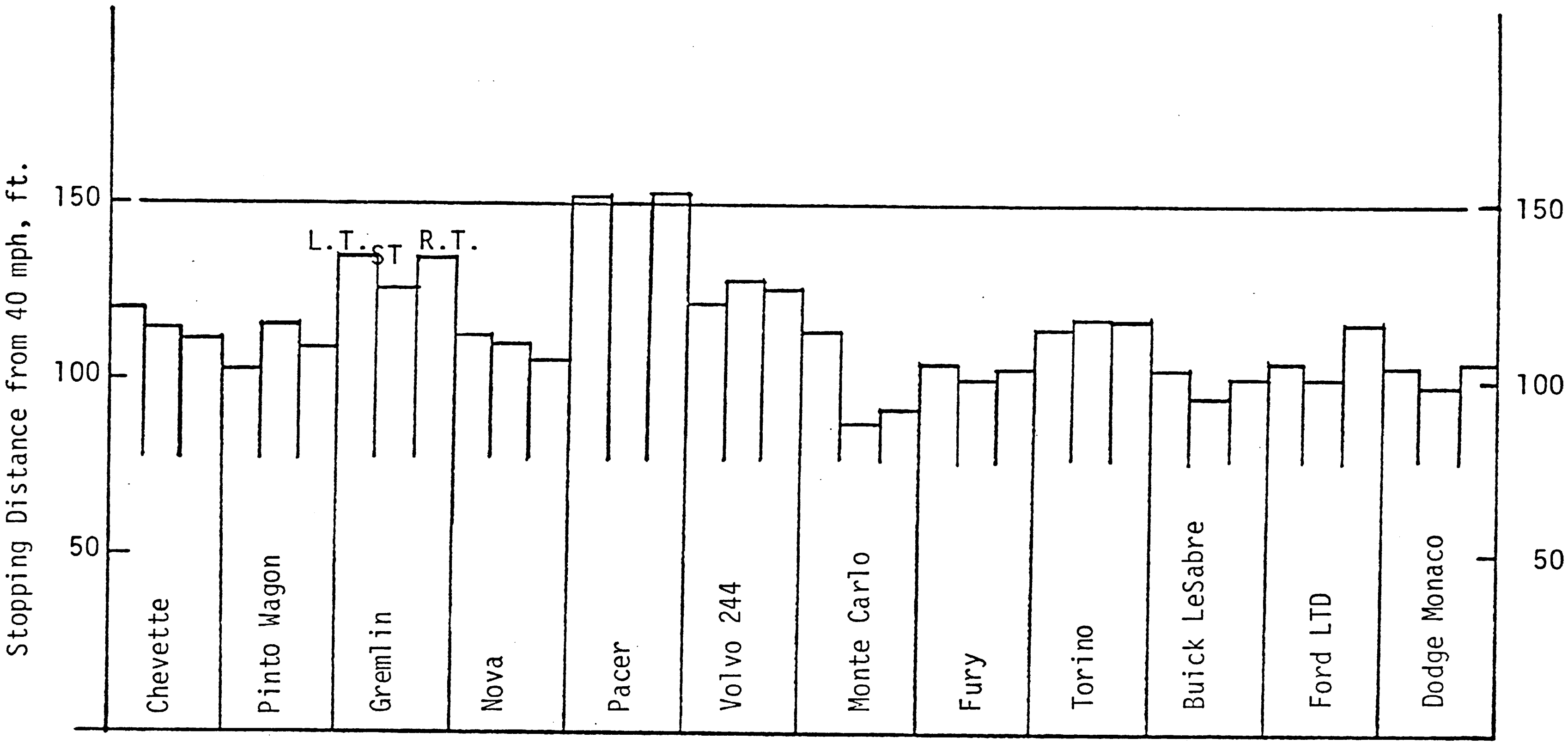


Figure 3.4. Minimum stopping distances for straight, left turning, and right turning stops from 40 mph on wet jennite. Initial lateral acceleration, 0.2 g. Turn radius 535 feet.

Table 3.3. Rank of the Twelve Vehicles in Order of Shortest to Longest Stopping Distance on Dry Concrete and on Wet Jennite for Straight-Line and In-A-Turn Stops.

Dry Concrete			Wet Jennite		
Turn	Straight		Straight		Turn
1	1	Torino	1	Monte Carlo	2
2	2	Monte Carlo	2	Buick LeSabre	1
5	3	Pinto Wagon	3	Dodge Monaco	3
3	4	Buick LeSabre	4	Fury	4
6	5	Volvo	5	Ford LTD	7
4	6	Dodge Monaco	6	Nova	6
7	7	Fury	7	Chevette	8
10	8	Chevette	8	Pinto Wagon	5
8	9	Ford LTD	9	Torino	9
11	10	Nova	10	Gremlin	11
9	11	Gremlin	11	Volvo	10
12	12	Pacer	12	Pacer	12

Comparing straight-line braking rank on dry concrete and wet jennite:

1 shows no change

8 show improvement of from 1 to 4 positions

3 show drastic drops of from 5 to 8 positions

Comparing straight to turn rank:

On dry concrete; the largest change is 2 positions
8 change 1 or 0 positions

On wet jennite; the largest change is 3 positions
10 change 1 or 0 positions

On the LO-CO surface (Figure 3.4), the difference in straight-line braking performance between large and small cars was more pronounced, with the average stopping distance for the small cars being 24 feet (24%) longer than for the large cars. The Monte Carlo stopped in the shortest distance, 87 feet, and the Pacer ranked 12th with 150 feet. Looking at the change in rank order which is seen to occur between the HI-CO and LO-CO straight-line braking tests (Table 3.3), eight vehicles show a difference ranging from 1 to 4 rank poisitions, one vehicle shows no change (Pacer) and three show rather large decreases in rank of from 5 to 8 positions. We also note that on the LO-CO surface, the Chevrolet Monte Carlo remained top ranked while the previously first-ranked Ford Torino dropped to ninth place.

Two items which are of interest relative to the braking-in-a-turn data are the matters of left-to-right asymmetry and the difference between stopping distance capabilities measured in straight-line and braking-in-a-turn tests. From Figures 3.3 and 3.4 it can be seen that both asymmetry in performance and straight versus turn differences were generally small. On the HI-CO surface the average asymmetry was 5.6 feet for the large cars and 9.3 feet for the small cars, or an overall average of 7.5 feet. Nine out of twelve vehicles exhibited their longest stopping distance turning right, but for three of these the difference was three feet or less. On the LO-CO surface, the average asymmetry was 7.1 feet for the large cars and 4.5 feet for the small cars, or 5.8 feet overall, and five out of twelve had their longest stopping distance turning right. The Nova had the largest asymmetry on the HI-CO surface (17 feet longer in a right turn), but on the LO-CO surface, the asymmetry was only six feet longer, in a left turn. On the LO-CO the high ranking Monte Carlo exhibited the largest asymmetry, 23 feet longer, in a left turn, but on the HI-CO it stopped in a distance only one foot longer, in a right turn.

To compare straight and in-a-turn stopping distances, the average value of the left and right turning stopping distances is used. The differences expressed as a percentage of the straight-line value are shown in Table 3.4. The small car average difference was 7.2% on HI-CO and 3.9% on LO-CO and the large car average difference was 1.4% on HI-CO and 7.6% on LO-CO. Over all twelve vehicles the average difference was 4.3% on HI-CO and 5.7% on LO-CO. On the HI-CO test, one car stopped in a shorter distance in the turn and four cars stopped shorter in the turn on the LO-CO test. Referring again to Table 3.3, where the vehicles are ranked according to stopping distance performance, it is seen that on the HI-CO test the largest change in rank between the straight and in-a-turn test is two positions and on the LO-CO test, three positions.

A final point of interest with respect to these data is the spread in stopping distance over the three runs which were conducted at the optimum pedal force level for each test; i.e., a measure of the data repeatability. Averaged over all tests, the difference in the three repeated stopping distances was 7.5 feet. Expressed as a percentage of the minimum stopping distance, the average was 5%. Note that this is very close to the average percentage difference between the straight-line and in-a-curve stopping distances.

These data indicate that little discrimination in performance is gained from braking-in-a-turn tests over what is learned from straight-line tests where stopping distance is the only performance measure.

Braking efficiency levels [1] were computed for each vehicle using the shortest measured stopping distances from the HI-CO and LO-CO straight-line braking tests and the peak surface friction characteristics which were plotted in Figure 3.1. The braking efficiency results are tabulated in Table 3.5. Braking efficiency is defined by the expression:

$$\text{Eff.} = \frac{\text{Ideal Stopping Distance of Reference Vehicle}}{\text{Measured Stopping Distance of Real Vehicle}} \times 100\%$$

Table 3.4. Percentage Difference Between Straight-Line and In-a-Turn Stopping Distance.

	Dry Concrete	Wet Jennite
Chevette	8.8%	0.9%
Pinto Wagon	7.6	*9.4
Gremlin	*2.7	7.2
Nova	9.6	*0.9
Pacer	9.9	0.7
Volvo 244	4.3	*3.9
Small Car Average	7.2%	3.9%
Monte Carlo	2.6	17.2
Fury	0.0	4.0
Torino	2.0	*1.7
Buick LaSabre	1.2	6.3
Ford LTD	1.2	10.0
Dodge Monaco	1.2	6.1
Large Car Average	1.4%	7.6%
Overall Average	4.3%	5.7%

*In-a-turn stopping distance shorter than straight-line stopping distance.

Table 3.5. Braking Efficiencies of 12 Vehicles on High Coefficient and on Low Coefficient Surfaces.

Vehicle	HI-CO Brushed Concrete	LO-CO Wet Jennite	Δ
Chevette	74.8%	67.5%	7.3%
Pinto Wagon	81.2	66.4	14.8
Gremlin	69.5	61.4	8.1
Nova	72.1	70.1	2.0
Pacer	63.5	50.9	12.6
Volvo 244	78.8	59.3	19.5
Monte Carlo	82.8	87.9	*5.1
Fury	74.8	76.7	*1.9
Torino	83.8	65.3	18.5
Buick LaSabre	79.1	80.3	*1.2
Ford LTD	73.7	77.5	*3.8
Dodge Monaco	77.7	78.8	*1.1
Average	76.0%	70.2%	

*Higher utilization of LO-CO surface than of HI-CO surface.

By this measure, five out of the six large vehicles achieved slightly higher utilization of the wet LO-CO surface than of the HI-CO surface. Overall, however, the vehicles averaged 5.8% higher utilization of the HI-CO surface than they did of the LO-CO surface. The Volvo registered the greatest difference (19.5%) with 78.8% efficiency on dry brushed concrete and 59.3% on wet jennite.

At the conclusion of the survey test program, straight-line split coefficient braking tests were made with the Dodge Monaco (from an initial velocity of 40 mph) on each of the three split coefficient surfaces provided by the junctions of the four test lanes. Minimum stopping distance tests were also made on each of the four surfaces taken singly to quantify their respective peak friction characteristics. The results obtained are plotted in Figure 3.5. Clearly, the stopping distances measured on homogeneous surfaces do not relate well to the surface skid numbers. This lack of correlation, of course, is largely due to the fact that peak friction coefficient rather than sliding friction or skid number is the characteristic of the tire/road interface which limits wheels-unlocked stopping distance performance.

Minimum stopping distance on the brushed concrete (SN_{40} = 50) was seen to be only six feet shorter than on the polished concrete (SN_{40} = 15), indicating that the peak friction levels on these surfaces were close to the same although the ratio of the skid numbers was greater than three to one. With approximately the same peak friction values on each side of the split, it follows that the stopping distance on the split was essentially the same as on the surfaces on each side. Furthermore, the wheel lock pattern was seen to be the same in the four cases with both rear wheels locking through the last ten to twenty feet.

The largest split in peak friction values was between the epoxy and brushed concrete surfaces as indicated by the large difference (38 feet) in the stopping distance results obtained on the separate

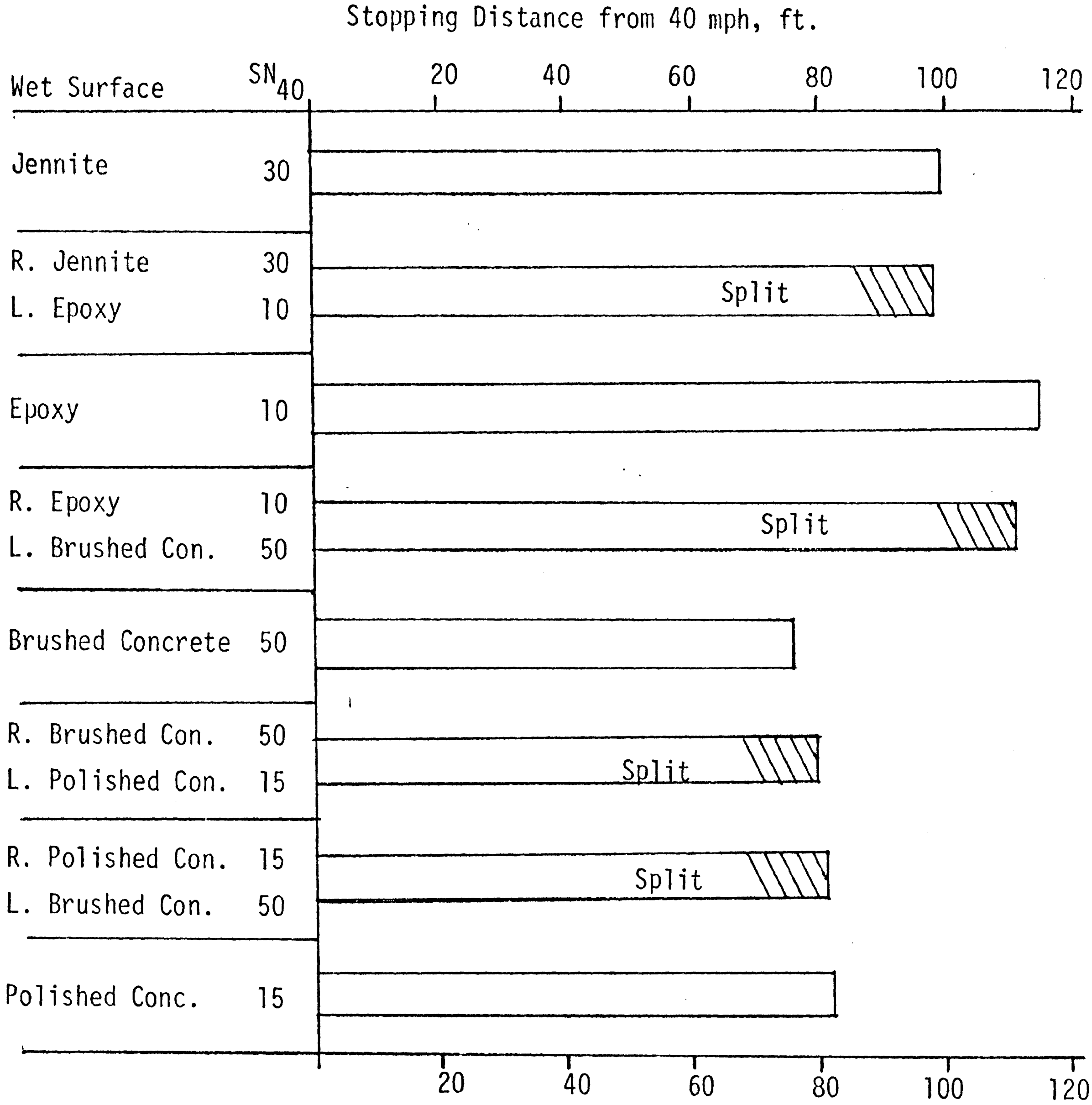


Figure 3.5. Straight-line stopping distances on split coefficient surfaces and on each surface forming the splits.

surfaces. On the split, the minimum stopping distance was achieved with both low coefficient-side wheels locked through the last forty feet of the run. On this split friction surface, the stopping distance was three feet shorter than was measured on the epoxy surface. With the wheels locked on the low coefficient side of the vehicle, a yaw moment was generated which caused the vehicle to drift toward the high coefficient side of the split, despite the driver's steering effort. An attempt to apply an even higher pedal force and thus achieve a shorter stopping distance through better utilization of the available traction on the high coefficient side simply resulted in loss of vehicle control. Thus, in this split friction surface condition, the minimum stopping distance was limited by vehicle controllability rather than by the maximum traction available.

The increment in peak friction existing between the jennite and epoxy surfaces is seen to be intermediate between the peak friction increments characterizing the other two split friction surfaces. Minimum stopping distances measured on the jennite and epoxy surfaces separately differed by only fourteen feet. On the jennite/epoxy split, the stopping distance was the same as on the jennite alone. Although this result is a few feet shorter than expected for the split condition, the result is reasonable in consideration of the typical data variability in these braking tests.

3.2 Quasi-Static Analysis

A series of calculations of braking efficiency, based on a quasi-static analysis, was conducted in order to understand some first-order sensitivities of vehicle performance to the types of braking maneuvers under consideration in this study. The results of the calculations were examined and used to plan the testing procedures described in the next section.

This section summarizes the quasi-static analysis by describing the model which was used, the various conditions which were considered, and the resulting observations on the different mechanisms involved in limit braking. A more complete documentation of this effort is included in Appendix B.

3.2.1 Quasi-Static Model. The model used in the quasi-static study constitutes a simple representation of a four-wheeled vehicle with conventional (non-antilock) brakes. The model is quasi-static in the sense that load transfer takes place instantaneously simply as a function of kinematic relationships. For the case of braking in a turn, the model considers steadily sustained lateral as well as longitudinal acceleration level. Thus the model is suitable for evaluating the incidence of wheel lockup which occurs immediately following pedal application, but does not account for the effects of diminishing lateral acceleration as the vehicle slows down.

The parameters needed to define a quasi-static model which can represent straight-line and in-a-turn braking maneuvers, on uniform and split-coefficient surfaces, are described in the following three groups:

1. Pitch Plane Parameters. The parameters involved are: A/L, which identifies the longitudinal position, A, of the c.g. (center of gravity) behind the front axle, ratioed to the length, L, of the wheelbase; H/L, which ratioes the vertical position of the c.g. to the wheelbase and acts as a mechanical "gain" which determines the amount of load transfer due to the vehicle deceleration; and P, the brake proportioning, which gives the portion of the total braking torque which acts on the front axle.

2. Tire/Road Frictional Characteristics. The braking force capability of a particular tire/surface combination is described by the friction coefficients μ_p and μ_s, which correspond to peak and slide values, respectively. The

braking force present at the tire/road interface is proportional to the brake torque, as long as the brake force/normal force ratio is less than μ_p. If the braking torque is such that the brake force/normal force ratio would be greater than μ_p, the wheel locks up, and the braking force is then equal to the product of the normal force and μ_s. A split-coefficient surface is represented when μ_p and μ_s differ between the right- and left-hand sides of the vehicle. For this analysis, six surface types were defined. The surface types comprised three uniform surfaces (a high coefficient, a medium coefficient, and a low coefficient surface) and three split coefficient surfaces (two of which had the same average of the right- and left-hand coefficients, and two of which had the same frictional increment between the right- and left-hand sides).

3. Roll Plane Parameters. The braking-in-a-turn maneuver involves definition of one input condition and two vehicle properties: A_y, the constant lateral acceleration level (in g's), H/W, the ratio of the c.g. height to the track width of the vehicle, and C_ϕ, the roll distribution, which gives the percentage of the total lateral load transfer which accrues at the front axle.

The limit braking performance for a particular set of parameter values was found by assuming a (small) brake torque level, balancing pitch and roll moments to determine the vertical load on each tire, checking each wheel for lockup, and calculating the resulting deceleration and stopping distance. The brake torque level was then increased by a small amount, and the calculation repeated. This sequence was continued until both wheels on the same axle were found to lock. At this time, the results which gave the lowest stopping distance were recorded. As this procedure was programmed on a computer, the large number of iterations could be accomplished economically and expediently.

The results of these calculations were ultimately expressed in terms of braking efficiency, thereby ratioing a given vehicle's performance to the best performance level achievable in straight-line braking by that vehicle on that surface if the vehicle were optimally proportioned. The efficiency notion is based on the following rationale: In actual braking situations, we cannot exercise any choice over surface friction conditions which prevail; surface conditions must simply be accepted and vehicle brake systems must be judged on their ability to maximize their utilization of the frictional constraint to minimize stopping distances. Since the best that a conventional non-antilock-equipped vehicle can do is be ideally proportioned in its front/rear brake torque distribution, it is reasonable to rate braking performance by ratioing a vehicle's measured performance to the "maximum utilization" case.

Because split coefficient surfaces were considered, some of the in-a-turn calculations resulted in efficiencies greater than 100%—an occurrence which is impossible with uniform friction surfaces. This anomaly occurs when the split friction surface is arranged such that the heavily loaded, outside tires run on the high friction side of the split. Ratioing minimum stopping distances obtained in this case to the reference minimum distance achievable in a straight line can yield efficiencies above 100%.

The quasi-static analysis was conducted in two distinct stages. First, the sensitivity of braking efficiency performance to major parameters such as the brake proportioning value was established. The results of these calculations were examined, and values for vehicle parameters and test initial conditions were selected. The selections were made to permit examination of a few vehicle configurations which represented the widest reasonable range of those parameters most influencing braking efficiency. Second, a matrix was formulated which contained 120 combinations of the parameter values selected in the first stage, and the braking efficiency associated with each combination was calculated.

3.2.2 Discussion of Calculated Results. The quasi-static calculations served to display the broad picture of vehicle braking capabilities over many operating conditions. The results of the calculations, presented in Appendix B, are the basis for the following observations:

1. The choice of front/rear proportioning is seen as the parameter most influential in determining braking efficiency over the straight and curved-path braking cases using high, low, and split friction conditions. The effects of other parameters often depend on the relation between the proportioning, the front/rear static loading, and the available peak friction levels.

2. As the combination of surface and maneuvering conditions becomes complex, braking efficiency becomes an increasingly complicated function of proportioning and other parameters. When braking in a straight line on a uniform surface, two limiting mechanisms exist—that is, performance is limited by the imminent lockup of either both front or both rear wheels. When braking in a turn, as many as eight different limiting mechanisms can be identified for a given vehicle over the range of brake proportioning. As a result of the added complexity, it was seen that the overall sensitivity of braking efficiency in a turn to the various vehicle parameters is decreased, since as one performance-limiting mechanism constrains efficiency in a certain friction level regime, it is possible for different, more efficient mechanisms to dominate in surface friction regimes that are either higher or lower than the regime in question.

3. The change in efficiency between straight-line and in-a-turn braking on a uniform surface varies considerably and depends on the relationship between the proportioning value and the available friction. With proportioning perfectly matched to the friction level for straight-line braking, braking efficiency was seen to reduce by as much as 18% when braking in a turn. On other surface friction levels, the same vehicle will exhibit broadly varying straight versus turn differences—in one case the in-a-turn efficiency was seen to be 16% higher than straight-line efficiency.

4. On split coefficient surfaces, the change from straight-line to in-a-turn braking generally resulted in a higher efficiency. The average level of braking efficiency on the split coefficient surfaces was observed to be about 10% higher than on the uniform surfaces.

5. The item most influencing braking efficiency in a turn on a split friction surface is the polarity of turn with respect to the right/left placement of hi/lo friction surfaces. Differences in efficiency between right and left turns on a split ranged from 0 to 18%.

6. Among split friction surfaces, the only descriptive parameter found to uniformly affect measured efficiencies is the increment in friction level across the split. This characteristic is seen to invariably exaggerate the asymmetry in performance between right- and left-hand turns.

7. The average friction level represented by the pair of surfaces comprising the split condition was not seen to methodically influence efficiency results. Thus the specification of the average μ level incorporated in a split friction test condition would appear to be more or less open to selection on the basis of practical

considerations of friction treatment techniques, and not constrained by any technical concerns over a discriminatory test practice.

8. The lateral acceleration level which exists during braking in a turn was found to impose a relatively small influence on braking efficiency between the values of 0.2 g and 0.3 g A_y. Thus a braking-in-a-turn test procedure could alternatively employ lateral accelerations between 0.2 and 0.3 g, obtaining generally representative, though not identical, results by either test condition.

9. The front/rear distribution of roll stiffness, C_ϕ, has a pronounced effect on the right/left asymmetry in braking-in-a-turn performance on a split coefficient surface—typically (though not invariably), increased bias in C_ϕ increases asymmetry.

10. The influence of C_ϕ on braking in a turn on a uniform friction surface is mixed; sometimes increasing, sometimes decreasing performance on various surfaces.

11. Vehicles with a relatively high value of brake torque proportioning, given the longitudinal location of the c.g., tend to fare consistently poorer on any lower friction surface, whether braking in a straight line or in a turn. The effect is generally reduced by the use of proportioning valves which change the proportioning as a function of line pressure.

3.3 In-Depth Test Program

A comprehensive test program was carried out on a sample of five passenger cars covering straight-line and curved-path braking on high, low, and split coefficient surfaces. In addition to providing a demonstration of the related test procedures and surface conditions which could be integrated with the existing 105-75 procedures, these tests yielded an additional data set characterizing representative braking performance levels of modern passenger vehicles under these test conditions.

3.3.1 Test Vehicle Selection. Four of the five vehicles were selected from among the twelve-vehicle sample tested earlier to provide a broad range in stopping distance performance. The fifth vehicle, by contractual requirement, was an antilock-equipped car. The four conventional vehicles selected were the Chevrolet Monte Carlo, Ford LTD, Ford Pinto station wagon, and the AMC Pacer, i.e., one from each of the four size classes. Table 3.6 gives an overall performance summary of the original twelve-vehicle sample from which the selection was made. Columns 1, 2, and 5 give sums of the minimum stopping distances in the several tests for each car, with performance rank (1 being the best) given in parentheses. In column 5 (Total Stopping Distance), the selected vehicles are seen to rank the highest (Monte Carlo and Pinto wagon) and lowest (Ford LTD and Pacer) of all vehicles in the respective large and small car groups. Considering the high and low friction tests independently (columns 1 and 2), the Monte Carlo and Torino have the maximum spread in performance on the LO-CO surface, but the smallest spread in performance on the HI-CO surface. Thus the selection of the Monte Carlo and LTD constitutes a compromise in the large car group. In the small car group the Pinto wagon and the Pacer give the maximum spread in performance for both surfaces. Looking at straight-line versus curved-path performance, the two large cars exhibit the largest difference values in column 4 for the LO-CO surface while on the HI-CO surface (column 3) the difference values are negligible for all the large

Table 3.6. Overall Performance Factors of Twelve Vehicles.

	Σ(ST, LT, RT)+Min. Stop. Distance, Ft.		Straight-Turn % Difference		Total Stop. Distance, Ft.
	Dry	Wet	Dry	Wet	
Chevette	544 (10)#	345 (8)	8.8	.9	889 (9)
Pinto Wagon	494 (3)	327 (6)	7.6	*9.4	821 (5)
Gremlin	543 (9)	393 (11)	*2.7	7.2	936 (11)
Nova	566 (11)	328 (7)	9.6	*0.9	894 (10)
Pacer	646 (12)	453 (12)	9.9	0.7	1099 (12)
Volvo 244	500 (5)	377 (10)	4.3	*3.9	872 (8)
Monte Carlo	471 (2)	292 (1)	2.6	17.2	763 (1)
Fury	513 (7)	308 (4)	0	4.0	821 (6)
Torino	462 (1)	348 (9)	2.0	*1.7	810 (4)
Buick LeSabre	491 (4)	297 (2)	1.2	6.3	788 (2)
Ford LTD	524 (8)	321 (5)	1.2	10.0	845 (7)
Dodge Monaco	500 (6)	306 (3)	1.2	6.1	806 (3)

*Stopping distance in a turn is shorter than that achieved in straight line.

+Numbers in these columns represent the summation of minimum stopping distance values measured in a straight line and in left- and right-hand turns.

#Numbers in parentheses represent a rank order with (1) being best performance.

cars. Among the small cars the Pinto exhibited the largest difference values in its performance on the LO-CO surface and the Pacer exhibited the largest difference values on the HI-CO surface. On the basis of these observations, it was expected that a broad range in stopping distance behavior would be covered by testing the selected sample.

Upon finding that a 1976 Pacer and a 1976 Pinto station wagon could not be rented for the in-depth test program, a 1977 Pacer and a 1977 Mercury Bobcat station wagon were substituted. These alternate choices were made after determining that essentially no changes were made from 1976 to 1977 in the design of the AMC Pacer and that the differences between the 1976 Pinto and the 1977 Bobcat wagons were principally cosmetic. The Monte Carlo and the Ford LTD were the same vehicles as had been used in the survey test program.

The fifth test vehicle, a four-wheel-antilock-equipped 1976 Nova, was loaned to the project by the Kelsey-Hayes Corporation. The Kelsey-Hayes antilock system installed on this vehicle was a two-modulator system employing separate axle control on the front and rear wheels. This system is comprised of two hydraulic-supported brake pressure modulators (hydraulic pressure is derived from the vehicle's power steering pump), a propellor shaft sensor providing average rear wheel velocity data, two front-wheel rotation sensors the outputs of which are used to obtain the average velocity of the front wheels, and the associated electronic modules.

3.3.2 Test Site and Experimental Procedures. This comprehensive test series was performed under subcontract by the personnel of the Bendix Automotive Proving Grounds at the Bendix test facility near South Bend, Indiana.

The six braking test courses required for the program were laid out on the Bendix Vehicle Dynamics Test Area, as shown in Figure 3.6. The LO-CO and HI-CO straight and curved courses and the SP-CO straight course were placed on existing surfaces of asphalt and jennite, as shown in the figure. A new, six-foot-wide strip of

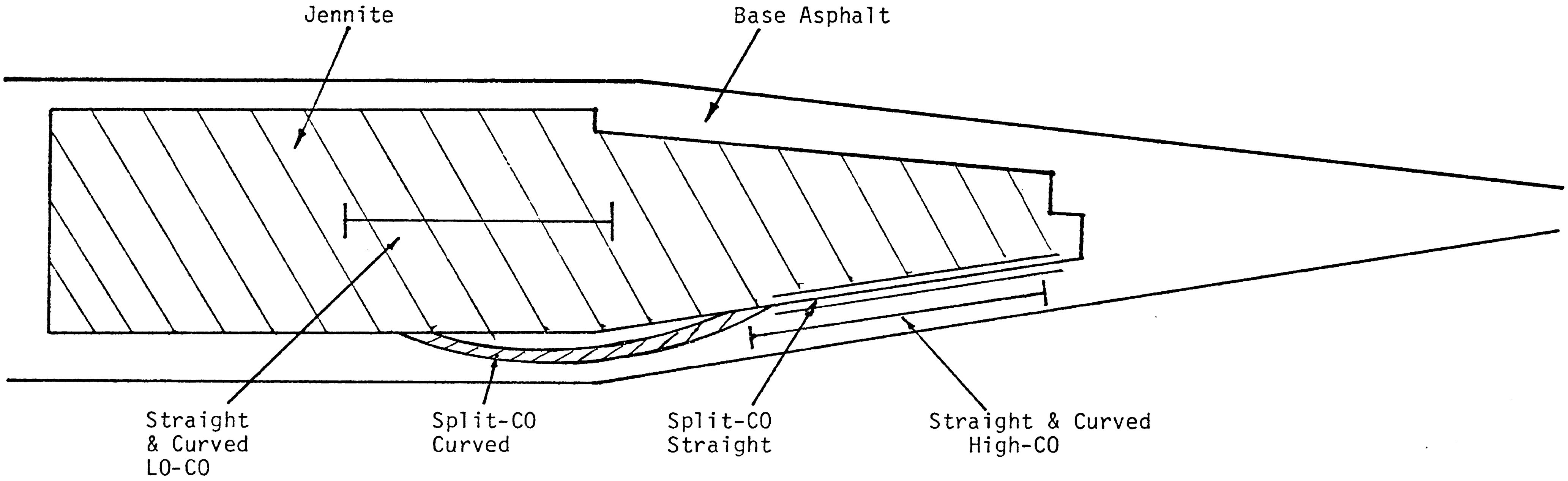


Figure 3.6. Location of the six test courses on the Vehicle Dynamics Test Area at Bendix Automotive Proving Grounds.

jennite coating was laid down over the existing base asphalt for the curved SP-CO course. As illustrated in Figure 3.7, the same jennite strip was used in four combinations of SP-CO in-a-turn tests, i.e., turning both left and right with the LO-CO surface on the inside and outside of the curve.

All of the test surfaces were characterized by peak and sliding friction measurements made once during each week in which vehicle braking tests were run. Measures were taken with the Bendix ASTM skid trailer and the DOT Surface Friction Dynamometer (SFD). A complete presentation of the obtained surface friction data is contained in Appendix C. Figure 3.8 is a plot of the data obtained on the HI-CO straight surface. The solid lines connect the high values and the low values of the two measures made on each date with the SFD. The broken lines connect the data points obtained with the Bendix skid trailer, each point representing the average of the readings from the left and right skid trailer tire; each is also averaged over as many as three runs. Data from the Bendix skid trailer is incomplete because of several breakdowns during the three-month period. Measured values of sliding friction from the two machines were in good agreement on the HI-CO surface, as seen in Figure 3.8, but agreement was not as good on the LO-CO surfaces (see Appendix C). Peak friction measures from the skid trailer were generally much higher than those from the SFD. This is, perhaps, not a surprising result since the SFD was designed specifically to produce accurate peak friction measures and the skid trailer was designed for skid number, or sliding friction, measurements. Surface friction values, peak and slide, referred to in the remainder of this report are from the SFD data only.

The temporal variations in measured surface friction were generally larger than the spread in the two values measured on each surface on a given day, indicating that a measurable variation in surface friction conditions did occur over the three-month period of the tests. However, in order to generally quantify the friction

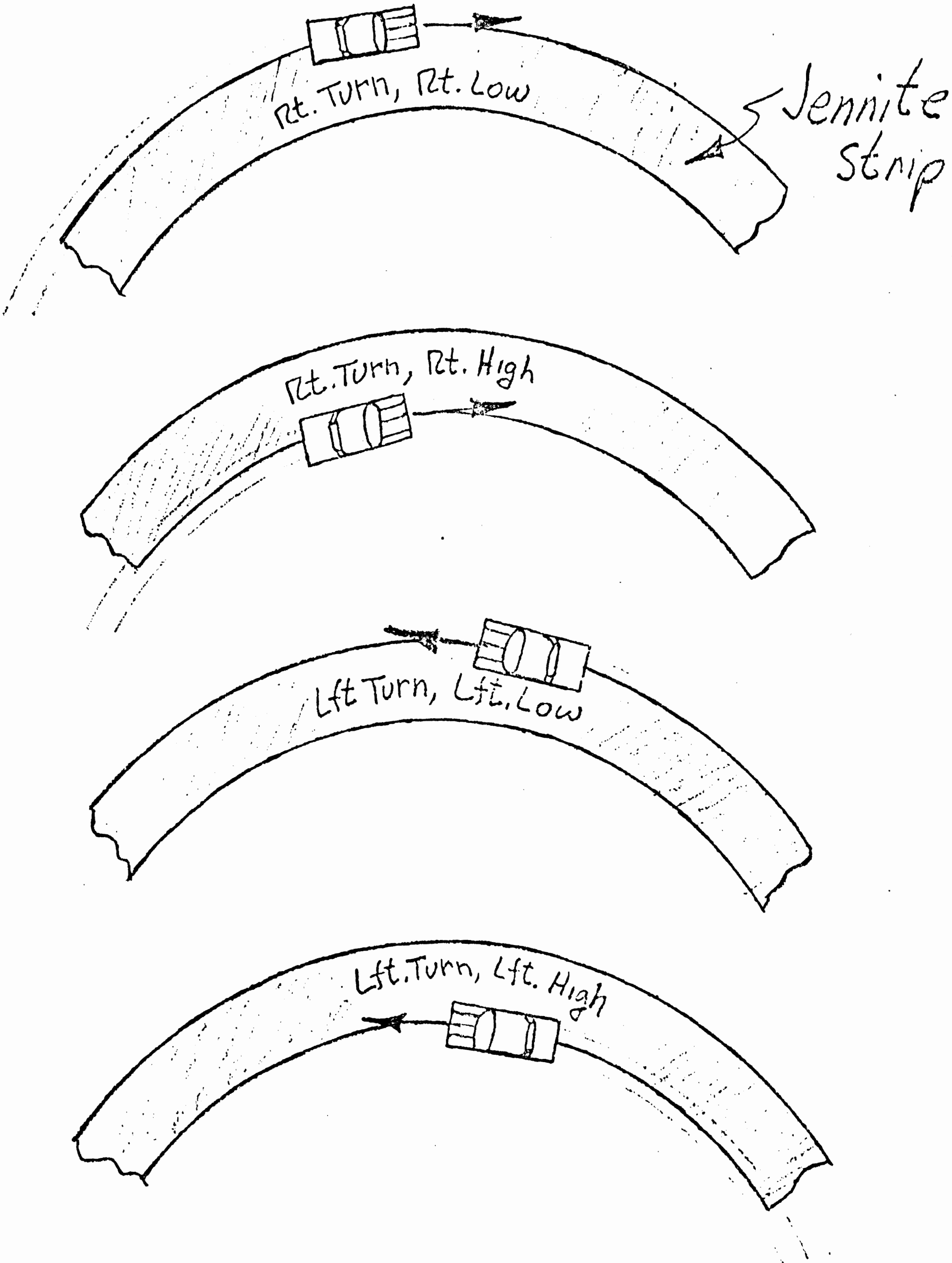


Figure 3.7. Curved jennite strip illustrating left and right turning paths with the high coefficient surface on the inside and outside of the curve.

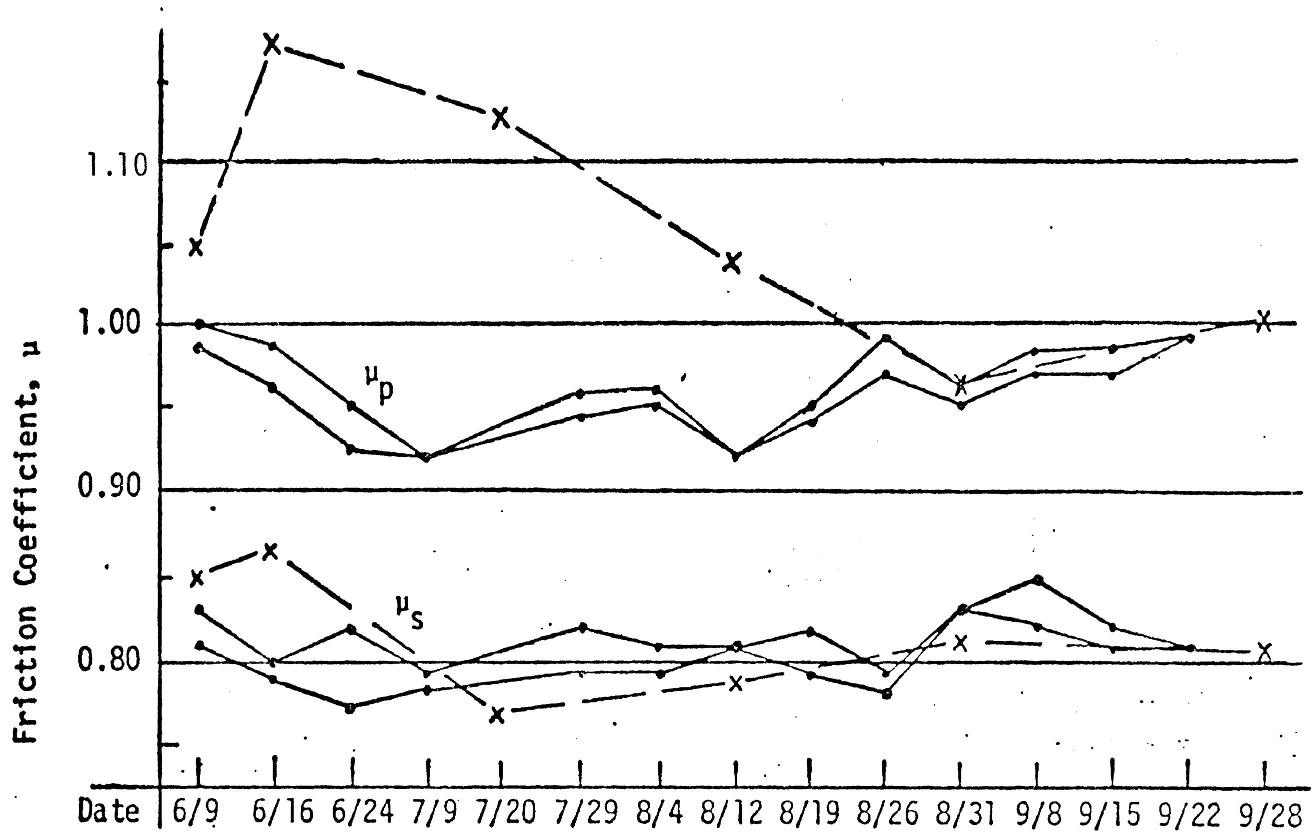


Figure 3.8. Peak and slide friction of the high coefficient straight test surface measured with the DOT Surface Friction Dynamometer (·) and the Bendix skid trailer (x).

of the several test surfaces, the average value and the maximum and minimum values over all measurements on a given surface are tabulated in Table 3.7. The HI-CO straight and HI-CO turn surfaces are seen to be essentially the same in both peak and sliding friction values. Thus a direct comparison of the HI-CO straight-line and braking-in-a-turn stopping performance is reasonable, as is the case also for the LO-CO straight and turn test since they were performed on the same surface area. On the split coefficient test areas, the peak and slide values on the HI-CO sides are nominally the same on the straight and curved areas. On the LO-CO side of the splits, the sliding friction values are nearly the same but the peak friction values differ by a factor of about 1.7. Thus for the peak friction values the degree of split is much higher on the straight split surface (about 3.1 to 1) than on the curved split surface (about 1.9 to 1).

The full matrix of surface friction measures required for computation of braking efficiency per the braking efficiency test method [1] were made with the SFD on the HI-CO and LO-CO straight test surfaces three times during the three-month test period. Plots of these data and the characteristic curves which they define are contained in Appendix C. These data, along with similar data taken at Chrysler during this contract and the data obtained during the Braking Efficiency Test Technique contract, constitute a fairly large data set with which the braking efficiency measure has been applied. This data set incorporates peak friction measures on surfaces with SN_{40} skid numbers ranging from about 20 to 80 and peak friction values from about 0.4 to 0.95. The surfaces included are: dry brushed concrete, dry asphalt, dry jennite, wet asphalt, and wet jennite. An analysis of this data set, presented in Appendix G of this report, suggests that the surface friction measurement phase of the Braking Efficiency Test Technique, requiring peak friction measurements at two tire loads and four velocities, could be simplified to measurements at one tire load and one velocity, while obtaining

Table 3.7. Peak and Slide Surface Friction Values of Test Surfaces at BAPG Measured with the SFD.

Surface		Average	Maximum	Minimum
HI-CO St. Dry Asphalt	μ_p	0.958	1.00	0.92
	μ_s	0.807	0.85	0.77
LO-CO St. & T. Wet Jennite	μ_p	0.414	0.52	0.34
	μ_s	0.227	0.26	0.19
HI-CO T. Dry Asphalt	μ_p	0.944	0.99	0.90
	μ_s	0.802	0.83	0.76
SP-CO St. HI Wet Asphalt	μ_p	0.804	0.88	0.72
	μ_s	0.463	0.55	0.37
SP-CO St. LO Wet Jennite	μ_p	0.252	0.31	0.18
	μ_s	0.140	0.19	0.11
SP-CO T. HI-I* Wet Asphalt	μ_p	0.831	0.93	0.74
	μ_s	0.489	0.58	0.33
SP-CO T. HI-O* Wet Asphalt	μ_p	0.794	0.92	0.67
	μ_s	0.481	0.62	0.38
SP-CO T. LO Wet Jennite	μ_p	0.425	0.53	0.34
	μ_s	0.165	0.23	0.12

*I = inside side of curve
O = outside side of curve

essentially the same computed ideal stopping distance as obtained from the much larger data set. This observation has provided the basis for a recommendation (Section 5.0) that the braking efficiency method may become much reduced in complexity as future research confirms the preliminary finding reported here.

Vehicle tests were conducted according to the basic procedures and conditions of FMVSS 105-75 with respect to the first (pre-burnish), second, and third effectiveness test with but a few exceptions. Exceptions were adopted concerning the topics of wind velocity, initial brake temperature, and brake pedal control forces. Test activity was permitted under prevailing steady-state wind velocities not exceeding 15 mph. Initial brake temperature was not to exceed 200°F, but the lower bound on initial brake temperature was dropped in recognition of the fact that tests on wet surfaces involving low energy stops and exposure to water spray imply low temperatures. Pedal force limits were not enforced in recognition of the fact that these were not compliance tests and that pedal forces less than 15 lbs might be encountered in tests on surfaces of low friction level.

The test sequence employed is given in Table 3.8. Burnish and re-burnish procedures per 105-75 were applied between the first and second effectiveness series and between the second and third effectiveness series, respectively. Tests on the high friction straight surface were run at 60 mph initial velocity so that results would relate directly to the current stopping distance requirement of 105-75 and would also then compare directly to the earlier test results obtained in this study at the Chrysler Proving Grounds. All other tests were conducted at an initial velocity of 40 mph. The reduced velocity condition was selected partially on the grounds of minimizing tire flat-spotting, a condition which occurred on several cars during the tests at Chrysler, particularly during the braking-in-a-turn test from 60 mph on the high coefficient surface. During braking in a turn, tires on the inside of the turn tend to lock

Table 3.8. Test Sequence Followed in Test of Five Cars at BAPG

No.	Speed	Surface	Direction	Load
		Pre-Burnish Effectiveness Series		
1	60 mph	Hi Friction	Straight	GVWR
2	40 mph	Lo Friction	Straight	GVWR
3	40 mph	Split (Hi-Rt)	Straight	GVWR
4	40 mph	Split (Hi-Lft)	Straight	GVWR
		Post-Burnish (2nd) Effectiveness Series		
5	60 mph	Hi Friction	Straight	GVWR
6	40 mph	Hi Friction	Turn-Right	GVWR
7	40 mph	Hi Friction	Turn-Left	GVWR
8	40 mph	Lo Friction	Straight	GVWR
9	40 mph	Split (Hi-Rt)	Straight	GVWR
10	40 mph	Split (Hi-Lft)	Straight	GVWR
11	40 mph	Lo Friction	Turn-Right	GVWR
12	40 mph	Lo Friction	Turn-Left	GVWR
13	40 mph	Split (Hi-Rt)	Turn-Right	GVWR
14	40 mph	Split (Hi-Rt)	Turn-Left	GVWR
15	40 mph	Split (Hi-Lft)	Turn-Right	GVWR
16	40 mph	Split (Hi-Lft)	Turn-Left	GVWR
		Third Effectiveness Series		
17	60 mph	Hi Friction	Straight	Light
18	40 mph	Hi Friction	Turn-Right	Light
19	40 mph	Hi Friction	Turn-Left	Light
20	40 mph	Lo Friction	Straight	Light
21	40 mph	Split (Hi-Rt)	Straight	Light
22	40 mph	Split (Hi-Lft)	Straight	Light
23	40 mph	Lo Friction	Turn-Right	Light
24	40 mph	Lo Friction	Turn-Left	Light
25	40 mph	Split (Hi-Rt)	Turn-Right	Light
26	40 mph	Split (Hi-Rt)	Turn-Left	Light
27	40 mph	Split (Hi-Lft)	Turn-Right	Light
28	40 mph	Split (Hi-Lft)	Turn-Left	Light

prematurely due to lateral load transfer and often minimum stopping distance is accrued with one or both inside tires locked. Successive repeat of such locked-wheel conditions leads to excessive tire wear and to a typical concentration of that wear on one spot. Regarding the initial velocity condition, it was also noted that previous NHTSA research studies [2, 3, 4] have yielded recommendations for 40 mph and 35 mph as the initial velocity condition for braking-in-a-turn tests. Furthermore, the 40-mph condition employed in the earlier tests at Chrysler showed good discrimination between vehicles as well as procedural advantages regarding test safety and repeatability of test results.

Certain of the arguments stated above also serve to rationalize the selection of 0.2 g for the initial lateral acceleration component for braking-in-a-turn tests. Although lower than the 0.3 g condition used in prior braking-in-a-turn procedures [2, 3, 4], the 0.2 g value is seen as reasonable considering the purely longitudinal interest of this study, and of 105-75. While higher values of lateral acceleration may be useful in emphasizing directional sensitivities to braking in a turn [3], the stopping distance result is adequately assessed in the more commonly encountered range of 0.2 g. This position is further supported by results from the quasi-static simulation effort, reported in Section 3.2, which showed very little differences in braking efficiencies, whether on uniform or split friction surfaces, while braking in turns of 0.2 g and 0.3 g lateral acceleration. Thus all braking-in-a-turn tests in this test series were run from an initial velocity of 40 mph, on a curve with a radius of 535 feet, giving an initial lateral acceleration of 0.2 g.

Twelve-foot-wide lanes were delineated by traffic cones for all tests. A valid test required that the vehicle remain within the traffic cone defined lanes throughout the run and, in the case of split coefficient surfaces, that the vehicle remain astride the split throughout the run. All stops were made with constant brake line pressure controlled by a Lebow Model 7610 brake machine. Pressure was incremented on successive runs to establish the minimum stopping distance performance with no more than one wheel lockup per axle, thus assuring vehicle controllability. At least two stops were required at that pressure yielding the minimum stopping distance. A maximum of ten stops were allowed in each of the 28 tests to limit the brake wear. All tests were run with the vehicle's transmission in neutral.

Each vehicle was outfitted with new OEM tires and brake linings and the brake systems were inspected to assure reasonable overall integrity and like-new condition. Tires were purchased in sets of six and, if flat-spotting occurred on the high coefficient test, the tire was replaced for the low coefficient test to prevent contamination of the data by premature wheel lockup on the flat spots.

Instrumentation was provided to permit measurement of brake lining temperature at the four wheels, brake pedal force, steering wheel displacement, initial velocity, stopping distance, and wheel lockup. Brake temperatures were sensed by Chromel-Alumel thermocouples, installed per 105-75, with readout on a digital display pyrometer. A strain gauge pedal force transducer sensed the machine-applied pedal force and a potentiometric transducer built into a replacement steering wheel sensed steering displacement. Pedal force and steering displacement were recorded on a two-channel stip chart. A standard fifth wheel with "latched-type" digital readout displays provided initial velocity and stopping distance data. D.C. tachometers mounted on each wheel were connected to lockup detection circuitry which displayed, at the end of each run, the wheel and axle lockup criteria which were satisfied. The "wheel lockup" condition was satisfied when the wheel speed dropped below the equivalent of 2 mph vehicle speed for 0.2 second or longer, with the overall vehicle speed remaining above 10 mph. Axle lockup was indicated if both wheels on an axle locked simultaneously for more than approximately 0.2 second.

3.3.3 Discussion of Full-Scale Test Results. Test results on the five vehicles tested at the BAPG are summarized in the bar graphs in Figures 3.9 through 3.14, covering the six sets of test conditions, viz., straight-line and curved-path stopping distance on high, low, and split coefficient surfaces. The range of peak and sliding friction values of each surface, measured with the SFD, are shown in each figure. The data plotted represent the shortest of the two best stopping distances obtained at optimum pedal force. These stopping distances include correction for differences in initial velocity by the application of the formula:

$$S_c = \left(\frac{V_o}{V_A}\right)^2 S_m$$

where

V_o is the intended initial velocity, i.e., 60 mph or 40 mph in these tests

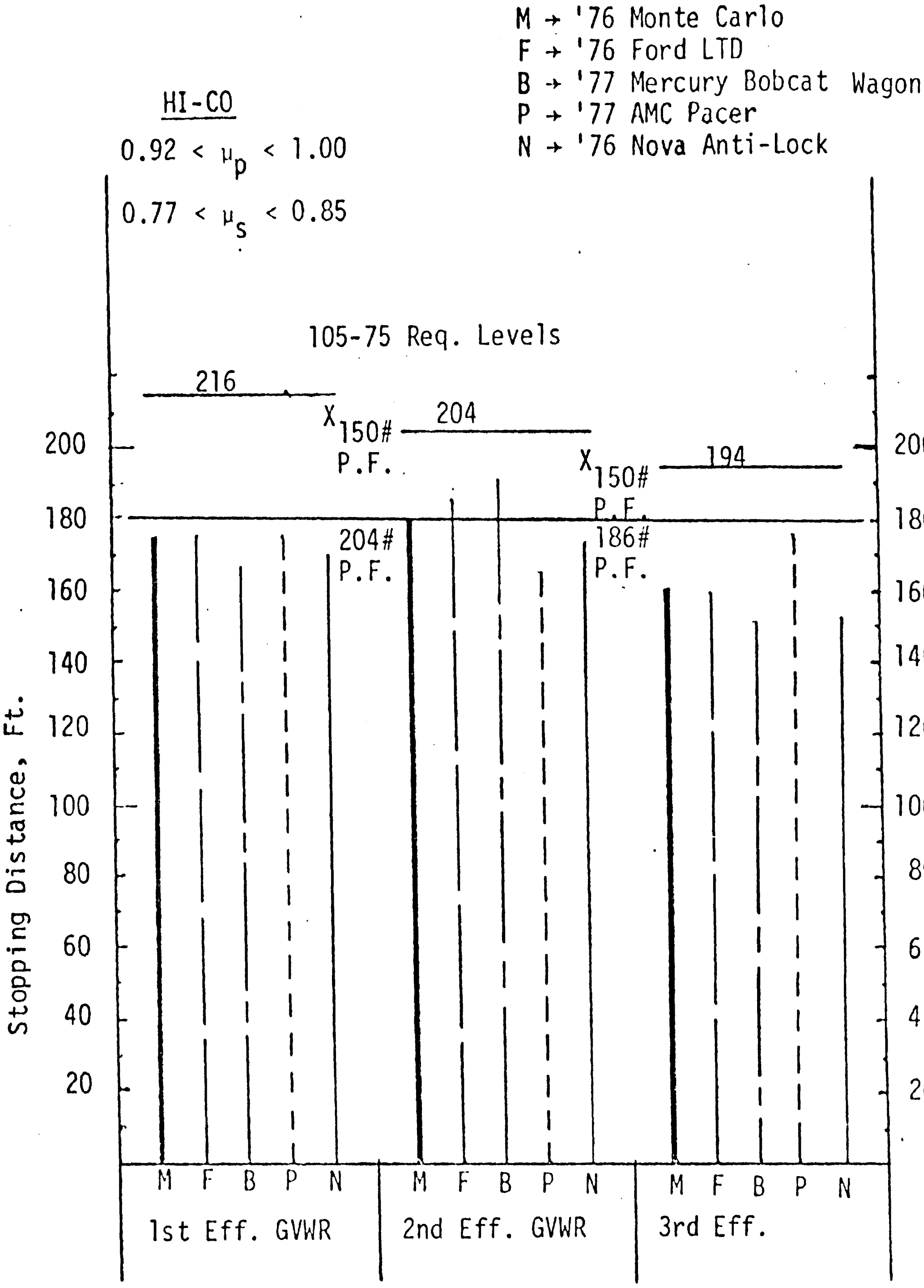
V_A is the actual initial velocity

S_m is the measured stopping distance

S_c is the corrected stopping distance

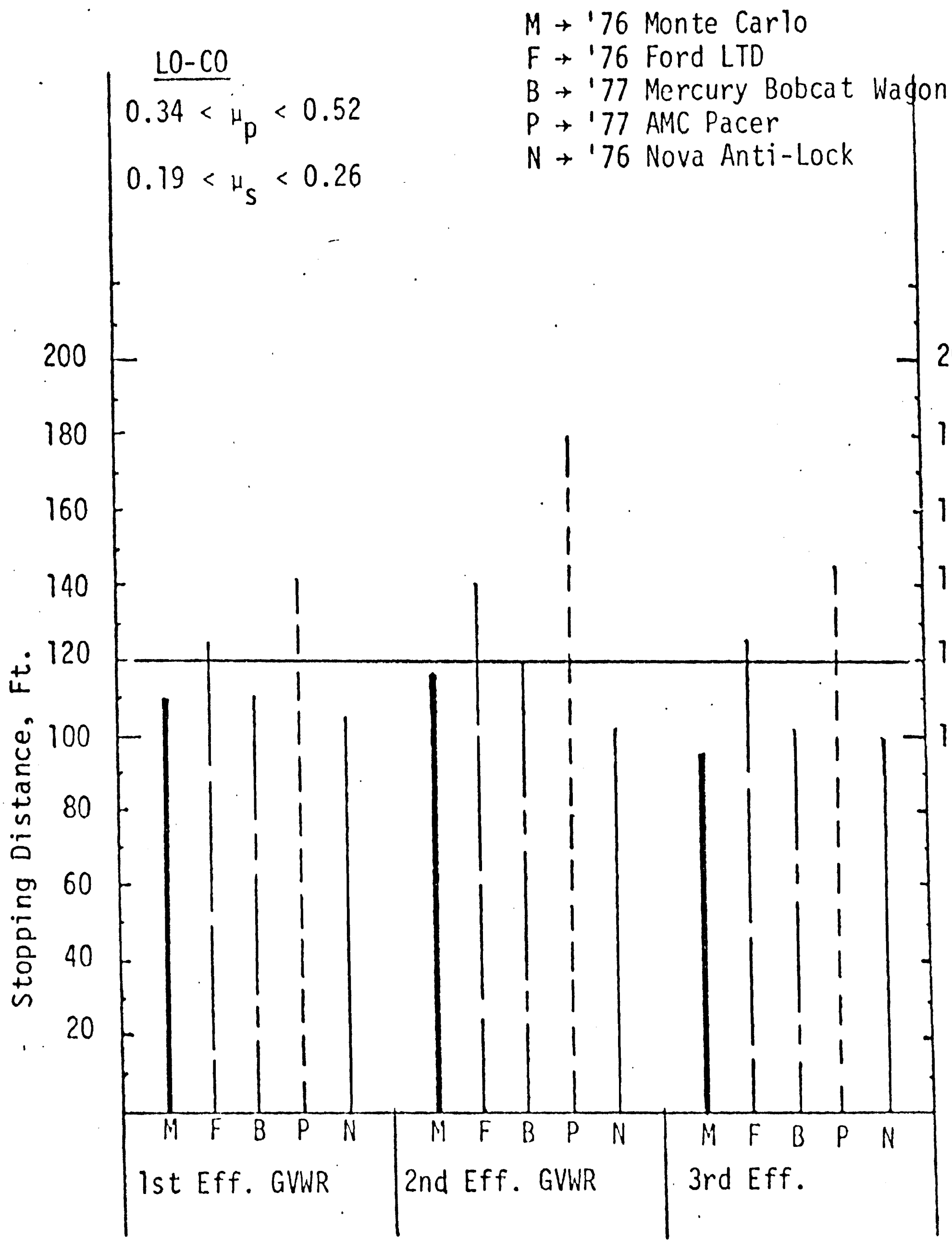
This formula is based on the assumption of constant deceleration, a condition closely approximated in constant pedal force stops.

The data plotted in Figures 3.9 through 3.14 are also presented in tabular form in Appendix C, along with wheel lockup data on the best performance stops. In the case of the antilock-equipped Nova, all best performance stops involved controlled lockup of either the front or both the front and the rear wheels, i.e., with antilock systems on the front axle or front and rear axles cycling. With the other four vehicles, most of the best performance stops involved lockup of one wheel or one wheel per axle. In particular, in straight and curved-path tests on split coefficient surfaces, 63 out of the 64 best performance stops involved lockup of one wheel or one wheel per axle. For braking in a turn on HI-CO and LO-CO



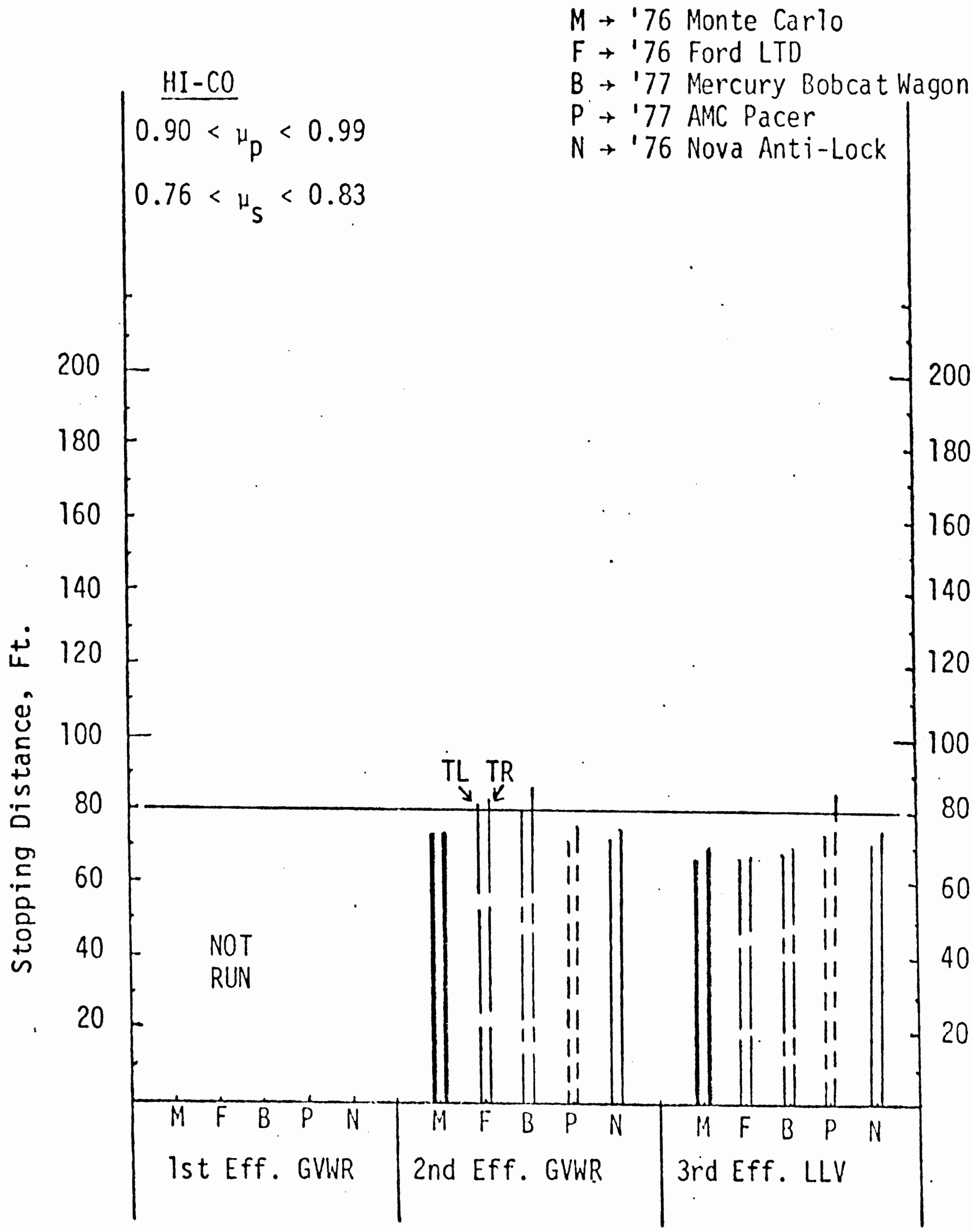
Straight-Line Stopping Distance on High Coefficient Surface. $V_0 = 60$ mph.

Figure 3.9



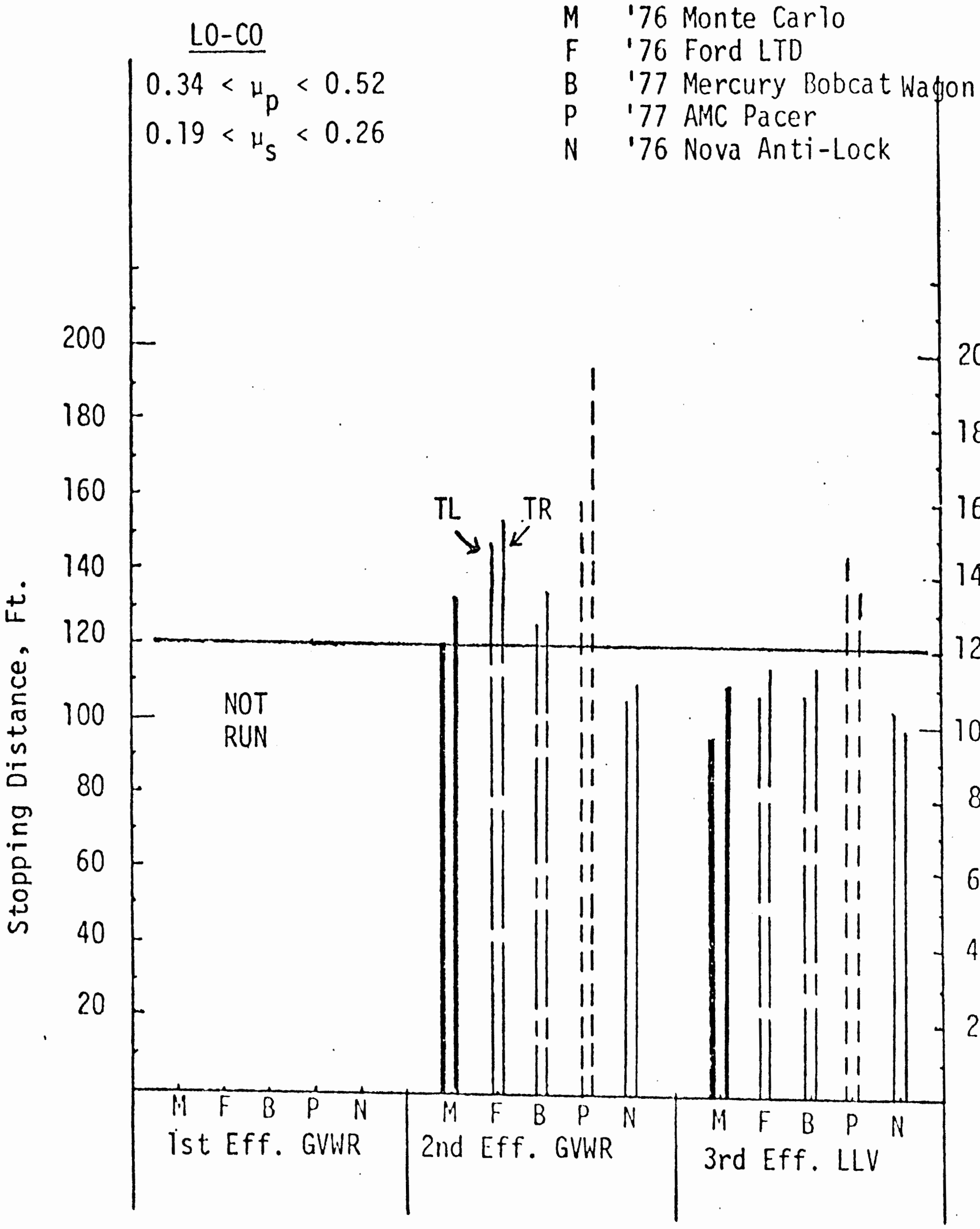
Straight-Line Stopping Distance on Low Coefficient Surface. V_0 = 40 mph.

Figure 3.10



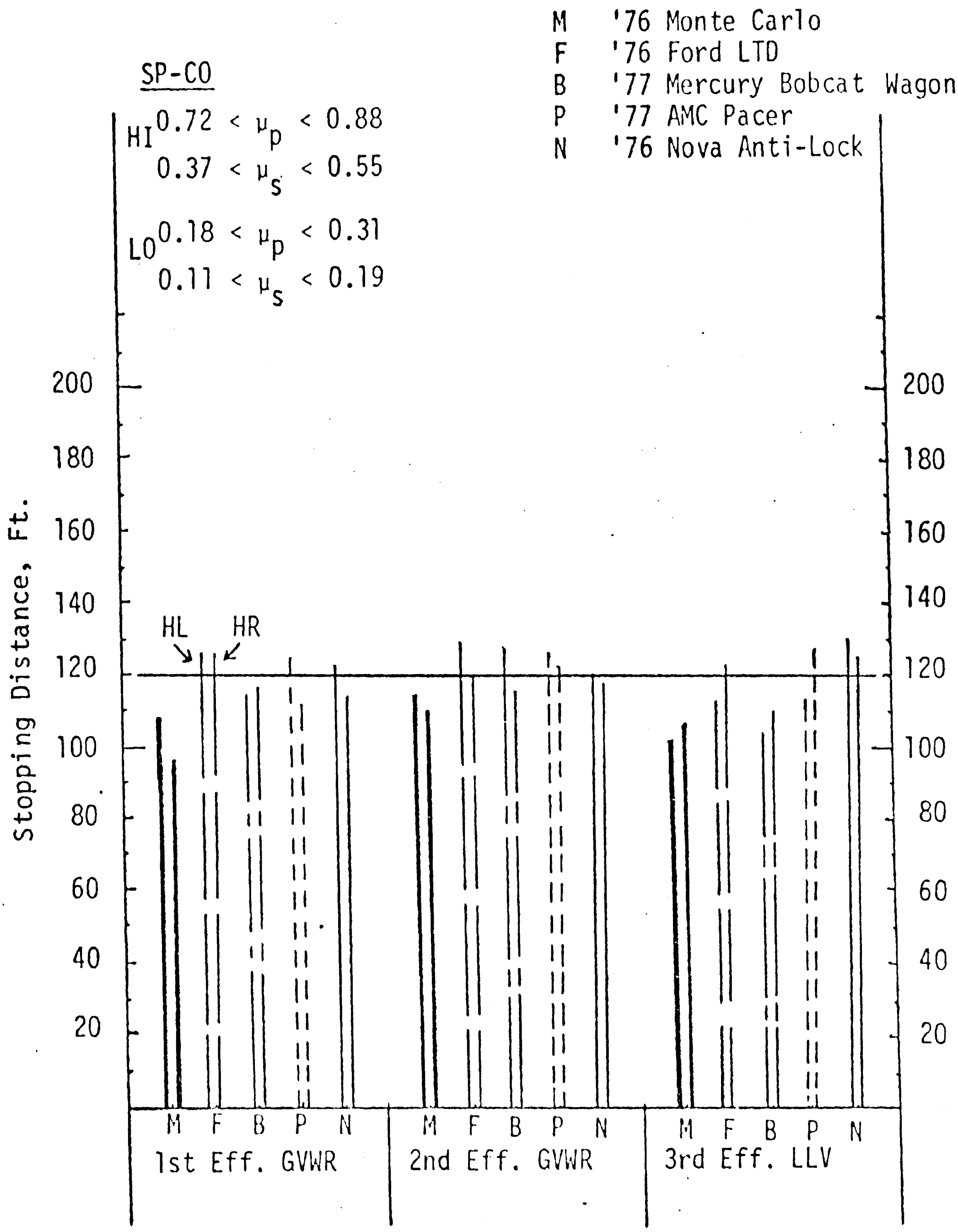
Braking-In-A-Turn Stopping Distance on High Coefficient Surface. V_o = 40 mph. A_{yo} = 0.2G. Turn Radius = 535 Ft. Turn Left - TL, Turn Right - TR.

Figure 3.11



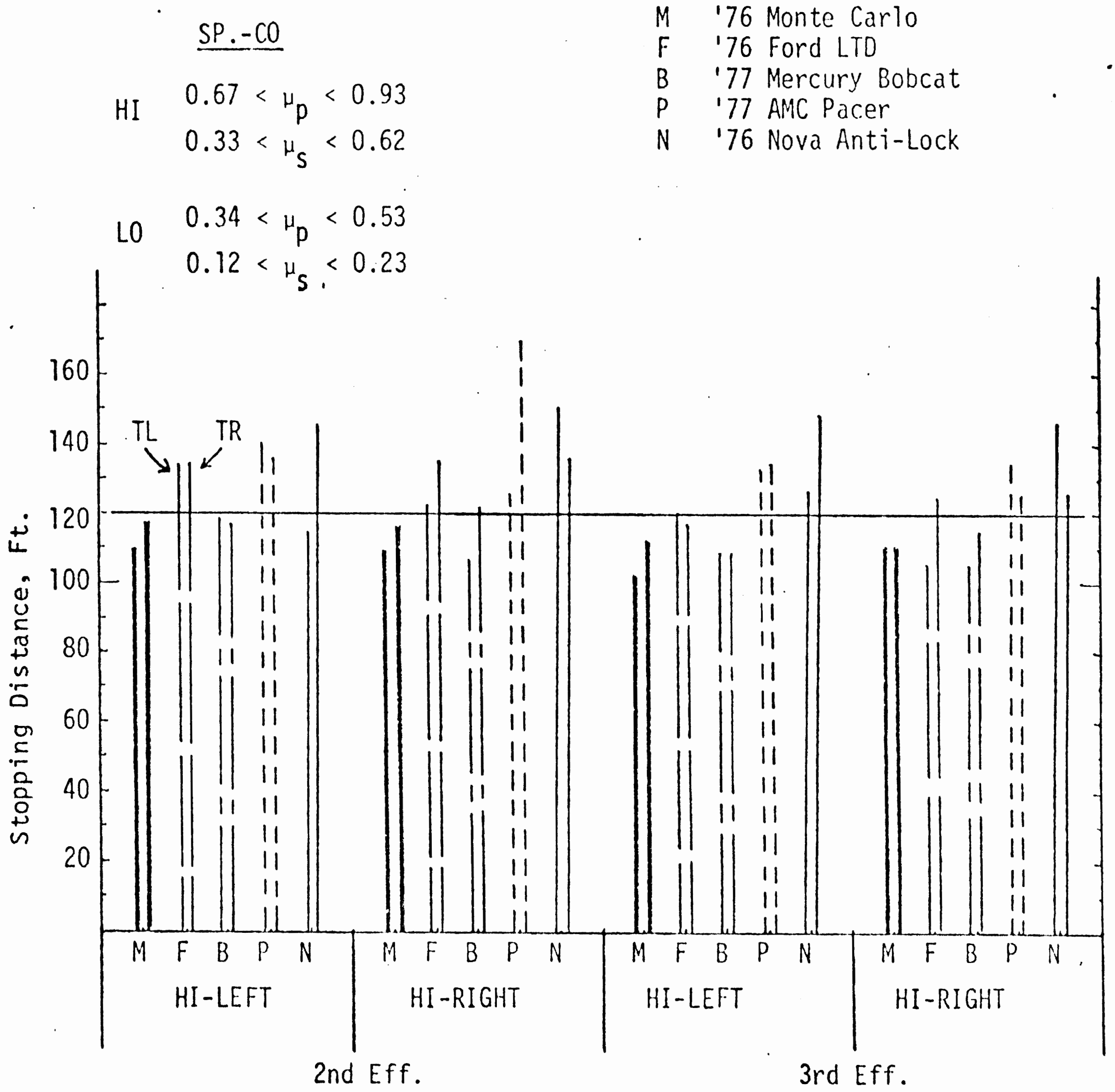
Braking-In-A-Turn Stopping Distance on Low Coefficient Surface. V_o = 40 mph. A_{yo} = 0.2G. Turn Radius = 535 Ft. Turn Left = TL, Turn Right = TR.

Figure 3.12



Straight-Line Stopping Distance on Split Coefficient Surface. V_o = 40 mph, High Left - HL, High Right - HR.

Figure 3.13



Braking-In-A-Turn Stopping Distance on Split Coefficient Surface. V_o = 40 mph, A_{yo} = 0.2G, Turn Radius = 535 Ft. Turn Left - TL, Turn Right - TR.

Figure 3.14

surfaces taken together, 22 out of 32 best stops involved lockup of at least one wheel and for straight-line stops on HI-CO and LO-CO surfaces taken together, at least one wheel was locked in 11 out of 24 best stops. A valid test required that lockup of no more than one wheel per axle occurred while the vehicle velocity was above 10 mph. However, on some of the straight-line braking tests on HI-CO and LO-CO surfaces, shorter stopping distances were obtained with either front or rear axle lockup with the vehicle remaining within the 12-foot-wide lane. This was particularly true of the Pacer, the only one of the five test vehicles which did not have a proportioning valve. The Pacer stopped in distances shorter by 10 feet, 8 feet, and 5 feet in the first, second, and third effectiveness tests, respectively, on the high coefficient surface with both rear wheels locked and by 19 feet, 47 feet, and 9 feet, respectively, on the low coefficient surface with both front wheels locked. These results indicate that the effectiveness of the unlocked axle was increasing significantly with increasing pedal force after lockup of the other axle had occurred. The results suggest that this vehicle's overall braking performance could be enhanced by an improved brake proportioning design incorporating a proportioning valve. Of the other vehicles, the Mercury Bobcat wagon exhibited a 3-foot shorter stopping distance in the HI-CO third effectiveness test with its rear axle locked and the Ford LTD exhibited a 3-foot shorter stopping distance in the LO-CO third effectiveness test with its front axle locked. Further, the wheel lock data indicated that the brake torque balance, front to rear, of the Monte Carlo, the Ford LTD, and the Bobcat wagon was close to optimum over the range of vehicle loads and surface friction conditions of these tests. These three vehicles exhibited slightly front-biased brake torque distribution in all cases with the exception that the Bobcat wagon exhibited a slight rear axle bias in the lightly loaded condition on the HI-CO surface.

Referring to the bar graphs of stopping distance, we see that on the 60-mph, high-coefficient, straight-line test (Fig. 3.9) all vehicles stopped in distances below that required by 105-75. The antilock-equipped Nova required surprisingly large pedal forces achieving its best performance on the HI-CO surface at pedal forces greater than the 150-lb limit of the 105-75 standard when loaded to the GVWR. This was due to the use of metallic brake linings on this vehicle which have a lower friction coefficient than conventional linings and therefore require higher pedal forces to achieve a given level of braking torque. The Nova antilock vehicle, loaned to the project by Kelsey-Hayes, was a police special package which Kelsey-Hayes normally operated with metallic linings because of their superior fade characteristics. The data of Figure 3.9 further show that while the Pacer exhibited the best stopping distance in the second effectiveness test, at GVWR it yielded the longest stopping distance in the lightly loaded condition (third effectiveness test). Also, it was seen that while the antilock Nova achieved the second shortest stopping distance in each of the three effectiveness series, it turned in the best average performance.

Considering the braking-in-a-turn data (Figs. 3.11 and 3.12), we see that the data spread on the HI-CO surface is not large. Further, the Pacer was again seen to perform very well relative to the other vehicles in the second effectiveness test but most poorly in the lightly loaded condition. On the LO-CO surface, the Pacer turned in the poorest performance and the antilock Nova the best. In addition, the difference between left turning and right turning performance was generally small. Note that the spread in vehicle performance is generally larger in the tests on low coefficient surfaces (Figs. 3.10 and 3.12) than it is on the high coefficient surfaces (Figs. 3.9 and 3.11).

Results of the straight-line braking test on the split coefficient surface are shown in Figure 3.13. Note that none of the vehicles show a significant asymmetry with regard to the right- or left-side placement of the high coefficient surface. Also note that all minimum distance stops involved the lockup (above 10 mph) of one or both of the wheels on the LO-CO side of the split.

Excluding the antilock Nova from the count, 13 out of 24 best stops occurred with only the LO-CO side front wheel locked and all others occurred with both the front and rear wheels on the LO-CO side locked.

Figure 3.14 shows the results of the braking-in-a-turn tests on the split coefficient surface. All but one of the minimum distance stops in these tests involved lockup of one or both of the wheels on the LO-CO side of the split. Again excluding the antilock Nova from the count, 21 out of 32 best stops occurred with only the LO-CO side front wheel locked and all others occurred with both the front and rear wheels on the LO-CO side locked. A somewhat surprising result here is that only the antilock Nova exhibited a consistent significant asymmetry in left and right turning stopping distance. In every case the Nova achieved the shortest stopping distance turning in the direction that placed its outside, or most heavily loaded, tires on the low coefficient side of the split. This is readily explained if we assume that the wheel (or wheels) on the LO-CO side of the split are always the first to lock, as was found to be the case with the non-antilock veicles, and if we consider that the antilock cycling control is derived from the average angular velocity of the two wheels on an axle. Thus a higher braking torque is achieved before wheel lockup occurs (and cycling is initiated) when the heavily-loaded tires are on the LO-CO side of the split rather than on the HI-CO side, and a shorter stopping distance is accrued.

Of the other four vehicles only the Pacer, in one test (second effectiveness, HI-Right), exhibited a large left to right turning asymmetry. In all other tests, the asymmetry was small (for all four non-antilock-equipped cars) and no consistent pattern was evident relating shorter stopping distances to the inside/outside location of the HI/LO-CO surfaces of the split in a turn.

In order to further elucidate the stopping distance data presented in the several bar charts, certain quantitative comparisons of the test results are presented below. First, as a measure of data variability, the difference in stopping distance between the two best stopping distance runs was tabulated (see Appendix C for the complete tabulation). The average difference (and average percentage difference) over all 28 tests for each car was: Monte Carlo, 2.3 feet (2.2%), Ford LTD, 4.3 feet (3.4%), Bobcat wagon, 1.6 feet (1.4%), Pacer, 4.5 feet (3.3%), and Nova (antilock), 3.4 feet (2.8%). The largest single difference observed was 36 feet (29%) in the LO-CO straight test with the Ford LTD.

The rank order of stopping distance performance of the five vehicles is readily seen in the bar graphs for each individual test. In a few cases a change in rank occurs from test to test, but large changes are rare. The AMC Pacer, for example, is seen in the HI-CO straight test to move from a rank position of (1) in the second effectiveness test to (5) in the third effectiveness test. Looking at overall performance factors, we find that the four conventional vehicles rank in the same order as they did overall in the previous tests conducted at Chrysler Proving Grounds (i.e., the ranking shown in Table 3.6, column 5). Table 3.9 contains three overall performance comparisons of the five vehicles: the sum of all minimum stopping distances for each vehicle on homogenous surfaces (i.e., the sum from the straight and in-a-turn test on both HI-CO and LO-CO surfaces); the sum of all minimum stopping distances for each vehicle on the split coefficient surfaces; and the sum of <u>all</u> minimum stopping distances for each vehicle from all 28 tests. The four conventional vehicles keep the same rank <u>with respect to each other</u> in all three columns in Table 3.9, which is also the same rank order they achieved in the previous tests, i.e., in order of best to poorest performance; Monte Carlo, Bobcat wagon, Ford LTD, and Pacer. Considering all five vehicles, the antilock Nova ranked first on the homogeneous surfaces (column 1), fourth on the split coefficient surfaces (column 2), and third overall (column 3).

Table 3.9. Overall Braking Performance Comparisons of Five Cars in Terms of the Sums of Minimum Stopping Distances in Several Tests.

	Σ on Homogeneous Surfaces	Σ on Split CO Surfaces	Total on All Tests
Monte Carlo	1578 (2)	1522 (1)	3100 (1)
Ford LTD	1729 (4)	1726 (3)	3455 (4)
Bobcat Wagon	1628 (3)	1582 (2)	3210 (2)
Pacer	1924 (5)	1818 (5)	3742 (5)
Nova Antilock	1513 (1)	1806 (4)	3319 (3)

Table 3.10 gives a comparison of the straight-line and in-a-turn braking performance of each vehicle on both the HI-CO and LO-CO surfaces in the second and third effectiveness tests. The average value of the left turn and right turn stopping distances was used. Since the straight and in-a-turn tests were run at different velocities (60 mph and 40 mph, respectively), the HI-CO test data measured at 60 mph was converted to an equivalent 40-mph stopping distance, in order to make the comparison, by application of the formula $S_c = (40/60)^2 S_m$ (see page 43). In 9 out of 20 cases, the in-a-turn stopping distance was shorter than the straight stopping distance. All differences were less than 11% in magnitude. The average differences were slightly larger on the LO-CO surface than on the HI-CO surface and are only slightly larger in magnitude than the data variability differences given above on page 53. These results are comparable to those obtained in the previous twelve-car sample tests.

Similar "difference measures" were computed for various additional cases as summarized in Table 3.11. This table presents various average and maximum percentage differences, where the average is taken over all stopping distance data gathered in the

Table 3.10. Percent Difference in Stopping Distance Between Straight-Line and Braking-in-a-Turn on HI-CO and LO-CO Surfaces from 40 mph.

	2nd Eff.		3rd Eff.	
Vehicle	HI-CO	LO-CO	HI-CO	LO-CO
Monte Carlo	*8.7%	8.4%	*4.3%	7.7%
Ford LTD	0.3	6.9	*5.1	*10.9
Mercury Bobcat	*4.0	8.5	2.5	8.4
AMC Pacer	*0.3	*2.1	0.8	*2.5
Nova (Antilock)	*3.9	5.6	7.5	1.3
Average	3.4	6.3	4.0	6.2

*In-a-turn stopping distance shorter than straight-line stopping distance.

Table 3.11. Percentage Differences in Stopping Distance Averaged Over All Five Cars and Over the 2nd and 3rd Effectiveness Test for Eight Different Combinations of Test Conditions.

Test Condition	Avg.	Max.
High Co. Δ (Turn Right-Turn Left)	4.8%	10.5% (Pacer)
Low Co. Δ (Turn Right-Turn Left)	8.2	12.6 (Monte Carlo)
Split Co. HI-Right Δ (Turn Right-Turn Left)	11.5	14.6 (LTD)
Split Co. HI-Left Δ (Turn Right-Turn Left)	6.5	7.8 (Nova)
Split Co. Δ (HI Left Turn Right - HI Right Turn Left)	5.3	10.2 (LTD)
Split Co. Δ (HI Right Turn Right - HI Left Turn Left)	6.0	8.3 (Nova)
Split Co. Straight. Δ (HI-Left - HI-Right)	1.8	4.6 (Pacer)
Split Co. Δ (Turn-Straight)	5.7	12.6 (Pacer)

second and third effectiveness tests. Tabulations of the differences for each car averaged over the second and third effectiveness tests are given in Appendix C. Certain of these differences are somewhat larger than the average variability found in the stopping distance measures. Considering the complex conditions in braking in a turn and on split coefficient surfaces, however, they do not appear significant as discriminating measures of vehicle performance.

A comparable means of viewing the absolute values of stopping distances measured on split coefficient surfaces is shown in Table 3.12. These tabulations show, for first, second, and third effectiveness tests, that the split coefficient condition imposed no unusual demands over that associated with straight-line braking on a low coefficient surface which is roughly comparable in friction level to the LO-CO side of the split. We see in these data that only the antilock Nova exhibits a significantly longer stopping distance on the split coefficient surface compared to its performance on the homogeneous low coefficient surface. Except for the Pacer which stopped in significantly shorter distances on the split surfaces, the other vehicles showed no significant differences. A full interpretation of these results, however, requires careful consideration of the vehicle wheel lockup status and of the peak and slide friction values of the several surfaces.

The quasi-static simulation effort reported in Section 3.2 demonstrates the complexity of the split coefficient/braking-in-a-turn situation resulting from factors such as lateral and longitudinal load transfer, front-to-rear proportioning, peak-to-slide friction ratios, degree of split between HI-CO and LO-CO sides, etc., showing that maximum braking efficiency could occur with various combinations of wheel lockup, depending on interactions of these factors. An understanding of the gathered data for split friction braking is facilitated by a generalized discussion of the braking force characteristics encountered on a split coefficient surface, as shown in Figure 3.15. This is a plot of the conceptual relationship between pedal force and right/left braking forces

Table 3.12. Percentage Differences in Stopping Distance Between LO-CO Straight and SP-CO Straight Tests and Between LO-CO Straight and SP-CO Turn Tests in 1st, 2nd, and 3rd Effectiveness Tests.

Vehicle	% Difference LO-CO Straight SP-CO Straight 1st	2nd	3rd	% Difference LO-CO Straight SP-CO Turn 2nd	3rd
Monte Carlo	*7.3%	*5.1%	10.4%	*4.2%	13.5%
Ford LTD	0.0	*12.1	*4.8	*7.1	*5.6
Bobcat Wagon	4.5	0.8	3.9	*2.5	5.8
Pacer	*17.6	*32.0	*16.7	*21.0	*8.3
Nova Antilock	11.4	13.0	27.0	31.1	36.0

*Stopping distance on the split coefficient surface shorter than on the low coefficient surface.

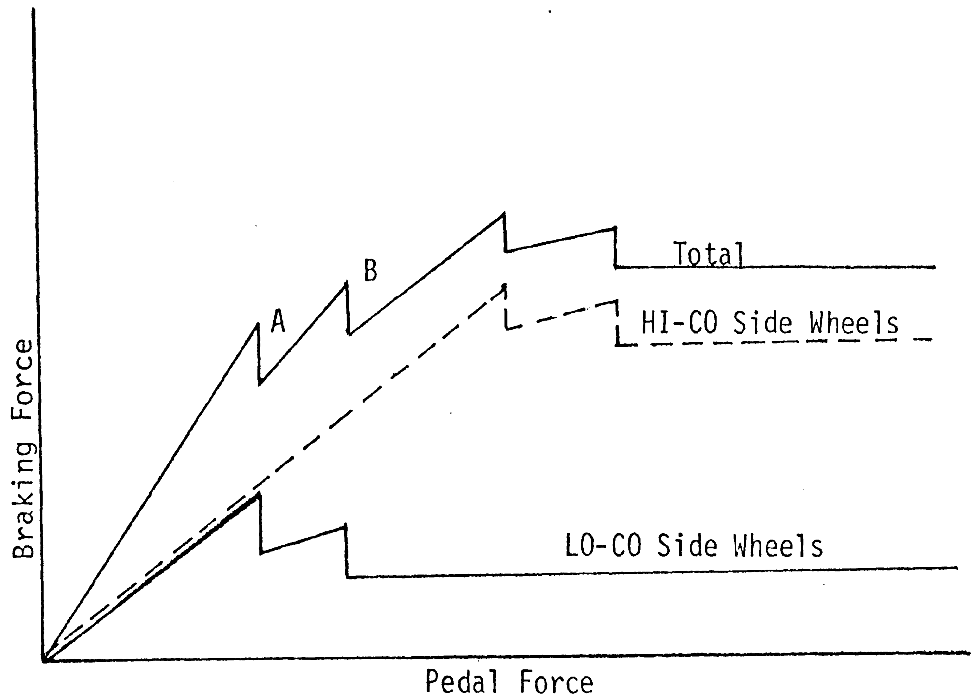


Figure 3.15. Illustration of pedal force vs. braking force on a split coefficient surface where both low coefficient side wheels lock before the high coefficient side wheels.

(such as derive from the tractive forces of tires on the high and low coefficient sides of the split) and total braking force. Each abrupt drop in the various curves represents a wheel lockup; thus the magnitude of the drop is determined by the peak and slide friction values of the surface and the instantaneous tire load. In this example we have shown both LO-CO side tires locking before either HI-CO side tire, as was generally the case in the tests at Bendix. The first peak in the total force curve is the maximum no-wheels-locked braking force. The relative positions (horizontally) and the amplitudes of the four peaks, as well as the slope of the curve between peaks, depends upon the several factors mentioned above, i.e., lateral and longitudinal load transfer (or instantaneous tire loads), proportioning, peak-to-slide ratios, etc. Immediately upon lockup of one wheel, a side-to-side braking force unbalance and a resulting yaw moment occurs which increases in magnitude as the pedal force increases. If the side force capability of the tires is exceeded or the driver is not able to provide adequate steering correction, the vehicle will depart from the desired path. Note that if optimum four-wheel dynamic proportioning could be achieved, all wheels would arrive at incipient wheel lock at the same pedal force and maximum utilization of the available friction or braking force would be achieved. On a surface with a large increment in friction across the split, however, a large yaw moment would still result which could result in loss of vehicle control.

From the curves in Figure 3.15, it appears that on a surface with a large split and a relatively small difference in peak-to-slide friction values on the LO-CO side, the largest net braking forces (shortest stopping distances) would be achieved with the LO-CO side wheels locked and the HI-CO side wheels operating near their peak traction limits. In such cases, the best stopping distance would be limited by the driver's ability to steer such that the vehicle would stay astride of the split. One could also conjure up a set of split friction conditions such that the difference between the maximum no-wheels-locked braking force would approximate the net brake force achievable with the LO-CO side wheels locked and the HI-CO side wheels operating near their traction limits. A close look at the Bendix stopping distance and surface friction data suggests that the latter condition existed in the split coefficient braking-in-a-turn tests and the former condition existed in the split coefficient straight-line braking tests. In both cases, the vehicle was controllable with one or both inside (LO-CO) wheels locked and with outside wheels operating near their traction limit. However, a much larger driver steering effort was required in these tests than in the straight and in-a-turn tests on the homogeneous surfaces.

From the wheel lockup summary given earlier, it is clear that the best performance stops in the split coefficient tests occurred with the vehicle operating in the regions indicated in Figure 3.15 as A (one LO-CO-side wheel locked) and B (both LO-CO side wheels locked).

Surface friction measures gathered in these tests appear to be related to the stopping distance data only by considering averages. The pertinent average surface friction values (taken from Table 3.7) are given in Table 3.13. Also in Table 3.13 are listed the average of the peak friction on the HI-CO side and the sliding friction on the LO-CO side of the split. This numeric relates to the maximum braking force available under the condition of the LO-CO wheels locked and the HI-CO wheels operating near their traction limits. Assuming this average to be the traction limit in the split coefficient tests, we see that the values on the split surfaces for the straight-line and turning courses are almost identical (0.47-straight and 0.49-turn). Further, we see that these values are only slightly larger than the traction limit for the LO-CO homogeneous surface which had a nominal peak value of 0.41. Considering these simple friction measures alone, we would expect the stopping distances achieved on these three surfaces to be about

Table 3.13. Average Surface Friction Values on Low Coefficient Straight Surface and on the Split Coefficient Straight and Curved Surfaces.

LO-CO Straight	SP-CO Straight		SP-CO Turn	
	HI-SIDE	$\bar{\mu}_{PH}=0.80$	HI-SIDE	$\bar{\mu}_{PH}=0.81$
	LO-SIDE	$\bar{\mu}_{PL}=0.25$	LO-SIDE	$\bar{\mu}_{PL}=0.43$
		$\bar{\mu}_{SL}=0.14$		$\bar{\mu}_{SL}=0.17$
$\bar{\mu}_{P}=0.41$	Avg.	$\frac{\bar{\mu}_{PH}+\bar{\mu}_{SL}}{2}=0.47$		$\frac{\bar{\mu}_{PH}+\bar{\mu}_{SL}}{2}=0.49$

the same; perhaps slightly shorter on the split surface. This expectation is indeed confirmed by the data in Table 3.12. (If, on the other hand, a no-wheels-locked test constraint were employed, the traction limit on the split surfaces would have been determined simply by the peak friction value on the LO-CO side of the split.)

It was mentioned in the above discussion that the driver effort required to maintain vehicle control was considerably greater on the split friction surfaces than on the homogeneous surfaces. A quantitative measure of this effort is given by the steering displacement data summarized in Table 3.14. Since the four vehicles with conventional brake systems displayed very similar results, data from these vehicles is averaged together. The antilock vehicle consistently required less steering effort and average values for it are shown separately. For the conventional vehicles the peak-to-peak steering displacement on the split surfaces was about twice that required on the homogeneous surfaces. Also, we see that only slightly larger levels of steering effort were required for the case of braking in a turn on homogeneous surfaces over that required during straight-line stops. On the average, the steering effort with the antilock Nova was less than half that required with the conventional vehicles.

Table 3.14. Average Peak-to-Peak Steer Angle and Average Number of Steering Reversals.

	Conventional Vehicles		Antilock Nova	
	Average P-P Steer	Average No. of Reversals	Average P-P Steer	Average No. of Reversals
HI-CO Straight	85°	3.6	16°	5.0
LO-CO Straight	81°	3.4	11°	2.3
HI-CO Turn	107°	2.9	48°	2.0
LO-CO Turn	110°	3.3	40°	4.0
SP-CO Straight	191°	3.9	30°	4.7
SP-CO Turn	195°	3.4	82°	6.7

The braking efficiency numeric was computed for each vehicle using the minimum stopping distance data from the high and low coefficient straight-line tests and for each of the three sets of surface friction measurements taken with the SFD during the three-month test period. The complete surface friction data is presented in Appendix C. Table 3.15 contains the results of the braking efficiency computations. The efficiency values deriving from the surface friction data labeled number "II" (columns 2 and 5 in Table 3.15) are plotted in Figure 3.16 for ease of comparison.

All vehicles except the Pacer achieved significantly higher utilization of wet low coefficient surfaces than of the dry high coefficient surface and the top ranked vehicles (Monte Carlo, Bobcat wagon, and Nova) exhibited larger differences than the bottom ranked Ford LTD and Pacer. One would expect that in general the higher utilization of the wet surface traction derives largely from poorer wet surface traction of the ASTM-501 test tire, which has a relatively smooth, unsiped tread structure, compared to the typical passenger car tire.

Comparing the efficiency values computed from the same stopping distance value but for each of the three sets of surface

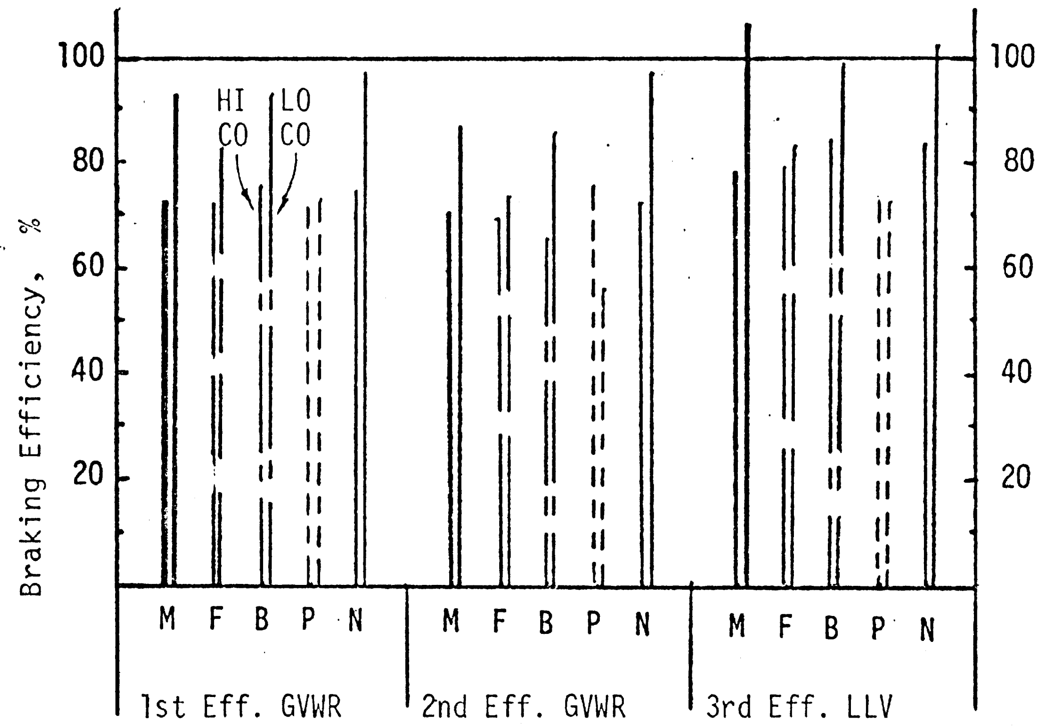
Table 3.15. Braking Efficiencies of Five Vehicles on High Coefficient and on Low Coefficient Surfaces, Straight-Line Stops.

Vehicle	Eff. Test	HI-CO Dry Asphalt I	II	III	LO-CO Wet Jennite I	II	III
Monte Carlo	1st	71.7%	71.9%	70.6%	87.7%	92.6%	86.0%
	2nd	69.6	69.9	68.6	82.5	87.0	80.9
	3rd	77.4	77.8	76.4	101.0	106.7	99.1
Ford LTD	1st	71.4	71.8	70.4	76.9	81.2	75.4
	2nd	67.8	68.2	66.9	68.5	72.3	67.2
	3rd	78.3	78.8	77.3	77.7	82.1	76.3
Bobcat Wagon	1st	74.6	75.0	73.6	87.1	91.9	85.4
	2nd	64.8	65.0	63.8	80.7	85.2	79.2
	3rd	82.6	83.4	81.8	94.0	99.3	92.2
Pacer	1st	70.6	71.4	70.0	68.2	72.1	66.9
	2nd	75.2	75.6	74.2	53.4	56.5	52.4
	3rd	70.6	71.0	69.6	67.3	71.2	66.0
Antilock Nova	1st	73.5	73.7	72.3	92.1	97.2	90.3
	2nd	72.4	72.8	71.4	92.6	97.2	91.1
	3rd	82.0	82.6	81.0	96.9	102.5	95.1

I Based on SFD Measurements from June 6, 1977

II Based on SFD Measurements from July 19, 1977

III Based on SFD Measurements from October 3, 1977



M → '76 Monte Carlo

F → '76 Ford LTD

B → '77 Mercury Bobcat

P → '77 AMC Pacer

N → '76 Nova Antilock

Figure 3.16. Braking efficiencies on high coefficient and low coefficient surfaces, straight-line stops.

friction measurements (Table 3.15), we find that the maximum and average spread in the efficiency numeric is 1.6% and 1.4%, respectively, on the dry surface and 7.1% and 6.1%, respectively, on the wet surface. This is a clear demonstration of the larger variability to be expected from wet surface friction measurements compared to dry surface friction measurements.

In summary, we find that where stopping distance is the only performance measure, little or no information is gained by performing braking-in-a-turn and split-coefficient braking tests which cannot be deduced from straight-line braking tests on high and low coefficient homogeneous surfaces. Antilock-equipped vehicles can be an exception, perhaps, as evidenced by the right/left turning asymmetry revealed in the split coefficient in-a-turn tests of the antilock Nova and the longer stopping distances exhibited by this vehicle on the split surfaces. In the case of split coefficient surfaces, if no wheel lockups are permitted, the stopping distance will be equivalent to that achieved on a uniform surface with peak friction equal to that of the low friction side of the split. If wheel lockup on the low side is permitted, the stopping distance will approximate that which would be achieved on a uniform surface with a peak friction value equal to the average of the peak friction value on the high friction side and the sliding friction value on the low friction side. A large steering effort is required to compensate for the yaw moment which is generated when one wheel per axle is allowed to lock on the split coefficient surface. Consequently, it is doubtful that the average driver would achieve the minimum stopping distances on the split surfaces achieved by the professional test driver in the tests at Bendix. In reality the stopping distance would generally be limited by vehicle controllability rather than the available traction on the split coefficient surface.

The antilock-equipped vehicle exhibited the best overall performance on homogeneous surfaces, but ranked fourth in performance on the split coefficient surfaces. However, the antilock system relieves the driver of the task of precisely modulating pedal force and significantly eases the steering task during maximum performance stops.

3.4 Simulation of Vehicles Used in the In-Depth Test Program

The five vehicles which were tested as described in the previous section were also simulated by computer, in order to investigate the sensitivity of the braking performance to small deviations in the test conditions. An existing vehicle handling program run at the Applied Physics Laboratory (APL) at John Hopkins University was used for this purpose, and was modified only to include a model of the antilock system which was installed on the Nova test car.

This section briefly describes the model employed and some emphasis is given to the measurement and estimation of the various parameter values needed for the simulation. The computed sensitivities of stopping distance performance to variations in the test conditions are then presented and discussed.

The parameter listings for the five vehicles are included in Appendix D, as are some of the other simulation results not included in this section. The mathematical model developed to represent the antilock system is presented in Appendix F, which also lists the FORTRAN subroutine which was added to the APL program.

3.4.1 Computer Model. The mathematical model which was used for this study is documented in References [5] and [6]. The general character of the model is summarized by the following exerpts from Reference [6]:

> The simulated vehicle is represented by a 17-degree-of-freedom model that consists of

- A basic 6-DOF model of the vehicle sprung mass
- A 2-DOF front wheel or front axle model
- A 2-DOF rear wheel or rear axle model
- A 3-DOF steering system model
- A 4-DOF wheel rotational dynamics model

The 6 DOF's of the sprung mass model are the 6 standard translational and rotational DOF's of a body moving in inertial space: translation along three axes and rotation about each axis. The 2 front wheel DOF's represent the motion of the front unsprung masses. For an independent front suspension, the 2 DOF's are the vertical motion of each front wheel. If the front suspension represents a solid axle configuration, the 2 DOF's are the rotation and vertical motion of the front axle. A corresponding discussion describes the 2 rear wheel DOF's.

.... Four additional DOF's ... are contained in the rotational equations of motion about the spin axis of each wheel. These equations, which include the differential effects of the rear wheels, yield the wheel rotation rates from which slip and, in turn, the circumferential and lateral friction coefficients are computed...

.... The suspension forces include spring effects, shock absorbers, auxiliary roll stiffness, coulomb friction, and "anti" forces such as antipitch and antiroll...

The force of gravity, aerodynamic forces and moments, and tire forces and moments generated at the tire-road interface are considered the only important externally applied forces and moments acting on the vehicle. The tire forces include the radial force arising from radial tire deflection, the side force due to slip angle and inclination angle, and the circumferential force arising from applied torque. Since the roadway is treated as a flat horizontal plane, a "point-contact" representation of the tire is used to obtain the radial loading. The circumferential force calculation uses a two-straight-line approximation of friction coefficient-slip behavior. The side force calculations are based on slip-angle and inclination-angle properties of the tires which are saturated at large angles. Interaction between circumferential and side forces utilizes a modified "friction-ellipse" concept that "rolls off" side force as a function of tire slip. The "rolloff" characteristic is an emperical relationship obtained from tire test data. The tire aligning torque and overturning moment are modeled as nonlinear functions of lateral force, normal force, and inclination angle....

Open-loop control inputs can be entered in the form of steering wheel angle and drive or brake torque.... The brake torque magnitude is determined from input data functions of brake line pressure at the front and rear wheels.

It should be clear from the above description that a large amount of information is required to describe the vehicle being simulated. Most of the needed parameter values were either taken from published specifications or estimated from measurements made by HSRI and other groups on comparable vehicles. Many of the parameters of lesser significance in the computer model, such as the aerodynamic coefficients, the anti-pitch and anti-roll coefficients, were neglected completely given the stopping distance-only character of our investigation.

All of the tire data was gathered by Calspan on its indoor tire testing setup, and can be found in Reference [7]. Plots were made from these data and are included in Appendix D. Since Calspan provided the tire model used at APL, the data were presented in terms of the various coefficients and thresholds required by the computer program.

The mechanical characteristics of the test cars which were measured at HSRI for this study included the following:

1) The center-of-gravity (c.g.) location of the entire vehicle. This was found by using a large swing which had been fabricated at HSRI for this purpose. The longitudinal position was found by balancing the vehicle, and the vertical location was found by pitching the swing, measuring the static angle and resultant torque, and calculating the c.g. position based on these measurements, the vehicle weight, and the known properties of the swing. The c.g. location

of the sprung mass was then estimated for the input to the computer program.

2) The pitch moment of inertia of the entire vehicle. The same swing apparatus used to locate the c.g. was also used to establish the pitch inertia. This was done by allowing the vehicle to oscillate, pendulum fashion, obtaining a measure of the period of the oscillations. The pitch inertia was then calculated from the measured period, the c.g. location, the weight of the vehicle, and the swing properties. The sprung mass pitch moment of inertia was then separated from this figure.

3) The suspension spring rates and coulomb friction levels. The suspension parameter measurement setup at HSRI was utilized for these measurements. The sprung mass of the vehicle was fixed to ground, while the two wheels of either the front or rear suspension were moved together vertically. Force versus deflection curves were plotted to enable the observation of the spring rates and friction levels.

4) The auxiliary roll stiffnesses due to anti-sway bars. The same setup needed to determine the spring rates was used again, although this time right- and left-side wheels were moved in opposite directions from their equilibrium positions. The total roll stiffness was calculated from the force and displacement histories, and the contribution from the springs was subtracted to produce net auxiliary roll stiffness values.

5) Vehicle weight during testing. Immediately before the physical tests for each car were started at BAPG, the load at each axle was measured. The results were used to establish the sprung mass value and its

c.g. location for the loaded and unloaded car, with instrumentation and driver.

6) The brake proportioning. Each vehicle employed a front/rear brake line circuit split, which made it possible to disable the front brakes and to conduct tests characterizing stopping distance as a function of "front-directed" pedal force. Similar data were taken with the rear brakes disabled, which then gave us the proportioning as a function of pedal force.

7) Antilock parameters. All of the parameter values needed to simulate the antilock system on the Nova were initially calculated from strip chart traces which were provided by the Kelsey-Hayes Company. We did not have a great deal of confidence in some of the estimates. A refined set of parameters was chosen empirically so as to obtain a close correlation between measured and simulated stopping distances. In the physical testing activty, it was seen that the Nova performed similarly to the Bobcat during straight-line braking on high coefficient surfaces. Thus the antilock parameter values were adjusted until the simulated Nova performance was close to the simulated Bobcat performance, while at the same time the contrast between the traces provided by Kelsey-Hayes and the simulated traces was minimized. The antilock simulations are discussed further in Section 3.5.1.

After these measurements had been made for each vehicle, it was seen that the inertial and suspension parameter values closely confirmed those values which can be derived from empirical relations, such as those given in Reference [8].

3.4.2 Computed Sensitivities of Vehicle Braking Performance. In any experimental test program, certain variations in the testing conditions are inevitable. A set of computer simulations was implemented to determine the effect of such variations on stopping distance performance. A matrix of conditions was formulated, as listed in Table 3.16, together with the results of the simulations. For each case, the basic performance is defined by the shortest stopping distance which was found for the conventionally-braked vehicles by a series of simulation runs in which the brake line pressure was increased step-by-step until axle lockup occurred. In all of the simulations involving the conventionally-braked vehicles, the front axle locked before the rear.

The simulations involved two surfaces representing high and low coefficient friction levels. These are indicated in the table by the two skid numbers 30 and 81. The "baseline" test conditions for each vehicle were: loading to achieve the gross vehicle weight (GVW), an initial speed of 40 mph, a steer angle such that a steady-state lateral acceleration of 0.2 g's was reached immediately before the brake application, and a rise time of 0.2 seconds for the brake line pressure. The minimum stopping distance for each vehicle serves as a reference against which the stopping distance occurring under different simulated conditions are compared on a percentage basis. The sign convention used in the table assigns a positive value when the performance is improved (shorter stopping distance) with respect to the reference.

The first three lines after the baseline condition contrast the performance found under the different loadings and during straight-line braking. It can be seen that the four conventional cars performed significantly better when unloaded. This can be understood by noting that:

1) the center of gravity of the vehicle is more forward in the unloaded state, and

Table 3.16. Simulated Sensitivities of Vehicle Braking Performance.

Baseline Condition: V = 40 mph, GVW, A_y = .2g
Rise Time = .2s.

	Monte Carlo		Ford LTD		Pacer		Bobcat		Nova with Antilock	
Skid Number	30	81	30	81	30	81	30	81	30	81
Baseline (Stopping Distance in ft.)	282	106	250	96	318	102	285	97	264	97
Straight (A_y=0)	+7%	+4%	-1%	+4%	-2%	-1%	+36%	+2%	+5%	+5%
Lightly Loaded	+12%	+16%	+9%	+15%	+8%	+9%	+13%	+11%	-1%	-1%
Straight & Lightly Loaded	+18%	+16%	+6%	+20%	+12%	+12%	+20%	+13%	+8%	+2%
V=42 mph (A_y=.22g)	0%	+3%	-6%	0%	+1%	0%	0%	0%	-5%	+1%
V=38 mph (A_y=.18g)	0%	0%	0%	0%	-1%	-1%	+4%	-1%	+4%	0%
110% Peak Friction	+7%	+7%	+4%	+7%	+13%	+6%	+10%	+9%	+17%	+10%
90% Peak Friction	-10%	-6%	-16%	-9%	-9%	-10%	-6%	-11%	-22%	-9%
+20% Torque Imbalance	+5%	+1%	-4%	0%	+1%	-4%	+1%	0%	-4%	-2%
-20% Torque Imbalance	+3%	-6%	-4%	+2%	+9%	+5%	+7%	+9%	-1%	-1%
Rise Time=.1 sec.	+1%	+4%	+1%	+3%	+1%	+3%	+1%	0%	+6%	-1%
Rise Time=.3 sec.	-1%	-3%	-2%	-3%	-1%	-3%	-1%	-3%	+3%	+4%

2) the front axle was always found to lock before the rear, indicating that the front/rear proportioning was optimum for either a surface having a high friction coefficient or a more forward c.g. location.

The contrast between the performance achieved under the in-a-turn and straight-line conditions is variable, although the tendency is towards better performance during the straight-line maneuver. We can also see that the effects of being lightly loaded and of braking in a straight line are not simply additive, but rather that, when combined, they form an independent third category. In all three categories, the Nova showed markedly less sensitivity than the other cars.

The effect of a variation in the initial vehicle speed is twofold: first, the expected stopping distance (in a straight-line maneuver) increases in proportion to the square of the initial velocity, and second, the lateral acceleration resulting in a constant-radius turn (or in the simulations, a constant steer angle) is also proportional to the velocity squares. Simulations were run with the initial velocity varied by ± 2 mph to quantify the performance sensitivity to initial velocity, and the results are shown in the table. The calculated stopping distances were adjusted in each case by the correction factor,

$$K = \frac{V_0^2}{V^2}$$

to eliminate the first of the two speed effects. In general, we see that the effect of the variation in the initial lateral acceleration condition is indeed small.

The friction that is available at the tire-road interface is addressed in FMVSS 105-75 only by the skid number of the track surface. Thus the peak friction coefficient of a given tire may differ from surface to surface or even from day to day with the same surface, while at all times satisfying the 105-75 condition.

During straight-line braking, if the rise time is neglected, we would expect that a 10% change in the available peak friction would cause a 10% change in the stopping distance. The results of a 10% change which are shown in the table indicate that this simple assumption does not hold when braking in a turn is considered. The deviations from the 10% figure show no generalized trends in behavior for the four conventional cars. The Nova behaved exactly as expected on the high coefficient surface, but was about twice as sensitive on the low coefficient surface as would be expected.

The next condition shown in the table is a 20% brake torque imbalance (that is, the brake torque/line pressure gain on one side of the car was increased by 20%.) (A brake torque increase on the side loaded by the initial lateral acceleration in the turn is designated as the +20% condition.) An imbalance of this sort might be due to production tolerances or to differing work histories for right- and left-hand brake pads (or shoes) during braking-in-a-turn tests. Scanning the results, we see that the change in performance varies from -4% to +9%, and that no trends are evident, even in the polarity of the imbalance. The inconsistent trend of the torque imbalance sensitivity serves to further illustrate the complexity of the interactions between the parameters involved in braking-in-a-turn maneuvers.

The final sensitivity shown in Table 3.16 is that of the stopping distance involved in variation of the rise time in brake line pressure. If the vehicle deceleration is assumed to reach a constant value at the same time as the line pressure, the portion of the overall stopping distance due to the rise time is

$$\Delta S \cong \frac{1}{2} V_0 \, \Delta T$$

where ΔT is the rise time. From this relation, we would expect a change of 1% on the low coefficient surface and 3% on the high coefficient surface, for the rise times shown in the table. And we can see that this approximation is adequate for all of the vehicles except the Nova, in which a steady deceleration is never reached due to the antilock system.

Before concluding the discussion of the computer simulations, some observations regarding the validity of the results is in order. While examining the computer output for this study, certain discrepancies were noted between the test and simulation results. The main differences are the following:

1) The simulated stopping distances were longer than those actually measured. If the computer results for the high coefficient surface are extrapolated to include an initial velocity of 60 mph, it can be seen that none of the simulated vehicles meets the provisions of 105-75.

2) The front axle locked before the rear in all of the simulations involving non-antilock-equipped cars. However, rear axle lockup occurred frequently during the actual tests on the high coefficient surfaces.

3) The antilock cycling rate was much higher in the simulation than in the tests.

It is suspected that the major reasons for these differences are the following:

1) Inadequate tire data. The moments and forces occurring at the tire-road interface are calculated using traction coefficients which had been experimentally-determined on Calspan's TIRF machine [7]. The traction limits are all "corrected" by a coefficient,

$$K = \frac{SN_S}{SN_T}$$

where SN_S is the skid number of the surface to be simulated and SN_T is the skid number of the test surface on which the tire data were taken. Two problems stem from this procedure.

a) The Calspan test surface had a $SN_T = 85$, however, the measured traction limits were less than those measured with a mobile tire tester on dry asphalt and concrete (SN ~ 81) with similar tires (same model and manufacturer). All of the traction coefficients seem to be too low, by about 20%. The effect of this error is longer stopping distances and lower deceleration levels. Since the deceleration is reduced, so is the front/rear load transfer. Thus the simulated rear axle loadings are too high, the simulated front axle loadings are too low, and the front wheels have a greater than actual tendency to lock before the rear wheels.

b) Since only dry surface traction data was available, it was necessary to extrapolate these data, per the scheme of "skid number ratios" which is employed in the APL simulation, to obtain wet surface traction data for each tire. The extrapolation procedure does not shape the peak/slide relationship authentically, however. Thus the peak friction level employed in simulation of the low coefficient surfaces is too low. Because the Nova showed such a high sensitivity to the peak friction level on the low coefficient surface, all of the simulations involving the Nova yield long stopping distances.

Simulating the original equipment tires correctly would have required measuring their properties on the actual test surfaces with a mobile tire tester—an endeavor beyond the scope of this study. The tire properties might have been estimated better, however, by abandoning the concept of using TIRF data taken on the original equipment tires and using existing mobile tester data pertaining to tires of the same size and construction, and test surfaces employing representative materials (i.e., dry asphalt, wet jennite, etc.).

2) No suspension kinematics data. The kinematics of suspension linkages is represented in the model by anti-pitch and anti-roll coefficients. The measurement or even the estimation of these coefficients lay outside the scope of the study. However, the pitch transients observed in the simulated results may have been significantly influenced by the inclusion of accurate anti-pitch coefficients. It might be expected that if the transient motions were reduced by including the proper coefficients, greater brake torque levels might be achieved without lockup which would result in shorter stopping distances.

3) Estimated shock absorber characteristics. The actual shock absorber coefficients may have differed significantly from the estimated values. Since shock absorbers influence the character of transient responses, the incidence of wheel locking may be determined by the relative accuracy of shock absorber parameters.

4) A simplified antilock model, along with approximate parameter values. It is difficult to say what inaccuracy in stopping distance prediction is due to the antilock model and what is due to inadequacies in the tire representation, as the two are intimately related. Nevertheless, it was necessary to adjust a few parameters in the antilock model to bring the

cycling behavior to the expected form and frequency.

5) "Noise" inherent in the APL hybrid computer. Because many of the differential equations and vehicle-component functions were programmed on an analog computer, noise was introduced into the simulations, both from the analog electronic circuits and from the ADC's (analog-to-digital converters) and DAC's (digital-to-analog converters). Normally, the errors which result from these influences are completely negligible. However, during the antilock simulations, an unwanted oscillation in the wheel spin accelerations was seen to disrupt the antilock logic. As a result, it was seen that the brake release and re-apply conditions were being established in the logic elements somewhat prematurely due to random noise in the program.

6) The simulation program encountered computational instabilities when vehicle speed reached levels much less than 10 mph. Accordingly, each simulation was stopped when the vehicle speed reached 10 mph, and the average deceleration from 35 mph to 10 mph was used to extrapolate the entire stopping distance. Errors due to this approximation should have been negligible for the conventional cars, and small for the Nova.

Overall, simulations served the purpose of showing relative sensitivities, but should not be construed to represent absolute levels of performance. Also, it should be noted that the problems numbered (2) through (6) were quite small, especially in comparison to item (1). Computer results which covered other areas of performance than stopping distance, but were still felt to be of general interest, are included in Appendix D.

3.5 Consideration of Advanced Braking Systems

In this section a limited discussion of advanced braking systems will be presented, as a means to support the generality of recommendations regarding possible future rulemaking. The basic proposition of the study was that certain braking conditions may exist which need to be considered in an extended version of FMVSS 105-75. To the degree that such extensions might become promulgated as law, concern arises that future evolution of brake system technology would render the requirements unenforceable or even restrictive in a sense which discourages safety improvement. As a partial step to avoid such narrow-sightedness, the prominent technological advancement, antilock braking systems, is considered as a special interest. In Section 3.5.1, a simulation of the various basic antilock system arrangements is reported—primarily in terms of stopping distance performance, but also with brief reference to directional response features. In Section 3.5.2, the cost implications of various antilock system installations is examined—as supported by input from various manufacturing organizations.

3.5.1 Simulations of Advanced Braking Concepts. The simulation activity involving the advanced braking concepts was essentially a straightforward extension of the simulation work reported in the previous section, using the same computer program, the Nova vehicle parameters, and the same antilock model, developed in Appendix F.

Five variations of the Nova were considered, viz.:

1) no antilock systems,

2) one antilock module, controlling both brakes on the rear axle,

3) two antilock modules, each controlling the brakes on one axle,

4) three antilock modules, one controlling both brakes on the rear axle and the other two each controlling one front wheel brake,

5) four antilock modules, each independently controlling one wheel brake.

The conditions simulated covered a low friction (skid number of 30) surface and a high friction surface (skid number of 81), straight-line and in-a-turn braking, and lightly-laden and GVW loadings. In addition, the effects of variation in the peak friction coefficient were found for the case of the heavily-laden vehicle braking in a turn. In all simulations, the rise times of the brake line pressure was 0.2 second, and the initial speed was 40 mph.

The braking performance of the basic Nova without any antilock system was characterized by the front axle always locking before the rear. (See Section 3.4.2 for a discussion of the validity of this simulated behavior.) Thus, in antilock configuration number (2) front axle lockup was always experienced before the rear wheels were braked to a sufficient level to start the cycling of the antilock system. Because axle lockup violates one of the stipulations assumed throughout this study, the results involving one module are not included in the results.

The stopping distance performances of the four remaining vehicle configurations are shown in Figures 3.17 and 3.18. In the straight-line maneuvers, the performance of the three antilock-equipped vehicles should be nominally the same due to the assumed symmetry of the vehicles and the commonality of the antilock modules. We do, however, see up to a 4% difference in the simulated stopping distances. This is due to certain computational "noise" in the hybrid computer, a problem discussed briefly in Section 3.4.2. Thus the straight-line braking simulations serve to define the repeatability of the simulations of the antilock-equipped vehicles.

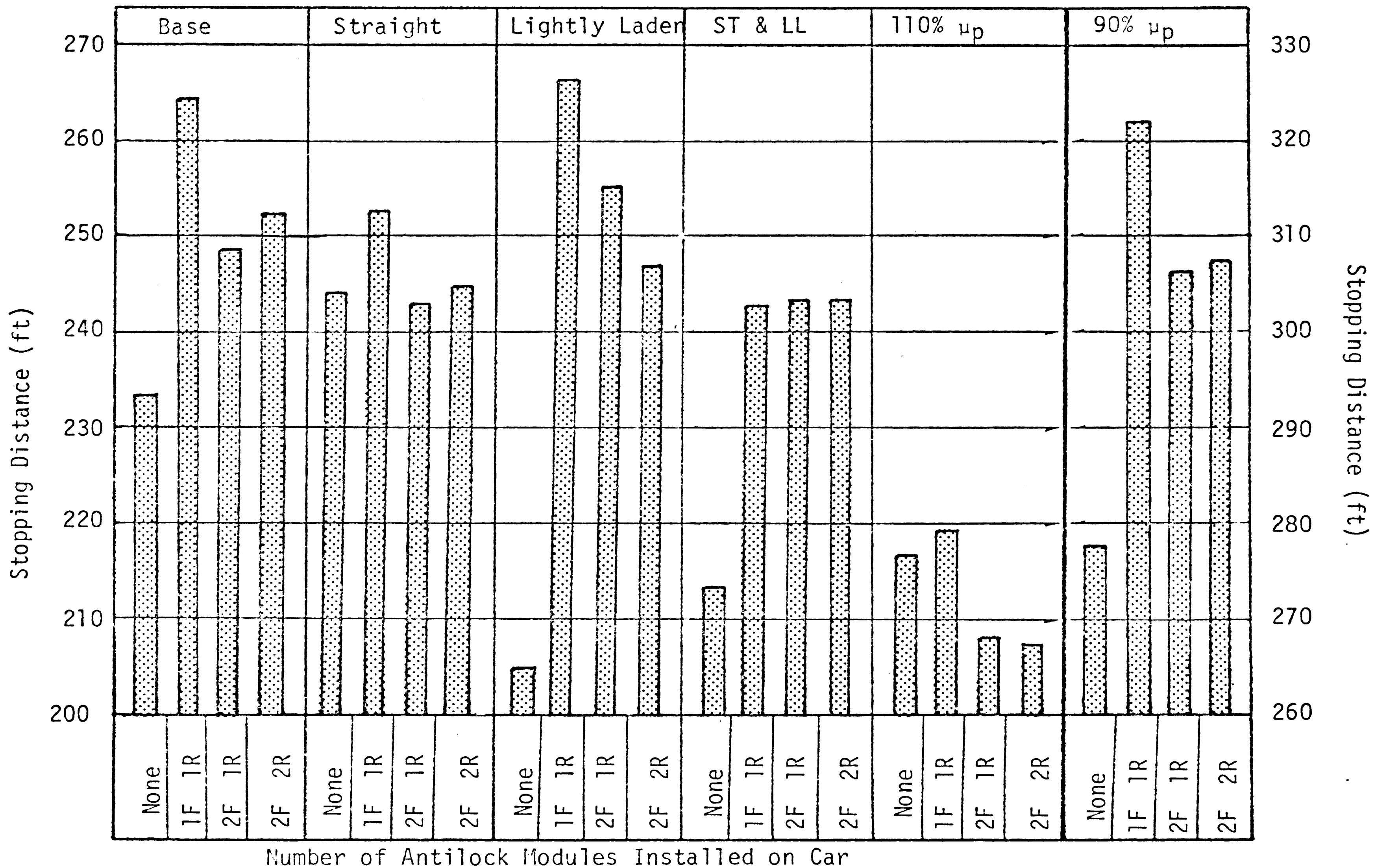


Figure 3.17. Simulated stopping distance performance of various antilock-equipped cars on a low friction surface.

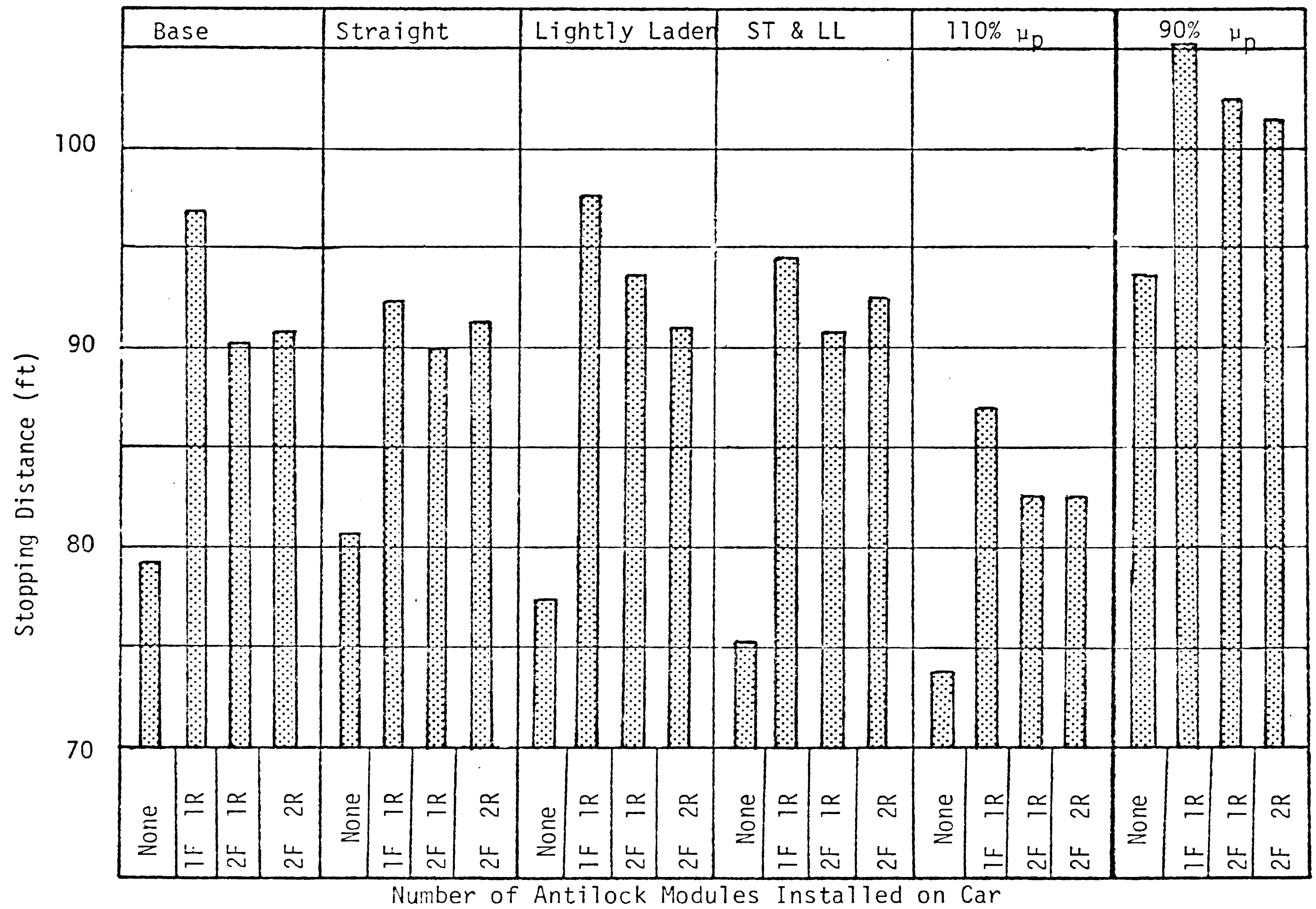


Figure 3.18. Simulated stopping distance performance of various antilock-equipped cars on a high friction surface.

An examination of the information presented in Figures 3.17 and 3.18 leads us to make the following conclusions:

1) The differences in performance between the three-module antilock system and the four-module system are negligible.

2) The three- and four-module systems lead to slightly better performance during braking-in-a-turn maneuvers, when compared with the two-module system. The differences are consistently about 5%.

3) In nearly all cases, the vehicle without any antilock system provided shorter stopping distances those with the antilock systems; the differences ranged from -4% to +20%, and averaged a +10% improvement.

It is generally held that the most important aspect of braking-in-a-turn performance is the ability of the vehicle to hold its path curvature without also suffering large sideslip. Although directional performance was not to be directly addressed during this research, the curvature-holding ability of the simulated vehicles was evaluated and is presented below.

A directional measure of the braking-in-a-turn response is presented in a cross-plot of normalized path curvature [2] and deceleration in Figure 3.19. To calculate the average path curvature, $(R_0 1/R)_{ave}$, the simulated curvature was averaged over the first one second, while the average deceleration covers the time in which the vehicle speed was between 35 and 10 mph. Average curvature values greater than unity indicate that the radius of the turn decreased during the first second, while values less than unity indicate an increase in the radius of the turn, or a loss in the steering ability of the vehicle. Representative path trajectories illustrating these responses are plotted and included in Appendix D.

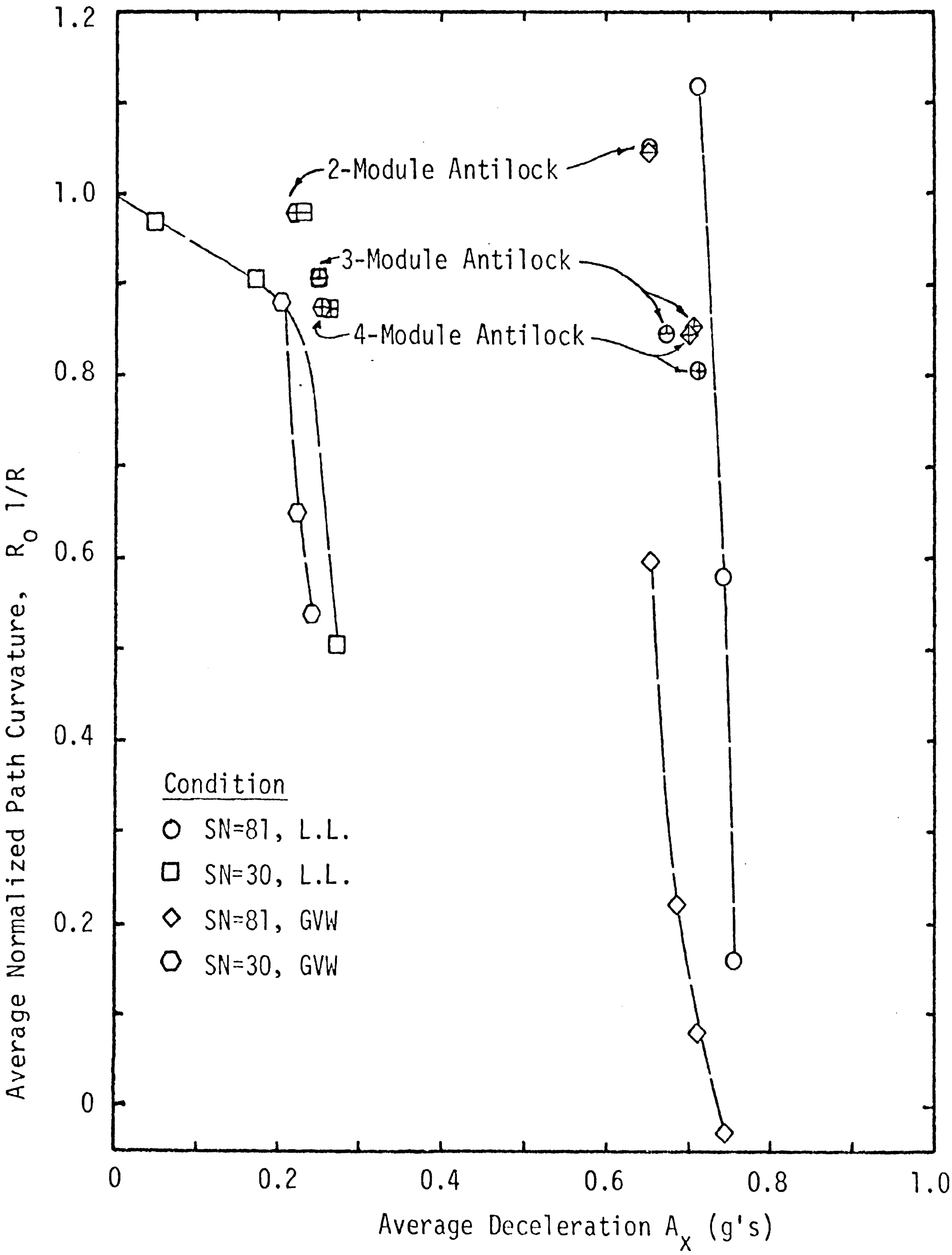


Figure 3.19. Simulated deceleration and curved-path-holding performance of Nova with and without antilock systems.

The limit braking performance levels for the conventionally-braked vehicle are indicated in Figure 3.19 by the open points, showing the largest decelerations. The differences between the average path curvatures of the vehicles equipped with different braking systems are dramatic for the limiting cases—as the anti-lock-equipped vehicles nearly retain their original curvature, while the non-antilock-equipped vehicle loses most of its path curvature, and in one case even reverses slightly. It is seen, however, that the conventionally- and antilock-braked vehicles exhibit comparable path curvature performance at sub-limit deceleration levels.

The trends shown in Figure 3.19 are briefly summarized below.

1) The differences in average path curvature shown by the three- and four-module antilock systems are negligible under all of the simulated conditions.

2) The three- and four-module systems consistently display less average path curvature than the two-module system. This is due in part to the ability of the more complicated systems to increase the braking forces produced by the tires on the side of the vehicle that is loaded by centrifugal force. Thus a right-to-left force imbalance exists, which produces a yaw moment, turning the vehicle out of the curve.

3) On the high friction surface, under the GVW loading condition, the three- and four-module systems were capable of higher deceleration levels than the two-module system at equivalent average path curvature levels. Further, the two-module system did slightly better than the vehicle with a conventional braking system. When lightly laden, the conventionally-brake vehicle was capable of the best stopping distance performance. Directional control, however,

was clearly retained only by the antilock-equipped vehicles, when maximum deceleration levels were reached.

4) On the low friction surface, the differences between the deceleration/path curvature capabilities of the antilock-equipped vehicles were small, and all of them performed markedly better than the non-antilock-equipped vehicle, retaining the "steerability" of the vehicle even at the maximum braking level condition.

3.5.2 Advanced Braking Systems Cost Estimates. In this section, the results of a small scale economics analysis of the costs associated with advanced braking system elements are presented. Two basic concepts in brake system advancement are considered: antilock controller systems and load-sensitive proportioning valves. In the foregoing section, a computerized simulation was used to examine the relative performance advantages which accrue from antilock systems of various configurations. Load-sensitive proportioning was not included as an element of those simulations, of course, since such devices merely serve to redistribute brake torque to account for static loading of the vehicle—a function whose significance is represented directly in the quasi-static calculations of Section 3.2.

In the development of estimates for the consumer costs of the two advanced concepts, the following procedure was utilized:*

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*The original proposal called for a range of technical experts to be interviewed for the determination of the cost data. The procedure, called the Delphi interview technique, was to have been a series of interviews, each following a very precise interview pattern. It was soon discovered, however, that the costing procedures and investment criteria vary so widely between the four automobile firms that it would not be beneficial to follow the Delphi procedures on a basis that would include all four U.S. passenger auto firms. Rather, the most appropriate procedure would be to concentrate attention on the smallest of the four firms--American Motors. The logic of this procedure was that the costs that were developed would represent the high side of the cost curve because of the economics of scale. The analyst could be confident that all firms could produce systems at the projected cost figures.

The selling price of brake system components from the OEM brake suppliers was determined; this selling price was considered a component cost in determining the manufacturer's "mill door" cost (i.e., the manufacturer's actual cost as the product leaves his shipping point) for the brake system.

The auto manufacturer's mill door cost for the additional brake system element was determined, then various burdens were added to the cost, and finally the dealer's markup was included. The resulting figure was considered as the list price equivalent for the additional element or sub-system.

In the case of antilock brake systems, the cost analysis was applied to four system configurations, viz.,

one modulator, two-wheel system (similar to the Kelsey-Hayes/Ford Sure-Trac system)

two modulator system - both two- and four-wheel sensor arrangements

three modulator, four-wheel system (similar to the Bendix/Chrysler system)

four modulator, four-wheel system

Antilock Braking System

It was discovered at an early stage of this investigation that a fairly precise definition of candidate systems was necessary before definitive cost evaluations could be made. Because of this, it was decided to develop cost data on the basis of two antilock brake control systems that have been developed and offered to the public on a limited basis. The first of these was the Kelsey-Hayes package, offered by Ford on its Continental automobiles. The system employed a single sensor at the differential, with both rear wheel braking units being operated by a single hydraulic modulator. The second system was the Bendix package offered as optional equipment in the luxury class Chrysler products. This system had three sensors, one on each of the front wheels, and a third sensor on the drive shaft for control of the rear wheels. Each of the sensors had one modulator. The Bendix system was vacuum-actuated.

OEM Delivery Price. Interviews indicated that there was a general relationship between the OEM delivery prices and the number of modulators utilized in a system. This relationship was as follows:

Number of Modulators	Price Multiple (Range)
one modulator	base price
two modulators	1.5-2.0 times base price
three modulators	2.0-2.5 times base price
four modulators	2.5-3.0 times base price

Table 3.7 shows the OEM sale price range for the typical antilock controllers as a function of OEM annual sales volume of an identical system. The numbers were developed from a variety of price estimates obtained during discussions with individuals in OEM firms. The range of prices at a specific volume derived from a range of potential complexities of the system.*

Analysis of the relationships between price and production volume indicated that the volume-induced cost reductions are generally consistent with that found for most manufactured products: as the cumulative volume is doubled, the average cost (selling price) for all units is reduced 15-25 percent. In the case of an auto supplier, the reduction is about 16 percent. Table 3.7 translates a 16 percent "learning curve" reduction into the OEM selling price ranges for the antiskid braking systems at annual production volumes from 10 thousand units to 700 thousand units. The numbers indicate that with production runs of 10 thousand units the OEM selling price to the auto manufacturer for a one-modulator system

*As an example of the system complexity variances that could occur, consider the differences in the modulator construction if it were possible to have any type of fluid in the modulator. Certain modulators operate with two incompatible fluids: brake and power steering. If a fluid were developed that could be operated in both systems, the modulator complexity would be reduced, resulting in a cheaper price.

Table 3.17. OEM Sales Price Estimate for Antilock Brake Control Systems at Varying Levels of Production.

	Estimated OEM Selling Price Range			
Annual Production Value	One Modulator	Two Modulators	Three Modulators	Four Modulators
10K	$80-100	$120-200	$160-250	$200-300
50K	52-65	75-125	100-160	130-200
150K	37-50	55-100	75-125	90-150
350K	30-37	45-75	60-90	75-110
700K	25-32	37-65	50-80	60-100

Source: Estimate

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will be in the $80-100 range. The price would drop to $25-32 if the production runs were in the 700 thousand/year range. Similarly, very large reductions in cost as a function of volume are projected for the other candidate systems.

Auto Manufacturer's Add-On Costs (including dealer markup). The auto manufacturer is the assembler of the brake components. The cost of any specific component is a function of the cost of materials plus the assembly time required to incorporate the parts into the vehicle. To these two costs are added the corporate overhead and profit. Table 3.8 shows the relationships for the major cost categories as a percentage of the manufacturing costs. The percentages were based on production volumes of 350 thousand units/ year [11, 12].

Variable costs of production of automotive components are those incremental costs associated with that component. The major categorical contributions to variable costs are direct labor, direct materials, and variable burden. Other minor contributors to variable cost, such as setup costs, are included in the variable burden rate.

Table 3.18. Relationships Between Manufacturing Cost and Consumer Costs for Automobile Components.

Category	Percent
Manufacturing Costs	
Variable Costs	80-81
Fixed Costs	19-20
Total Manufacturing Costs	100
Tooling Costs	3
Other Costs and Profit	10
Dealer Wholesale	113
Dealer Markup	20-30
Customer Cost	133-143

[11] DOT Contracts HS-5-01153 and [12] HS-5-01081 performed by Pioneer Engineering Corporation.

The portion of total manufacturing costs, known as fixed cost, is the accumulation of costs incurred in the manufacturing of a product that does not vary regardless of the volume. Major categorical contributors to fixed cost are indirect labor, indirect materials, and fixed burden.

Tooling costs are apportionments for special tooling to manufacture a component over the life production volume of that component.

Other costs plus profit included such items as engineering and warranty costs, selling and administrative costs, corporate burden and taxes (excluding factory burden and taxes), corporate depreciation and maintenance (excluding factory depreciation and maintenance), and other corporate costs and profits.

Dealer markup is the summation of all costs incurred in the operation of a dealership and the dealer's profit.

The percentages shown in Table 3.8 were developed by Pioneer Engineering Corporation on contracts for the Department of Transportation. In these studies vehicles were totally disassembled so that production engineers could estimate the production costs of each part (assuming an annual production run of 350,000 vehicles). The estimates were totaled to equal the known consumer cost of the vehicle.

Table 3.9 shows the estimates developed for a variety of antilock modulator concepts at assumed production runs of 10 thousand/year and at 350 thousand/year. The estimates were reviewed by a representative of one of the smaller U.S. auto manufacturers; he agreed with the range with one exception. He felt the estimates for units with a projected volume of 350 thousand units/year were unreasonably low. He indicated, however, his firm had not developed estimates for 350 thousand production runs and his criticism was totally subjective.*

In Table 3.9, the Bendix/Chrysler system (taken as typical of a three-modulator installation) and the Kelsey-Hayes/Ford system (taken as typical of the one-modulator system) were evaluated. In both instances, it was assumed that current states of the art were being used in the technology of electronic microprocessors and wheel or drive shaft rotation sensors.

Finally, Table 3.10 shows the final consumer price range that might be expected for each of the modulator concepts at various production levels. This table clearly shows the price range that can exist within each of the generic antiskid systems. One could reasonably expect, for example, that the one-modulator system could range in price from $45 to $215, depending on the production volume and the system complexity. The variable most strongly affecting price, however, is production volume.

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*Personal interview (confidential) with manager of brake systems, U.S. auto firm, February 1978.

Table 3.19. Estimates of Cost Buildup for a Variety of Antilock Brake Control Systems at Different Production Levels.

Type of System	1 Module	1 Module	3 Module	3 Module	1 Module	1 Module	3 Module	3 Module
Production Volume (units/yr)	350K	350K	350K	350K	10K	10K	10K	10K
Estimate Range	Low	High	Low	High	Low	High	Low	High
Variable Costs								
OEM Supplier Parts Cost	30.00	50.00	89.00	105.00	80.00	100.00	179.00	213.00
Direct Labor[1]	2.07[2]	2.07[2]	2.76[3]	2.76[3]	2.76[3]	2.76[3]	3.45[4]	3.45[4]
Variable Burden[5]	4.16	4.16	5.52	5.52	5.52	5.52	6.90	6.90
Total Variable Costs	36.23	56.23	97.38	113.28	88.28	108.28	189.35	223.35
Fixed Costs								
Non-Variable Burden[6]	5.18	5.18	6.90	6.90	6.90	6.90	8.63	8.63
Total Fixed Costs	5.18	5.18	6.90	6.90	6.90	6.90	8.63	8.63
Manufacturing Costs	41.41	61.41	104.18	120.18	95.18	115.18	197.98	231.98
Tooling Costs[7]	2.85[8]	2.85[8]	2.85[8]	2.85[8]	50.00[9]	50.00[9]	50.00[9]	50.00[9]
Other Direct Cost and Profit (10 percent of mfg. costs)	4.14	6.14	10.41	12.02	9.52	11.52	19.80	23.20
Dealer Markup (20 to 30 percent of mfg. costs)	8.28[10]	18.42[11]	20.82[10]	36.06[11]	19.02[10]	34.56[11]	39.60[10]	69.60[11]
Consumer Cost	56.68	88.82	138.26	171.11	173.72	211.26	307.38	374.78

[1]Based on average labor costs in 1977 for U.S. auto firms of $8.28/hr (13.8¢/min.)
[2]Based on installation time requirement of 15 person minutes/vehicle
[3]Based on installation time requirement of 20 person minutes/vehicle
[4]Based on installation time requirement of 25 person minutes/vehicle
[5]Estimated at 200 percent of direct labor (Source: a U.S. auto manufacturer)
[6]Estimated at 250 percent of direct labor (Source: a U.S. auto manufacturer)
[7]Based on $1.5 million for special tooling and $1.5 million for special equipment
[8]3-year amortization
[9]6-year amortization
[10]20 percent of mfg. costs
[11]30 percent of mfg. costs

Table 3.20. Estimates of Consumer Costs for Various Antilock Brake Systems at Different Production Levels.

	Annual Production Level (Thousands)			
Type	10	100	350	700
One Modulator	$170-215	$ 80-125	$ 55-90	$ 45-75
Two Modulator	235-300	130-180	95-135	80-115
Three Modulator	305-375	180-230	135-175	115-150
Four Modulator	375-450	230-280	175-215	150-185

Source: HSRI estimate

Load-Sensitive Proportioning System

The OEM manufacturing costs for a load-sensitive proportioning valve system would have the same "learning curve" cost reductions as that described in the preceding section for antilock braking systems. Thus, it would be expected that each time the cumulative volume was doubled, the average cost for all units produced would be reduced about 16 percent.

OEM Delivery Price. Responses to field inquiry indicated that the OEM selling price for a linkless load-sensitive proportioning valve with metering and differential pressure functions would be $18-19, in 350 thousand annual volumes. Using this price as a reference point, Table 3.11 reflects the price estimate for the system under other annual production volumes. Also shown in the table is the price for the currently-used three-way valves that would be replaced by the linkless, load-control system. The final column in the table shows the net added cost that the auto manufacturer would encounter by switching from the present system to the load-sensitive proportioning system.

Auto Manufacturer Add-On Costs (including dealer markup). The auto manufacturers would use the same technique of adding on costs for the load-sensitive proportioning valve as was described in the

Table 3.21. Estimate of OEM Sale Price for Load-Control Systems at Various Production Volumes.

Annual Volume (Thousand)	Load-Control Proportional System	Average Cost Presently-Used System (Replaced)	Net Added Cost
10	$48-55	$10-14	$38-41
100	25-28	6-7	19-21
350	18-20	4-5	14-15
700	16-18	3.50-4.50	12.50-13.50

Source: Calculated

preceding section. The OEM parts will be considered part of the materials cost. Variable burden will be about 200 percent of fixed labor costs; non-variable burden will be about 250 percent of labor costs. Corporate general and administrative expense will be about 10 percent of all manufacturing costs, and dealer markup will be 20-30 percent of manufacturing costs.

Table 3.12 compares the estimated cost of the load-sensitive proportioning valve system with the valve arrangement that would be replaced. As seen, it is projected that the new system would cost $31.99 and would replace a system costing $12.32. The net increase would be $19.67.

The data in Table 3.12 were developed on the basis of an annual production volume of 350 thousand units. Table 3.13 shows the net consumer costs that might be expected with the addition of load-sensitive proportioning systems into a manufacturer's product line, under varying annual production volumes. Again, the data shows the significant effect that production volume has on cost, or selling price.

Table 3.22. Consumer Cost Estimate of Load-Control Proportioning System (350 Thousand Annual Production)

Item	Load-Control Proportioning System	Current 3-Way Valve (To Be Replaced)	Net Cost
Variable Costs			
OEM Supplier Parts Cost	18.50	5.00	
Direct Labor	.41[1]	.69[2]	
Variable Burden[3]	.82	1.38	
	19.73	7.07	12.66
Fixed Cost			
Non-Variable Burden[4]	1.03	1.73	
Total Fixed Cost	1.03	1.73	(.70)
Manufacturing Costs	20.76	8.80	11.96
Tooling Costs	3.00[5]	-0-	3.00
Other Direct Cost & Profit (10% of Mfg. Cost)	2.00	.88	1.12
Dealer Markup (20 to 30 Percent of Mfg. Cost)	6.23[6]	2.64[6]	3.59
Consumer Cost	31.99	12.32	19.67

Source: Calculated

[1]Assume 3 minutes installation time @ 13.8¢/minute

[2]Assume 5 minutes installation time

[3]Assumed at 200 percent of direct labor

[4]Assumed at 250 percent of direct labor

[5]Based on $1.5 million for special tooling and $1.5 million for special equipment

[6]Based on 30 percent of manufacturing costs

Table 3.23. Estimates of Consumer Costs for Load-Sensitive Control Systems at Various Levels of Production.

Annual Production Volume (Thousands)	Load-Sensitive Control System	Current System (To Be Replaced)	Net Cost
10	$95-100	$37-40	$55-63
100	45-50	17-20	25-33
350	31-33	12-13	18-21
700	25-28	10-12	13-18

Source: Calculated

4.0 DISCUSSION OF CANDIDATE MODIFICATIONS IN FMVSS 105-75

The underlying interest of this research has been to determine those additional braking performance requirements which would meaningfully augment Federal Standard FMVSS 105-75. As has been mentioned, the candidate augmentations were to be limited to measures of performance concerning stopping capability, per se, and were not to include measures of directional response during braking. Candidate modifications to the standard will be discussed in this section in three main topic areas, viz., low coefficient braking, braking on split friction surfaces, and braking in a turn.

4.1 Low Coefficient Braking

The most basic treatment of the mechanics of braking for any road vehicle will reveal that, if wheel lockup is to be avoided, surface friction level and brake torque proportioning will play first-order roles in determining braking efficiency. For simple, constant proportioning brake systems, it is classically illustrated that only a single surface friction level exists at which the vehicle is ideally proportioned. For higher or lower friction levels than the design, or ideal, value, we find the simple vehicle to have a reduced efficiency. If a vehicle were to be designed with the ideal occurring at a high friction value, we would expect a reduced efficiency, or friction utilization, on low friction surfaces. Thus, the issue with a performance standard which specifies only a single test surface friction level is that braking efficiency can become significantly compromised for cases of other surfaces which are removed in friction level from the single specified condition.

For the most part, however, vehicles being sold in the United States since promulgation of FMVSS 105-75 incorporate some sort of proportioning valve such that efficiency levels achieved over a range of surface friction levels are significantly enhanced.

Furthermore, such brake systems can yield two ideal or optimum points over the friction range, with little efficiency loss in between.

Despite this state of affairs, namely, that most contemporary vehicles incorporate a non-constant proportioning mechanism which provides for efficient braking over a broad friction range, it can nevertheless be argued that a test at the low friction condition may be in order if the mix of constant and variably-proportioned vehicles in the population are to be meaningfully regulated.

Concerning methodological issues, it has long been recognized that the achievement of a stable, low friction test condition is a difficult assignment. In this study, confirmation of this point was obtained in the measurements of peak friction level on high and low friction surfaces. As shown in Figure 4.1, it was found that week-to-week variation in peak friction level involved standard deviations on the low friction (wet) surface which were twice as high as those found in the data for the high friction (dry) surface. Insofar as the two repeat data samples for each friction measure tend to follow one another rather well, we might hypothesize that the deviations do not derive from the spatial non-uniformity of the water film on the test surface as much as from the variability in the nominal depth of water film, and perhaps other factors, from week to week.

Vehicle stopping data, however, tend to suggest problems of spatial non-uniformity by the indication of greater run-to-run variability on a wetted, low coefficient surface than on dry. Shown below, for example, are averages of the percentage difference in the two shortest stopping distances measured on all of the cars tested in this study at Chrysler and Bendix Proving Grounds. These data show that wet surface tests exhibit 2.5 times the tolerance in stopping distance repeats as is found in measures taken on a dry surface.

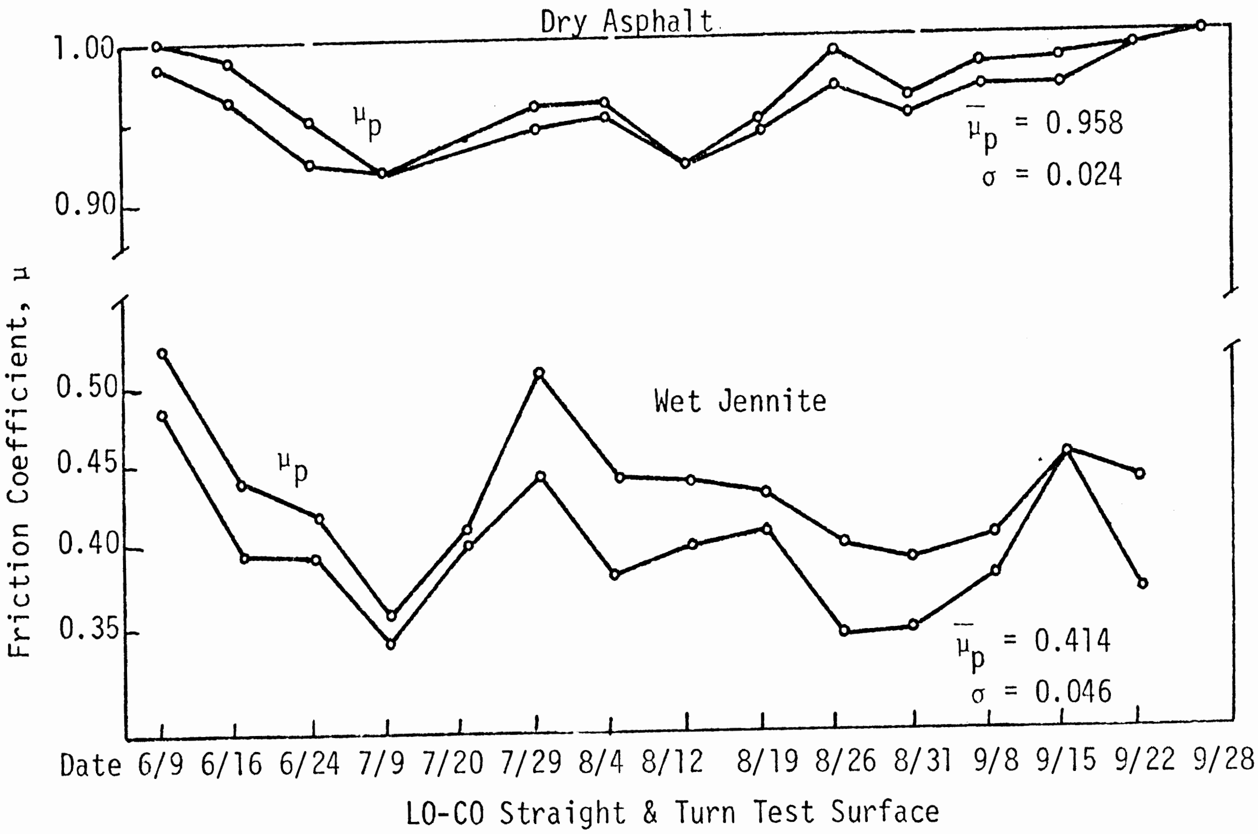


Figure 4.1. Measured peak friction coefficients vs. date for dry asphalt and wet jennite pavements.

Table 4.1. Average Percentage Differences Between the Two Shortest Stopping Distance Repeats Obtained in Straight-Line Braking Tests.

	Chrysler		Bendix	
	Dry	Wet	Dry	Wet
First Effectiveness	-	-	2.2%	8.7%
Second Effectiveness	-	-	1.2	2.1
Third Effectiveness	3.3%	5.2%	0.9	3.0

Aside from the issue of data repeatability, of course, there always remains the problem of the absolute friction levels pertaining to the test surface being used. For limit braking tests in which wheel lockup constitutes a performance constraint, minimum stopping distances will be heavily influenced by the peak traction condition. On dry surfaces, the μ-slip curve typical of passenger car tires is rather flat, with little observable peaking (Fig. 4.2). In contrast, coated or polished surfaces will, when

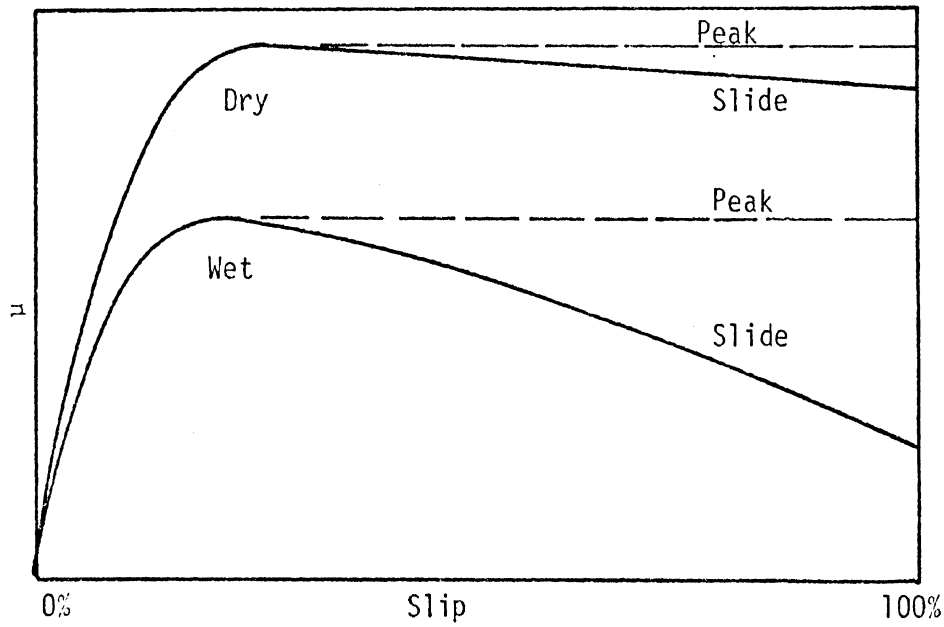


Figure 4.2. Typical μ-slip curve shapes on dry pavement, and wet, coated, or polished pavements.

wetted, typically exhibit a decidedly peaked response [9]. Thus, although a locked wheel friction value such as the ASTM skid number may serve to closely establish both the peak and slide range of the reference tire on dry surfaces, it is clear that a skid number obtained on a wetted low friction surface may only crudely pin down the peak traction potential of the test surface. To put it another way, the "peakiness" in the traction potential of various wet, coated pavements can vary significantly from one another in absolute level and can also vary with respect to the slide traction value.

Accordingly, we see the need for two methodological features which, although desirable for upgrading braking performance measurements on any type of surface, seem <u>especially</u> needed for testing on low coefficient, wet surfaces. The two features are:

1) the measurement of the peak traction potential of the test surface using a standard tire, and

2) the normalization of minimum stopping distance results using the measured peak traction levels to effect a braking efficiency numeric.

Braking efficiency determinations were made in this study and were shown, in Section 3.3, to hold promise for being considerably simplified in contrast to the methods developed under NHTSA sponsorship in Reference [1].

If stopping on a low friction surface was to be added as a requirement of FMVSS 105-75, it would appear warranted for both the LLV (lightly-loaded vehicle) and GVW (gross vehicle weight) loading conditions. It can be anticipated, of course, that for a given vehicle, either the forward or rearward translation of the vehicle mass center as results from loading will produce a single worst case behavior on a known friction condition. Vehicles whose mass center moves forward with loading, for example, will be most challenged during braking on a low friction surface when they are unloaded. Conversely, vehicles whose mass center moves rearward with loading will be most challenged on a low friction surface when they are loaded. Practically speaking, however, a federal requirement must cover test conditions which include those cases of service loading and pavement friction which are most challenging to each vehicle, regardless of its configuration.

An additional practical matter concerns the sequence of tests in any compliance test series which includes wet surface conditions. As in the test sequence of FMVSS 121, the wet surface tests should be placed to follow the effectiveness measurement on the high friction surfaces in each (first, second, and third) effectiveness series. This practice minimizes the water contamination of brakes and tires prevailing at the time of the dry test conditions.

If we turn to the issue of the likely benefits which would accrue from the specification of vehicle braking performance on low friction surfaces, we see a mixed basis for forecast in the data gathered in this study. For example, looking at braking efficiency data taken for the twelve-car sample at Chrysler Proving Grounds, Figure 4.3, we find that vehicles on the average registered efficiency levels on the low friction surface which were within 7 percent of the levels achieved on the high friction pavement. It is seen, however, that certain individual vehicles exhibited markedly lower efficiency levels on the lower friction surface. These data, gathered for vehicles in their lightly-loaded condition, contrast markedly with measurements made over the three series of effectiveness tests using five vehicles at the Bendix Proving Grounds. As shown in Figure 4.4, the data from the Bendix tests indicate generally higher efficiencies being accrued on lower friction surfaces. The apparent conflict in data sets taken at Chrysler and Bendix test sites seems to derive from differences in the low friction surfaces involved. In particular, it was found in the Reference [1] study that the ASTM tire (with which the traction data is obtained to compute braking efficiency) produces a significantly

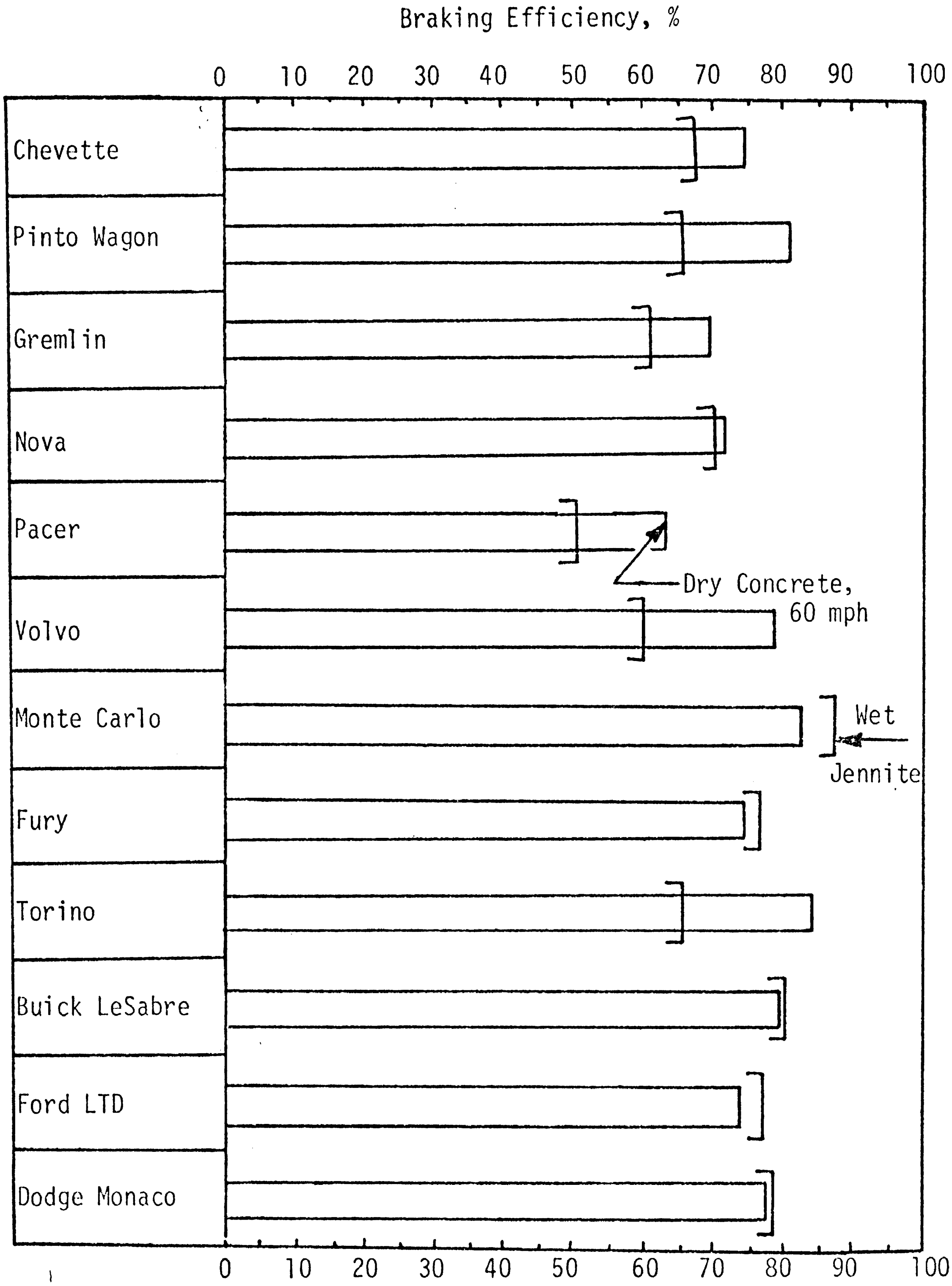
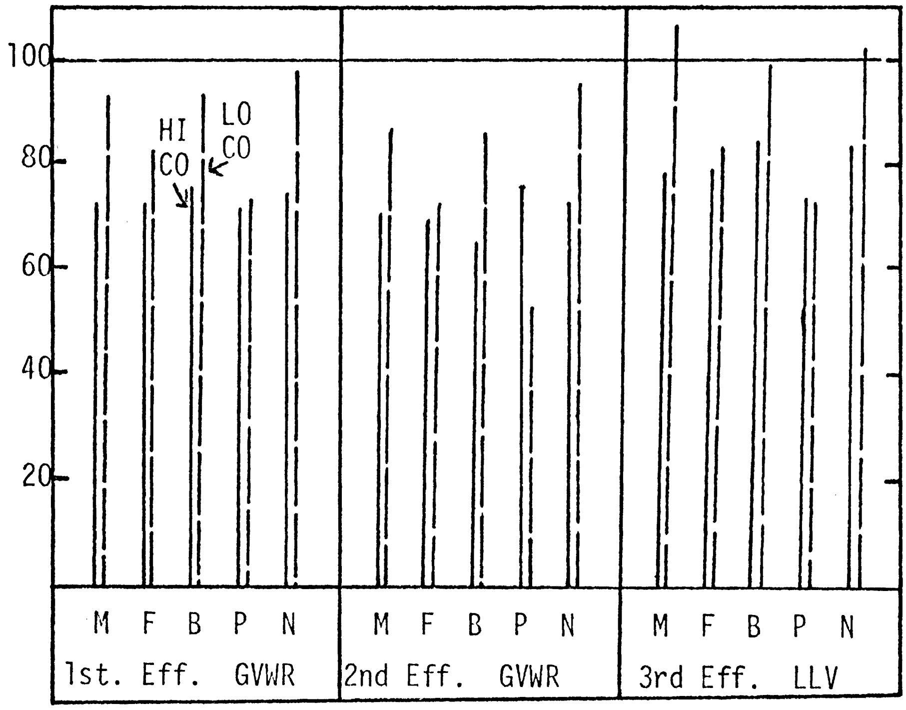


Figure 4.3. Braking efficiency per braking efficiency test technique.



M → '76 Monte Carlo

F → '76 Ford LTD

B → '77 Mercury Bobcat

P → '77 AMC Pacer

N → '76 Nova Antilock

Figure 4.4. Braking efficiencies on high coefficient and low coefficient surfaces, straight-line stops.

reduced traction performance on very low friction surfaces. Since the Bendix facility's low friction pavement registered peak traction levels in the 0.4 to 0.5 range compared to the 0.6 to 0.7 regime on the respective Chrysler surface, it seems likely that the ASTM tire produced a decidedly substandard traction performance in its characterization of the Bendix low friction pavement. Accordingly, we find the superior braking efficiencies on low friction surfaces in the data of Figure 4.4 to be anomalous. These data require another adjustment based upon the non-representativeness of the ASTM tire for such a reduced friction level condition. If we apply an adjustment of 20 percent to the traction measurements of the ASTM tire on the Bendix low friction surface (thereby effectively reducing braking efficiencies by 20 percent on that surface), we find that measurements made at Chrysler and Bendix sites more closely agree. The 20 percent adjustment, estimated from the data of Reference [1], produces the result that braking efficiencies on the low friction surface at Bendix average 12 percent lower than braking efficiencies measured on dry asphalt.

Moreover, we conclude from the review of straight-line braking performance data, that vehicles complying with FMVSS 105-75 yield on the order of 10 percent lower braking efficiencies on low friction surfaces than they do on the currently specified high friction surface. Of perhaps equal significance to considerations for rulemaking, however, braking efficiencies on low friction surfaces show approximately twice as wide a range of performance as braking efficiency levels measured on high friction surfaces.

4.2 Split Friction Surface Conditions

In this study consideration has been given to the examination of split friction surfaces as a generic condition under which braking performance may be evaluated. Because of the constraint of this study to issues relating to stopping distance performance, we have confined our examination to avoid directional response matters—recognizing, however, that directional, or yaw, perturbation has classically been seen as the primary problem involved with braking on a split friction surface [10, 11]. In considering the potential application of a split friction condition to federal braking regulations, certain conceptual problems can be identified. Most generally stated, it does not seem reasonable to hypothesize that the braking performance of four-wheeled, non-antilock-equipped vehicles on split friction surfaces would bear any generally significant relationship to the brake system design. This statement can be expanded upon in the following observations:

1) For the case of minimum stopping distance on a split friction surface <u>without wheel locking</u>, the vehicle will exhibit a braking performance identical to that which would be achieved on a homogeneous surface of friction level equal to the low-μ side of the split.

2) After the first wheel has locked, the order of successive wheel lockups depends both upon the vehicle's brake proportioning and load transfer kinematics, as well as to the friction level increment of the split. Since the friction increment across the split is one of the first-order determinants of the vehicle's further braking behavior, the need to identify a generally safety-significant split is paramount. Since we see no grounds for identifying a specific split friction condition as generally characterizing the U.S. road system, and since the regulation of performance on any specifically selected split surface will beget brake system designs which are peculiarly tuned to a specific order of wheel lockup on that surface, the general notion of regulating performance for split friction surfaces seems to be conceptually weak.

3) If we consider the vehicle which has achieved lockup of all four wheels on a split friction surface, we see that its braking performance is simply determined by the average of the two existing friction values, while its yaw response will be related to the friction increment, the vehicle's track width, wheelbase, and its yaw moment of inertia. Accordingly, the regulation of braking performance under this condition would not impact on brake system design.

4) The split friction condition can be seen as a singularly powerful discriminator of tire characteristics—a discrimination, however, that places an emphasis on the tire's ability to minimize the increment in traction between two surfaces. A "good" tire, from the viewpoint of minimizing vehicle performance sensitivity to a split, is one which tends to do poorly on high friction surfaces and very well on low friction surfaces. Moreover, the split friction condition tends to emphasize the importance of the tire's traction peculiarities—but not in a fashion which is compatible with the general desire for high traction quality on all road surfaces.

5) Regarding the significance of braking in a turn on split friction surfaces, the dilemma appears to be that one would design a completely different set of brake system and cornering/rolling properties if he presumed that the high friction side of the split would always be on the outside of the turn than if the high friction was presumed to be on the inside of the turn. Accordingly, it would again seem reasonable to hypothesize that no generally meaningful requirement could be placed upon braking-in-a-turn performance on a split. Aside from the earlier points generally pertaining to split friction

surfaces, it is not thought possible to influence vehicle braking performance in a turn on a split by any generally applicable design technique—other than the gross adjustment of track width, wheelbase, and yaw moment of inertia to retard the spinout divergency under four-wheel lock.

6) The specification of the braking efficiency of an antilock-equipped vehicle on a split friction surface does have technical merit, although the desirability of a purely antilock-relevant provision in a hydraulic brake system rule may be questionable. As is generally known, the interest of the antilock developer in split friction testing is one of assessing the ability of his control strategy to improve upon the braking capability afforded by the low friction condition alone. Thus the "select low" control strategy is less than optimum only insofar as it reduces the antilock-equipped vehicle to a braking performance no better than that provided by a non-antilock-equipped vehicle. Considering that note, however, it would be somewhat nontraditional for a regulation to be applied to a special vehicle configuration which tended to demand more of that system than is otherwise demandable of all other vehicles in the class.

Going beyond conceptual issues, it has been demonstrated in this study that a relatively straightforward test procedure is available for conduct of braking tests on a split friction surface. Further, the split condition that is achieved has a certain intuitive appeal as constituting a reasonable analog of a demanding wet weather condition involving polished and unpolished pavement sections. This condition, achieved by wetting the junction between coated and uncoated asphalt, is seen, then, as providing a practicable approach toward a split friction test methodology, notwithstanding conceptual issues of representativeness to the nation's split friction conditions, and meaningfulness of stopping measures obtained under such a test condition.

Given conceptual problems about the meaningfulness of stopping distance measurements on split coefficient surfaces, it may be pointless to comment further on the particular characteristics of the results of the split friction tests which were conducted in this study. Nevertheless, certain interesting observations were presented in the discussion of Section 3.3. A particularly curious aspect of the split friction test results was that the respective test vehicles yielded stopping distance performances which were markedly insensitive to the right/left location of the split when braking in a turn. Indeed, overall, results of the split friction tests showed a very undramatic spread in performance, in fact, comparing very closely with straight-line stopping distance values measured for each vehicle on a low friction homogeneous surface. The powerful "normalizing" influence which renders split friction test results so non-discriminating appears to be the complicated array of wheel lockup possibilities providing vehicle decelerations which are determined from a variety of wheel load/peak-slide friction combinations around the vehicle. This situation was shown rather clearly in the simulation results discussed in Section 3.2—illustrating that split friction braking, while complex because of wheel lockup possibilities, does not peculiarly challenge stopping distance performance of conventional vehicle types. An exception to this rule, of course, is found in the case of the antilock-equipped vehicle which does not modulate each wheel brake independently.

4.3 Braking In A Turn

As a generic class of maneuvers, braking in a turn can apply to a broad variety of combinations of control inputs and initial conditions. In this study, however, and in previous NHTSA-sponsored studies, the relevant maneuver scenario has been described as limit braking in an initially steady turn of intermediate cornering severity. In this scenario, a limit condition is reached when the braking input level is such that directional controllability is unacceptably degraded. In virtually all previous research which has adopted this basic maneuver scenario, the measurement and characterization of directional response to braking has constituted the primary task. Indeed, it has been implicit in this prior research that loss of directional control is the primary hazard associated with emergency braking in a turn.

In the present study, however, we consider the same type of maneuver scenario for braking in a turn, but from the viewpoint of evaluating the loss, if any, in limit stopping capability which may derive from the simultaneous requirement of tracking an intermediate severity curve. The fundamentals of tire mechanics reveal that a tradeoff in tire shear force capability does exist such that stopping capability is compromised by the existence of the simultaneous cornering requirement. However, the loss in stopping capability due to low level cornering is a very weak function of the lateral slip condition while the loss in cornering capability with increased braking level near the traction limit is a very powerful function of the prevailing longitudinal slip condition. Thus we are clearly addressing the less significant aspect of the safety problems associated with braking in a turn when we address stopping capability per se.

Looking at data gathered in the two test phases of this project, we can evaluate the degree of significance of elongation of stopping distance in a turn compared to stopping distance in a straight line. As outlined in tabular form in Sections 3.1 and 3.3, it was found that values of stopping distance in a turn registered very close to stopping distance values measured in a straight line on the same surface. In the data taken at Chrysler with twelve vehicles in the LLV loading condition, the average vehicle produced a value of in-a-turn stopping distance which was within 3.2% of its straight-line stopping distance value. In tests of the five vehicles at Bendix, in both the LLV and GVW loading states, the average value of in-a-turn stopping distance falls within 0.75% of the straight-line stopping distance for each vehicle. Individual vehicles, however, exhibited differences in "turn vs. straight" stopping distances which were as high as 17.2% in the test series at Chrysler and 10.9% in the series at Bendix.

Regarding these "flyers" in the data sample which suggest that certain vehicles may exist (albeit few in number) which provide substantially reduced in-a-turn stopping capability, we observe that a basic unreliability of the data tends to leave this question open. For example, in Table 4.2 we see that the percentage differences between in-a-turn and straight-line stopping distances are not well correlated in the data gathered on the four vehicles which were tested at both the Chrysler and Bendix facilities. In fact, this table shows that some of the larger percentage differences were registered with a (+) polarity at one test facility and (-) at the other. We interpret this table as suggesting that there may be no bona fide anomalous vehicles in terms of stopping distance in-a-turn performance compared to straight-line stopping distance. Rather, the larger-sized percentage differences may simply derive from random processes in the methodology. Needless to say, the data are too sparse to conclude anything other than the fact that the "flyer" data points are inconclusive.

Table 4.2. Percentage Differences in Stopping Distance Between Straight-Line and Braking-In-A-Turn for Vehicles Tested at Two Facilities in the LLV Condition.

	HI-CO		LO-CO	
	Chrysler	Bendix	Chrysler	Bendix
Monte Carlo	-2.6	+4.3	-17.2	-7.7
Ford LTD	-1.2	+5.1	-10.0	+10.9
Bobcat (Pinto) Wagon	-7.6	-2.5	+9.4	-8.4
AMC Pacer	-9.9	-0.8	-0.7	+2.5

Nevertheless, the general finding is that a conventional straight-line stopping distance measure constitutes a rather good approximation of the stopping distance capability of the same vehicle for the case of braking in an intermediate severity turn on the same surface. Thus, from the viewpoint of a candidate stopping distance in-a-turn regulation, there is a question as to the potential "payoff" in braking performance standardization which would be added to that already accrued through regulating straight-line braking performance.

5.0 CONCLUSIONS AND RECOMMENDATIONS

This research has examined various aspects of passenger car braking performance in an effort to determine whether FMVSS - 105-75 might be meaningfully extended to include stopping distance requirements covering other braking conditions than those currently specified. We find, in general, that the answer to this inquiry is, "NO". The single exception concerns the specification of a stopping distance-based braking efficiency performance for the case of braking in a straight line, on a low friction pavement. In regard to this conclusion we offer the following remarks:

1) The conclusion that 105-75 may be meaningfully extended by adding a low friction test does not derive from a general discovery that 105-compliant vehicles are peculiarly deficient in braking capability on low friction surfaces. Rather, the conclusion is based upon the observation that a "single point" braking performance requirement does not, of itself, constitute a means of "standardizing" vehicle braking capability over the range of possible friction levels. Thus the suggestion of extending 105-75 to include a low friction test is based upon the matter of conceptual adequacy rather than upon a demonstrated safety need.

2) Regarding safety needs, it was seen in tests on low friction surfaces, that current vehicles provide braking efficiencies which average approximately 7-12% lower than the efficiency levels attained on a high friction surface. This relatively small difference between efficiencies achieved on low and high friction surfaces suggests that the vehicle manufacturing industry is generally designing its brake systems to provide adequate low friction performance - even though this performance is currently unregulated.

3) The larger scale problems associated with maintaining and specifying a low friction test pavement suggest in our view, that any extension of FMVSS 105-75 to include a low friction test should also incorporate a "braking efficiency" approach toward performance normalization.

4) If a low friction braking efficiency requirement were to be promulgated, any other existing stopping distance requirements should be reformulated in terms of "efficiency", also. Further, the setting of requirement levels will necessitate that NHTSA proceed with certain previously recommended research [1] regarding development of braking efficiency into a rulemaking-suitable method.

In addition to this basic conclusion and recommendation, other findings supported by the study results are as follows:

1) The minimum stopping distance exhibited by vehicles while braking in a medium-severity turn does not appear to differ significantly from minimum stopping distances measured on equivalent surfaces while braking in a straight line. Accordingly, one may generally assume that measures of straight line stopping distance suffice as a close approximation of stopping distances achievable in medium severity turns.

2) Measures of minimum stopping distance obtained while braking on split-friction surfaces do not appear to be generally useful as characterizations of vehicle safety quality. Because of many possible lockup combinations, split friction stopping distances are generally seen to be rather comparable to the performance limits imposed by the low friction side of the split, alone. Moreover, the potential for a yaw perturbation during split friction braking was seen to be the overwhelming reality — tending to suggest that any stopping distance compromises are by far the smaller part of the safety issue.

3) The extra costs of anti-lock braking systems would not appear to be generally justifiable simply on the basis of stopping distance improvements. Although both improvements and degradations in stopping distance were seen to derive from anti-lock system operation, the conventional wisdom, borne out by the data, is that anti-lock systems contribute profoundly to directional controllability but only minimally to stopping distance performance.

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