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Foreword

Our highway transportation system consists of three basic elements—the vehicle, the driver, and the roadway—and the complex interactions of each upon the other. Greater knowledge of each of these elements and their interactions is essential to the development of action programs designed to correct transportation deficiencies.

This RECORD contains four papers that shed additional light on these transportation system elements. This publication should appeal to a wide span of interests, including the commercial truck operator and safety researcher on the one hand, to the automotive manufacturer and medical doctor on the other.

The first paper presents off-tracking calculation charts for trailer truck combinations. Using scale models, three Bureau of Public Roads employees investigated steering, and off-tracking characteristics of power vehicles, and trailer combinations and developed charts for 90 degree and 270 degree turns. This information is extremely helpful in laying out proper intersections for trucks.

The next paper, by an Ohio State researcher, describes a study of the ability of drivers to estimate the speed of their vehicle without using their speedometers. The research investigates many of the factors that bear on the driver's decision to travel at a given speed.

Another paper, also by Ohio State researchers, indicates how a vehicle lighting display affected performance using car-following studies. Indications are that relative headways and velocities can be improved if vehicles are equipped with different lighting configurations that convey additional information to drivers.

In the last paper, a Maryland researcher has studied performance requirements of a passenger airbag restraint system that becomes operable in the event of an automobile crash. Analyses and laboratory tests are reported and some preliminary indications of what might be ultimately required are stated. This is a frontier area in safety research and is an interesting example of an aerospace application to the highway scene.

Contents

Offtracking Calculation Charts for Trailer Combinations

Offtracking, Turning Track Widths and Curb Radii for Single-Unit Vehicles and Trailer Combinations on Turns of Various Degrees and Radii

HOY STEVENS, SAMUEL C. TIGNOR, and JAMES F. LOJACONO
Respectively, Highway Transport Research Engineer, Highway Research Engineer, and Engineering Technician, Traffic Systems Division, Office of Research and Development, U. S. Bureau of Public Roads

In this report the offtracking characteristics of single unit vehicles and combination vehicles are described. Offtracking values were obtained with scale models of vehicles making turns on turning radii varying from 25 to 225 feet. The offtracking data are shown in two charts, one for 90-degree turns and one for 270-degree turns, for vehicle wheelbases ranging from 5 to 55 feet.

Individual vehicle offtrackings are influenced by three variables: the degree of turn, the length of vehicle wheelbase, and the turning radius. It was found that the offtracking measurements of a trailer combination may be calculated by adding the offtracking measurements of the individual vehicles in the combination. The research also found that the offtracking is greatest when the projection of the rear axle axis passes through the turning radius center, even though the projections of the other axles on the vehicle or trailer combination do not, at the same time, pass through the turning radius center.

•WITH THE expansion of the Interstate System and increasing usage of the nation's highways, new demands are being made on highway designers. Strong emphasis is directed to designing highways which provide good traffic flow, traveling ease, and maximum safety.

Changes are also taking place in the vehicles which use the highways. Size and weight regulations for commercial vehicles are being reevaluated and changed, and more and bigger vehicle combinations are using the highways. With these changes comes the need for more information on the handling characteristics of such vehicles.

The turning characteristics and offtracking behavior of single-unit trucks and trailer combinations are of particular interest to the highway engineer for use in the design of highway curves, city street turns and freeway entrance and exit ramps. Until recently only limited information has been readily available for turns of different degrees and turning radii.

Paper sponsored by Committee on Vehicle Characteristics and presented at the 45th Annual Meeting.

Data from the vehicle manufacturers have been sparse. There have been several other studies made on vehicle steering performance, but most of the material reported thus far has provided information on the operation of a few specific vehicles and combinations on minimum radius turns. The data presented usually relate only to maximum resulting offtracking values without regard to the degrees of turn made by the vehicle before exiting onto a tangent.

There have also been a number of sketches, drawings and detailed descriptions of the minimum turning paths of various specific vehicles. Such information is of value, but is inadequate because it does not allow for easy interpolation of offtracking measurements between different classes of vehicles, or for comparisons of vehicles of different wheelbases operated on turns of different radii. Even the SAE offtracking formulas (1) require the use of specific vehicle dimensions and yield only maximum offtracking values for a particular trailer combination. Thus, in order to make a comparison of the offtracking characteristics of a number of different trailer combinations, or to determine a range of values for some particular turning radius, a long and tedious process of individually calculating the offtracking for each variation in vehicle dimensions is required.

In addition, previously reported offtracking data have been based on measurements taken from the center of the axles of the vehicle. This may be satisfactory for automotive engineering uses but for the highway engineer it necessitates the adding or subtracting of additional factors. In this report, all offtracking values and turning radius measurements are to the outside of the outer tire of an axle.

The research reported in this paper was planned to develop a simpler, quicker and more comprehensive method of calculating offtracking. The technique used to accomplish this is a series of charts which are so constructed as to allow for direct reading and calculation of offtracking values for almost all practical highway vehicles and trailer combinations. The information is reported for the important turns of 90 and 270 degrees and for outer front wheel turning radii from 25 to 225 feet. The range of turns and turning radii covers most of the vehicle turns made on city streets and at rural intersections, including at-grade intersections, diamond interchanges and separated cloverleaf interchange ramps.

The fundamental premise on which the data and methods contained in the report are based is that the sum of the offtracking of the individual vehicles of a highway trailer combination closely approximates the total offtracking of the combination. Starting with this premise, the pattern of research included experiments with vehicle models which led to the establishment of patterns and values of vehicle offtracking behavior which are related to variations in wheelbase length, turning radius and degree of turn. The final purpose was to develop methods of plotting these data for rapid use and comparison.

In approaching the data contained in this report there is a need for preciseness in definitions. This is because in some instances two or more engineering organizations have defined the same terms differently. There are also a few terms used in the paper which have not been previously defined by others. A definition of terms is given in the Appendix, and it is recommended that it be reviewed carefully before continuing.

FUNDAMENTALS OF OFFTRACKING

Offtracking is the phenomenon in which the paths of the wheels of a rear axle of a single-unit power vehicle, or of a trailer combination, deviate inward toward the center of a turn from the circular turning path of the outside front wheel. When operating on uniform radius turns individual vehicles, whether in combinations or single-unit vehicles, offtrack in similar patterns of turns. The front wheels of a power unit do not offtrack, but all other axles on a vehicle, or on a trailer combination, do offtrack on a turn.

While the preponderance of highway vehicles have non-steerable rear axles, there is a small minority of vehicles which have various methods of rear steering. In this study only vehicles with non-steering rear axles were considered, and the data refer only to such vehicles.

For practical vehicle-highway geometrics, and important value of offtracking is the greatest offtracking that occurs when a single-unit vehicle or a trailer combination makes a turn of 270 degrees. With short wheelbase single-unit vehicles, the greatest offtracking may occur early in the first 90-degree segment of a turn, but with very long trailer combinations, it may take the full 270 degrees of turn before the greatest offtracking is attained.

With the longer trailer combinations, the offtracking during a 90-degree turn will be substantially less than their greatest offtracking on a 270-degree turn. Further, on 90-degree turns the front wheels of the power vehicle of a long trailer combination will run for some distance on the exit tangent before its greatest offtracking is attained. It is difficult to calculate the offtracking of long wheelbase trailer combinations when the front wheels of the power vehicle travel on the exit tangent, but the solution can be obtained with scale models of vehicles. Of course, with very short wheelbase single-unit vehicles, which reach their greatest offtracking before 90 degrees of turn, any travel on the exit tangent does not increase the amount of offtracking.

On turns, the offtracking characteristics of single-unit vehicles, and of the individual vehicles in trailer combinations, are affected by several interlocking factors. The factors are:

1. The degree of a turn.
2. The wheelbase of each individual vehicle in a trailer combination.
3. The uniform turning radius of the outside of the outer front tire of the power vehicle. (This turning path of the outside of the outer front tire usually is the outer pavement or curb radius on a specific turn.)
4. The radius of the outside of the outer front tire on a trailer's real or virtual front axle with reference to the turning radius center when the towing vehicle is at its point of greatest offtracking on a specific turn.
5. In trailer combinations, the rear trailing axle of each leading vehicle acts as a real or a virtual front axle of the following trailing vehicle. The virtual front axles of trailing vehicles are as follows: (a) On semitrailers, the tractor rear axle is the semitrailer's virtual front axle. (b) On trailer converter dollies, or on non-detachable front axle assemblies of full trailers, the virtual front axle of such semitrailer-type assemblies is located on the centerline of the towing vehicle's pintle hook, which is the same location as the center of the pintle hook eye of the towbar. (c) On full trailers, the axle of the trailer converter dolly is its virtual front axle; however, with non-detachable front axle assemblies, such a front axle of a full trailer is its real front axle. Both types of front axles for full trailers perform similarly.

ILLUSTRATIONS OF TURNING AND OFFTRACKING PHENOMENA

Single-Unit Vehicles

The principles of offtracking phenomena are illustrated in Figures 1, 2, 3, and 4, which show the action of vehicles with Ackerman steering on turns.

In Figure 1 a long single-unit vehicle is shown at its entrance tangent position just before entering a curve. It will be noticed that the projections of the two stub axles of the front wheels and of the rear axle are parallel, and do not intersect. During a turn the projections of the front wheel stub axles and the rear axle of long vehicles vary from parallel when on the entrance tangent, to various intersecting positions during a turn (Fig. 2), and the projections reverse towards parallelism when the front wheels leave the turn on an exit tangent (Fig. 3). Thus, the offtracking rear wheels travel in a double spiral curve.

In Figure 2 the vehicle has not attained its greatest offtracking on a 90-degree turn; in fact, it is still in transition from its starting position, even though the front wheels are at the exit tangent. It will be noted that, while the projections of the stub axles of the front wheels pass through the turning radius center, the axis of the rear axle does not pass through the turning radius center.

In such situations, the greatest offtracking will occur after the outer front tire of the vehicle is on its exit tangent, as shown in Figure 3, where the front end of the

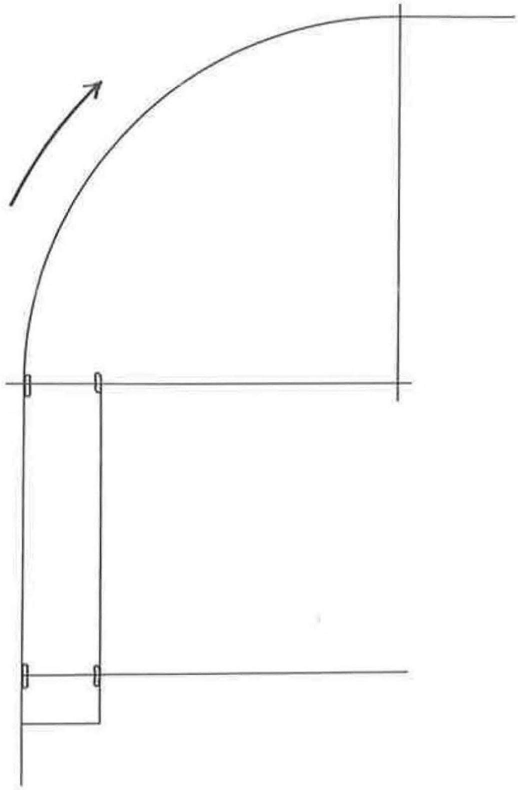

Figure 1. A long wheelbase vehicle in tangent position about to enter turn.

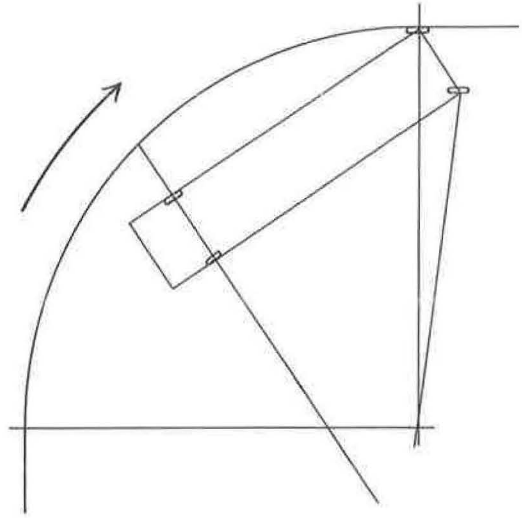

Figure 2. A long wheelbase vehicle which has completed 90 degrees of turn but which has not yet obtained its greatest offtracking.

vehicle has moved down the exit tangent until the projected axis of the rear axle passes through the turning radius center. This point of greatest offtracking during a 90-degree turn was observed in the operation of the vehicle models. It will be noted that the axes of the front wheels no longer pass through the original turning radius

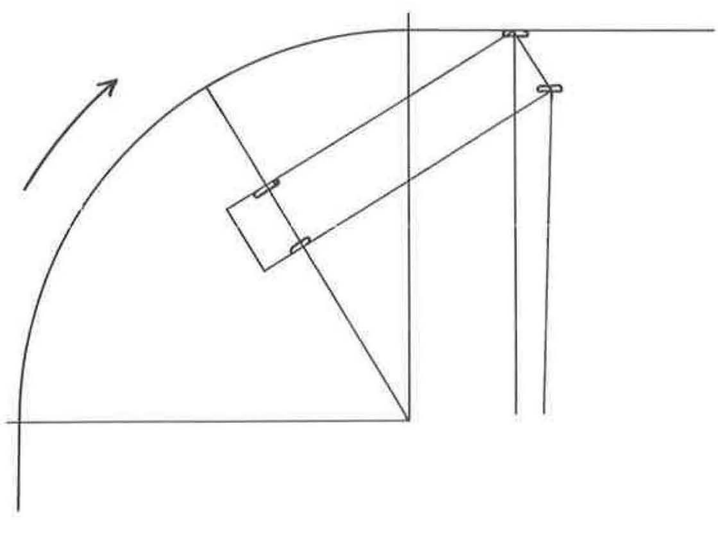

Figure 3. A long wheelbase vehicle which has reached its point of greatest offtracking after traveling some distance on an exit tangent.

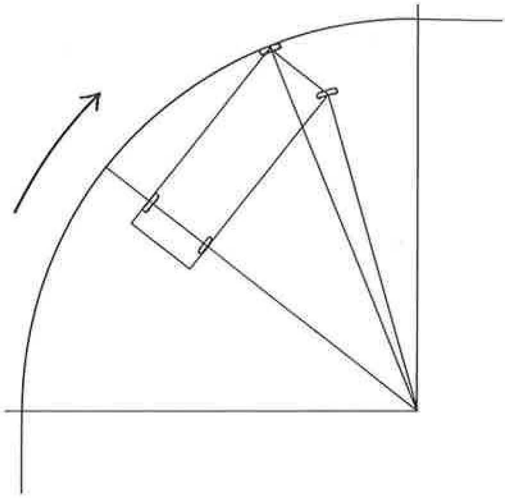

Figure 4. A short wheelbase vehicle which has reached its point of maximum offtracking.

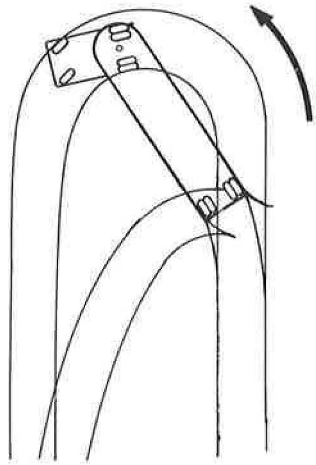

Figure 5. A long wheelbase combination on a short radius turn in which the semitrailer backs up and pivots behind the turning radius center.

center, but the axis of all axles will intersect at some distance behind the turning radius center. The amount of greatest offtracking in such situations was measured with the vehicle models, but cannot be calculated by the SAE equations.

With short wheelbase, single-unit vehicles, such as passenger cars and small trucks, the vehicle's maximum offtracking usually will occur during the first 90-degree segment of a turn. This situation is illustrated in Figure 4. Here the axes of all axles intersect at the turning radius center. The offtracking values of such vehicles were measured with the vehicle models and are included in the offtracking charts (Figs. 10-12). However, the offtracking values of these short wheelbase vehicles can be solved by the SAE equations.

Trailer Combinations

It is desirable that trailer combinations, when negotiating highway curves or at-grade intersections, move continuously and progressively forward at a reasonably rapid speed. Because of their articulated construction, however, trailer combinations may not travel in a continuous, smooth path when the turning radius is shorter than the trailer wheelbase. Such nonuniform type of travel is possible with trailer combinations because fifth wheel pivot type steering permits a trailer to turn 90 degrees or more from the longitudinal axle of the towing vehicle. The angle through which the power vehicle can turn is limited, of course, by its steering cramp angle and its wheelbase.

An example of a trailer combination offtracking in a noncontinuous, irregular manner is shown in Figure 5. When trailer combinations are negotiating 180-degree turns and the turning radius is less than the length of the trailer's wheelbase the rear axle will pass behind the turning radius center and will pivot and travel backwards in an irregular path. The rear axle of long trailer combinations traveling short radius 270-degree turns also exhibits similar backing and pivoting characteristics. Such reverse travel and pivoting of the rear axle can only be considered in very close quarters, as in buildings where the drivers carefully manipulate the trailer combinations at creep speeds. The charts included in this report are not applicable to this type of irregular offtracking.

Trailer combinations negotiating 90-degree turns, however, travel in a continuous and smooth path regardless of which side of the turning radius center the semitrailer passes on. Figure 6 shows a long wheelbase trailer combination following a relatively

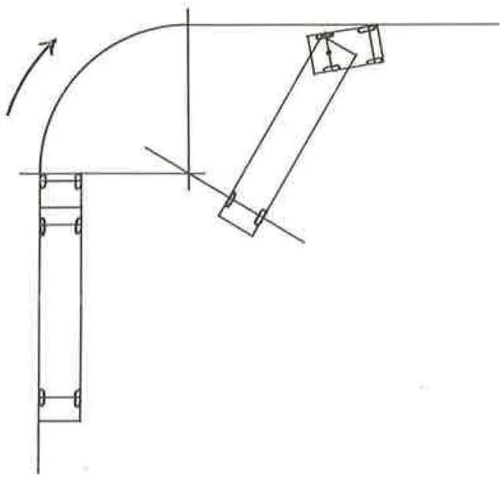

Figure 6. A long wheelbase combination on a short radius turn in which the semitrailer passes in back of the turning radius center.

short turning radius, a situation typical of city street operations. Because the outer rear tire on the rear axle passes behind the turning radius center, the greatest offtracking cannot be calculated with the SAE equations but can be and was measured with the vehicle models. The greatest offtracking on such a turn occurs when the projection of the rear axle, as shown in Figure 6, passes through the turning radius center even though the front wheels of the power vehicle are on the exit tangent.

The problems associated with long trailer combinations negotiating curves having short turning radii are troublesome, particularly on city streets and diamond approaches to controlled access highways. Such problems will be further magnified if, in the future, longer single trailer combinations are permitted. In general, double trailer combinations offtrack less than long single trailer combinations, as is shown later.

FACTORS IN OFFTRACKING DETERMINATIONS

One important feature of vehicle offtracking is that the greatest offtracking for any degree of smooth and continuous turn occurs when a projection of the axis of the rear axle of a vehicle is on a radial passing through the turning radius center. This phenomenon was observed with the vehicle models, which were equipped with a scale that projected from the outer end of the trailing rear axle. It was observed that the greatest outer rear tire offtracking occurred when the rear axle was parallel with a radial line passing through the turning radius center on the model test pattern.

The various measurements of offtracking data which are of interest and use to the highway design engineer are: (a) dimensions of vehicles and trailer combinations; (b) turning radius of specified turn; (c) offtracking of trailing rear axle; (d) turning track width; and (e) inside curb radius (for zero clearance with tire).

The dimensions of vehicles and trailer combination are needed so that the design engineer will know the sizes of vehicles to be considered in a specific turn situation. Dimensions of the individual vehicles in a trailer combination needed are: (a) wheelbase of each vehicle, (b) width over the tires, and (c) on double cargo vehicle combinations, the rear overhang of each towing vehicle and the spacing between the vehicles.

The outer curb radius of a specific turn usually is determined by the location and terrain situation in the turning area.

Offtracking is the radial distance between the outer front wheel turning radius of the outside of the outer front tire of a vehicle, and the radius of the outside of the outer rear tire of a rear trailing axle, at its point of greatest offtracking. Offtracking amounts for single-unit vehicles and individual vehicles of trailer combinations can be obtained from the offtracking charts.

The turning track width is the amount of offtracking plus the width over the tires of the dual tires on a rear axle, or the width of the cargo body if it is significantly greater than the width over the dual tires. This dimension was assumed as 8.0 feet in this study because it is the predominant width at present. However, the newly revised 1965 AASHO Size and Weight Recommendations carry a provision for over-the-tire widths of 8.5 feet.

The inside pavement or curb radius on a turn is the radius from the turning radius center to the outside of the innermost rear tire on the rear axle at the point of its greatest offtracking for a specific turn. The inside curb radius equals the original front wheel turning radius minus the turning track width. This inside curb radius will permit a perfectly driven trailer combination, following the specified outer curb turning radius, to just clear the inner curb at its point of greatest offtracking. Practically, the actual inner curb radius should be some amount shorter in order to permit variations in driver manipulation.

Offtracking Values of Single-Unit Power Vehicles

The procedures for determining the offtracking amounts of individual single-unit vehicles from the offtracking charts require only a single reference to either the 90-degree or the 270-degree chart. The procedures are described later.

Offtracking of a Trailer Combination

The offtracking of a trailer combination on a specific turn is a summation of the offtracking of the individual vehicles making up the trailer combination. Each vehicle in a trailer combination offtracks individually in accordance with its wheelbase, and the radius from the turning radius center to the outside of the outer front tire on its real or virtual front axle. The problem is to determine the turning radius of the real or virtual outer front tire of each individual trailer vehicle in the train. Because all trailing vehicles (semitrailers, trailer converter dollies or non-detachable full trailer front axle assemblies, and full trailers) offtrack and steer like semitrailers, it is necessary to assume a virtual, or real, front axle for each such semitrailer-like unit. The point of greatest offtracking for the rear axle of each towing vehicle on a specific turn will prescribe the turning radius of each following semitrailer-like unit. As one proceeds from the front axle of a trailer combination, there is a progressive series of changed, usually reduced, turning radii for the outer tire of the virtual front axle for each semitrailer-like unit. Thus by analyzing each semitrailer-like unit in the order it appears in the trailer combination, using in sequence the turning radius of the outer front tire on each real or virtual front axle, it is possible to obtain a series of separate offtracking measurements for each vehicle, which can be added together to give the greatest overall offtracking of the complete trailer combination. The procedure for making these progressive steps in calculations is described later.

When determining the offtracking of trailer combinations having full trailers, the phenomenon of negative offtracking must be considered. Negative offtracking occurs when the outer rear corner of the cargo body, opposite the pintle hook, swings outside of the path of the outside of the outer rear tire on a turn as shown in Figure 7. In effect, negative offtracking increases the turning radius of the following semitrailer-like unit. The magnitude of negative offtracking depends on the wheelbase of the towing vehicle, the length of rear overhang to the centerline of the pintle hook, the turning radius, and the degree of turn. The negative offtracking values for practical power vehicles and towing semitrailers are given later in Tables 1 and 2. The tables were used because of the difficulty of presenting four variables in graph form.

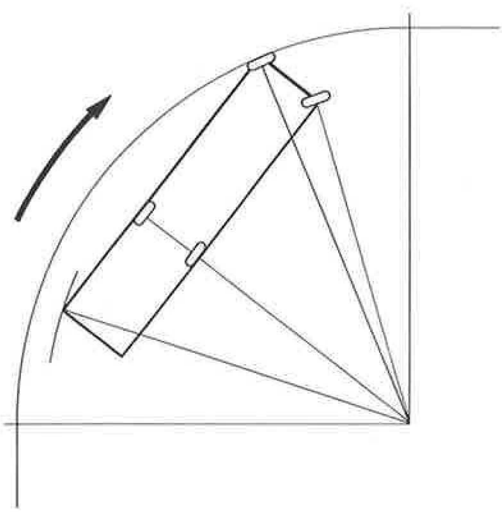

Figure 7. Negative offtracking in which the path of the outer rear corner of the body has a greater radius than the path of the outer rear wheel.

HIGHWAY VEHICLE STEERING SYSTEMS

Ackerman Steering

In order to understand the various aspects of vehicle turning and offtracking, it is first necessary to know something about the systems of steering used on most highway vehicles. Single-unit vehicles, automobiles, light trucks, tractive trucks, and tractors are equipped with Ackerman type steering. The Ackerman system was invented in Germany about 1817 and first patented by an Englishman in 1818. It is preferred over other steering systems because it provides better stability to the front end of the vehicle during a turn.

In the Ackerman system the two front wheels are mounted on short stub axles which are in turn connected to the steering kingpins. The kingpins are connected to the front wheel spring suspension and are supported by the vehicle chassis, or in some cases by a rigid "beam type" front axle. During a turn the front wheels are pivoted on the kingpins by the steering linkage and other mechanisms which are connected to the driver's steering wheel.

Vehicles equipped with Ackerman steering are limited in the amount of offtracking which they can attain by the minimum turning radius curve which can be followed by the outer front wheel. This minimum turning radius usually is limited by the degree to which the inner front wheel may be turned because of mechanical obstructions. This limit to the turning capability of the inner front wheel is commonly called the "cramp angle." On most over-the-road trucks the maximum cramp angle is between 30 and 35 degrees. Recently, however, the manufacturers of city delivery trucks have been widening the distance between front wheels and are obtaining cramp angles of 45-50 degrees. The offtracking charts in this report are designed to include such vehicles.

Fifth Wheel Steering

Semitrailers, full trailers, and trailer converter dollies all operate with a fifth wheel pivot steering principle which is different from the previously discussed Ackerman system. Since trailers are never operated by themselves it is not necessary for them to have the front end stability required for power vehicles. In the fifth wheel pivot type of steering system the front wheels are mounted at the ends of a rigid one-piece axle. This axle is pivoted about a kingpin which is mounted above the lateral center of the axle, where it is connected to the trailer body.

In the case of semitrailers the rear axle of the tractor acts as the virtual front axle of the trailer. In most designs the trailer kingpin, surrounded by a lubricated bearing plate, is attached to the underside of the semitrailer, usually about 3 feet back from the front end of the trailer. Another bearing plate, equipped with a kingpin locking device, is mounted on the tractor chassis, just over the rear axle. This device, known as the fifth wheel, engages and holds the trailer kingpin and allows the trailer to be pulled and steered by the tractor. This system permits easy coupling or uncoupling and the interchanging of trailers.

Full trailers are basically semitrailers which have one of two types of front axle assemblies. In one type the front axle is permanently attached to the trailer, and in the other the front axle may be removed. The removable front axle assemblies are known as trailer converter dollies. They consist of one or more one-piece axles supported by a spring suspension system and with a fifth wheel mounted above the center of the axle. Both the trailer converter dollies and the permanently attached type front axle assemblies have towbars affixed at a 90 degree angle to the axle. This towbar has an eye which engages a vertical pintle hook on the rear end of its towing vehicle. Once engaged, the towbar may pivot freely about its pintle hook, limited only by any interferences with rear frame parts of the towing vehicle. Because of this free pivoting action of the towbar, both types of front axle assemblies of full trailers act as short wheelbase semitrailers in making a turn. Thus a full trailer turns and offtracks like a semitrailer connected in tandem to another semitrailer, both of which have pivot steering.

In considering the steering and offtracking behavior of these full trailer front axle assemblies it may be assumed that their virtual front axle is located at the center of the pintle hook. Thus, the wheelbase of such devices is measured from the center of the towbar eye to the center of the axle.

With fifth wheel pivot steering there is no cramp angle problem and the angular relationship between the towing vehicle and the semitrailer is not restricted and may be as much as 90 degrees or more.

VEHICLE MODELS AND INSTRUMENTATION

The relationships on offtracking contained in this report were obtained primarily through the use of scale models of highway vehicles. The models were designed to provide a good simulation of actual vehicle turning characteristics for a wide variety of different types and lengths of single unit vehicles and trailer combinations. In order to expedite the study, models were designed as detachable components which could be quickly assembled or disassembled. The models, which were equipped with an Ackerman steering mechanism, were constructed to a scale of 0.75 in. = 1 ft and with the width over the tires equal to 8 model feet. For simplicity in manipulation and measurements of the models, the width over the tires on the front wheels was made 8 model feet, the same as the model width over the rear tires. These few inches of additional width over the front tires are not significant to the highway design engineer, as he must allow a far greater width clearance for differences in vehicle manipulation on the road.

The models were operated on a smooth surface composed of 4- by 8-foot panels placed on a level concrete floor. The panels were assembled into a 16- by 16-foot square, centered on which were painted circles simulating highway curves ranging from turning radii of 25 to 100 model feet. Radial lines, at 10-degree intervals, and tangents were then superimposed upon the test layout as shown in Figure 8. Turning radii of 165 and 225 model feet were obtained by placing 8 additional panels about the original 16- by 16-foot square.

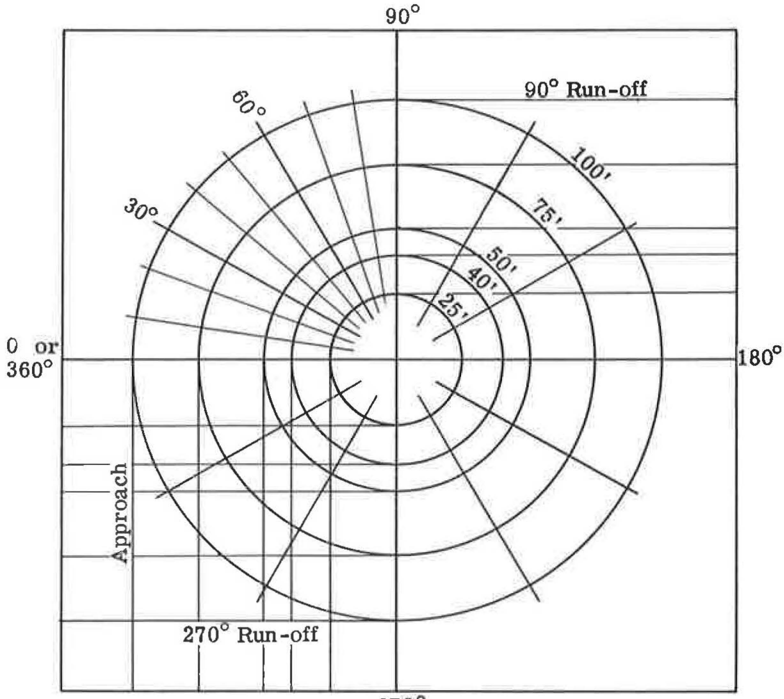

Figure 8. Schematic arrangement of guidelines on floor panels.

Before each individual test was conducted, the vehicle and axle alignment of the model was first checked on an 8-foot long approach tangent. If the model followed the tangent without any perceptible deviation, it was then guided so that the outside of the outer front wheel followed the circular curve selected for the test.

Offtracking tests were conducted with models representing various types of single unit vehicles and trailer combinations. Included were models of power vehicles with wheelbases ranging from 5 to 30 model feet, tractive truck models with considerable rear overhang, and semitrailer models with wheelbases ranging from 5 to 55 model feet. Full trailer model tests were not conducted since full trailers offtrack like semitrailers, regardless of whether they are equipped with trailer converter dollies or non-detachable front axles.

It should be noted that in the semitrailer model tests the trailer kingpin was positioned directly over the center of the front axle of a short wheelbase tractor model as shown in Figure 9. With the kingpin in this position any offtracking of the tractor did not affect the trailer offtracking; however, the tractor did provide model stability.

In all of the model tests the greatest offtracking was measured at the rear or trailing axle of a vehicle or trailer combination. In order to ascertain the magnitude of negative offtracking on tractive trucks having long rear overhangs, an additional offtracking measurement was taken at the outer rear corner of the model opposite the pintle hook centerline. In order to expedite the determination of the greatest offtracking, a scale was mounted on the vehicle models as shown in Figure 9. Offtracking data were obtained by the model assemblies for both 90- and 270-degree turns. Single tires were used on the models at the place of the outer tires in order to give best consistency for various turning paths of the model.

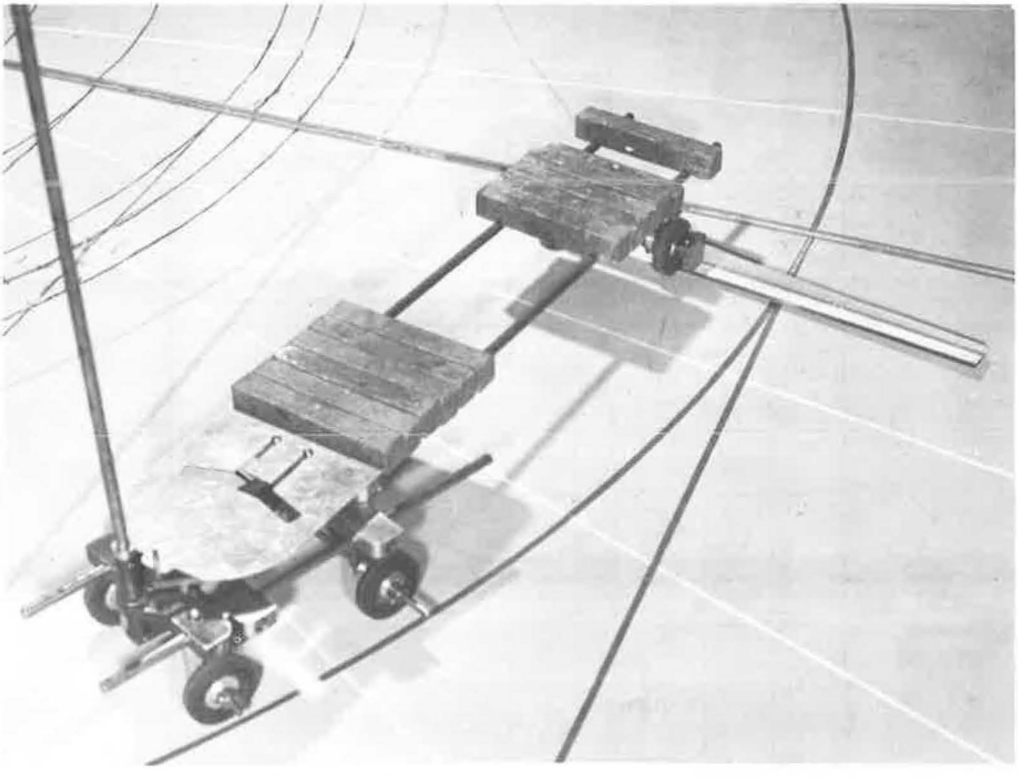

Figure 9. Semitrailer model with fifth wheel pivot steering.

THE USE OF THE OFFTRACKING CALCULATION CHARTS

The results obtained from the vehicle models are presented on two charts, Figure 10 for 90-degree turns and Figure 11 for 270-degree turns. The charts have been designed to permit the rapid determination of offtracking for both single unit vehicles and trailer combinations. For single unit vehicles the offtracking is determined directly. Determination of the offtracking for combination vehicles is accomplished by adding together the offtracking of the individual units of the combinations.

Semilogarithmic graph paper has been used in preparing both charts. The ordinate, in logarithmic scale, represents offtracking in feet. The logarithmic scale was selected in order to reduce the height of the ordinate for publication. The abscissa represents the turning radius in feet and it is shown on an equal interval scale. Wheelbase curves have been drawn in 5-foot increments.

With the use of the charts, vehicle offtracking may be evaluated for turning radii of 25 to 225 feet and for wheelbase lengths of 5 to 55 feet. The 25-foot turning radius represents the shortest radius turn studied with the models. Above a 225-foot turning radius, the offtracking of single-unit vehicles and trailer combinations approaches the "maximum" offtracking as calculated by the SAE equations (1). The charts also indicate the approximate limits of the minimum radii of turns possible when an Ackerman type steering system is employed and when the front wheel cramp angle is 50 degrees. The following examples explain how the charts are to be used.

Single Unit Vehicles

The offtracking for single-unit vehicles can be determined directly from the charts. For example, the greatest offtracking for a 2-axle truck negotiating a turning radius of 70 feet on a 90-degree turn would be found from Figure 10. Assuming the 2-axle truck had a wheelbase of 30 feet, the greatest offtracking would be 6. 4 feet. If the same 2-axle truck was negotiating a 70-foot radius curve through a 270-degree turn, then the greatest offtracking would be 7. 0 feet as determined from Figure 11.

If the minimum turning radius for this 2-axle truck is desired, it can be approximated by the dashed curve shown on the charts. Assuming the front wheel cramp angle is 50 degrees, Figure 10 indicates that a 30-foot wheelbase single-unit truck cannot negotiate a curve having a turning radius less than 45 feet.

If the offtracking is desired for a vehicle having a wheelbase between those represented by the wheelbase curves on either Figure 10 or 11, then Figure 12 may be used to interpolate between the wheelbase curves. For example, if in the above problem the offtracking had been desired for an 11-foot wheelbase single-unit truck, the following procedure would be employed. Take the vertical distance found on either Figure 10 or 11 between the 10- and 15-foot wheelbase curves and locate the same distance vertically on Figure 12 between the 10- and 15-foot lines. At this location on Figure 12, the vertical distance between the 10- and 11-foot lines is then carried back to the initial chart, and located vertically above the 10-foot wheelbase curve. The offtracking is then read from either Figure 10 or 11 on the ordinate horizontally opposite the point representing the 11-foot wheelbase. When negotiating a 90-degree turn, the greatest offtracking for this single-unit vehicle is 0. 9 feet.

Tractor Semitrailers

Offtracking is determined for tractor semitrailers by adding together the offtracking of the individual vehicles of the combinations. For example, the greatest offtracking for a 2-S2 vehicle combination negotiating a turning radius of 100 feet through a 270-degree turn would be found from Figure 11. The dimensions of the sample 2-S2 trailer combination are given in Figure 13.

First, the greatest amount of offtracking is determined for the tractor having a 10-foot wheelbase. Reference to Figure 11 indicates that the greatest tractor offtracking will be 0. 54 feet. After finding the tractor offtracking, the semitrailer offtracking is determined. In arriving at the semitrailer offtracking, it is required that its turning radius and wheelbase be known. Both Figures 10 and 11 have been prepared to take

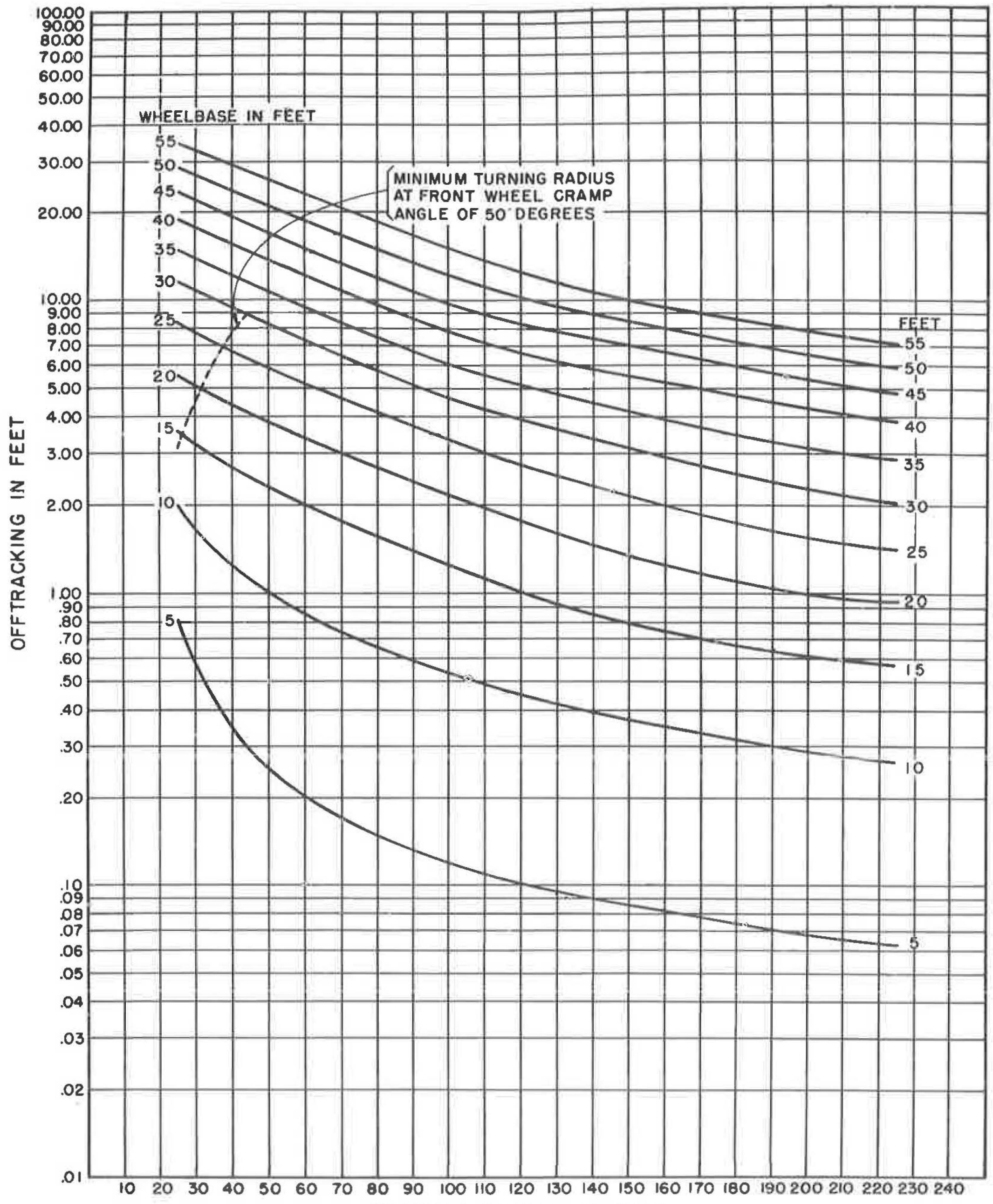

Figure 10. Offtracking vs turning radius for 90-degree turns, various wheelbases.

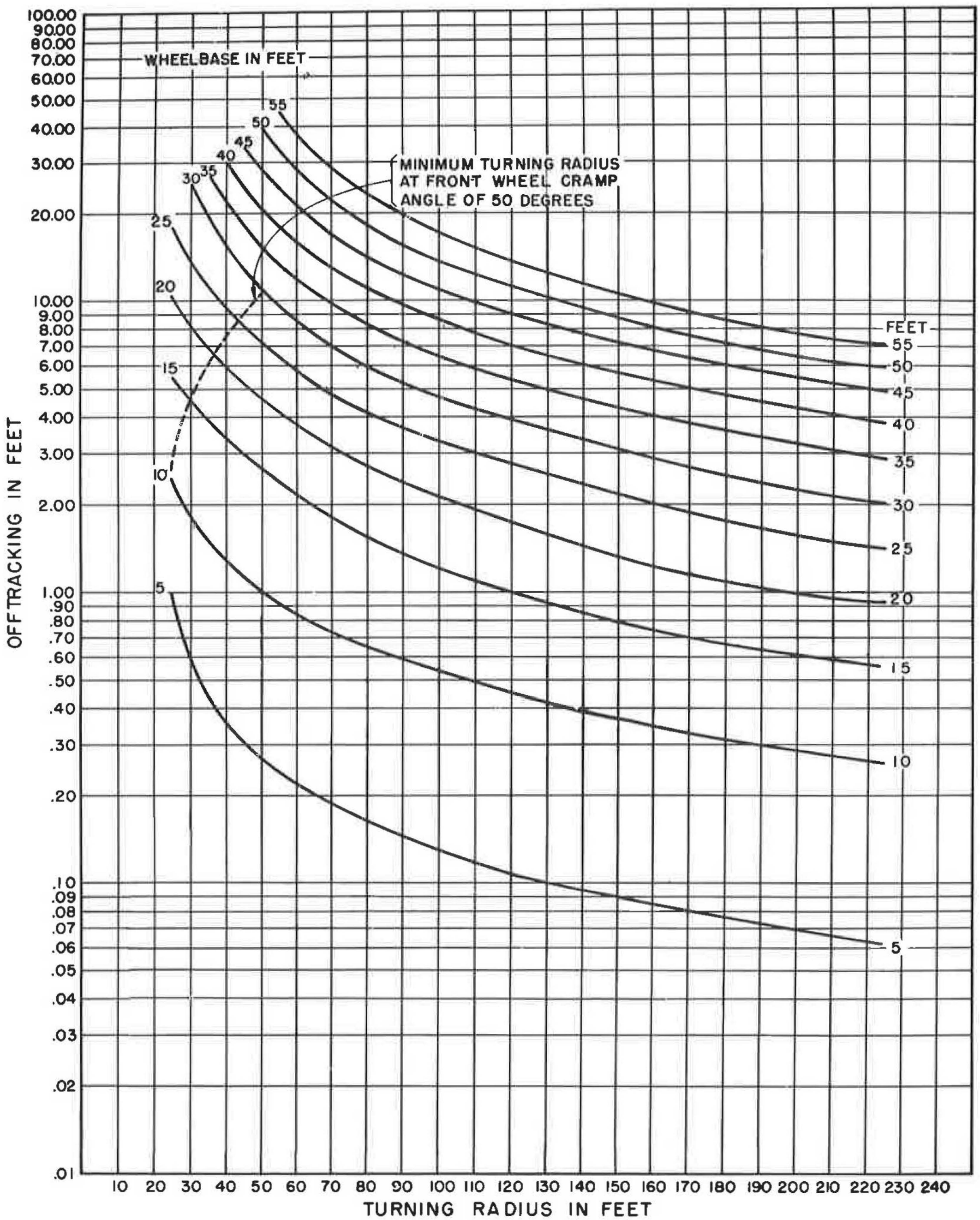

Figure 11. Offtracking vs turning radius for 270-degree turns, various wheelbases.

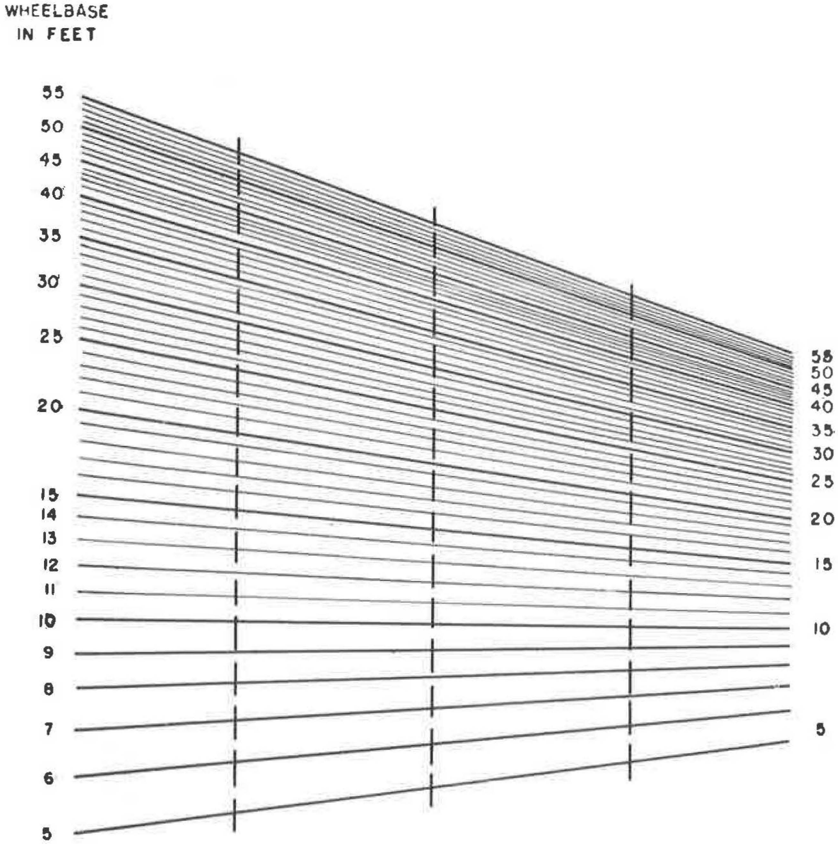

Figure 12. Interpolation guide for wheelbase lengths between 5-foot interval wheelbase curves.

into consideration the assumption that the kingpin is located "directly above" the centerline of the rear axle of the tractor. Then in effect the rear axle of the tractor becomes the virtual front axle of the semitrailer.

The semitrailer turning radius is computed by subtracting the tractor offtracking from the tractor turning radius and is found to be 99.5 feet. The semitrailer wheelbase is the distance from the kingpin to the centerline of the rear axle on the semitrailer. In this example, the semitrailer has a tandem rear axle. Therefore, the wheelbase, which is 29 feet, is the distance from its kingpin to the centerline between the tandem axles. Figure 11 indicates that the greatest semitrailer offtracking will be 4.4 feet when the turning radius is 99.5 feet and the wheelbase is 29 feet.

The offtracking of the entire 2-S2 tractor semitrailer, with an overall length of 50 feet, negotiating a turning radius of 100 feet through a 270-degree turn is the sum of the tractor offtracking and the semitrailer offtracking which is 0.54 plus 4.4 or 4.94 feet. The turning track width is the sum of the offtracking and the width over the tires. In this example the width over the tires is eight feet; therefore, the turning track width is 4.94 plus 8.00 or 12.94 feet. The radial distance from the turning radius center to the inside curb is equal to the turning radius minus the turning track width. In this example, the inside curb radius equals 100.00 minus 12.94, or 87.06 feet.

The values for both the turning track width and the inside curb radius are computed values obtained from the charts. However, in design problems highway engineers may desire to use a turning track width greater than computed from the charts in order to assure proper traffic operations. Thus, in the above example, an inside curb radius may be designed somewhat less than 87 feet.

Tractor Semitrailers and Full Trailers

The offtracking for tractor semitrailers and full trailers is determined from Figures 10 and 11 in much the same way as for tractor semitrailers. For example, assume the 2-S2 tractor semitrailer used in the previous example is connected to a 30-foot long full trailer as shown in Figure 13.

In determining the offtracking of the 2-S2-2 type tractor semitrailer and full trailer combination, the offtracking of the individual vehicles of the combination are added together. As previously determined the tractor semitrailer's greatest offtracking, in reference to the centerline between the tandem axles, was 4.94 feet when negotiating a 100-foot turning radius through 270 degrees. But before the offtracking is known for the entire 2-S2-2 type vehicle combination, the offtracking for both the trailer converter dolly and the full trailer must be determined.

The trailer converter dolly is connected to the semitrailer at a pintle hook located 7 feet behind the centerline between the tandem axles. It is apparent that the phenomenon of "negative offtracking" is present in that the path of the outer rear corner of the semitrailer swings outward from the turning radius center. The magnitude of negative offtracking is found from Table 1 or 2. With a wheelbase of approximately 30 feet, a turning radius of nearly 100 feet and a 7-foot rear overhang to the pintle hook, the negative offtracking of the virtual front axle of the dolly is 0.26 feet for a 270-degree turn.

The pintle hook was assumed to be in the center of the virtual front axle of the trailer converter dolly. Knowing the offtracking for the tractor semitrailer, the turning radius can be determined for the virtual front axle of the trailer converter dolly. The trailer converter dolly turning radius is found by subtracting the tractor semitrailer offtracking, 4.94 feet, from the turning radius of the tractor and adding to that quantity the value of negative offtracking. Thus, the turning radius of the trailer converter dolly's virtual front axle would be 100.00 minus 4.94 plus 0.26 or 95.32 feet. With the trailer converter dolly having a wheelbase of 7 feet and with a turning radius of 95.32 feet, the dolly offtracking can be determined from Figure 11 as 0.28 feet.

After finding the greatest offtracking of the trailer converter dolly, the offtracking for the full trailer is determined. The turning radius for the full trailer is computed in the same way as for the semitrailer. Thus, the turning radius of the virtual front axle of the full trailer is 95.32 minus 0.28 or 95.04 feet. The kingpin on the full trailer is assumed to be "directly above" the centerline of the dolly axle. If a tandem axle dolly had been used, the kingpin would be located "directly above" the centerline between the tandem axles. In effect the dolly axle is the virtual front axle of the full trailer. With a full trailer wheelbase of 24 feet and with a turning radius of 95.04 feet, the full trailer offtracking is determined from Figure 11 as 3.2 feet.

The offtracking of the entire 2-S2-2 tractor semitrailer and full trailer with an overall length of 83 feet is the sum of the offtracking of the individual vehicles minus the negative offtracking. The greatest offtracking for this 2-S2-2 combination would be 0.54 + 4.40 + 0.28 + 3.20 - 0.26 or 8.16 feet when negotiating a 100-foot turning

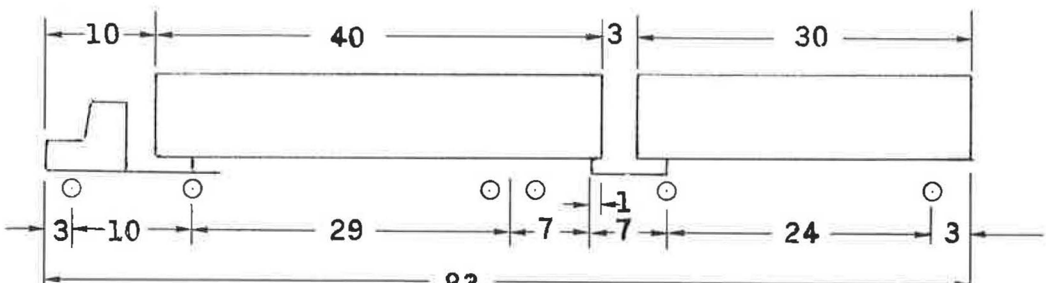

Figure 13. Dimensions in feet of trailer combination used in demonstrating calculation charts.

TABLE 1

NEGATIVE OFFTRACKING VALUES FOR 90-DEGREE TURNS

Wheelbase, Feet	Turning Radius of Outside of Outer Front Wheel, Feet	Offtracking of Outside of Outer Rear Wheel, Feet	Negative Offtracking, Feet[a]				
			3-Foot Overhang	5-Foot Overhang	7-Foot Overhang	9-Foot Overhang	11-Foot Overhang
10	25	2.00	0.19	0.54	1.04	1.70	
	30	1.71	0.16	0.44	0.85	1.40	
	40	1.27	0.12	0.32	0.63	1.03	
	50	1.01	0.00	0.26	0.50	0.82	
	60	0.85	0.00	0.21	0.41	0.68	
	70	0.75	0.00	0.18	0.35	0.58	
	80	0.67	0.00	0.16	0.31	0.51	
	90	0.60	0.00	0.15	0.27	0.45	
	100	0.54	0.00	0.13	0.25	0.39	
15	25	3.62	0.20	0.55	1.11	1.78	2.75
	30	3.24	0.17	0.46	0.88	1.45	2.17
	40	2.70	0.12	0.33	0.64	1.07	1.59
	50	2.30	0.00	0.26	0.51	0.84	1.25
	60	2.00	0.00	0.22	0.42	0.69	1.03
	70	1.75	0.00	0.18	0.36	0.59	0.88
	80	1.55	0.00	0.16	0.31	0.51	0.77
	90	1.38	0.00	0.14	0.28	0.46	0.68
	100	1.23	0.00	0.13	0.25	0.41	0.61
20	25	5.43	0.22	0.62	1.23	1.96	2.97
	30	5.05	0.18	0.51	0.96	1.56	2.33
	40	4.40	0.13	0.35	0.68	1.12	1.66
	50	3.89	0.10	0.27	0.53	0.87	1.29
	60	3.45	0.00	0.22	0.43	0.71	1.15
	70	3.08	0.00	0.19	0.37	0.60	0.90
	80	2.72	0.00	0.16	0.32	0.52	0.78
	90	2.41	0.00	0.14	0.28	0.46	0.69
	100	2.17	0.00	0.13	0.25	0.41	0.62
25	25	8.37	0.27	0.74	1.43	2.28	3.31
	30	7.80	0.20	0.58	1.08	1.75	2.58
	40	6.80	0.14	0.37	0.73	1.20	1.77
	50	5.93	0.10	0.28	0.55	0.91	1.35
	60	5.28	0.00	0.23	0.45	0.74	1.09
	70	4.61	0.00	0.19	0.37	0.62	0.92
	80	4.10	0.00	0.16	0.32	0.53	0.79
	90	3.70	0.00	0.14	0.28	0.47	0.70
	100	3.34	0.00	0.13	0.25	0.42	0.62
30	25	11.71	0.33	0.92	1.76	2.73	3.96
	30	10.90	0.23	0.66	1.24	2.01	2.94
	40	9.50	0.15	0.41	0.79	1.30	1.92
	50	8.30	0.11	0.30	0.58	0.96	1.43
	60	7.27	0.00	0.24	0.46	0.76	1.14
	70	6.41	0.00	0.20	0.38	0.63	0.94
	80	5.70	0.00	0.17	0.33	0.55	0.81
	90	5.11	0.00	0.15	0.29	0.48	0.71
	100	4.62	0.00	0.13	0.26	0.42	0.63
35	25	15.08	0.44	1.19	2.22	3.47	4.89
	30	14.20	0.28	0.77	1.48	2.38	3.45
	40	12.55	0.16	0.46	0.88	1.44	2.12
	50	11.10	0.12	0.32	0.62	1.04	1.53
	60	9.60	0.00	0.25	0.48	0.80	1.19
	70	8.49	0.00	0.20	0.40	0.65	0.98
	80	7.52	0.00	0.17	0.38	0.56	0.83
	90	6.78	0.00	0.15	0.29	0.49	0.72
	100	6.10	0.00	0.13	0.26	0.43	0.64

[a]Outer rear corner opposite pintle hook.

TABLE 2

NEGATIVE OFFTRACKING VALUES FOR 270-DEGREE TURNS

Wheelbase, Feet	Turning Radius of Outside of Outer Front Wheel, Feet	Offtracking of Outside of Outer Rear Wheel, Feet	Negative Offtracking, Feet[a]				
			3-Foot Overhang	5-Foot Overhang	7-Foot Overhang	9-Foot Overhang	11-Foot Overhang
10	25	2.58	0.20	0.55	1.07	1.74	
	30	1.90	0.16	0.44	0.86	1.41	
	40	1.27	0.12	0.32	0.63	1.03	
	50	1.01	0.00	0.25	0.50	0.82	
	60	0.85	0.00	0.21	0.41	0.68	
	70	0.75	0.00	0.18	0.35	0.58	
	80	0.67	0.00	0.16	0.31	0.51	
	90	0.60	0.00	0.14	0.27	0.45	
	100	0.54	0.00	0.13	0.25	0.41	
15	25	10.40	0.31	0.83	1.59	2.55	3.68
	30	4.60	0.18	0.49	0.95	1.55	2.28
	40	3.43	0.12	0.33	0.67	1.09	1.63
	50	2.68	0.00	0.26	0.51	0.85	1.26
	60	2.18	0.00	0.22	0.42	0.70	1.03
	70	1.83	0.00	0.18	0.36	0.59	0.88
	80	1.61	0.00	0.16	0.31	0.51	0.77
	90	1.38	0.00	0.14	0.28	0.46	0.68
	100	1.23	0.00	0.13	0.25	0.41	0.61
20	25	18.93	0.70	1.79	3.19	4.79	4.49
	30	8.70	0.21	0.58	1.12	1.82	2.67
	40	6.24	0.13	0.37	0.72	1.18	1.75
	50	4.72	0.10	0.27	0.54	0.89	1.30
	60	3.71	0.00	0.22	0.43	0.71	1.06
	70	3.10	0.00	0.19	0.37	0.60	0.90
	80	2.73	0.00	0.16	0.32	0.52	0.78
	90	2.41	0.00	0.14	0.28	0.46	0.69
	100	2.17	0.00	0.13	0.25	0.41	0.62
25	25						
	30	14.25	0.28	0.77	1.55	2.39	3.46
	40	9.70	0.15	0.41	0.80	1.31	1.93
	50	7.27	0.11	0.29	0.57	0.95	1.39
	60	5.80	0.00	0.23	0.45	0.74	1.10
	70	4.81	0.00	0.19	0.37	0.62	0.92
	80	4.17	0.00	0.16	0.32	0.53	0.79
	90	3.70	0.00	0.14	0.28	0.47	0.70
	100	3.34	0.00	0.13	0.25	0.42	0.62
30	25						
	30	26.01	1.00	1.80	4.07	5.85	7.71
	40	14.85	0.18	0.49	0.88	1.52	2.20
	50	11.21	0.12	0.32	0.63	1.03	1.53
	60	8.67	0.00	0.24	0.48	0.78	1.17
	70	7.18	0.00	0.20	0.39	0.64	0.96
	80	6.07	0.00	0.17	0.33	0.55	0.81
	90	5.26	0.00	0.15	0.29	0.48	0.71
	100	4.70	0.00	0.13	0.26	0.42	0.63
35	25						
	30	38.00					
	40	18.80	0.20	0.58	1.13	1.83	2.68
	50	15.65	0.13	0.36	0.71	1.16	1.72
	60	12.50	0.00	0.26	0.51	0.85	1.26
	70	10.15	0.00	0.21	0.41	0.67	1.00
	80	8.50	0.00	0.17	0.34	0.56	0.84
	90	7.35	0.00	0.15	0.30	0.49	0.73
	100	6.57	0.00	0.13	0.26	0.43	0.65

[a]Outer rear corner opposite pintle hook.

radius curve through a 270-degree turn. The turning track width would be 8.16 + 8.00 or 16.16 feet.

In summary, the following computations were performed:

Turning radius on curve	100.00 feet	
Tractor offtracking		0.54 feet
Semitrailer turning radius	99.46	
Semitrailer offtracking		4.40
Negative offtracking		-0.26
Turning radius of dolly	95.32	
Dolly offtracking		0.28
Turning radius of full trailer	95.04	
Full trailer offtracking		3.20
Total combination offtracking		8.16 feet

Truck-Full Trailers

The greatest offtracking for truck full trailers can also be determined from Figures 10 and 11. The same techniques are used for determining the offtracking of the individual vehicles of a truck-full trailer combination as are used for determining the offtracking of a tractor semitrailer full trailer combination.

VEHICLE OFFTRACKING COMPARISONS

To illustrate that different types and sizes of vehicle combinations offtrack differently, several representative long trailer combinations have been selected. The dimensions of the trailer combinations are given in Table 3, and their offtracking characteristics are shown in Table 4. It can be seen in Table 3 that the 2-S1, 2-S2, and 3-S2 combinations have overall lengths shorter than either of the 2-S1-2 combinations. It should be noted in Table 4, however, that the 65 foot long 2-S1-2 combination offtracks less than either of the tractor semitrailer combinations and the other 2-S1-2 combination has approximately the same offtracking as the tractor semitrailers.

In view of the fact that vehicles do offtrack differently, highway design engineers use as guides the highway design vehicles recommended by the American Association of State Highway Officials. As a further illustration, the offtracking characteristics of the 1965 proposed revision of the AASHO highway design vehicles are also given in Table 4. Table 5 gives the dimensions of these design vehicles.

MODEL AND SAE OFFTRACKING COMPARISONS

Model offtracking results were compared to values computed by the SAE offtracking equations. Comparisons were made for 90- and 270-degree turns on 50- and 150-foot

TABLE 3

DIMENSIONS IN FEET OF TRAILER COMBINATIONS SHOWN IN TABLE 4

1. Type of trailer combination	2-S1	2-S2	3-S2	2-S1-2	2-S1-2	3-S2-4
2. Length of each trailer	40.0	40.0	40.0	27.0	30.0	40.0
3. Front bumper to nose of first trailer	10.0	15.0	15.0	8.0	8.0	16.0
4. Space between trailers	—	—	—	3.0	3.0	3.0
5. Overall length	50.0	55.0	55.0	65.0	71.0	99.0
6. Width over tires	8.0	8.0	8.0	8.0	8.0	8.0
7. Wheelbase, tractor (to center line of tandem axle)	10.0	15.0	15.0	8.0	8.0	16.0
8. Front bumper to front axle of tractor	3.0	3.0	3.0	3.0	3.0	3.0
9. Wheelbase, semitrailer	34.0	29.0	32.0	21.0	24.0	32.0
10. Rear pintle hook overhang of semitrailer	—	—	—	3.0	3.0	5.0
11. Wheelbase of trailer converter dolly	—	—	—	6.0	6.0	6.0
12. Wheelbase, full trailer	—	—	—	21.0	24.0	32.0
13. Rear overhang of trailer	3.0	8.0	5.0	3.0	3.0	5.0
14. Overall length	50.0	55.0	55.0	65.0	71.0	99.0

TABLE 4

VEHICLE OFFTRACKING COMPARISONS

Class of Trailer Combination	Overall Length (ft)	90-Degree Turn 50-Foot Turning Radius			270-Degree Turn 150-Foot Turning Radius		
		Offtracking (ft)	Turning Track Width (ft)	Inside Curb Radius (ft)	Offtracking (ft)	Turning Track Width (ft)	Inside Curb Radius (ft)
Long Trailer Combinations							
2-S1	50	11.3	19.3	30.7	4.5	12.5	137.5
2-S2	55	10.3	18.3	31.7	3.7	11.7	138.3
3-S2	55	11.7	19.7	30.3	4.3	12.3	137.7
2-S1-2	65	9.4	17.4	32.6	3.3	11.3	138.7
2-S1-2	71	12.5	20.5	30.5	4.4	12.4	137.6
3-S2-4	99	22.0	30.0	20.0	8.1	16.1	133.9
Proposed 1965 Revision of AASHO Highway Design Vehicles							
Passenger Car	19	1.1	7.1	42.9	0.4	6.4	143.6
2	30	3.8	12.3	37.7	1.2	9.7	140.3
2-S2	50	7.8	16.3	33.7	2.7	11.2	138.8
3-S2	55	11.8	20.3	29.7	4.2	12.7	137.3

TABLE 5

DIMENSIONS IN FEET OF PROPOSED 1965 REVISION OF ASSHO HIGHWAY DESIGN VEHICLES SHOWN IN TABLE 4

1. Type of design vehicle	Passenger Car	Single Unit Truck or Bus	2-S2 Combination WB-40	3-S2 Combination WB-50
2. Length of trailer	—	—	36	37
3. Front bumper to nose of trailer	—	—	14	18
4. Overall length	19	30	50	55
5. Width over tires	6	8.5	8.5	8.5
6. Wheelbase, passenger car, single unit, or tractor	11	20	13	18
7. Front bumper to front axle	3	4	4	3
8. Wheelbase, semitrailer	—	—	25	30
9. Rear overhang	5	6	8	4
10. Overall length	19	30	50	55

TABLE 6

MODEL AND SAE OFFTRACKING COMPARISONS

Trailer Combination	Overall Length in Feet	Trailer Length in Feet	Offtracking in Feet: 90-deg Turn, 50-ft Turning Radius		Offtracking in Feet: 270-deg Turn, 150-ft Turning Radius	
			Model	SAE Equation	Model	SAE Equation
2-S1	50	40	11.30	16.62	4.47	4.36
2-S1	55	40	13.00	18.75	4.96	4.81
2-S2	50	40	8.90	11.69	3.28	3.26
2-S1	55	40	10.30	13.51	3.72	3.69
3-S2	50	40	10.00	14.46	3.87	3.90
3-S2	55	40	11.65	16.45	4.30	4.34
2-S1-2	65	2X27	9.38	—	3.28	3.06
2-S1-2	71	2X30	12.48	—	4.41	4.32
3-S2-4	99	2X40	21.97	—	8.12	8.16

turning radii, respectively. In that most of the vehicle models had obtained their maximum offtracking prior to the 270-degree exit tangent, the results could be validated by comparing them to the maximum offtracking value computed by the SAE equations. The results of some of the comparisons are given in Table 6.

It should be noted in Table 6 that the tractor-semitrailer models negotiating the 90-degree turns on the 50-foot turning radius curve did not obtain SAE maximum offtracking. For the tractor semitrailer and full trailer models negotiating the same turns, the SAE offtracking equation is not applicable and the maximum offtracking could not be computed in that the trailing rear axle of the combination passed behind the turning center.

FURTHER RESEARCH PROBLEMS

1. The work reported in this study included only 90-degree and 270-degree turns; hence, if similar data are needed on turns of different degrees, models and procedures similar to those employed in this study can be used.
2. The amount of variation in maneuvering of long trailer combinations may be worthy of study at a variety of cloverleaf intersections.
3. It is believed that the width over-the-tires has almost no effect on the offtracking characteristics of the outside of the outer tires of a trailer combination. This appears certainly to be the case with trailers, but for power vehicles with Ackerman steering a considerable increase in width over the front tires may have some further limiting effects, as discussed briefly in the paragraphs describing Ackerman steering.
4. It is necessary to determine more precisely the percentage relationships between the turning radii and the wheelbases of various trailers in order to prevent any trailer backup and pivot motion on 180- and 270-degree turns.

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Appendix

GLOSSARY OF DEFINITIONS AND TRADE TERMS

Angle of Turn. —The angle through which a vehicle travels in making a turn (AASHO, 2).

Axle. —For simplification, only the singular term "axle" is used in the text and it may designate either a single axle or the centerline between tandem axles, depending on the vehicles being investigated. The reason for this simplification is that on the centerline between tandem axles lies the theoretical turning center of a tandem axle assembly.

Cramp Angle. —The cramp angle is the limit of the turning ability of the front wheels of an Ackerman type front axle, and is limited by the construction of the mechanical parts around the front axle kingpin pivot mechanisms. This construction limits the degree to which the inner front wheel may be turned and also limits the turning of the outer front wheel.

Fifth Wheel. —The fifth wheel is a lubricated bearing plate, mounted on a tractor chassis, or on a trailer converter dolly chassis, and arranged with an internal clutch device to engage and hold the kingpin of a trailer. The fifth wheel clutch engages and locks on contact with the trailer kingpin, but requires a manual release by the driver in order to separate the trailer kingpin and the fifth wheel.

The manual fifth wheel is in predominate use at present. In earlier developments of an "automatic" fifth wheel, the fifth wheel was attached to the trailer and connected to its landing gear, while the kingpin was mounted on the towing vehicle. A few of these automatic fifth wheels are in use today, but only in local cartage service with a captive fleet of semitrailers.

Kingpin. —The term kingpin has two different meanings in automotive design. Hence, the precise meaning of the term is determined by the context in which the term is used:

1. A kingpin of a front axle of a power vehicle is a vertical or near-vertical shaft that is the pivot which connects each stub axle that carries a front wheel of a power vehicle to the rigid center of an Ackerman type front axle. All Ackerman type front axles have two kingpins, one at each end of the rigid center of the front axle.

2. A kingpin of a trailer is a vertical pivot shaft attached near the front of and on the centerline of the underside of a trailer chassis, and is surrounded by a lubricated bearing plate. It engages a fifth wheel on either a towing tractor or on a trailer converter dolly, or is permanently connected to the center of a non-detachable front axle of a full trailer. A trailer is pulled by and pivots around its kingpin.

Radius of Inside Curb. —The radius of the inside curb is the radial difference between the turning radius and the turning track width, when the offtracking of the vehicle is at its greatest amount for a given turn. Note: The shortest radial distance from the turning center to the inside curb may occur at only one point (instantaneous) on a 90-degree turn. On a 270-degree turn, the shortest curb radius may remain constant for some interval or distance before the exit tangent.

Minimum Turning Radius. —The radius of the minimum turning path of the outside of the outer front tire. Note: Vehicle manufacturers' data books usually give "minimum turning radius" to the centerline of the outer front tire (AASHO, 2).

Negative Offtracking. —Negative offtracking is the phenomenon during a turn that causes the radius of the path of the outer rear corner of a vehicle to be greater than the radius of the turning path of the outside of the vehicle's outer rear tire. For example, this phenomenon occurs on a tractive truck, or a trailer, with a cargo body extending back of the rear axle, as the outer rear corner of the cargo body swings outside of the path of the outside of the outer rear tire on a turn.

Offtracking. —Offtracking is the phenomenon in which the path of the outside of the outer tire on a rear, or trailing, axle of a vehicle or a trailer combination deviates inward toward the center of a turn from the circular path of the outside of the outer front tire, while the vehicle or trailer combination is making a turn.

Outside of Tire. —The outside of a tire means the external side of a tire farthest away from the vehicle chassis.

Outside of Outer Tire. —The outside of the outer tire of a vehicle means the outside of the outermost tire (on an axle) on the outer side of a turn.

Outside of Innermost Rear Tire. —The outside of the innermost rear tire of a vehicle means the outside of the rear tire nearest the turning radius center.

Overall Length. —The overall length of a vehicle or trailer combination is the distance between the front bumper of the power vehicle and the rear bumper or guard on the rear vehicle.

Pintle Hook. —A vertical hook device attached to the rear of a tractive truck or to the rear of a leading (towing) semitrailer in a double trailer combination. The pintle hook engages the towing eye (ring eyelet) at the front end of the towbar of a trailer converter dolly, or the towbar of a full trailer non-detachable front axle assembly.

Power Vehicles. —There are three general classes of power vehicles:

1. Single-unit trucks are power vehicles with Ackerman front axle steering, and equipped with a cargo body but not equipped to pull a trailer.

2. Tractive trucks are power vehicles with Ackerman front steering which are equipped with a cargo body and a pintle hook attached to and recessed into their rear frame members in order to pull a full trailer.

3. Tractors for commercial freight use are legally defined as "truck-tractors" to differentiate them from farm or industrial tractors. However, the single term, tractor, is used in this report to mean a power vehicle of short wheelbase that is equipped with Ackerman front wheel steering, and with a fifth wheel to engage and pull a semitrailer.

Rear Axles of Trailers. —Rear axles of trailers predominantly are attached through springing suspensions and mechanisms to the trailer chassis so as to be in a fixed alignment with the longitudinal centerline of the trailer.

Rear Overhang. —The rear overhang of a tractive truck, of a semitrailer, or of a full trailer is the distance between the centerline of the vehicle's rear axle, and the centerline of its pintle hook.

Steering System. —There are two generally used types of steering systems:

1. The Ackerman steering system for front axles of power vehicles consists of a three-piece articulated axle with two front wheels which are mounted on short stub axles. The stub axles are attached to opposite ends of the rigid center section of the front axle by the front axle kingpins. The short stub axles are pivoted about the axle kingpins by steering arms and mechanisms connected to the driver's steering wheel.

2. Fifth wheel pivot steering is similar to that used at the front ends of two-axle, horse-drawn wagons. The front axle is a one-piece, rigid axle with the front wheels at each end of the axle. The rigid axle pivots about a kingpin located above the lateral center of the axle. Surrounding the kingpin are two lubricated bearing surfaces; the lower one is attached to the axle assembly, the upper bearing plate is attached to the underside of the vehicle chassis on its longitudinal centerline. These bearing plates give lateral and longitudinal stability to the cargo vehicle, and enable the trailer to be pulled by means of the kingpin. This type of steering is predominantly used at the front end of trailers.

3. Pintle hook steering through a towbar is similar in action to fifth wheel pivot steering except that no vehicle weight rests on the pintle hook.

Trailers. —There are three classes of trailers:

1. A semitrailer is a cargo trailer equipped with one or more axles at or near its rear, and so constructed that a substantial part of its tare weight and its cargo weight rests upon a tractor through the tractor fifth wheel.

2. A full trailer is basically a semitrailer which has been converted into a full trailer by one of two methods. In one method, the front axle and spring suspension are permanently connected to the chassis of the trailer. The second method is to combine a semitrailer with a trailer converter dolly.

3. A trailer converter dolly is a very short wheelbase semitrailer. It consists of an axle which is attached through a spring suspension system to a platform (chassis) which carries a lower fifth-wheel plate. It has a towbar mechanism affixed at 90 degrees to its axle. The front end of the towbar is equipped with a towing eye which engages with a pintle hook on the rear of the towing vehicle.

Towbar. —A bar, or a V-shaped assembly of two bars, attached to the chassis of a trailer converter dolly, or to the non-detachable front axle assembly of a full trailer, and so constructed that it has a towing eye at its forward end, and exerts a pulling force in the middle of and at 90 degrees to the axle of a trailer converter dolly, or to a full trailer non-detachable front axle.

Trailer Combinations. —For ease of identification, trailer combinations are identified by the numerical axle arrangement code defined in the SAE Handbook (3). In this code, the first single separate numeral means the number of axles on the tractor or tractive truck. Semitrailers are identified by a capital S in front of the second code axle number. A final single number without a prefix in the axle number code indicates a full trailer. Samples of this code are as follows:

2-S1 means a 2-axle tractor and a 1-axle semitrailer.
3-S2 means a 3-axle tractor and a 2-axle semitrailer.
2-S1-2 means a 2-axle tractor and a 1-axle semitrailer pulling a 2-axle full trailer; also called a double trailer combination.
3-2 means a 3-axle tractive truck pulling a 2-axle full trailer.
4-6 means a 4-axle tractive truck pulling a 6-axle full trailer.
3-S2-4 means a 3-axle tractor and a 2-axle semitrailer pulling a 4-axle full trailer; also called a double trailer combination.

Turning Path. —The path of a designated point on a vehicle making a turn (AASHO, 2).

Turning Radius. —The radius of the circular turning path of the outside of the outer front tire from the turning radius center.

Turning Radius Center. —The turning radius center is that point which is the center of the circular turning path that is followed by the outside of the outer front tire of the power vehicle.

Turning Track Width. —The radial distance between the turning paths of the outside of the outer front tire and the outside of the rear tire which is nearest the center of the turn (AASHO, 2).

Wheelbase. —There are several measures of wheelbase depending upon the vehicle. These are defined below.

1. The wheelbase of a single-unit power vehicle (truck or tractor) is the distance between the centerline of the front axle and the centerline of the rear axle. As mentioned earlier, the centerline between any tandem axles always is used as the reference point for wheelbase measurements shown in the charts.

2. On semitrailers, the wheelbase is the distance between the trailer kingpin and the centerline of the rear axle.

3. On trailer converter dollies, which are in effect short semitrailers, the wheelbase is the distance between the center of the towing eye of the towbar and the centerline of the dolly's axle.

4. On full trailers, the wheelbase is the distance between the kingpin of the trailer and the centerline of the rear axle, as in the case of semitrailers.

5. On complete trailer combinations, the overall wheelbase is the distance between the front axle of the power vehicle and the rearmost axle when the trailer combination is strung out in a straight line. This overall wheelbase may differ from the sum of the wheelbases defined above.

Width Over Tires. —The width over tires is the outside-to-outside distance over the tires on an axle.

Note: The meanings of the two following SAE terms are different from the definitions and measurements of offtracking that are used in this report. In order that care may be taken not to confuse the meanings, the SAE definitions are given below.

Turning Center, SAE. —The turning center is that point about which all parts of a vehicle or combination of vehicles revolve in describing a turn of constant radius and to which all wheel spindles are normally radial. In the case of two-axled bogies or tandems in which the axles are constrained to parallelism, the interaxle trunnion or its equivalent is assumed to be radial from this point (SAE, 3). Note: The location of this turning center moves around as a trailer combination enters a curve from a tangent, proceeds around the curve, and leaves on an exit tangent. This turning center should not be confused with outer front wheel turning radius center used in this study.

Offtracking, SAE. —Offtracking is the difference in radii from the turning center to the vehicle centerlines at the foremost and rearmost axles of a vehicle or combination and represents the increase beyond the tangent track occasioned by a turn (SAE, 3).

Capability of Automobile Drivers To Sense Vehicle Velocity

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This paper presents the results of an investigation into velocity sensing. The psychophysical techniques of magnitude estimation and production were employed. Results are presented in the form of regression lines and confidence intervals.

•ONE IMPORTANT element in the control of a motor vehicle is velocity regulation. The proliferation of speed limits and their attendant radar velocity detectors gives mute testimony to this fact, as does the effect of poor velocity control on traffic flow during periods of high density.

An analysis of the driving task suggests that the driver must employ information other than that from the speedometer in regulating velocity. Also indicative of this is the rather frequent occurrence of people who operate vehicles for extended periods of time without the assistance of a speed display device.

The cues available for judgments about vehicle velocity are indeed many. Various aspects of the information associated with vision, audition, proprioception, and kinesthesis are known to change as a function of velocity or from the behavior of the vehicle due to its velocity.

There are two aspects of velocity sensing which are of concern, velocity estimation and velocity change detection. Basically, velocity estimation may be investigated with the analogous techniques of estimation and production. In velocity estimation, the driver-subject is guided to the desired velocity and asked to estimate it, whereas in velocity production, the subject-driver is given a velocity to produce (in both cases without the aid of a speedometer). In the case of velocity change detection, the vehicle is accelerated or decelerated from a reference velocity at a rate which is less than the subject's acceleration threshold. The subject then responds when he can detect the change in velocity. This paper considers only the velocity estimation and production cases.

With the velocity estimation and production techniques, the scaling factor used by the subjects will be reflected in the regression function as fitted to their data. Variance about this regression function will indicate the consistency with which that response can be made.

Conceptually, there should be no difference between the performance of the subjects in velocity estimation and in velocity production if the subject makes his judgment on the basis of some long-term "memory" pattern. Of course, it is expected that biases will be associated with these two techniques as a result of the techniques themselves.

It should be mentioned that factors such as adaptation, forgetting, etc., may play an important role in velocity control. However, we have chosen to fix the magnitude of most of these variables because of the problems associated with an investigation of their effects without a broad base of information about the basic phenomena.

In an effort to investigate the possible effects of "short-term memory," it was decided to include a series of velocity estimations and productions in which knowledge of results was provided. In this fashion, the role of short-term memory could be investigated and compared with the effects of other factors.

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EXPERIMENTAL PROCEDURE

The general form of this study was established as that of a three-factor, switchback design. Two groups of subjects were involved, one receiving first estimation then production trials without feedback, followed by estimation then production with feedback (EPEP order). The second group received production then estimation trials without feedback followed by production then estimation trials with feedback (PEPE order). The two groups of subjects were used so that the effect of the order or presentation could be analyzed (assuming no differential effects due to subjects). The design is balanced with respect to time and presence or absence of feedback. Each of the two groups consisted of six subjects (male college students), giving a total of twelve subjects for the experiment. Each subject was run through a fixed sequence of 48 velocity treatments in each of the four phases of the experiment. This sequence of velocities consisted of two replications of each of 24 different velocities ranging between 18 and 65 mph. This sequence was randomly selected with the restriction that no two consecutive velocities be less than 5 mph apart.

In conducting the experiment, the subject drove the research vehicle (a 1965 eight-cylinder Plymouth station wagon with automatic transmission) to the test site. During this 25-minute drive, the subject had visual access to a Stewart-Warner police speedometer which replaced the vehicle's standard speedometer. After arriving at the test site, the subject was denied direct access to the speedometer in that it was moved to the back seat where the experimenter sat. All data recording was done by hand with an accuracy to the nearest mph.

The instructions given the subject were as follows: (a) For velocity estimation,

> In this part of the study, you will be estimating different speeds. I will help guide you to the speed I desire by saying faster or slower. When we reach the speed, try to hold the car's speed constant, and I will ask you to estimate the speed to the nearest one mph. We will be using a number of different speeds so we will not be going 10-20-30, for example.

(b) For velocity production,

> In this part of the study, you will be producing various speeds. I will ask you to reach a speed and when you feel that you have reached it, you are to hold your car's speed steady there and repeat that speed to me in order to tell me when you have reached it. When I give you the next speed, you will repeat the procedure.

During phases 3 and 4, the subjects were told their true speed to the nearest mph after completing each judgment.

In operation, the car was stopped before beginning each phase of the research, and the appropriate instructions for that phase were read to the subjects. After repeating the instructions if there were questions, the actual trials were begun.

During the execution of this study, no attempt was made to accurately control the period of time in which the subject was allowed to make his response. Also, no acceleration restrictions were imposed on the subjects when making the velocity changes. At the conclusion of each specific trial, the subject was immediately instructed or guided to the next velocity without stopping the vehicle.

Two specific types of analysis were performed on this data. These were (a) an analysis of variance of the error data, and (b) a series of regression procedures.

ANALYSIS OF VARIANCE OF ERROR DATA

To conduct an analysis of variance on the data, they were grouped into six velocity ranges. These ranges were:

d_1	d_2	d_3	d_4	d_5	d_6
18 mph	26	35	44	52	60
19	27	37	45	54	61
21	30	39	48	55	63
23	31	40	49	57	65

Each cell entry in this analysis was the algebraic sum of that particular subject's errors in producing or estimating the four velocities in that velocity group.

Velocity was the only factor found significant at the 0.01 level. The order by time interaction and the order-time-feedback interactions were found to be significant at the 0.01 level while the order-velocity, order-time-velocity, and the order-feedback-time-velocity ABCD interactions were significant at the 0.05 level ($F_{0.95}$).

Figure 1 shows graphically the effect of velocity. Each of the six data points shown here is the mean error for that velocity range summed over all other conditions. It is apparent that the subjects tend to overestimate or overproduce in the lower velocity ranges and underestimate or underproduce in the higher velocity ranges. It is also interesting to note that the mean error appears to be zero somewhere in the d_3 (35 to 40 mph) range. A Newman-Keuls Sequential Range Test on the six velocity group means showed the following at the 0.01 level:

$d_1 > d_4$, d_5, and d_6

$d_2 > d_4$, d_5, and d_6

$d_3 > d_5$.

No other differences were found at this or the 0.05 level.

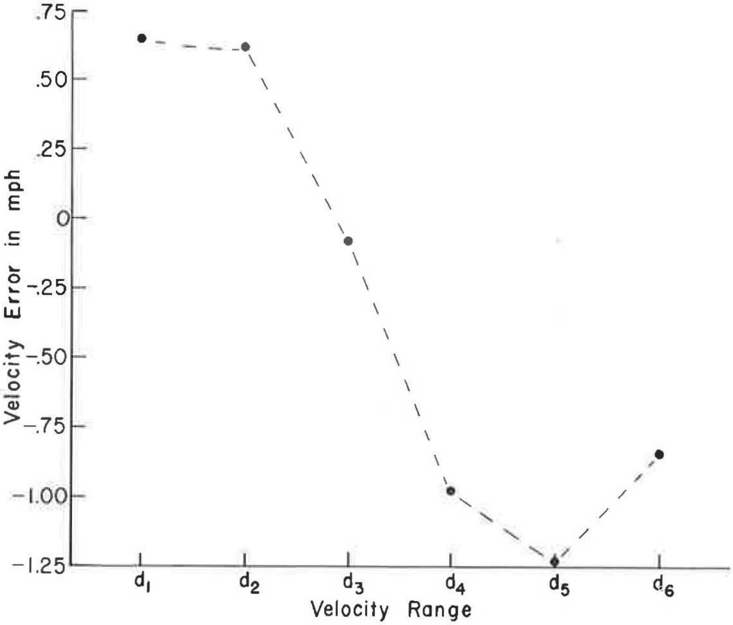

Figure 1. Mean velocity error for each of the six velocity ranges for data summed over all conditions (velocity effect).

The order-time interaction (AC) is quite apparent as shown in Figure 2. Here the positive error for velocity production and the negative error for velocity estimation is obvious as is the relative downward ("more negative") drift of the data in time C_2 with respect to C_1. This interaction is interpreted as being a method of presentation effect. The basis for this interpretation rests primarily on the fact that a differential time effect due only to the two groups of subjects is quite unlikely. Additional evidence for this comes from a Newman-Keuls Sequential Range Test which showed, at the 0.05 level, differences between means from the same group and from the same time but none from the same period. In other words, the Newman-Keuls Sequential Range Test found no difference between group means under the same method.

Figure 3 shows the order-time-feedback (ABC) interaction. Note that these data are identical with those in Figure 2 except that they have been split into two groups on the basis of the presence or absence of feedback information. Because of the interpretation given the order-time interaction, this order-time-feedback (ABC) interaction will be interpreted as that of a method by feedback condition interaction. It is apparent that the feedback conditions exert a powerful influence in that feedback greatly reduces the magnitude of the error but does not change the direction of the method effect or that of the slight time effect. A Newman-Keuls Sequential Range Test on the data showed all means to be greater than the two no feedback-estimate means while no difference could be found between these two no feedback-estimate means. This test also showed that, while the two no feedback-produce means did not differ, the no feedback-produce mean at time C_1 was greater than the two no feedback-estimate means but not greater than the four feedback means (all at the 0.05 level).

The order-velocity (AD) interaction is shown in Figure 4. This interaction was found significant at the 0.05 level. A Newman-Keuls Sequential Range Test showed the group A_2 means at both the d_1 and d_2 velocity range levels to be greater than the A_2 means at the d_4, d_5, and d_6 velocity ranges and greater than the group A_1 mean at the d_5 velocity range. No other differences were found significant. This interaction is quite interesting in that it appears that most of the velocity effect is associated with group A_2. This group received the PEPE treatment order while group A_1 received the EPEP treatment order. Apparently, some sort of transfer effect produced this result—it is unlikely that this effect is due to innate differences in the subjects.

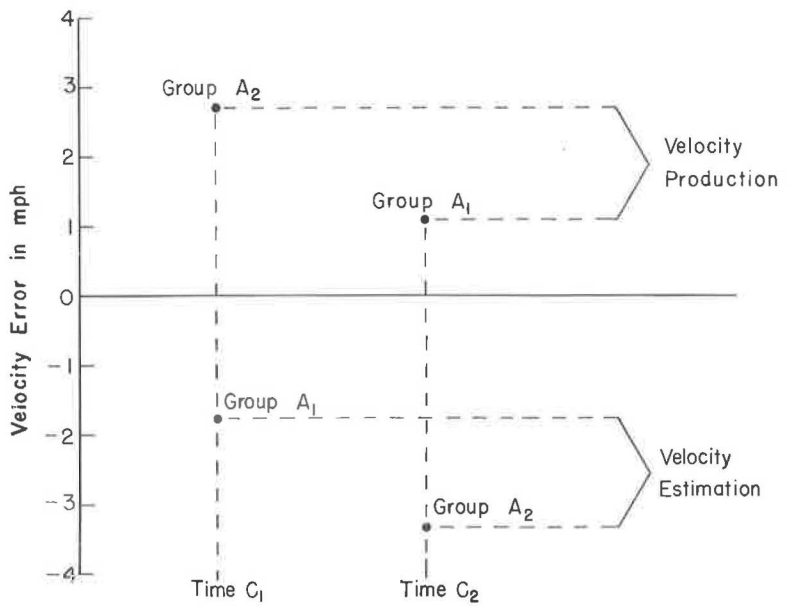

Figure 2. Mean velocity error for data grouped to show order of presentation and time interaction.

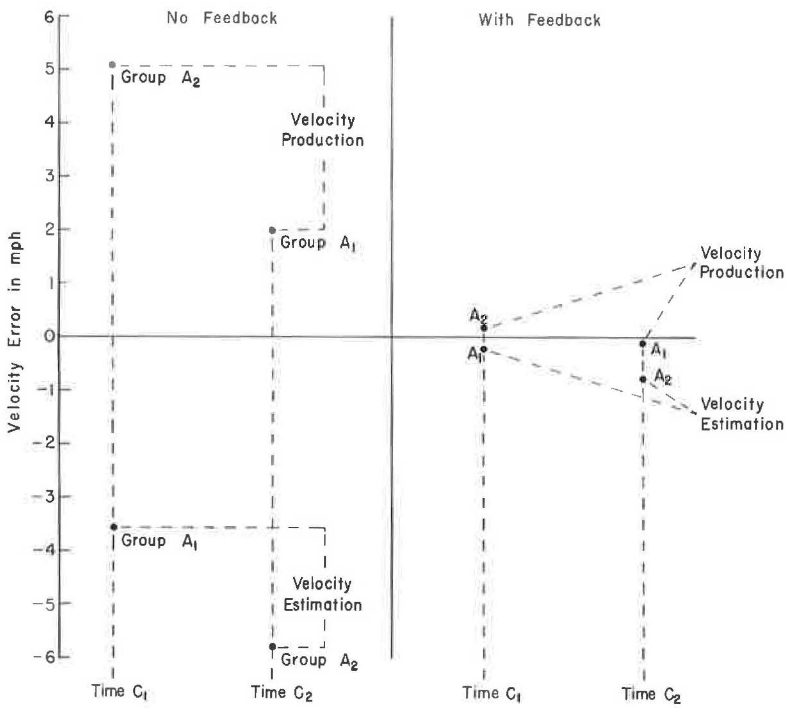

Figure 3. Mean velocity error for data grouped to show order of presentation, feedback, and time interaction effects.

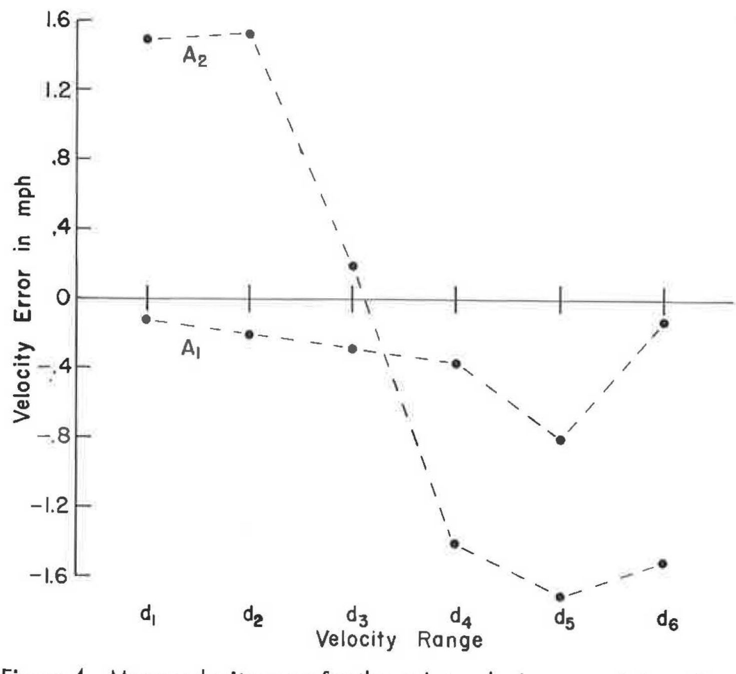

Figure 4. Mean velocity error for the order-velocity range interaction.

Figure 5 shows the order-time-velocity interaction (ACD). Here, both groups (A_1 and A_2) approach zero error in the higher velocity ranges for time C_1 while only group A_1 does so in time C_2. (Again, we are probably seeing the effects of order of presentation.)

The order-time-feedback-velocity interaction (ABCD) is shown in Figure 6. This is a method-feedback-velocity interaction, but it is presented without summing over the two groups (A_1 and A_2). The performance of group A_2 under the no feedback conditions is in sharp contrast with the mirror-image performance of group A_1 under the same conditions.

Figure 7 shows the order-feedback-velocity interaction. Here again, note the relation of group A_1 and group A_2 under the no feedback condition.

REGRESSION ANALYSIS

A standard linear regression was applied to the data from each of the 12 subject's four runs and to the data when pooled across subjects but arranged by experimental condition. Standard error scores were also calculated for these regression lines. These regression lines were fitted with a linear regression procedure which simultaneously evaluated the fit of higher order polynomials through a piecewise linear approach. This analysis indicated that the linear regression provided a satisfactory fit.

Figure 8 shows the regression lines and their corresponding 95 percent intervals for the E and P trials without feedback for group A_1 (EPEP). Note that the approximate slope of the E line is > 1 while that for P is < 1. Also, note that the two regression lines cross at approximately 40 mph.

Figure 9 shows the regression function for the group A_1 trials with feedback. It is apparent that the feedback reduced the variability about the regression line and shifted the slopes of the regression line toward a value of one.

Figure 10 shows the regression functions for the no feedback cases for group A_2 (PEPE group). Note the wider deviation of the slopes of these regression lines than for the corresponding data for group A_1. Figure 11 shows the group A_2 data for the feedback conditions. Again, note how the slope has returned to approximately one,

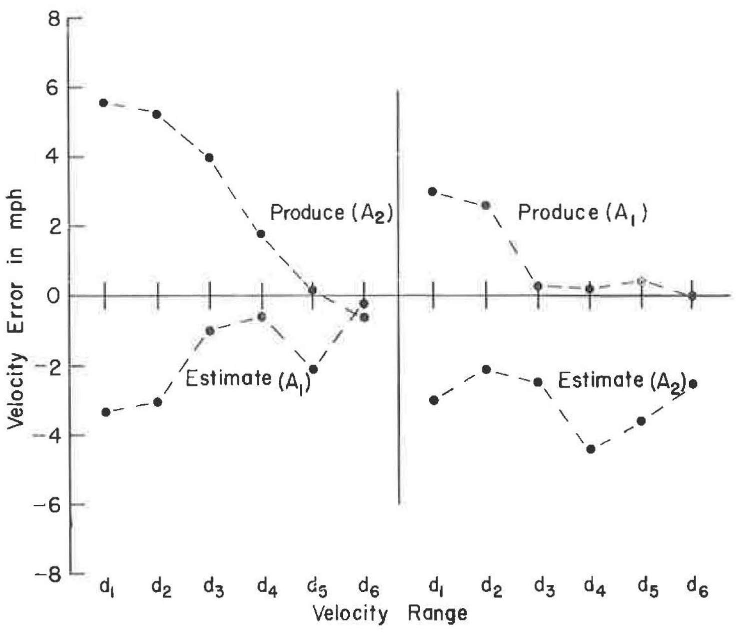

Figure 5. Mean velocity error for the order-time-velocity range interaction.

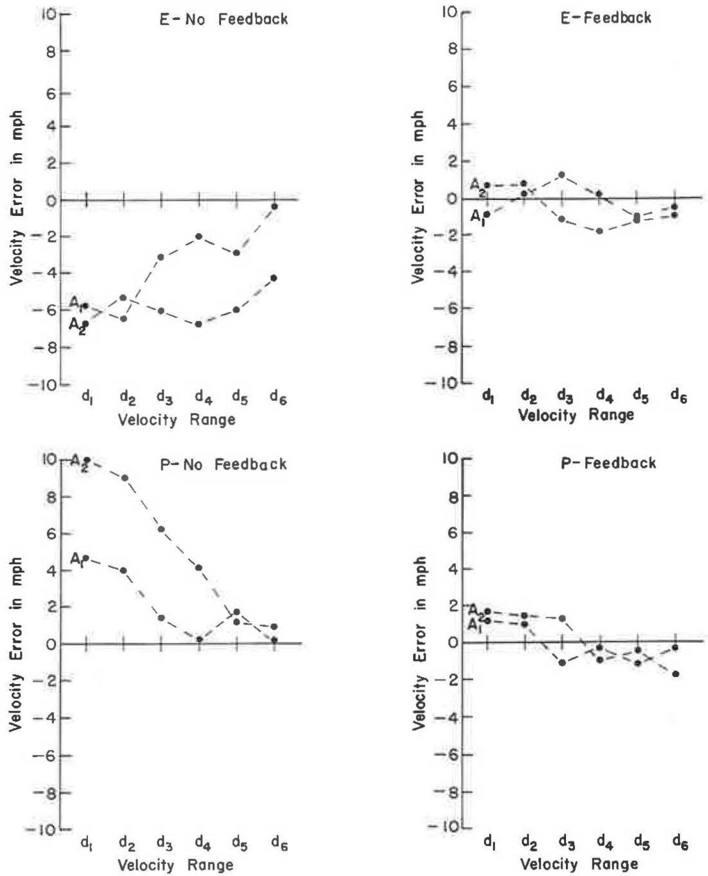

Figure 6. Mean velocity error for order-feedback-time-velocity range interaction.

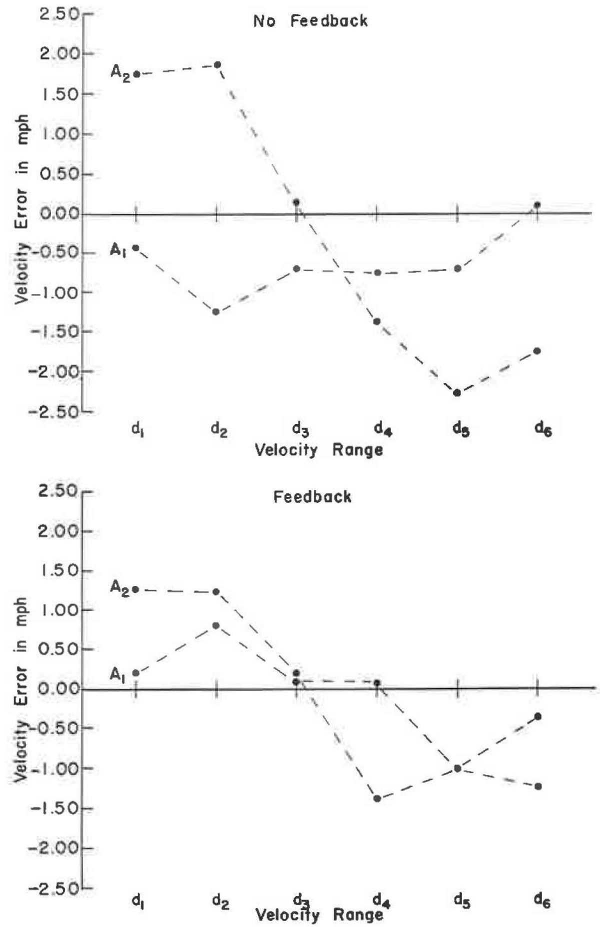

Figure 7. Mean velocity error for data grouped to indicate order-feedback-velocity range interaction.

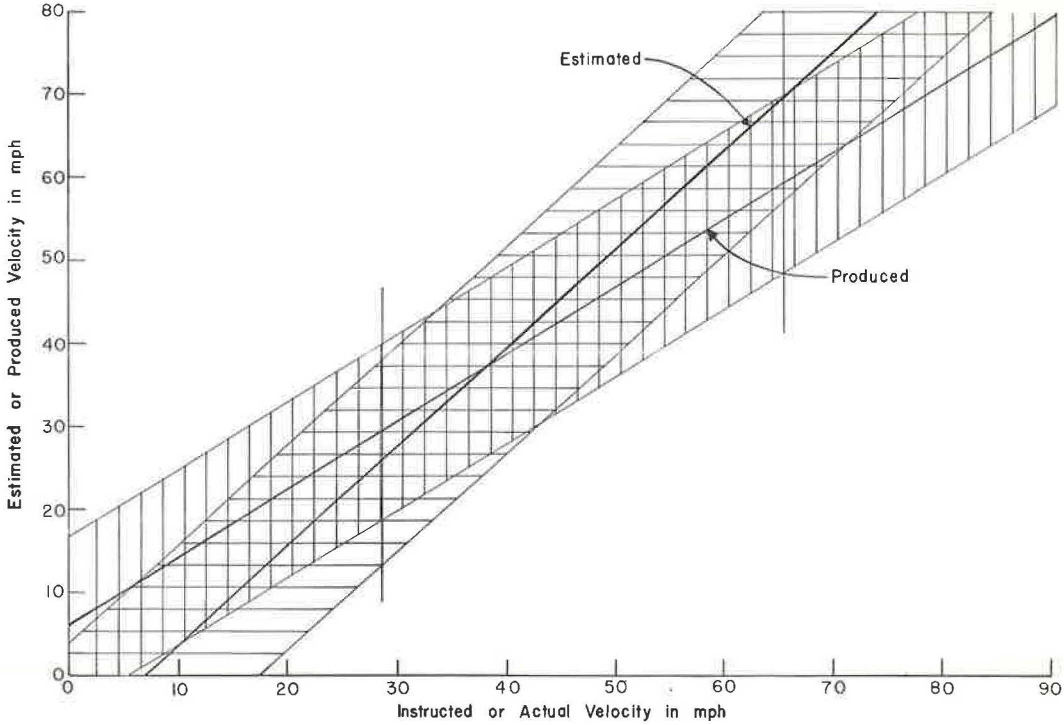

Figure 8. Regression lines and 95 percent confidence intervals for data for estimated and produced velocities, no feedback conditions for subjects 1 through 6 pooled.

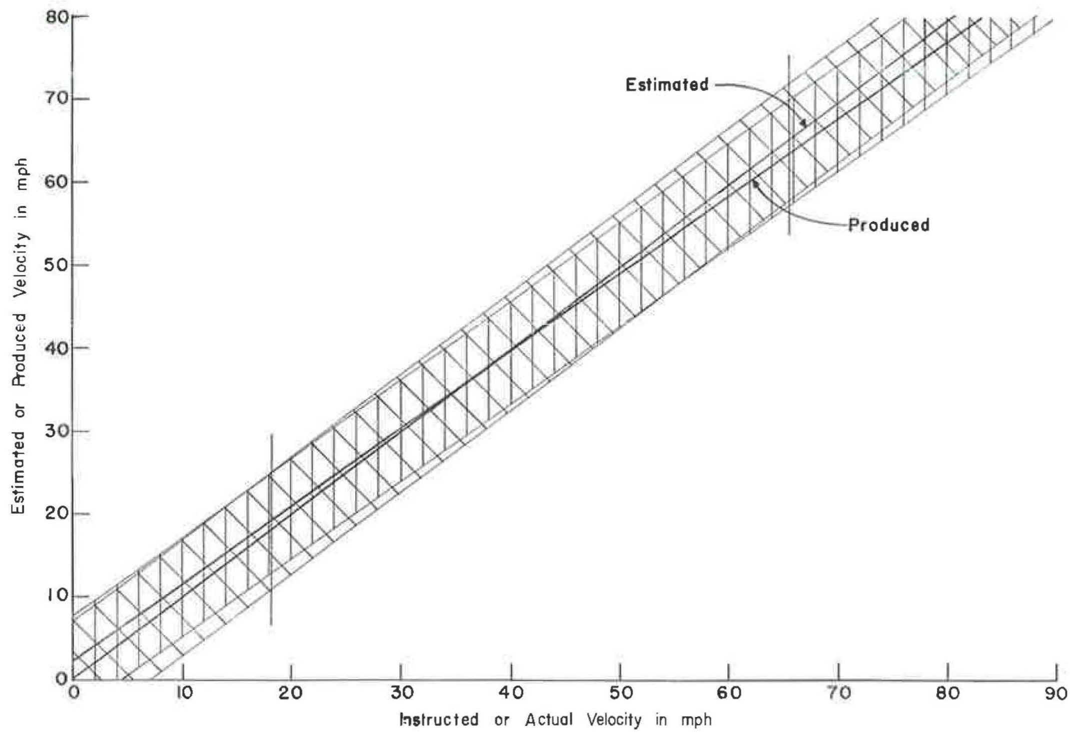

Figure 9. Regression lines and 95 percent confidence intervals for estimated and produced velocities, feedback conditions for subjects 1 through 6 pooled.

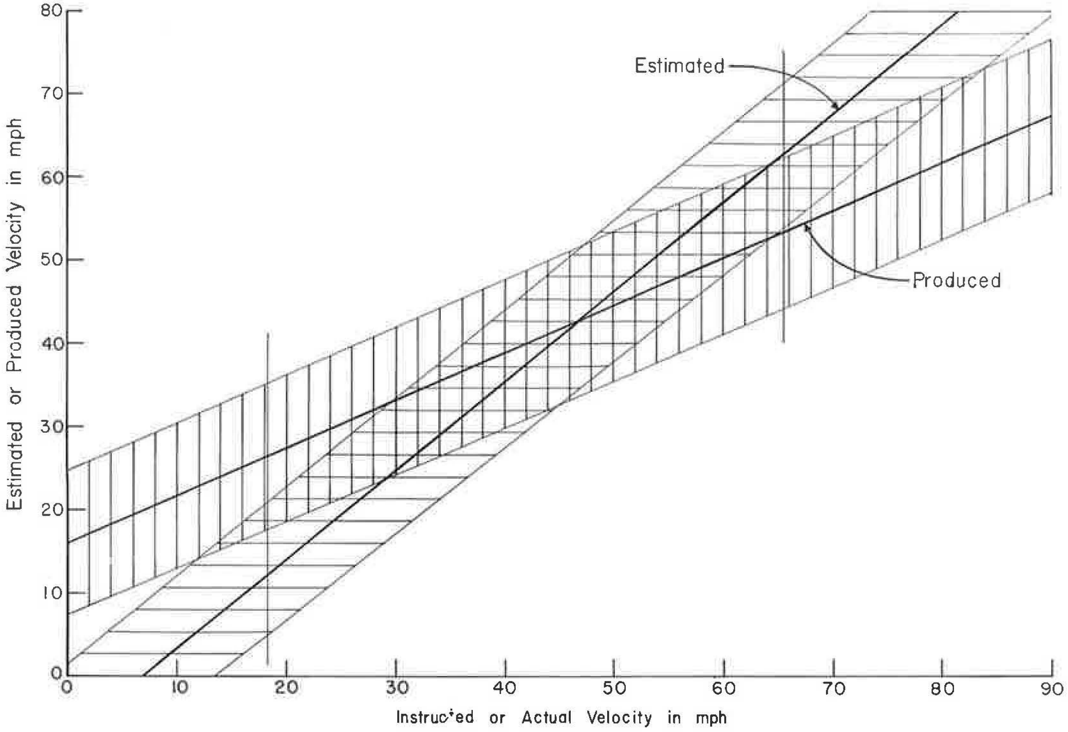

Figure 10. Regression lines and 95 percent confidence intervals for estimated and produced velocities, no feedback conditions for subjects 7 through 12 pooled.

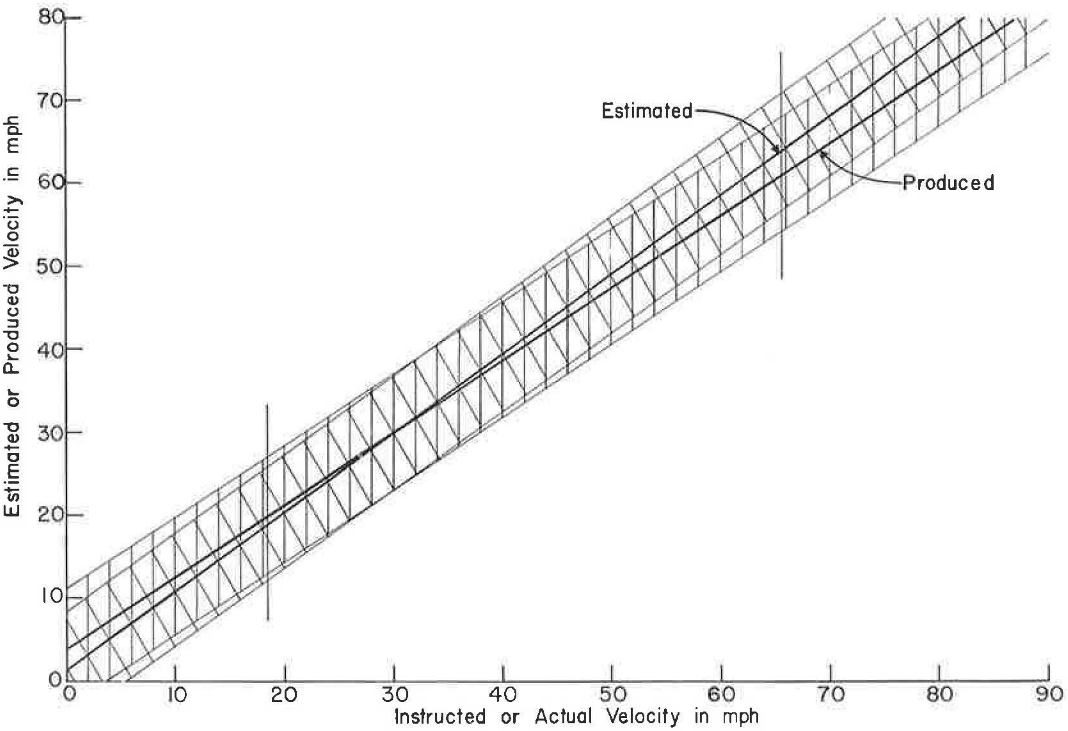

Figure 11. Regression lines and 95 percent confidence intervals for estimated and produced velocities, feedback conditions for subjects 7 through 12 pooled.

and the variance is reduced. It is interesting that in all cases the estimated values are greater than the produced values in the high velocity range (the opposite being the case in the lower velocity ranges) and that the two regression lines intersect somewhere in the 40-mph range.

The range of the standard error scores for all of the individual subject's treatment regression lines were from 1.92 to 5.38 mph with a mean score of 3.09 mph. This compares with a range of from 3.28 mph to 6.25 mph with a mean of 4.27 for all data when pooled. In addition to these analyses, a number preference analysis and a transition analysis were performed. The number preference analysis failed to find a significant bias on the part of the subjects for particular numbers. The transition analysis also failed to find a significant effect due to the magnitude of the various velocity changes which were required from trial to trial.

In general, this study has involved the construction of scales for man's sensing of velocity. Two psychophysical methods have been employed which are roughly the antithesis of each other. These are the method of magnitude estimation and the method of magnitude production. Stevens (1) reports that these two techniques may well be used together in such a way that the biases innate to each are offset by the other. It would appear that this is the case.

For those subjects who received the EPEP treatment order, the two no feedback regression lines (Figs. 8 and 9) do, in fact, intersect the "correct response" line at approximately equal angles (i. e., have the same error magnitude for a given instructed or actual velocity). For the two cases with feedback, little error exists between the two regression lines and the slope 1,0 intercept line. These observations do not appear to hold for those subjects who received the PEPE treatment order. With this order, under the no feedback condition, the slope of the regression line for the P instruction is very small, reflecting an extreme tendency to underproduce in the high velocity range, whereas the regression line for the E instruction is approximately where it fell in the EPEP order. There is little difference between the performance of those two groups under the feedback condition. Also, little difference is observed in the S. D. error of the data about these regression lines, other than for the large reduction associated with feedback.

The small slope of the P no feedback regression line for group A_2 may well be due to a mistaken use of the cues available to the subject. The subjects may rely only on cues that are associated with the vehicle until they are required to estimate the vehicle speed. When the subject estimates rather than produces, he apparently makes better use of the information which he has available to him. By so doing, he "learns" to better perform the task and continues to do so. Thus, this effect would not be apparent for those who received the EPEP combination because they estimated the vehicle velocity before producing it. This explanation would also account for other effects because under the feedback trials, both the group A_1 and group A_2 subjects would have previously experienced an E trial.

A characteristic illustrated in Figures 8 through 11 is that the regression lines all cross in the mid-velocity range (40 mph). This may be accounted for by the fact that the vast majority of driving is at speeds in this range. Thus, experience may well have produced "anchor points" in this speed range.

CONCLUSION

This paper has presented a study of the ability of automobile drivers to sense one of the many types of information which is available to them. The use of other techniques, such as stimulus change detection, is necessary to fully define this sensory capability. This study is one of a series of investigations aimed at defining the capability of drivers to sense that information which is necessary for longitudinal vehicle control. Investigations are currently being conducted on the ability of drivers to sense velocity (as reported here), acceleration, jerk, headway, and relative velocity. It is felt that by knowing the sensory capability of drivers, it will be possible to better predict the performance of drivers under such conditions as the high density freeway and that it will be possible to evaluate the possible benefits of augmenting the driver's sensory capability.

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1. Stevens. Problems and Methods of Psychophysics. Psychological Bull., Vol. 55, No. 4, July 1958.

Effects of Discrete Headway and Relative Velocity Information on Car-Following Performance

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•ONE WAY to increase the traffic flow on crowded highways is to decrease the time headway between cars. However, as spacing between cars is made smaller, the probability of rear-end collision and resulting chain reactions could increase because the cumulated reaction time of the drivers could induce the so-called shock wave in the flow. The reaction time here consists of the driver making a control movement after he perceives a change in spacing and then making a decision which will correct any deviation from his desired spacing.

It is hypothesized that the time it takes the driver to perceive a change in the car-following environment could be decreased by presenting the driver with information before he visually perceives such a change. Stated differently, our concern is whether or not subthreshold information and/or higher derivative information would be advantageous to the driver in the car-following mode. In recent research at Ohio State University, Rockwell and Snider (4) showed that there are driver thresholds for headway, velocity, and acceleration. The problem of determining the threshold of headway, for example, is complicated by the fact that the threshold is dependent on the initial headway (headway is defined as the separation distance between the rear of the lead car and the front of the following car) and also the acceleration and possibly higher derivatives of lead car performance. If some type of display could present information of a change in the relative spacing of the two cars before the driver could perceive it, perhaps quicker control action could be made by the driver of the following car and thus the spacing of the cars could be maintained more closely to their target level. If this display also gave information as to the required directionality of control motions to be taken, then the driver decision time could be further decreased and the hazards associated with closer spacing reduced.

Relatively few studies have been conducted in this area of driver aids. Bierley (1) conducted a study to establish the effect on spacing control of more information about vehicle headway and relative speed. The effects of two types of information displays were tested. In the first, a meter showed the driver current headway. The driver was asked to maintain a constant headway of 80 feet. It was concluded that the spacing display significantly reduced the average spacing error and reduced even more the error variability over that of the no-display condition. The fact that error variability was decreased suggests that this would help increase traffic flow by maintaining a more stable flow condition. In the second display, the relative velocity between the two cars was algebraically added to the relative spacing signal. With the velocity-aided spacing display, Bierley found that performance was improved over the spacing display. There was a greater reduction in the average spacing error variance, maximum absolute spacing change, and driver reaction or detection time. One interesting result noted in this study is that drivers tended to undercorrect more often during deceleration than during acceleration conditions.

Wright and Sleight (5) conducted car-following experiments with the use of simple judgment aids. One aid was a board in which acuity fusion stimuli were used. At close

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distances the driver could differentiate members of pairs of bars on a board mounted on the rear of the lead car. At farther distances each pair seemed to fuse into a single strip. This aid resulted in significantly lessening the tendency to follow at a greater than requested distance.

Michaels and Solomon (3) conducted a study in which advance warning of acceleration or deceleration maneuvers was given the driver of the following car. The advance speed information was given at one of four time intervals before the onset of speed change. The display consisted of four lights, two for acceleration and two for deceleration, mounted on top of the lead car. It was found that such communications reduced the variation of headway. It was further shown that minimum headway and headway variance were obtained at warning intervals of 1 to 3 seconds depending on the speed at which the test cars were traveling.

EXPERIMENTAL PROCEDURE AND EQUIPMENT

The above studies indicated that a display presenting changing information before the driver can directly sense the change, might improve car-following performance. Accordingly, a research program was established to begin exploratory investigations of an information display for car following. This display would provide information of a discrete nature as to whether the drivers' headway or relative velocity performance were within a bandwidth or error tolerance established by the experimenter or elected by the driver himself. A red light gave the driver information that his performance was outside the lower limit of the bandwidth and required a release of pressure on the gas pedal in order to reestablish the target headway and/or relative velocity. Similarly, a green light indicated an error in the opposite direction. Figure 1 shows the general display information given to the driver for the close target headway condition.

The main task of the driver was to hold the given headway throughout the experimental task, which in terms of the discrete display was a null or keep the lights out during the maneuvers. Primary research interests were whether too small a bandwidth might cause oscillations about the target value and lead to over-response on the part of the driver and conversely, whether too wide a range would not give enough information to improve system performance over a no-display condition. Table 1 shows the various experimental conditions that were employed in the research using both the discrete relative velocity signal information and the discrete headway information.

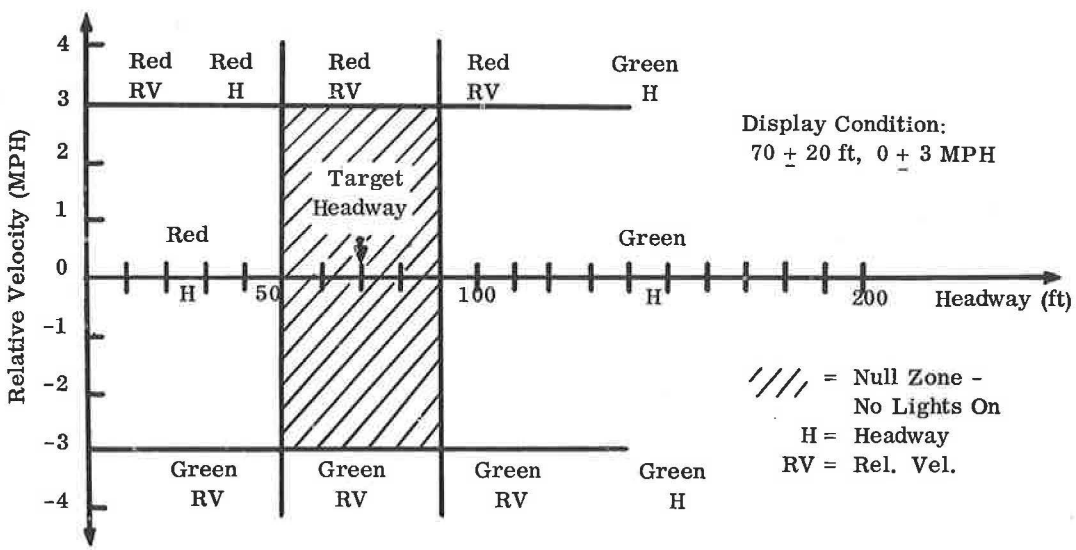

Figure 1. Schematic description of when the display lights come on.

TABLE 1

TREATMENT COMBINATIONS FOR THE EXPERIMENT AND SYMBOL EXPLANATIONS

RV Display	H Display: Close SH	Close MH	Close LH	Close NH	Far SH	Far MH	Far LH	Far NH
SR	x		x		x		x	
MR	x		x		x		x	
LR	x		x		x		x	
NR	x	x	x	x[a]	x	x	x	x[a]

H - Headway
NH - No H Bandwidth
LH - Large H Bandwidth (±27 or 34 ft)
MH - Medium H Bandwidth (±17 or 22 ft)
SH - Small H Bandwidth (±6 or 9 ft)

RV - Relative Velocity
NR - No RV Bandwidth
LR - Large RV Bandwidth (±5 mph)
MR - Medium RV Bandwidth (±3 mph)
SR - Small RV Bandwidth (±1 mph)

[a]The No-display condition was replicated once.

The equipment used in this experiment included a lead car which was equipped with a Stewart Warner Auto Co-Pilot device, mounted near the driver and used to dial in the speed required for the lead car maneuver during each trial run. This device can usually maintain relatively uniform acceleration, deceleration, and constant speed with less variance than a driver can obtain. It is used to keep the lead car maneuvers as fixed as possible from trial to trial and subject to subject. The lead car went through a series of four different random velocity profile maneuvers which were randomly sequenced throughout the experiment so that lead car velocity variance did not affect experimental results. The velocity range was from 35 to 65 mph for the lead car. The following car was a 1963 Chevrolet (Fig. 2) equipped with an automatic transmission. In the back seat of this car were located the recording equipment, control console, and experimenter seat (Fig. 3). Speed, acceleration, gas and brake pedal movements, steering wheel movements, headway and relative velocity were recorded on the 51-channel oscillograph recorder. Figure 4 illustrates the primary recorder data—velocity headway, target bandwidth, and light actuation periods. Experimental difficulties resulted in the loss of gas pedal and steering wheel movement data. The loss of the former was unfortunate as it prohibited any analysis of the frequency and magnitude of driver response to headway change and signal light actuation. Thus, performance is limited to the "driver-vehicle" complex as opposed to the effect of information on specific driver control changes.

A modified General Motors mechanical take-up reel was used to measure headway and relative velocity. This device, mounted to the front bumper of the following car, used a fine wire about 500 feet in length. The wire was fastened to the rear bumper of the lead car and constant tension was maintained by a slipping friction clutch. The distance between the two cars was measured by a set of potentiometers geared to the reel shaft. In addition, the relative velocity of the two vehicles was measured through a tachometer generator operating off the shaft. The two cars maintained communication by two-way radio; the experimenter wore a headset and throat microphones so that his instructions to the lead car would not be heard by the subject in the following car.

Figure 2. Following car with mechanical take-up reel.

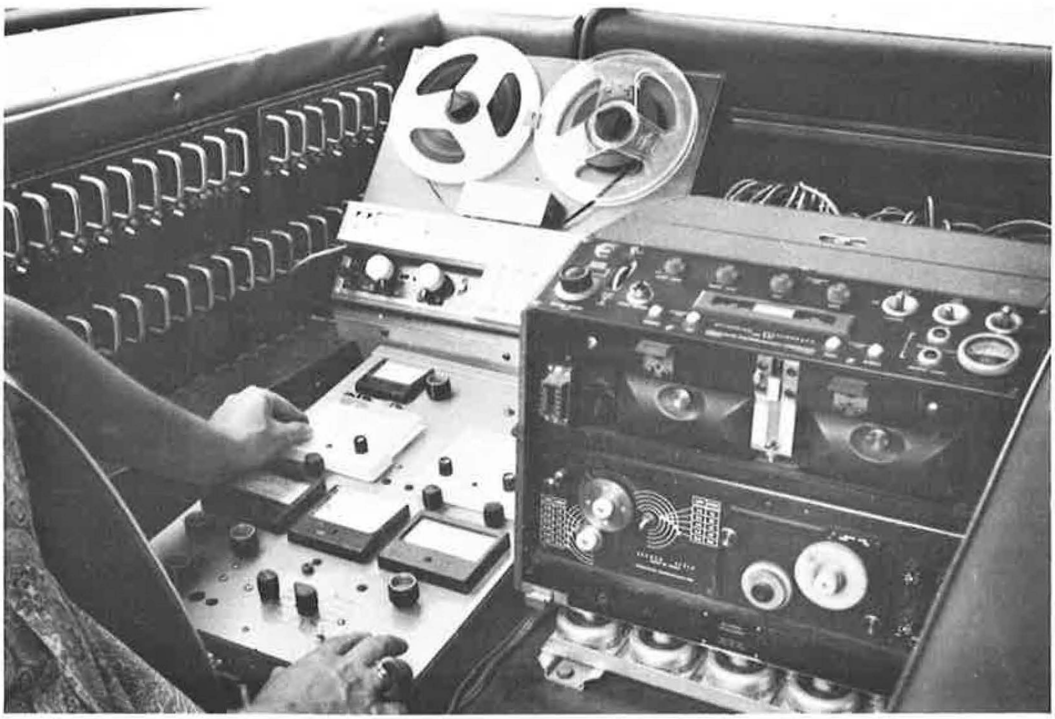

Figure 3. Control console in back seat of following car.

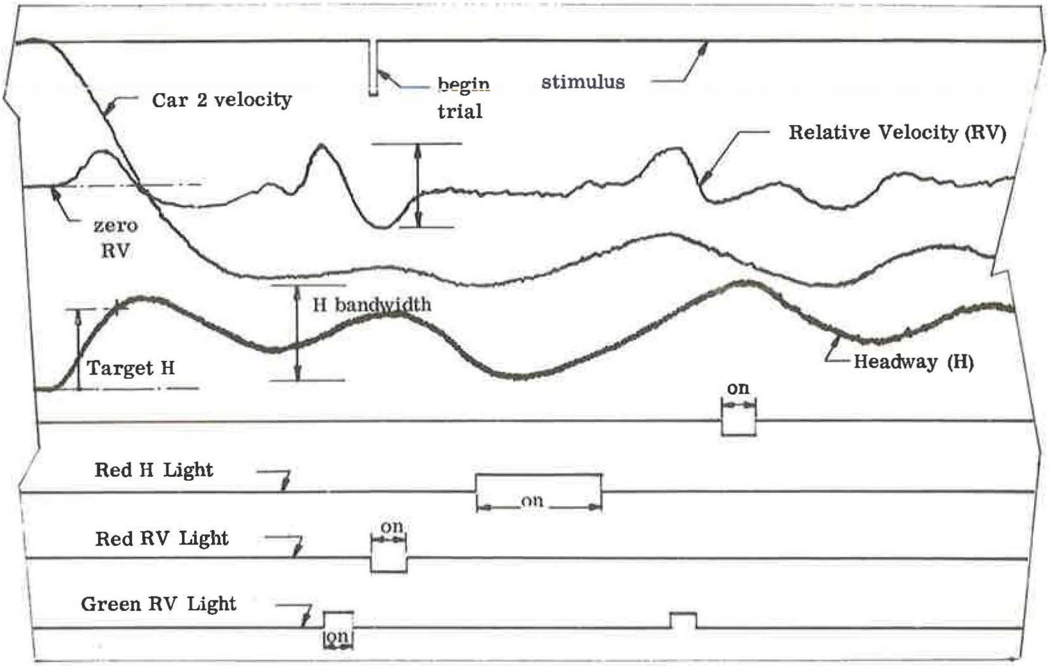

Figure 4. Example of chart paper output (not to scale).

Figure 5. Display lights on dashboard of following car.

With his control console the experimenter had relative velocity and headway information and could set the bandwidth selection for any given trial. Tests were conducted in light traffic on a limited-access highway.

The display (Fig. 5) consisted of four lights located in front of the subject on the dash board in the following car. The research was conducted at two target headways, a close headway which was approximately 70 ft and a far headway which was approximately 170 ft. The bandwidth for the headway display was approximately ±6 ft, ±17 ft, and ±27 ft for the small, medium, and large headway bandwidth at the close target headway and ±9 ft, ±22 ft, and ±34 ft for the far target headway. The relative velocity bandwidths, for both close and far target headways, were established around 0 miles per hour at ±1 mph, ±3 mph, and ±5 mph. It should also be noted that for a control condition, the drivers drove without the benefit of the displays.

The experiment was not a complete orthogonal design because all combinations of variables of interest were not included; specifically, the relative velocity display was not used by itself, as the main task for the driver was to maintain a constant headway. The no-display condition for both the close and far target headway conditions were the first two trials for each subject. The next 18 trials were in random order of the treatment combinations of target headway, headway display bandwidth, and relative velocity display trials. In the last two trials, runs 23 and 24, the subject was asked to select the headway bandwidth that he preferred for both the close and far conditions (Table 2). Each trial lasted approximately 9 minutes while the lead car went through the velocity change maneuvers. Data were sampled every 2 seconds during this period. Four college-age male drivers were used as subjects in this research. The intent of the research was to explore the effects of these display aids in some depth rather than to try to establish the effect of displays across a more representative sample of the population.

The subject driver was given approximately a half hour to get the feel of the car, driving from the laboratory to the test site. The speedometer was covered throughout the experiment. The subject for each trial was homed in, at either the close or the far target headways, and when he replied that he felt he could establish this distance visually, the experimental trial would begin. Five-minute rest periods were introduced approximately each half hour of driving. Although the subject was given information as to which display he would receive, he was not given information on the bandwidth of the headway and relative velocity aids, except for the last two trials in which the subject would establish by trial and error basis the bandwidth most comfortable to him. The accuracy in the headway bandwidth was less than ±2 feet. The percent of time the lights were on during the various trials for subject 2 is shown in Table 3.

TABLE 2

H BANDWIDTH SELECTED BY SUBJECTS

Subject	Target Headway	
	Close	Far
1	17 (MH) feet	16 (SH/MH) feet
2	6 (SH)	16 (SH/MH)
3	6 (SH)	22 (MH)
4	6 (SH)	16 (SH/MH)

Note: ( ) indicates approximate experimental H bandwidth condition.

TABLE 3

PERCENT OF TIME DISPLAY LIGHTS WERE ON DURING TRIALS—SUBJECT 2

Bandwidth Combinations	Close		Far	
	H	RV	H	RV
SH-SR	24%	16%	65%	61%
SH-MR	27	2	42	16
SH-LR	8	0	55	2
SH-NR	38	—	54	—
MH-NR	2	—	39	—
LH-SR	3	24	9	42
LH-MR	2	5	19	12
LH-LR	3	0	22	4
LH-NR	3	—	27	—

Note: Each trial is approximately a 9-minute run.

RESULTS

Before interpreting the results of this experiment, analysis was made to determine whether or not the lead car programs were effectively the same for each driver on each trial. These statistical tests indicated that effectively there was no significant difference among the lead car velocity programs from subject to subject. Statistical tests were also conducted to determine whether or not there were any significant learning effects during the research testing period. Tests were made to compare the first two no-display runs at close and far distance target headway with the replication later near the end of the experiment. There were no statistically significant differences in headway variances. With subject 4 the second 100 sample points of each of the 18 display combination trials were used to calculate the headway variance. These were then compared with the headway variance of the same trials calculated from all sample points available (samples were taken at 2-second intervals). There was no significant difference between the headway variances computed by these two techniques. Again, while we cannot be absolutely sure that there was no learning effect, it appeared to be minimal, particularly when viewed against the main effects of the experiment.

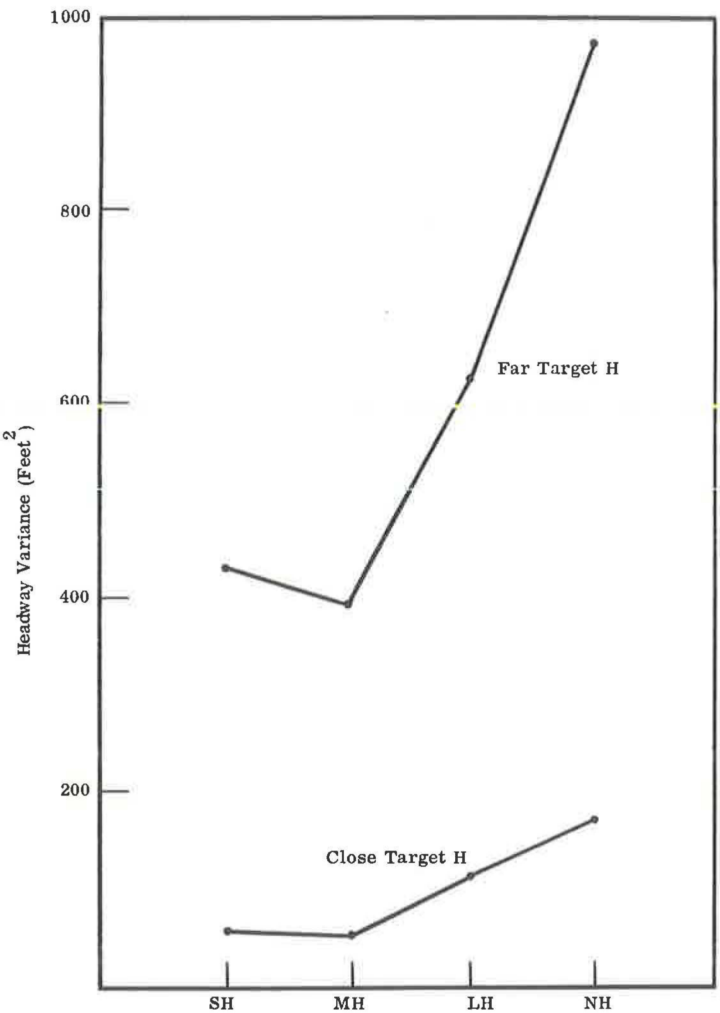

Figure 6. Headway variance for H bandwidth and target H over all subjects.

Analysis of variance was used to test the data. Subjects, bandwidth, and type of display were considered as major independent variables in the experiment as well as the initial target headway. The dependent variables were the headway variance and relative velocity variance over the trial. Figure 6 shows the general results of headway display bandwidth selection on headway variance for the two target conditions over all subjects. It had previously been established through statistical tests that subjects were not a significant effect in the analysis using headway variance and, thus, subject data were pooled in the case of headway bandwidth. It was interesting to note that there is a significant reduction in headway variance with the use of the headway display alone (over 60 percent for both target headways). Figure 7 shows the effects of the various headway bandwidths on relative velocity variance. Since in this case the individual subjects were slightly statistically significant, their data are shown individually. A general trend exists in which the relative velocity variance is actually larger with decreasing error bandwidth. This is probably explained by the frequent corrections made by the driver in maintaining his primary task, that of target headway, with subsequent resultant changes in relative velocity. Figure 8 shows the effects of relative velocity bandwidth displays on headway variance for all subjects. The variance is the average for the two headway bandwidths used with each RV display. Again it is noted that the headway variance decreases with smaller bandwidth for the far-target condition. At the close-target condition, there appears to be little advantage to the use of the RV display system.

Figure 9 depicts the general trends of results of the entire experiment showing the two criteria, the two target headways, and the display conditions involved. For this figure, the data are pooled for all subjects even though subjects were found to be statistically significant for a few of the cells. Results are described in comparison with performance under the no-display condition. As can be noted in a few instances, particularly at close headways with relative velocity variances as the criterion, the displays reduced performance (i.e., increased the variance). However, for the large

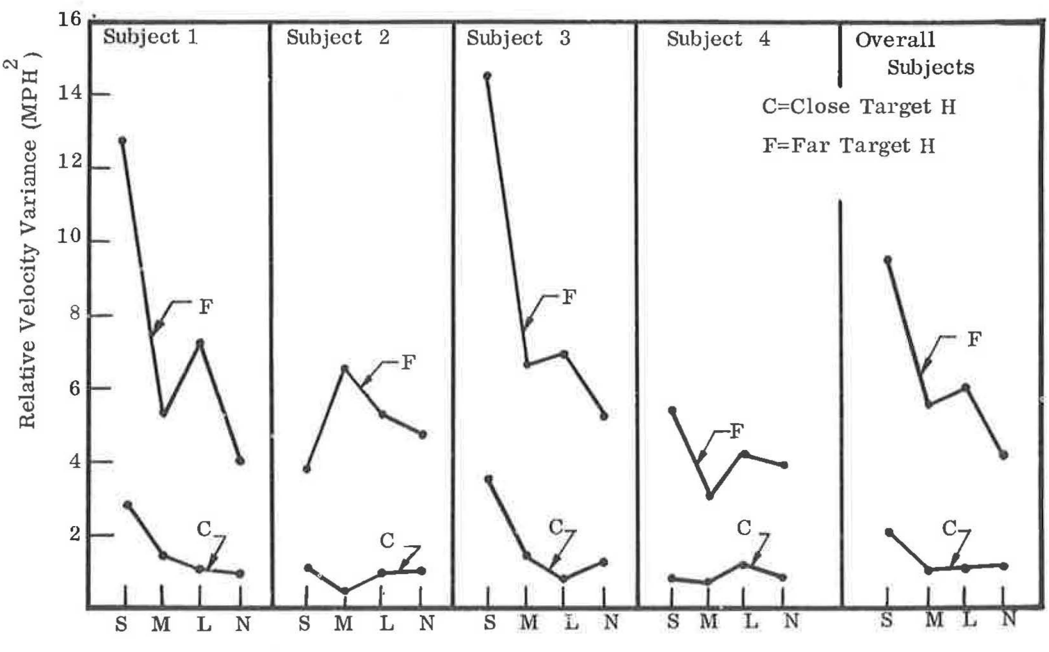

Figure 7. Relative velocity variance for H bandwidth and target H for each.

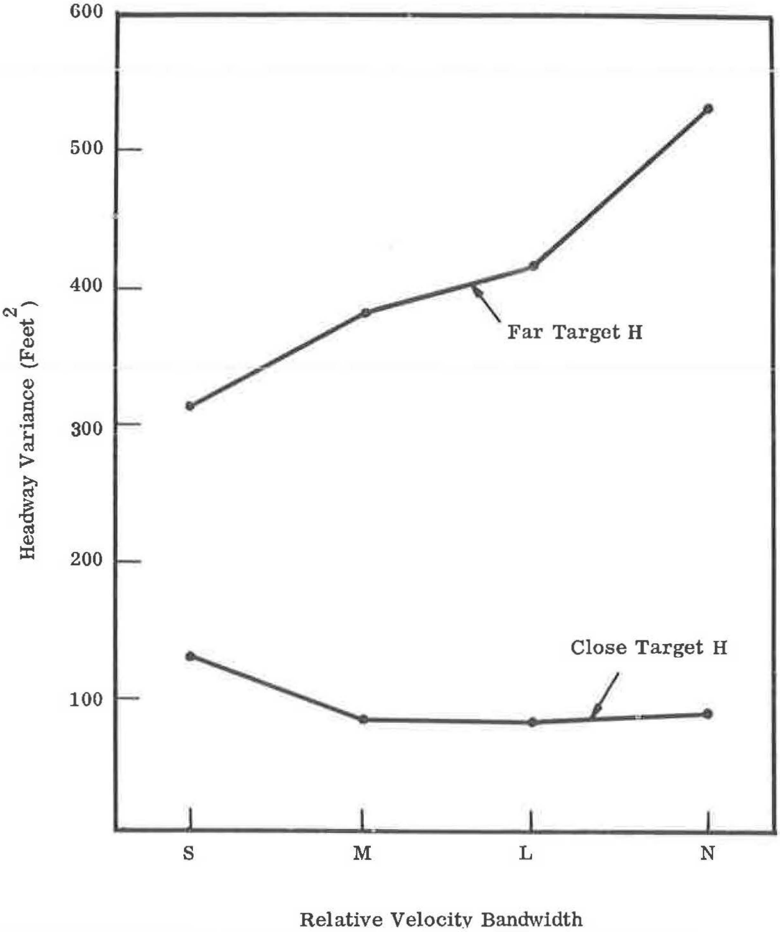

Figure 8. Headway variances for target H and RV display interaction (variances averaged for the 2 headway bandwidths used).

majority of conditions, the use of displays led to improved car-following performance. It is also evident that average car-following performance is much better at close target headways than at far headways and that display improvement appears to be less evident here. This may come from the fact that at close headways the driver's sensory capability for headway and relative velocity detection is better and that he tends to rely more on his perceptual processes and less on the display at these more risky car-following conditions. In general, for the entire experiment target headway was highly significant, bandwidth was usually significant, and subjects were slightly significant as found in the analysis of variance.

Finally, in regard to the elected bandwidth that the subjects would select voluntarily (Table 2), it is interesting to note that although the subjects need more headway information at the far distance, they are apparently willing to sacrifice sensing information for comfort. That is, they will accept less headway error information so that they will not have to make so many control corrections.

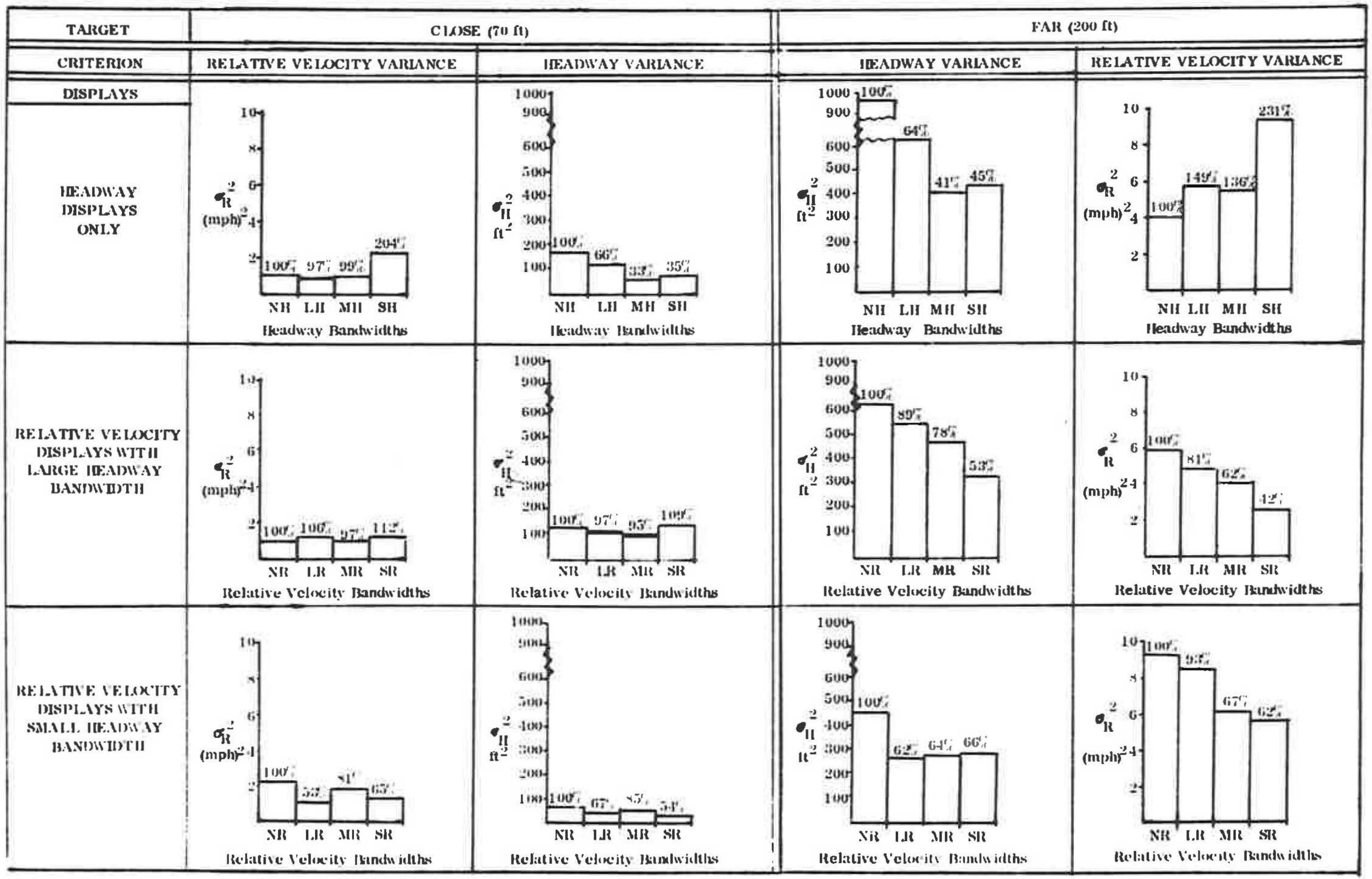

Figure 9. General trends of experimental results (data pooled for all subjects; all percentages in terms of the no-display condition).

CONCLUSION

On the basis of this exploratory work any generalizations are tenuous. The research does indicate that a discrete light display presenting headway and relative velocity information can improve car-following system performance. Over 60 percent reduction of headway variance can be obtained by using a headway bandwidth display alone at both target headways. Overall, the medium headway bandwidth display produced the best reduction in headway variance. The addition of the relative velocity bandwidth display reduced headway and relative velocity variances up to 47 percent and 58 percent, respectively, at the far distance. The relative velocity display in combination with large headway bandwidth appeared to have little effect at the close target distance. However, with small headway bandwidth the same general improvement as found earlier is seen. There appears to be no single display combination which is clearly optimal for the experimental conditions, although the SH-SR combination would be a leading candidate.

If it is assumed that headway variance is an important performance criterion and that close target distance is more realistic for increased traffic flow, then the best display combination would appear to be the small bandwidth for both displays. However, the absolute performance improvement at the close target headways seems rather small. It appears that at the close target headway conditions, the driver may prefer to use other cues than those provided by the display.

Future research suggests a larger treatment of this experiment and, perhaps, the use of time-delayed information to prohibit oscillations in control whereby the driver would be out of phase with the actual conditions between the two vehicles. It might also be fruitful to provide higher order derivative information on preview-type displays which measure traffic dynamics upstream.

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Analytical Performance of an Airstop Restraint System in An Automobile Crash

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An analytical representation, based on a haversine acceleration function, is presented for: (a) the passenger compartment in automobile-barrier crashes; (b) car-to-car, both moving head-on; (c) car-to-car one-stationary side and (d) car-to-car one-stationary rear crashes. Characteristics of the airstop restraint (to be used initially for passengers but not the driver), consisting of a chest airbag automatically inflated just before a crash to 0.1 to 0.3 psig, an inflated airseat, and in some applications a lap belt and inflated foot airbag, are presented on the basis of data for swing and drop tests, and experimental airplane crash tests. In this early period of restraint improvement without modifying car controls, we suggest that the driver wear a more rigid restraint—lap belt and shoulder straps—permanently in place. A preliminary damped elastic model of the airstop system, having 1820 lb/ft spring constant, 3-cps resonance frequency, and 22 percent of critical damping, is utilized. For the most severe crash case, the barrier impact, it is shown on the basis of these models that the airstop restraint would convert a 30-mph "probably fatal" impact with present structures and lap belt (but without upper body support) into a crash without injury if two feet of motion into the airbag system without bottoming were available. If cars cannot have this distance for controlled load isolation, it is suggested that a lap belt be used with a less deformable airseat, so that the system becomes essentially an upper body restraint addition to present lap belt restraints.

•FOR THE past three years we have been experimenting with the use of inflated airbag restraint systems (1), partly under NASA contract, for crash and vibration load isolation, in space and aircraft applications. We have shown the possibilities for significant protection from these loads by tests in four vertical drop devices, three vibration devices, a swing impact device, a DC-7 experimentally crashed for the FAA by the Aviation Safety Engineering and Research (AvSER) group, Flight Safety Foundation, and two C-45's and a CH-21 helicopter experimentally crashed for the Army by AvSER.

We have not been able to get support for automobile airbag restraint work, in part perhaps because government support for car safety research for the 150 million or so Americans who use cars is only a small percentage of the support for airplane safety research, or, indeed, for research on the safety of the few dozen astronauts. Yet clearly, the deaths in car accidents of some 50,000 Americans a year, the significant

Paper sponsored by Committee on Highway Safety.

injury of perhaps 40 times this number, and the statistics that cars kill far more people than disease in the age span from about five to twenty-nine, and that one-fourth of all cars will be bloodied in an accident—all these facts make it the responsibility of all those doing crash research to look for applications of their work to automobiles. We are therefore developing an analytical method for estimating what airbag restraints, and the airseat restraint included in the full "airstop" system, might do for passenger protection in automobile crashes. In this preliminary period, we recommend that the driver wear more rigid restraints (lap belt and shoulder straps at present) so that inflation of the bags will not jeopardize his efforts to avoid the crash. This early system is to "grab the wife and kids" (the passengers) with full body protection, instead of counting on the crumpling dash and instrument panel, pop-out windshield, etc., to turn death into injury decelerating these people even supported by lap belts. Airbags provide upper body support.

We understand that both Ford and General Motors have considered airbag restraint systems or inflated dashboards, etc., and have rejected them, due in part to the hazard to a child of being thrown into the back seat if leaning on the dash when it is explosively inflated following bumper contact in the crash. This work apparently has not been published. We expect to utilize signals from abrupt braking or turning, wheel slipping off the road, or the car position information of the highways of the future to anticipate a potential crash situation and inflate the bags more slowly, with switch-actuated automatic roll-up if the crash is avoided. A theoretical analysis by the Road Research Laboratory of Great Britain (2), using what we call the rigid piston adiabatic compression model of airbag performance, led to their airbag system rejection. Our bags are not rigid. We find a little experimentation highly salutary to preliminary and too often pompous theory. We hence warn that what follows on automobiles is a work of the head and not of the field—reader beware.

LOAD ISOLATION BY AN ELASTIC SYSTEM

Figure 1 shows the load on a mass mounted by a spring and damper to a support displaced by varying periodic motions. For the resonance period, the load on the mass is several times that on the support. For periods more than two or three times the resonance period, the mass and support will have the same peak loads. It is for periods of impact less than one-half or one-third of the resonance period, or for frequencies two or three times the resonance frequency, that load isolation occurs, with the load on the mass being less than the load on the support due to its greater distance of travel.

For passenger compartment impact periods of automobiles of 0.1 to 0.2 sec we need, for passenger load isolation, an elastic restraint resonance period longer than 0.3 to 0.6 sec or a resonance frequency lower than 3 to 1.8 cps.

This isolation is attained only by motion into the restraint system, and the motion must not "bottom" on support structures. If it appears that the applied load is such that bottoming will occur, then the elastic restraint must be stiffened, shortening the resonance period. If this shortening is such that the resonance period approaches the duration of the impact period, then we must abruptly stiffen the restraint so that τ_{res} is about one-third of τ_{impact}, so that the man is essentially in a rigid restraint and receives no more load than the vehicle. We would like the restraint to be totally out of the way when not needed, yet automatically, rapidly, and safely applied when a crash is imminent.

The elastic system will also rebound. In our early work, we experimentally compared crash sensations when using a large valve system that dumped bag pressure near peak displacement, reducing rebound, and sensations when leaving the bags inflated. Because of the low resonance frequencies of these bag systems (below 3 cps and hence below internal body resonance frequencies) and the observation that damping even with unobstructed bag interior design was such that only two or three cycles of rebound occurred, the crash is experienced not as the slam of the external structural but a slight multiple pat by the restraint, of very short duration. Hence, it is quite acceptable to leave the restraint inflated, giving us the major advantage of being pro-

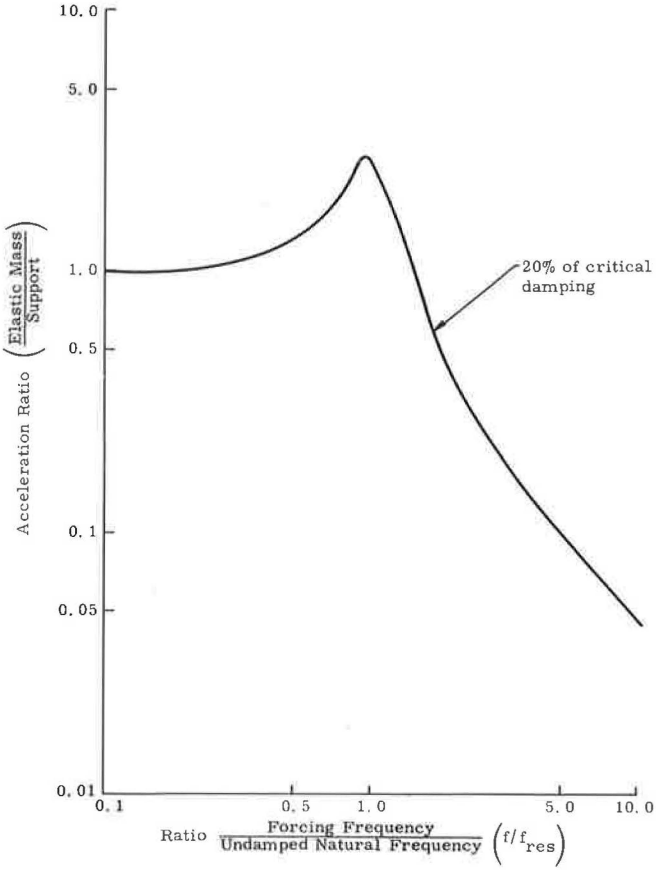


Figure 1. Acceleration ratio vs resonant frequency ratio for continuous sine wave vibration input.

tected for multiple impacts. The automatic bag deflation system would have a time delay for several seconds to ensure that the motion is stopped before deflation.

Rebound sensations during crashes have been instructive for us. In an early swing crash at 11 mph into a barrier, with the swing receiving $-40G_X$, one of us experienced a chest load of $-10G_X$ but then on rebound, a load again of $+10G_X$, with a higher head load due to bottoming on the rigid frame of the airplane seat. This made us annoyed at rigid structures in the seats, which also gave poor down load ($+G_Z$) protection, which protection is of increasing importance in aircraft crashes, and we proceeded to develop the inflated airseat, which in goal looks and feels like an ordinary seat at 1G, yet in crashes provides controlled yielding load isolation for the passenger. We propose that this airseat, with a transparent plastic section above the shoulders to maintain visibility, also be used in automobiles to provide controlled deformation load isolation, particularly for rear-end collisions.

Inflated airbag restraints, evidently first systematically conceived early in 1952 and constructed by Assen Jordanoff, and experimentally verified as to usefulness by our work, are the only devices we know that offer the required versatility for a passenger restraint system for the general public. Airbags can be

1. Stowed out of the way when not in use.
2. Rapidly and automatically brought into position by inflation when an automatic signal is provided in anticipation of a crash. (This is a separate but also solvable engineering problem.)

3. Capable of load isolation by controlled motion of the passenger with respect to the vehicle if the crash load will not produce bottoming.
4. Utilized as broad surface support of the man during the crash loading.
5. Capable of resonance frequency adjustment by inflation pressure change—if the crash load (anticipated from vehicle velocity) might be such that bottoming could occur, the inflation pressure could be increased to prevent bottoming.
6. Capable of rapid automatic deflation removal or roll-up, after the crash event or if the crash event does not develop.

ANALYTICAL REPRESENTATION OF AUTOMOBILE CRASHES

Not having an operating proving ground at hand, we have attempted to mathematically model crash results (3, 4, 5) of Haynes, Stonex, Fredericks, and others, particularly Severy et al of the University of California Institute of Transportation and Traffic Engineering. This has proven to be a long job, in part because of the changing characteristics of cars. Earlier barrier crashes gave impact periods near 0.12 sec, as one example, whereas some more recent cars give barrier crash periods near 0.07 sec, and much higher acceleration, for the same impact velocity.

Crash acceleration wave forms, of course, are also not uniform, though they must all start and stop at 0 G_x and rise to some peak or peaks during the event. (We are utilizing the aerospace acceleration terminology (6); if the passenger is thrown forward by the crash he experiences $-G_x$; thrown left, $+G_y$; and thrown down, $+G_z$.) We might use a triangular ramp up-and-down function for the accelerations on the floor by the seat in the passenger compartment, but the haversine is more easily handled. For the decelerating car, we use

Acceleration

$$\ddot{x} = \frac{a_{max}}{2}\left(1 - \cos\frac{2\pi t}{\tau}\right) \tag{1}$$

Velocity

$$\dot{x} = \frac{a_{max}}{2}\left(t - \frac{\tau}{2\pi}\sin\frac{2\pi t}{\tau} - \tau\right) \tag{2}$$

Distance

$$x = \frac{a_{max}}{2}\left(\frac{t^2}{2} + \frac{\tau^2}{4\pi^2}\cos\frac{2\pi t}{\tau} - \tau t - \frac{\tau^2}{4\pi^2}\right) \tag{3}$$

Initial velocity

$$\dot{x}_0 = \frac{-a_{max}\,\tau}{2} = v_0 \tag{4}$$

Passenger compartment stopping distance

$$x_{max} = \frac{a_{max}\,\tau^2}{4} = \frac{v_0^2}{a_{max}} \tag{5}$$

For crash conditions in which all of the initial velocity is not lost during the primary impact period, we replace v_0 by Δv, the part of the crash velocity lost by the rear car, or gained by the front car. Adding dimensional constants, we then have

Impact period

$$\tau_{sec} = \frac{2(\Delta v_{mph})}{a_{max,\ G}}\ (0.0455\ \text{G-sec/mph}) \tag{6}$$

Passenger Compartment stopping distance

$$\begin{aligned} x_{max,\ ft} &= \frac{(\Delta v_{mph})^2}{a_{max,\ G}}\ (0.0667\ \text{ft-G/mph}^2) \\ &= \frac{\Delta v_{mph}\,\tau_{sec}}{2}\ (1.465\ \text{ft/sec-mph}) \end{aligned} \tag{7}$$

We have not seen similar modeling representations of automobile crashes, although Kulowski (4, p. 75) quotes "Midgette's equation," developed for Cornell Injury Research in 1944, as

Mean acceleration

$$G = \frac{(\text{mph})^2}{\text{stopping distance in feet}} (0.0333) \tag{8}$$

Since the peak acceleration, a_{max}, of a haversine is twice the mean acceleration, this is the same as Eq. 7. We emphasize, however, that the passenger compartment stopping distance is, in general, not directly measurable after an accident, since it includes some anterior car structure crumpling and, for car-to-car crashes, some travel of the impact interface.

Table 1 presents the constants for our preliminary Automobile Haversine Acceleration Collision Model for various crash conditions. This model includes many approximations to the "real world." We use impact periods independent of velocity, and accelerations varying in proportion to velocity. Barrier crashes particularly have accelerations increasing more rapidly than velocities. It can be seen that the aligned car-to-car crash, with one stationary, has the longest period and lowest acceleration, and the barrier crashes the shortest period and highest acceleration. Note that the head-on, both-cars-moving collision has a lower acceleration than the barrier crash—apparently because parts of the cars interpenetrate, with some of each passing beyond the initial impact surface.

We have also marked off on Table 1 a boundary of crash conditions "marginal or unsurvivable with present structures and lap belts only." Note that the 30-mph barrier (tree or bridge abutment) crash is considered marginal. Indeed, almost half of all car deaths occur with impact speeds below 40 mph—although most of these do not involve collision with a rigid object. At 30 mph, with a passenger compartment acceleration of $-30G_x$, a 165-lb passenger (fiftieth percentile) would develop a load of 30 x 165 lb or 4950 lb even in a rigid restraint which decelerates him perfectly with the vehicle. A rule of thumb is that seat belt deceleration will be $1\frac{1}{2}$ to 2 times the passenger compartment deceleration, due particularly to slack in the belt. Clearly some of the belts, designed to take up to 5000 lb, will fail. Seat latchings and attachments may also fail. In all twelve automobiles crashed head on at speeds of 21 to 52 mph by the Institute for Transportation and Traffic Engineering (4, 7), all front seats and some rear seats tore loose from their anchorages, contributing to the injury-producing loads. Chayne (8) notes that "The seat track latching mechanisms must withstand a shock load in their test of 20 g's." He also says about barrier crashes, "Suppose the collision occurred at 40 mph. . . . The amount of kinetic energy to be dissipated amounts to. . . 216,000 ft-lb. Faced with these tremendously high values, any spectacular accomplishments could be achieved only by repealing the natural physical laws pertaining to motion and energy."

Stonex (9) also tends to feel that these are hopeless conditions. Speaking of a 33-mph barrier impact giving a peak power dissipation of 3800 hp, he states, "While the occupant doesn't actually participate in this inferno, he is in the immediate vicinity, somewhat like sitting in the boiler room while the boiler is exploding. We can't appraise these conditions realistically, but we can conclude that this environment is not conducive to serene comfort or long life." He goes on, "While continued design improvements of the vehicle will raise the injury threshold and reduce the severity of injuries, it is clear that there is a positive limit—a low ceiling—and there is no possibility of a total solution by vehicle design or superpackaging. The only total solution is to avoid these impacts where tremendous amounts of energy must be absorbed in a few milliseconds. The obvious solution is to eliminate the obstacles against which the car may impact." His preliminary study suggests that "80 percent of the fatal, public highway, single car, off-the-road accidents would be eliminated if the roadside were traversable for a distance of 30 feet from the edge of the pavement." He, disparing of driver education, selection, or supervision, opts for eliminating barriers (rigid sign posts, bridge abutments, etc.) at least to this distance, and making all roads one way, as "the only approach apparent which has the potential of turning the total-fatality-trend curve down materially."

TABLE 1

PEAK ACCELERATIONS, PERIODS AND PASSENGER COMPARTMENT ACCELERATION DISTANCES FOR VARIOUS SPEEDS AND COLLISION CONDITIONS
(Automobile Haversine Acceleration Collision Model)

	Barrier[1]	Head-On, Both-Moving[1]	Front Car Stationary Left Side Collision[2]		Front Car Stationary (Front or) Rear Collision[3]	
			Rear car $\Delta v=0.9v$	Front car $\Delta v=0.7v$	Rear car $\Delta v=0.7v$	Front car $\Delta v=0.6v$
Car speed 15 mi/hr (v)						
Accel.	$-15G_x$	$-14G_x$	$-11G_x$	$+9G_y$	$-6G_x$	$+5G_x$
Period	0.091 sec	0.098 sec	0.11 sec	0.11 sec	0.16 sec	0.16 sec
Distance[4]	1.0 ft	1.0 ft	1.1 ft	0.8 ft	1.2 ft	1.1 ft
Car speed 30 mi/hr						
Marginal or unsurvivable with present structures & lap belt only						
Accel.	$-30G_x$	$-28G_x$	$-22G_x$	$+18G_y$	$-12G_x$	$+10G_x$
Period	0.091 sec	0.098 sec	0.11 sec	0.11 sec	0.16 sec	0.16 sec
Distance	2.0 ft	2.1 ft	2.2 ft	1.6 ft	2.4 ft	2.2 ft
Car speed 45 mi/hr						
Accel.	$-45G_x$	$-42G_x$	$-33G_x$	$+27G_y$	$-18G_x$	$+15G_x$
Period	0.091 sec	0.098 sec	0.11 sec	0.11 sec	0.16 sec	0.16 sec
Distance	3.0 ft	3.1 ft	3.3 ft	2.4 ft	3.6 ft	3.3 ft
Car speed 60 mi/hr						
Accel.	$-60G_x$	$-56G_x$	$-44G_x$	$+36G_y$	$-24G_x$	$+20G_x$
Period	0.091 sec	0.098 sec	0.11 sec	0.11 sec	0.16 sec	0.16 sec
Distance	4.0 ft	4.2 ft	4.4 ft	3.2 ft	4.8 ft	4.4 ft

1. For barrier and head-on, both-moving, collisions, this preliminary model includes no rebound velocity. A 10% rebound velocity might be a better model, but this complicates the equations and so is not used here.
2. After the initial acceleration period, due to inelastic interaction with crumpling, in this model the rear car following a side collision still has 10% of the initial velocity and the front car has attained only 70% of the initial velocity (to the side). The rear car has a greater residual velocity and the front car less, and both cars rotate, if the initial velocity vector is not aligned through the centers of gravity of the two cars.
3. After the initial acceleration period, due to inelastic interaction with crumpling, for this model the rear car still has 30% of the initial velocity and the front car has attained only 60% of the initial velocity.
4. The passenger compartment or seat-base acceleration distance given here may include travel due to car deformation (impact surface stationary) and travel of the car (impact surface moving) during the collision acceleration period. For the rear or side collision, the front car seat acceleration distance represents seat travel which may in various cases be more or less than the front or rear car deformations involved in producing this acceleration.

We come from the aerospace business, fresh from designs to protect astronauts through malfunction explosions of their super-boilers, the space rockets, and reentry kinetic energy dissipation even in this early "sling shot" space era of 70 billion ft-lb, which in a 25-G abort reentry for which the astronaut is still protected may involve a peak power dissipation of a million horsepower. If the government would mobilize for automobile safety a more detectable fraction of what is spent on astronaut safety (a spending which has paid off in our astronaut safety record), we feel that automobile crash survival could be notably improved by cleaning up the roadways, by trying to clean up driver behavior, and by cleaning up the cars. We are dismayed to see the child's face torn on a door handle when the mother stops from five miles per hour. Instead of the 40-mph-barrier-collision survival being a "spectacular accomplishment," we feel it should be a routine requirement of proper car and restraint design. We recognize that this is a difficult task, and that putting airbags alone in present cars is not a sufficient solution. We do not aspire to repeal natural law, but we can appeal to laws other than those now operating in the automobile field. Indeed, lawsuit against the automobile manufacturers by people injured due to unsafe car design features, and insurance premiums modulated for particular cars by their safety ratings, can significantly speed our getting safer cars (10).

AIRSTOP RESTRAINT PERFORMANCE

We regretfully emphasize again that we have not yet carried out automobile tests with these restraints. Figure 2 shows a swing-barrier impact test, stopping from 11 mph. With only lap belt supporting the dummy, the head was thrown forward onto the seat in front. With the addition of chest and foot airbags (Fig. 3) the upper body and legs were supported, but the rebound into the seat was unnecessarily severe, and down-load isolation was not provided (Fig. 4). With the addition of the airseat (Fig. 5) the rebound load was largely eliminated (Fig. 6) and down-load protection was also provided. Figure 7 shows a 13.4-mph impact with a dummy. Note that the resonance frequency of the airstop system is near 3 cps, for the early part of the motion.

For the swing impact of Figure 5, the chest bag was below chin level at impact, that is, forward head support was not initially provided. These bags bulge laterally and

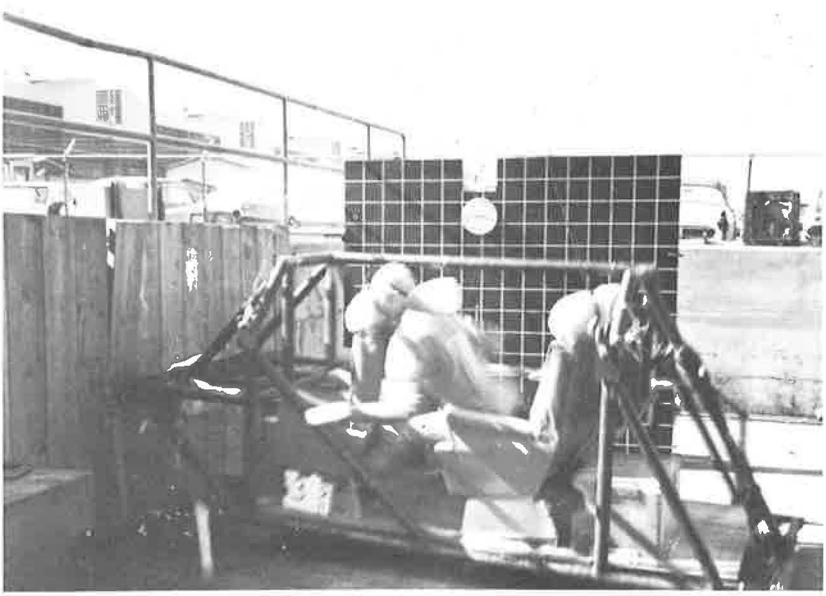


Figure 2. Swing impact with lap belt restraint (impact velocity 16 ft/sec; dummy subject).

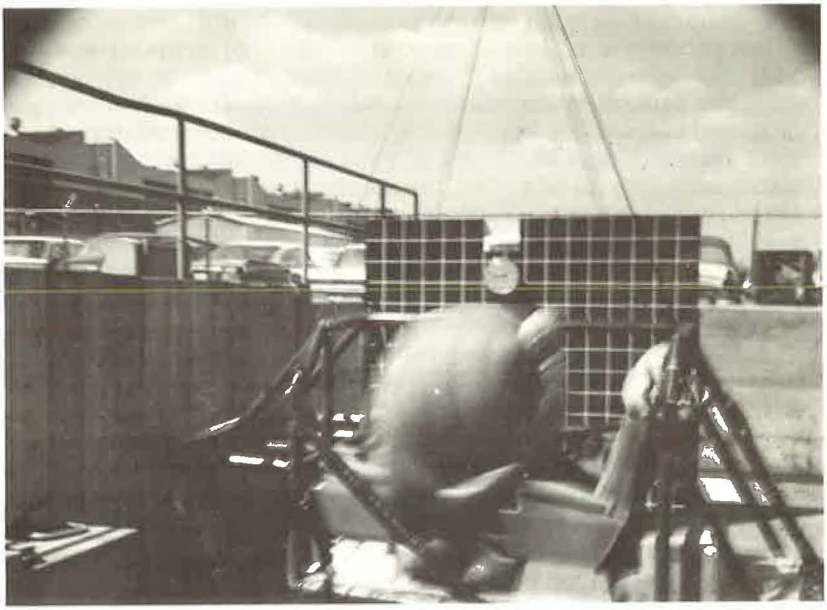


Figure 3. Swing impact of airstop restraint, without airseat (impact velocity 16 ft/sec; dummy subject).

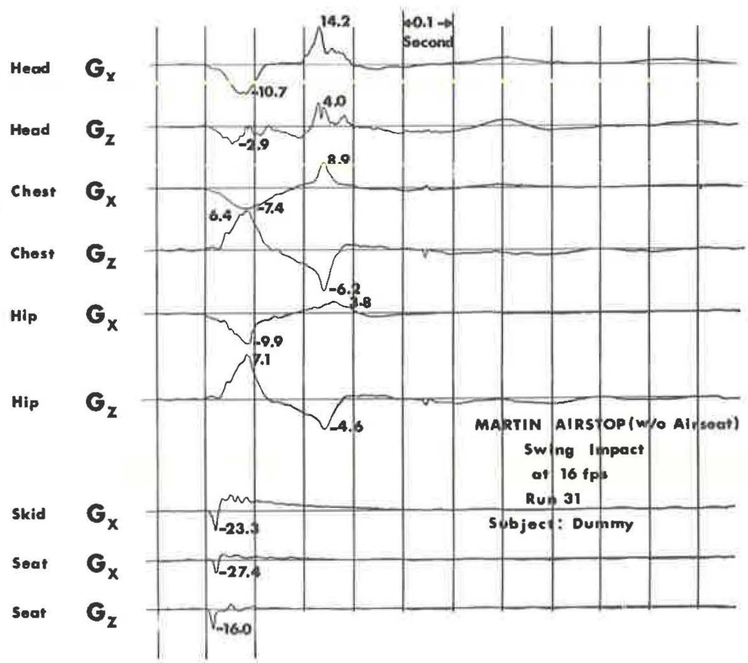


Figure 4. Acceleration time history.

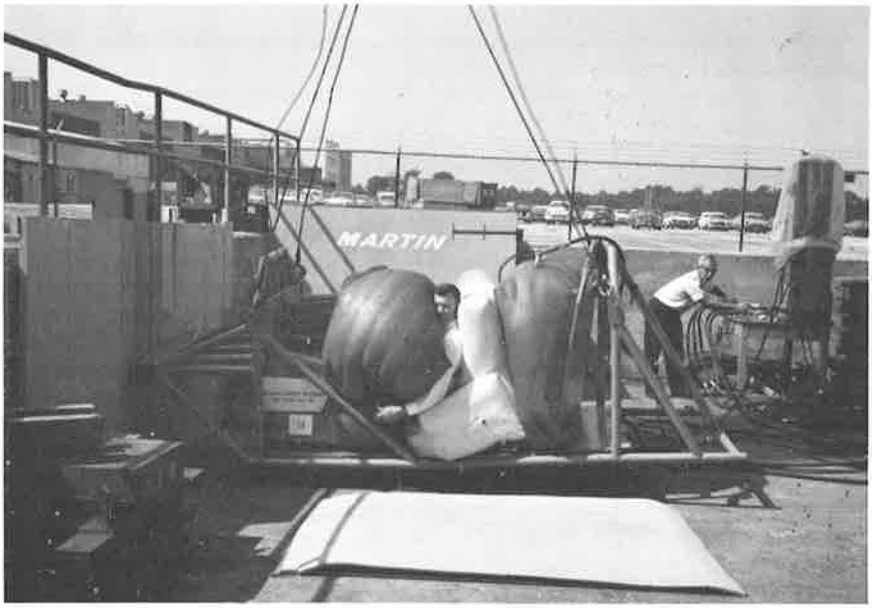


Figure 5. Swing impact of complete airstop system with airseat (impact velocity 16 ft/sec; human subject).

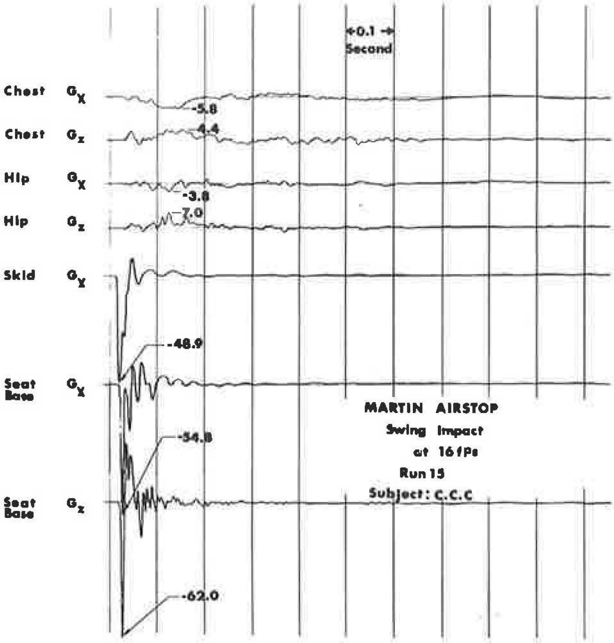


Figure 6. Acceleration time history of swing impact of complete airstop system (human subject).

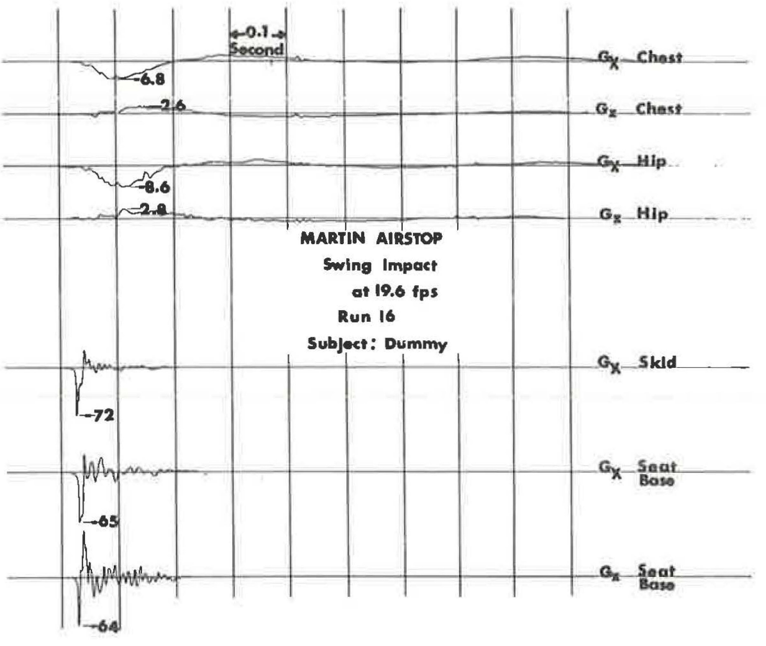


Figure 7. Acceleration time history of swing impact of complete airstop system (impact velocity 19.6 ft/sec; dummy subject).

also stretch, which is why the rigid piston adiabatic compression model of airbags is inadequate. Subsequent motions are at a sufficiently low frequency that there is no whiplash effect, even without the initial head support.

For the airstop system, a lap belt is worn, but one which passes through the seat back and then down, to allow a controlled deformation and load isolation rather than to provide a rigid support. Hence, in collisions the body is thrown forward ($-G_X$) then pivoted downward ($+G_Z$), so that compression of the seat will take up some of the energy. In many automobile collisions, the impact point is below the center of gravity of the vehicle, so that the rear wheels lift off the ground, also subjecting the body to down loads.

Our only laboratory experience with higher impact velocities is with the spacecraft simulator (Fig. 8) in which the subject lies horizontally with 2.5 ft of available motion into airbags in any direction. An impact at 22 mph with the vehicle tipped at a 45° angle gave the accelerations of Figure 9. Note the very minor consequences of the second impact, when the vehicle toppled down horizontally. A dummy impact at 29 mph (Fig. 10) still shows protection and load isolation, although the dummy stopping distance was less than would be available in a car, with 2 ft of car crumpling in addition to motion into the airbags.

Our field experience is with airbag and airseat restraints in a DC-7 (1, 11) and two C-45 (1) aircraft crashes, involving decelerations from about 160 mph, 100 mph, and 90 mph, respectively. In the DC-7 crash (Fig. 11) in which accelerations at our station were lost, it appears from motion pictures of dummy displacement into the chest airbags in comparison with displacement when we have measured accelerations, that the dummy supported by the latex airbags received less than $-10G_X$, while adjacent unsupported dummies were breaking seat belts, failing seat structures and losing a head on impact with the seat in front. Note that the controlled displacement into the airbags eliminates the snap loading when the unsupported dummy takes up the slack in the seat belt. Hence, with airbags not only is the load less on the passenger but also less on the seat, reducing seat failure and tear-up. For the fully designed system,

Figure 8. Test vehicle, pilot compartment airbag restraint system.

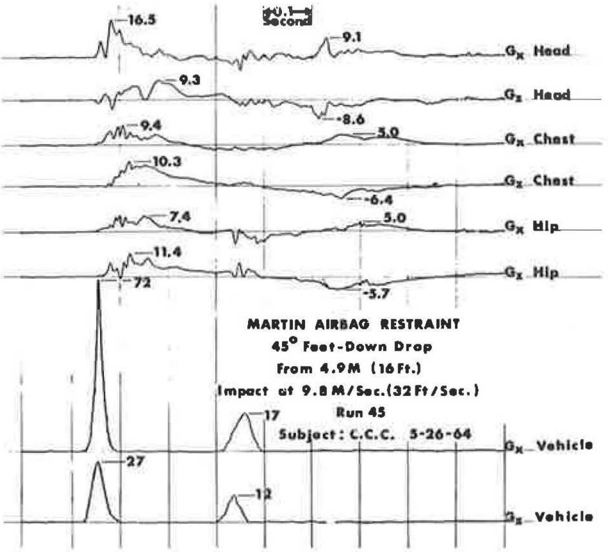


Figure 9. Acceleration time history (45-deg feet-down drop from 16 ft; human subject).

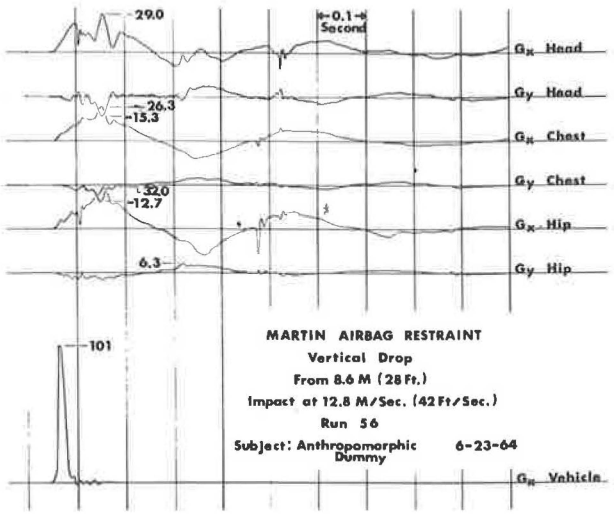


Figure 10. Acceleration time history (vertical drop from 28 ft; dummy subject).

Figure 11. DC-7 clearing second hill (courtesty of FAA).

after the crash and a few seconds time delay in case there are multiple impacts, the airbags and the airseats would be automatically deflated, opening up the entire vehicle for rapid egress following the crash.

In the C-45 crashes, the acceleration recordings were again inadequate. In the first crash, the aircraft hit a short 18-deg hill at 100 mph, bounced over it, and still came to rest in twice its length from the initial impact. Regular seats and dummies in back of our experiment were totally torn from the floor and ended up on top of the airseat dummies. During the hill impact the airseat dummies moved forward into their chest airbags, which failed, allowing them to contact the yielding airseat in front, yet after this impact they returned to the initial position, having lap belts on the airseats. After the final impact, the airseat was found torn in back and deflated, perhaps partially due to the additional load of the torn-out rear seat and dummies, yet the right-hand airseat dummy was in its initial position. If all seats in this crashing aircraft had been of regular design, they and their dummies after the crash would have been up against the forward bulkhead. With better materials than the canvas and latex rubber used by us, we feel that the airseats can have the leak and impact failure resistance and reliability of the automobile tire. In the second C-45 crash, with a dummy in a rearward facing airseat, without the chest or foot airbags, the plane turned over in bouncing over the hill at 90 mph, yet the dummy was still in the intact airseat after the crash. These initial field results have not given us much quantitative data, but a strong feeling that the airstop restraint in automobiles could allow barrier crashes, and other crashes, with survival at above 30 mph. Figures 12 through 14 show the first C-45 crash.

Preliminary force-displacement curves have also been measured in the laboratory (Fig. 15). These bags are not linear elastic systems, but because of the stretching and lateral bulging, a linear model, $df = -kdx$, is better than the rigid position adiabatic model, approximated by $df = -kdx^{1.3}$. With a 165-lb man and using a resonance frequency of the airstop system of 3 cps, for the linear model

$$f_{res} = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \tag{9}$$

we can solve

$$k = 4\pi^2 (3^2)\ 165/32.2 = 1820 \text{ lb/ft}, \tag{10}$$

Figure 12. First C-45 after 100-mph crash (airseat is at torn fuselage position).

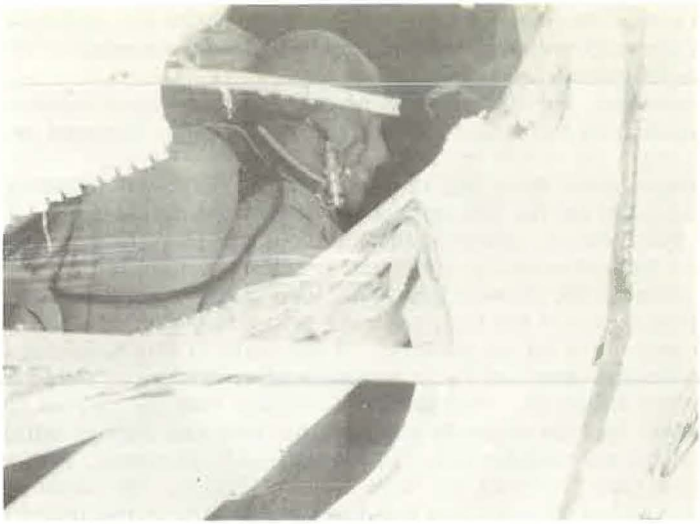


Figure 13. C-45 right airseat dummy, seen through torn fuselage, after 100-mph crash.

Figure 14. C-45 pilot and co-pilot dummies, restrained with lap belts and shoulder straps after 100-mph C-45 crash.

or about 11 G/ft of motion into this low pressure chest bag. By differences in bag volume, wall material, and pressure, this approximate "spring constant" can be increased or decreased.

An approximate model of airbag performance can be obtained by using a sine wave passenger acceleration function, but with an empirically lowered initial velocity representing the passenger velocity change which would be provided by an unforced cosine velocity wave response rather than the full velocity change which includes the initial forced displacement during the car deceleration (Fig. 16). The unforced response is adjusted in initial velocity Δv_1 to give the same velocity and displacement as the forced airstop system when the passenger compartment comes to rest.

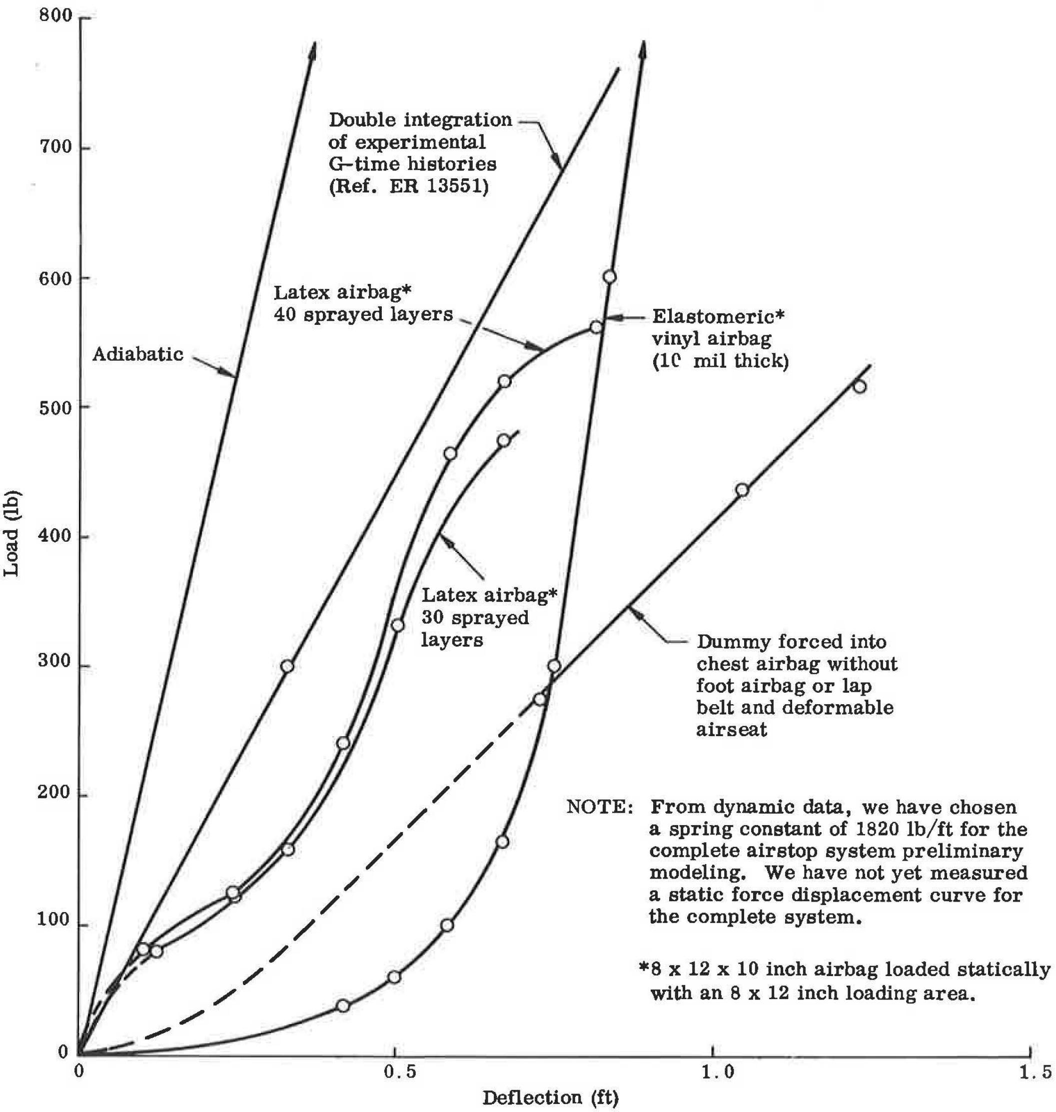


Figure 15. Load deflection characteristics.

By equations similar to Eqs. 1 to 7, it can be shown that for the sine acceleration function

Passenger acceleration distance (motion into the airbag)

$$x_{1,\ ft} = \frac{\tau_{sec}^{2}}{4\,\pi^{2}}\, a_{max,\ G}\ (32.2\ \mathrm{ft/sec^{2}\text{-}G})$$

$$= \frac{\Delta v_{mph}^{2}}{a_{max,\ G}}\ (0.0667\ \mathrm{G\text{-}ft/mph^{2}}) = \frac{\Delta v_{mph}\ \tau_{sec}}{2\pi}\ (1.465\ \mathrm{ft/mph\text{-}sec}) \tag{11}$$

and

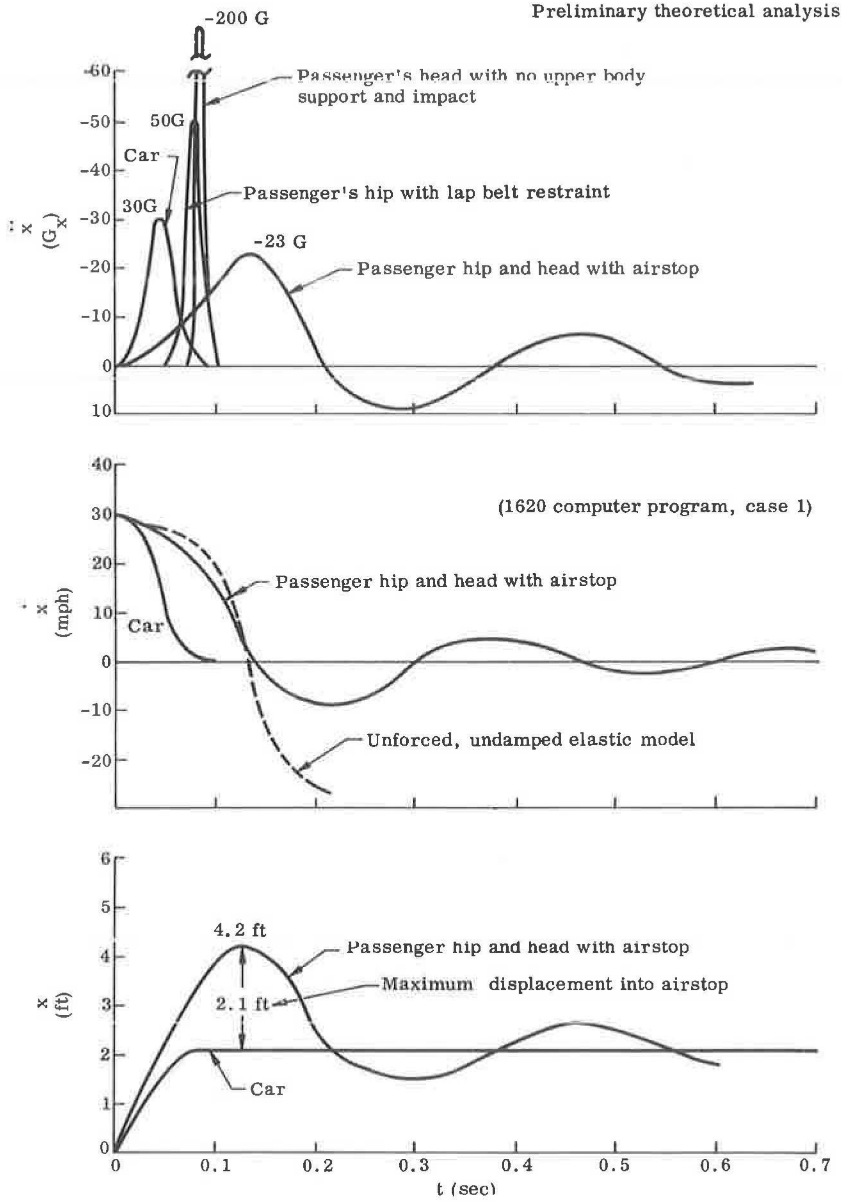


Figure 16. Acceleration, velocity and position of the passenger compartment and passenger for a 30-mph barrier collision.

$$\text{Peak acceleration} \quad a_{max,\ G} = \frac{2\pi\Delta v_{mph}}{\tau_{sec}} \ (0.0455\ \text{G-sec/mph}) \qquad (12)$$

It has been determined for a haversine barrier deceleration function of period 0.091 sec acting on a sine wave passenger restraint of period 0.333 sec that approximately 90 percent of the initial velocity would represent the equivalent load acting on the unforced restraint. Then for a model 30-mph-barrier collision, in which the passenger compartment moves 2 ft (front end crumpling of the automobile) and experiences a haversine loading of 0.091 sec period and $-30G_x$ peak (Table 1), the additional motion into the airbag by the passenger would be

$$x_1 = \frac{(0.9)(30)(0.333)}{6.28} (1.465) = 2.1 \text{ ft} \tag{13}$$

Either by using the spring constant of 11 G/ft or by Eq. 12, this airbag displacement gives a passenger acceleration of $-23G_x$, which is less severe than the passenger compartment's $-30G_x$, particularly when the latter is often accompanied by failure of present seat and seat belt supports. But this full-body load isolation is attained at the price of moving 2.1 ft in the compartment. To the extent that the passenger compartment cannot be opened up to allow such large full-body controlled displacements without bottoming, a more rigid restraint at the hips must be used, with the chest airbag essentially as upper body support, providing partial upper body and head load isolation only.

Table 2 shows the conditions for other collisions. It is clear that in terms of this simple modeling of airstop restraint performance, excessive displacements would be required to partially isolate from the higher velocity barrier crashes. To prevent bottoming on surrounding structures, we suggest controlling a bag pressure valve by the car speed, so that if the bag inflation is triggered, the bag will attain a pressure appropriate to the car speed at the time and sufficient to prevent bottoming. But before introducing such complexitities, direct experimentation of actual airbag systems in cars would be appropriate. The simple models are too simple. They do not show the down load consequences of the deformable lap belt on the airseat, for example. In addition, we have not yet modeled the "rigid" lap belt with deformable upper body support (airbag) case. If the bag "spring constant" were adjusted to upper body weight such that head motion remained at the airbag displacement values of Table 2, head load isolation would be as shown.

A mathematical description of the passenger and car as a coupled system has been attempted. If the deceleration of the car is a haversine independent of the motion of a passenger of mass M, x_1 and x are the distances of motion of the passenger and car from their positions at initial impact, and k is the spring constant of the linear airbag restraint model,

$$M \ddot{x}_1 + k(x_1 - x) = 0 \tag{14}$$

Substituting from Eq. 9 yields

$$M \ddot{x}_1 + kx_1 - \frac{ka_{max}}{2} \left(\frac{t^2}{2} - \frac{v_0 \tau}{2\pi^2} \cos \frac{2\pi t}{\tau} + v_0 t + \frac{v_0 \tau}{4\pi^2} \right) \tag{15}$$

This equation has the solution of the form

$$x_1 = A \sin\omega_1 t + B \cos\omega_1 t + Ct + Dt^2 + E + F \cos\omega_2 t \tag{16}$$

TABLE 2

BARRIER COLLISION (Preliminary Modeling)

Speed	Passenger Compartment		Passenger Accel.	Passenger Airstop Displacement
	Accel.	Stopping Dist.		
15 mph	$-15G_x$	1 ft	$-12G_x$	1.1 ft
30 mph	$-30G_x$	2 ft	$-23G_x$	2.1 ft
45 mph	$-45G_x$	3 ft	$-35G_x$	3.2 ft
60 mph	$-60G_x$	4 ft	$-46G_x$	4.2 ft

Hence

$$\dot{x}_1 = \omega_1 A \cos\omega_1 t - \omega_1 B \sin\omega_1 t + 2Dt - \omega_2 F \cos\omega_2 t \tag{17}$$

and

$$\ddot{x}_1 = -\omega_1^2 A \sin \omega_1 t - \omega_1^2 B \cos \omega_1 t + 2D - \omega_2^2 F \cos \omega_2 t \tag{18}$$

The constants of these equations for the conditions of passenger motion into a "linear spring" undamped airbag model and passenger compartment deceleration from velocity v_0 with a peak haversine acceleration a_{max} are

$$\begin{aligned}
\omega_1 &= \sqrt{\frac{k}{M}} \\
\omega_2 &= 2\pi/\tau \\
A &= 0 \\
B &= -\frac{v_0\tau}{4\pi^2} + \frac{a_{max}}{2\omega_1^2} - \frac{\omega_1^2 v_0 \tau}{(\omega_2^2 - \omega_1^2)\, 4\pi^2} \\
C &= v_0 \\
D &= \frac{+a_{max}}{4} \\
E &= \frac{v_0\tau}{4\pi^2} - \frac{a_{max}}{2\omega_1^2} \\
F &= \frac{\omega_1^2 v_0 \tau}{(\omega_2^2 - \omega_1^2)\, 4\pi^2}
\end{aligned} \tag{19}$$

These values may be used to evaluate the motion of the passenger with respect to the airbag at $t = \tau$, the end of the passenger compartment deceleration, to provide the initial conditions of the further free response of the unforced airbag system. However, at this point we can also easily introduce airbag damping. (More complex models, with damping time 0, could of course be handled by computer; we have not yet done this.) Our preliminary experience is that airbag damping, even without special internal baffles, etc., is about 11 percent of critical damping for the bag alone, and 22 percent for the bag with airseat and seat belts; this latter value is used for the model presented here. The response of the passenger-airbag system at time t after the car stops is thus given by

$$x_1 = e^{-\beta t}(A \sin \theta t + B \cos \theta t) \tag{20}$$

$$\dot{x}_1 = e^{-\beta t}(A\,\theta \cos \theta t - B\theta \sin \theta t) - \beta e^{-\beta t}(A \sin \theta t + B \cos \theta t) \tag{21}$$

$$\ddot{x}_1 = e^{-\beta t}(-A\theta^2 \sin \theta t - B\,\theta^2 \cos \theta t) - 2\,\beta e^{-\beta t}(A\theta \cos \theta t - B\theta \sin \theta t) + \beta^2 e^{-\beta t}(A \sin \theta t + B \cos \theta t) \tag{22}$$

For our case, with time now measured from the end of the car stopping period τ which gives the initial conditions for Eqs. 20 through 22, the constants are

$$A = \frac{\dot{x}_{1,\tau}}{\theta} + \frac{\beta}{\theta}(x_1 - x)_\tau \tag{23}$$

$$B = (x_1 - x)_\tau$$

$$\theta = \sqrt{\frac{k}{M} - \theta^2}$$

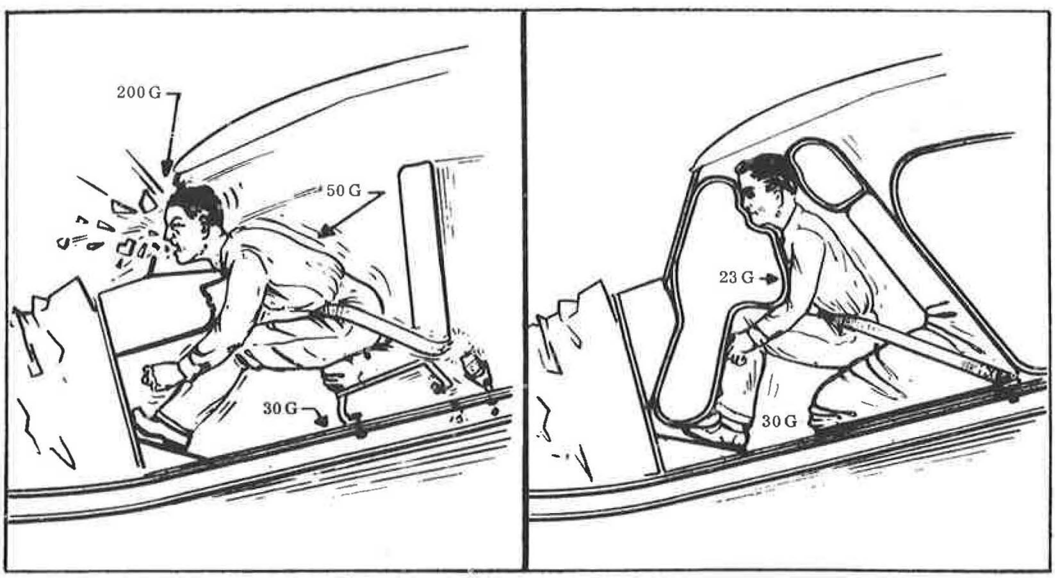
$$\beta = C/\,2M$$

$$C = (\%\text{ of critical damping})\ 2\sqrt{Mk}$$

The accelerations, velocities, and positions of the car and passenger for a 30-mph-barrier crash by this simple model are plotted in Figure 16, to which is added the expected front-seat passenger hip and head loads if supported by a lap belt that does not break, but with the head hitting surrounding structures, as it probably would for this crash condition. Severy and Mathewson (12) estimated a head mean deceleration in excess of 200G for a 26-mph barrier crash, with the passenger without lap belt. With lap belt, the head also impacted, and they consider the case probably fatal (at 29 mph) but do not give the head acceleration. We use a peak value of 200G to emphasize the magnitude of the upper body buckling problem, without upper body support.

Note that with the airstop restraint, the rate of change of acceleration is reduced over the "jolt" effect of the lap belt. For a haversine function of a_{max} acceleration peak and period τ

$$\dddot{x}_{max} = \frac{\pi a_{max}}{\tau} \quad \text{(in G/sec)} \tag{24}$$



Lap Belt 30 MPH → 0--Probably Fatal

Airstop 30 MPH → 0--Uninjured

Figure 17. Barrier collision.

For a 50G peak of 0.055 sec period with the lap belt alone

$$\dddot{x} = \frac{(3.14)(50)}{0.055} = 2850 \text{ G/sec} \tag{25}$$

a value which can contribute to the injury due to internal resonances (13). The airbags smooth out the load applied to the passenger, and distribute it over the entire body surface, so that with airbags (not bottoming) the load would be more tolerable even if of the same amplitude as the lap belt load. Our problem then is to prevent passenger bottoming through the airbags or airseat onto rigid or sharp surrounding structures. If bottoming can be prevented, the airstop restraint appears in this simple thoretical analysis to be very effective in reducing passenger injury and death in automobile crashes.

Figure 17 shows diagrams of the lap belt and airstop restraint conditions during impact, in the manner of Severy, et al (7). If cars can be modified to allow controlled displacement of the passenger during deceleration, without bottoming, load isolation by the airstop restraint can be utilized to prevent injury for crash conditions which now are fatal. If the cars cannot be modified, or bottoming might occur, we suggest using chest airbags at higher pressure, or preferably, since bag pressures above 10 in. of water are uncomfortable prior to deceleration, using a wide belt (3 in. or more) on a more rigid lower seat structure (high pressure airseat base), with the airseat back and chest airbag as the controlled deformation members for upper body support.

DISCUSSION

The foregoing analysis suggests that the airstop restraint offers some promise for the reduction of injury and death to passengers in automobile crashes—if indeed the restraint is in position at the time of the crash. If lap belts are not used, for example when "sweeping up" children playing on the floor of a station wagon by bag inflation when a potential crash situation develops, the restraint is most successful if "bottoming" does not occur. If bottoming might occur, for example when doing high speed driving with a heavy adult, the restraint should be used with a lap belt, with the chest bag essentially as an upper body support. We recommend 10,000-lb loop strength belts of 3-in. width, with seat support (or airseat) and floor strengthening to take $-50G_x$ loads without failure.

Cars would be safer if one were less likely to hit protruding structures in a crash. Discussing airplane accidents, DeHaven said over 20 years ago (14): "(a) The force of many accidents now fatal is well within physiological limits of survival. (b) Needless injuries—both serious and fatal—are caused by the unfortunate placement and design of certain objects and structures." Later he notes, "A more practical answer to the severe injuries caused by the instrument panel would be to space the panel slightly beyond range of the head." This plea has been repeated many times for automobiles.

DeHaven continues, "There is no intention of indicting this one type of plane, for many makes of planes have braces in this or other dangerous positions. In accident results of this kind it seems reasonable, however, to ask the question 'was this fatality caused by pilot error—or by an error in design?'" The "accident" may have many causes, but we emphasize that accident effects are not accidental—once the crash is under way, the variability of outcome is sharply reduced. It is interesting to note that the primary collision of the car with outside structures may be almost over before the secondary collision of the passengers with the inside of the car has occurred. With so many cars destined to be in accidents, more of the design should consider the crash consequences. Look around at your car (and at your highways and fellow drivers) and ask, "Am I ready for a crash in this environment?" If the average driver otherwise lives a normal life span, his chance of being killed in an automobile crash within this life span is something like one in forty with present structures and restraints, and with present mileage death rates, which show more tendency to increase than decrease.

DeHaven (and many since) continues (14), "A large expansion in use of aircraft by civilians will require more than mere concessions toward safety. It will require deliberate planning of safety with acknowledgement that this one factor—more than speed, efficiency, or cost—will govern the extent of future developments. Heavy and unnecessary penalties will be paid by the public—and the aircraft industry—until safety is considered foremost among design factors in aircraft for broad public use." For automobiles, heavy penalties are indeed being paid, with our emphasis on "tigers," not safety. With very little regulation as to automobile design, and the public not yet demanding safety advances, the manufacturers find little economic incentive in spending for safety. The automobile manufacturers are not now required, as are the pill manufacturers—and airplane manufacturers—to prove to a government agency the safety of their product before it is introduced to the public.

In this atmosphere, it is perhaps fruitless to consider the internal modifications of a car required for crash load isolation by controlled displacement within the car of perhaps 4 ft, to reduce the severity of the 60-mph barrier crash. Yet using present information, such a crash could be survivable. One might follow Ryan's lead (15) in putting the controlled displacement on the bumpers, with hydraulic shock absorbers, decelerating at 20 or 30G, not the 0.9G used by Chayne (8) in ruling out such designs. This indeed should be explored with far more support than has thus far been given to it, with of course the recognition of the interactions of hydraulic bumper compression and car body crumpling. Alternately, one might extend the car crumpling, or plan its characteristics as has been done for example in putting an intentional crinkle in the Mercedes frame.

Alternately again, might technology not be ready for considering emergency car deceleration by retro-rockets of perhaps $-5G_x$, since it is difficult with wheel brakes under the best friction conditions to exceed $-0.9G_x$? A car velocity of 44 mph is only 2G-sec, or 5G acting for 0.4 sec, provided for a 4000-lb car by a 36-lb retro-rocket of specific impulse of 220 lb-sec/lb of propellant. (But watch out for tailgaters. . . .) Having a lower thrust rocket canted up 30 deg might have the additional advantages of increasing braking and steering effectiveness. But these methods only reduce the primary collision of the car with its surroundings. The loads on the passengers must be determined with regard to their motions.

Ryan (16) has made another important suggestion, to alleviate the snap overloading of seat belts due to their initial slackness. He suggests using hydraulic seat belt tighteners to get some pre-loading of the passenger before he is thrown forward. He also shows the advantage of using a damped controlled stretching of the belts. Aldman (17) similarly shows the advantage of the long stretching diagonal chest strap in reducing passenger loads. Bierman's work (18) is the earliest experimental work that we know on load isolation by controlled stretching of the restraint. All of these authors seem to feel, with us, that a controlled use of whatever space is available for load relief by motion in the vehicle is highly advantageous. We go further than they in urging a consideration of the modification of automobile interior design to allow a greater distance for controlled motion load isolation.

The restraint pre-loading concept is a good one if it can be engineered. We would like to leave children free to move around in the car—not huddled in their rigid restraints waiting for a crash. With an automatic crash anticipation signal, the bags could be inflated at a low pressure (3-10 in. of water pressure, i.e., 0.1 to 0.3 psig) to get the passengers in a favorable restraint position before impact. Then if the impact occurred, the bumper or accelerometer signal could trigger open a higher pressure into the airbags, with this increased pressure set continuously proportional to speed, to preload the passengers and further reduce the accelerations. Alternately, a higher pressure bag might be inflated inside of the lower pressure bag, not quite touching the body, a concept we called in 1962 the "multiple gradient yield" restraint concept. We note that even with airbags we would like to have all interior fixtures of broad and yielding profile, to gentle our bottoming. Examine Kulowski (4) for the "signatures" of stylists' horn rings, light switches, gear shifts, rear view mirrors, hood ornaments, etc., ground into victim's bodies as a price of our culture's emphasis on style rather than safety.

Inflation triggering and rates are problems with which we have not yet experimented. If we can obtain a suitable "unusual behavior" signal from changes of wheel motion, steering rate, braking amplitude, radar or optical closing rate signals, road car position signals, etc., we suggest inflating the low pressure bags in perhaps 0.25 sec.

Since a body in a crash does not move off a car seat for perhaps 0.05 sec after bumper impact, we would hope that the bags would begin to inflate 0.2 sec before impact. A child pushed by these bags would expectedly not be thrown, but slide into restraint position. A car traveling at 60 mph would travel 18 ft in this time; this time is perhaps too long for some circumstances. On bumper impact we would release the higher pressure which would fill the bags in 0.05 sec even if they had not already been inflated at lower pressure. These problems cry for experimental exploration.

We wish to reduce the sense of claustrophobia and fright on chest airbag inflation, so that triggering can sensitively respond to small signs of odd car behavior with little consequence of an occasional unnecessary inflation (or one carried out manually to shush the children, not save their necks). Note again that the driver is not touched by airbag inflation, for this early plan. In aircraft for example, we propose that the airbags be inflated (in perhaps 5 sec) prior to every takeoff and landing. We have therefore experimented with transparent airbags (Fig. 18). These have ruptured in some crashes, but materials research and bag design (such as pop-out gores that would increase bag volume at certain pressures to flatten the force-displacement curve) would seem to be able to produce reliable transparent bags. We have particularly liked elastomeric vinyl bags, made by the Lindron Corp., Pawtucket, Rhode Island.

We have experimented with automatic roll-up devices (coil spring stock or bungees) in the bags (Fig. 19) and this too appears quite feasible. If the airbags inflated and a crash did not develop, the driver would actuate a switch, releasing bag pressure and allowing them to roll back into the dash or seat back. He would then check his air supply tanks (mounted in the wheel wells), and occasionally have these refilled when in the gas station, from the 80 psig air supply line. And he would hopefully try to drive more safely, with this automatic warning that his driving behavior had been sufficiently odd that a possible crash was predicted.

Figure 20 shows one of our aircraft airseats. This in an inflated structure, with fabric internal cross-bracing to prevent bulging. Controlled deformation down to the floor can occur without bottoming on rigid structures. Figure 21 shows a dummy in the airstop restraint on impact on a paved surface at 32 ft/sec, a crash condition we

Figure 18. Mockup of an automobile airstop restraint, which should include airseat.

Figure 19. Automatic roll-up of the airstop chest airbag.

Figure 20. An inflated airseat (for the automobile, the back above shoulder level would be transparent).

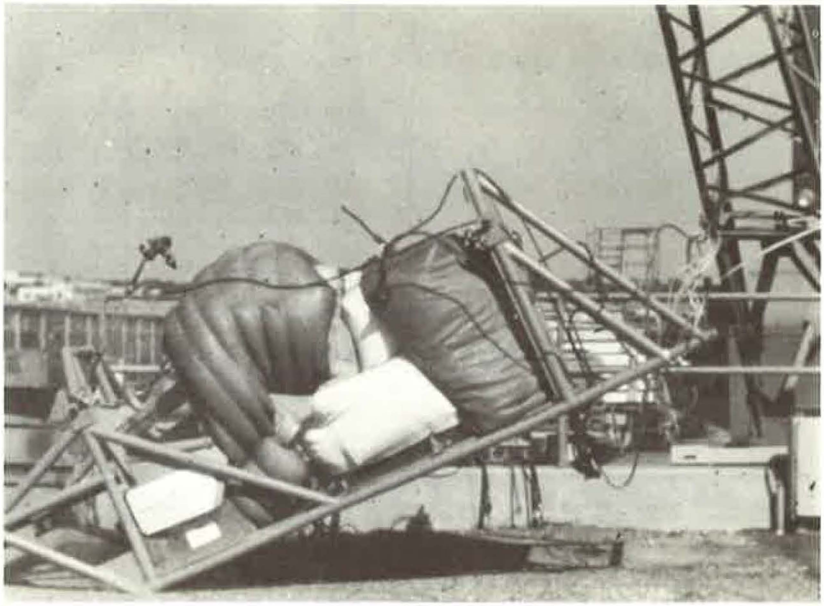


Figure 21. Crane drop of the complete system, during impact (impact velocity—32 ft/sec; dummy subject).

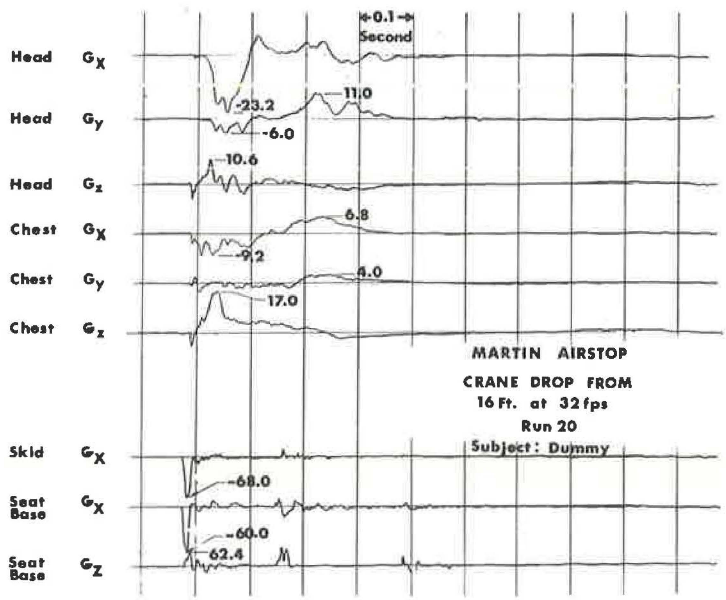


Figure 22. Acceleration time history of the crane drop of the complete airstop system (30-deg nose-down attitude; impact velocity—32 ft/sec; dummy subject.)

would not care to experience in any other restraint. Yet Figure 22 shows that this is survivable without injury in the airstop restraint.

The automobile airseat we would design rather differently, with multiple airbag compartments individually inflatable for height and position adjustments to suit individual drivers, a concept patented by W. J. Flajole, U.S. Patent 2938570. These multiple compartments could involve connections to the car air supply tank, so that they could be made anywhere. The seat back above shoulder level would be transparent. Alternatively, with the controlled deformation of the seat back (supported by the rear seat chest airbag) in a rear collision, the possibilities of neck whiplash injury are very much less than with present seats, so that this head support may not be necessary. (Where are the experiments to answer this question? We have not yet been able to do them.)

Likewise the automobile airseat could have arm-rests shaped more like Frank Crandell's Survival Car II (Liberty Mutual Insurance Company), and similar shoulder wings at least on the outside (perhaps inflated when the door is closed) to provide lateral support. A 3-in. lap belt of 10,000-lb loop strength would be on the seat, passing over the hips area down to the floor. This design would allow some forward motion with seat deformation, then increasing energy take-up by compressing the seat downward. Children, being better supported by the chest airbags, would not have to use the lap belt (though it would be better to have them do so, for added reliability). The car interior would be so padded that if mother jammed on the brakes at 5 mph, the children would not hit anything that would hurt them. The air seats would be designed not to fail at $-50G_X$. We discuss rear and side collisions further in another paper (19).

Kulowski (4) quotes Thomas Carlyle in summarizing his crash studies: "Our main business is not to see what lies dimly at a distance, but to do what lies clearly at hand." Research, vastly spurred by John Paul Stapp's challenging work, has shown what might be done in crash protection, with Stapp's conclusion (13), "The most reliable instrument for measuring the varied effects of dynamic force on man is man," and Stapp's survival under conditions that daily kill others. We, coming to the end of a contract without the support to continue at least until some of those 140 people killed by automobiles[1] in the United States every day are saved as a consequence of our work, sadly admit (with apologies for paraphrasing), "There's many an excuse between research and use."

Yet there is hope in the hustings. It is increasingly apparent to legislators (20) that the cost of automobile mayhem (perhaps \$9 billion a year) is far greater than the cost of automobile safety improvements which might cut this mayhem in half. Taking the systems approach, it is ridiculous for society not to obtain these improvements (seats which do not tear out, doors which do not open, steering wheels which are not pushed back, seat belts which do not fail, knobs and window cranks and instrument panels which yield under crash loads, use of shoulder straps or other upper body support, improved brakes, tires without defective materials, etc., etc.). Yet the individual is not in a position to get them if indeed he knows about them. (As Kulowski puts it, "Individual initiative is at best only second best.") The legislator (in our view) is increasingly coming to identify himself as the responsible agent of the "social systems," sensitive to the systems analyses and working to improve systems performance, in situations in which the individual cannot hope to do so. He hopes the voter can judge the success of systems performance; he is overwhelmed in his continuing efforts to teach the voter the details of systems operation that must affect the legislator's actions. And so we have Public Law 88-515 and a beginning set of safety specifications for government vehicles put out by the General Services Administration (with which the automobile manufacturers will now comply in cars for the public at an appropriate price), we have legislation for design of a safe automobile proposed at the federal level by Sen. Nelson and also proposed and passed by New York State, we have consideration of

[1]Daniel Moynihan (20) gives the estimate of 125 injuries by automobiles for each death (i.e., perhaps 6 million per year) and says, "Something like one out of every four automobiles manufactured ends up with blood on it" (p. 321).

the appropriate government role in traffic safety by Sen. Ribicoff's Subcommittee (20) with some suggesting a Federal Automobile Agency, we have proposed legislation for a National Accident Prevention Center, etc. Once a developed experimental safety car is available (with yearly improvements), we might require that the automobile manufacturers demonstrate to a federal agency by crash tests that their new designs are as safe as this federal standard, before they are handed to the American public cocked, tuned and tigerish. Drug manufacturers are required to do no less, yet (partly as a consequence) drugs kill far fewer people than automobiles.

Lawyers, too, see the court decisions moving us toward safety, with tort liability suits against the manufacturers perhaps beginning to apply pressure to reduce the practices of unsafe car designs. We can only hope that this will not end, as cigarettes might, with a little label on the windshield, "This car is unsafe to health," with all else going its jarring way, since the buyer has been warned.

And how can economics help? We suggest that there be fines (state and federal?) on the driver responsible for a crash and perhaps even on the manufacturer of his automobile, with these funds to support highway safety research, including improved safety car designs. Somehow, the revenue for safety research and development should be made commensurate with the magnitude of the safety problem. We suggest that those arrested for driving violations also be fined for the absence of safety features in their cars.

And we, with all due bias, suggest the use of the airstop restraint system—or first, research on its further applications to the automobile.

CONCLUSIONS

1. Upper body support is required in addition to the lap belt to reduce injury and death in automobile crashes. Even at 30 mph a barrier crash is probably fatal, due particularly to head impact, in cars of present structure and lap belts only.

2. Controlled motion into a low frequency (below 3 cps) damped elastic restraint system or other controlled displacement restraint can provide crash load isolation if bottoming does not occur, or occurs (into soft material) more gently than the primary crash itself.

3. The airstop restraint offers both upper body support and controlled load isolation. Inflation of the airbags to a low pressure (0.1-0.3 psig) just prior to an impact can push children and others previously unrestrained into positions of restraint. Airbag pressure can be adjusted to prevent bottoming. If the high pressure, proportional to car speed, is released into the inflated airbags just at bumper impact, through one-way valves in case the supply tanks are ruptured during impact, and automatically released after impact, discomfort due to bag pressure would expectedly be a small price for the protection provided.

4. Automobile experimentation is essential to extend this preliminary analysis of the potential of the airstop restraint. Many problems for research remain, including the quantification of the automatic signals triggering airbag inflation prior to a crash, engineering of the system for the car, materials and reliability analyses, passenger acceptance, and experimental validation of the protection provided by automobile crashes.

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