Safety and Crashworthiness of Dual-Mode Vehicles

A. E. Brown, H. Weinstock, and J. N. Rossettos,* Transportation Systems Center, U.S. Department of Transportation

Safety features and the degree of safety expected of dual-mode systems are reviewed. Some of the inherent advantages and disadvantages of dual-mode transportation are also outlined. Possible categories of vehicle safety are defined to aid in developing measures of collision survivability in terms of human tolerance. The available analytical tools for crashworthiness prediction are discussed, and the type of parameter studies that can be performed with computer programs of simplified simulation models are suggested. The importance of energy-absorption devices and impact energy management concepts is emphasized so that optimum design conditions can be attained. Finally, a review is made of some biomechanics dynamic models useful for the assessment of injury potential.

The increase in automobile population and automobile use during the past decade combined with a corresponding decline in the use of public transportation has resulted in a severe strain on urban road and highway systems. The resulting traffic congestion has produced increased air pollution, travel times, and fuel consumption. The reduction in public transportation services in urban areas has resulted in a decreased mobility for the young, elderly, poor, and handicapped who cannot drive or purchase an automobile. The conventional solution to traffic congestion is to construct new highways and increase the number of lanes of existing highways. This approach has not proved to be successful in urban areas and has frequently resulted in an increase in traffic congestion. Constructing new highways or widening existing highways is extremely expensive in terms of construction costs, land acquisition costs, and social and economic disruptions in urban communities.

The dual-mode and personal rapid transit system concepts offer another option for increasing the flow of people and goods. At the same time they provide the safety and convenience of the automobile and do not require excessive amounts of land and extensive new road construction. The use of automatic control of vehicles could result in a sevenfold increase in the capacity of a lane of traffic as discussed below. Two lanes of a dual-mode operation would provide the equivalent service of a 14-lane highway.

Highway capacity is currently limited by the driver's perception of his or her ability to detect an emergency condition and take appropriate corrective action. Figure 1 shows the relation between vehicle speed and vehicle density for a simplified model of driver behavior. This model assumes that the vehicle acceleration is a function of the difference between the maximum speed a driver would travel on a given road and the distance and closing velocity between the vehicle and the one in front of it. The parameter α is a driver caution factor that could depend on road conditions, weather conditions, or world tensions. Experimental data indicate that the curve for $\alpha = 4$ is a good approximation for freeway driving. The California highway rule recommends a separation of one car length for each 10-mph (16-km/h) speed increment.

As shown in Figure 2, this decrease in speed as a function of traffic density results in a maximum throughput of the highway lane for $\alpha = 4$. The maximum capacity is about 2000 automobiles/hour at an average speed of 30 mph (48 km/h). An attempt to increase the number of automobiles on the road will result in a further decrease in speed and a decrease in the net flow of traffic. Above this critical density the traffic flow becomes unstable with stop-and-go driving conditions, resulting in further decreases in throughput and increased likelihood of low-speed accidents due to traffic congestion. Current efforts toward metering of freeway traffic by control of on-ramps and in some cases traffic control signals interrupting traffic flow are directed toward keeping lane densities below this critical value.

In dual-mode operation the computer that controls the vehicle would have sufficient knowledge of the current and planned changes in the trajectories of the vehicles in front of another vehicle to permit headways closer than could be permitted were there a human operator. The close control of speed would permit headways approaching zero without risk of collision. The dual-mode system could therefore have potentially the speed throughput relation shown in Figure 2, making travel speed independent of density.

The dual-mode vehicle operating on its own guideway provides some strong potentials for improved highway safety.

1. Head-on collisions are impossible.
2. Human error (or vehicle operation by drivers under the influence of drugs or alcohol) is eliminated.
3. The computer can anticipate events that are miles ahead of the vehicle and take corrective action in a controlled and programmable fashion, thus eliminating surprises and near misses.
4. Vehicles are inspected regularly, and maintenance is under central control.

*When this research was performed, Mr. Rossettos was associated with the Department of Mechanical Engineering, Northeastern University, Boston.

The engineers concerned with the safety of such a system have asked the questions: What happens if something goes wrong? If there is a failure of a component in the control system, what is the likelihood of a collision? If the collision can occur, what injury will it produce? The permissible headways and component reliabilities are directly related to the crashworthiness of the vehicle.

Accordingly, Transportation Systems Center is conducting for the Urban Mass Transportation Administration design trade-off studies to evaluate the significant parameters affecting vehicle crashworthiness. The basic requirements for vehicle crashworthiness are that

1. The vehicle be able to sustain low-speed impacts (under 10 mph, 16 km/h) with no occupant injury or functional damage,
2. The vehicle provide the occupant with injury protection at least equivalent to the automobile, and
3. Overall system safety be at least equivalent to transit system standards and experience.

While operating off the guideway, the vehicle must also conform to highway safety standards.

This paper reviews some of the parameters relevant to this study and the status of the analytic tools to be applied in predicting dual-mode crashworthiness.

INFLUENCE OF HEADWAY ON VEHICLE IMPACT VELOCITIES

If an obstacle were to be introduced on the guideway and the maximum deceleration under emergency conditions was 0.34 g (11 ft/sec^2, 3.4 m/s^2), the minimum warning that a dual-mode vehicle traveling at 60 mph (96.6 km/h) would require to permit stopping with no impact velocity is

$$h = V^2/2(11) = 88^2/22 = 352 \text{ ft} \tag{1}$$

If there were a delay in detecting the obstacle and activating the vehicle emergency-braking system, the impact velocity would be given by

$$V_I^2 = (88)^2 - 22h \tag{2}$$

so that, if the emergency-braking system was activated at a distance of 176 ft (53.6 m) from the obstacle, the impact velocity would be 62.2 ft/sec (19 m/s) or 42.4 mph (68.2 km/h). If the braking system was activated at 350 ft (106.7 m), the impact velocity would be 6.6 ft/sec (2 m/s) or 4.5 mph (7.2 km/h).

For a 20-ft (6.1-m) automobile length, a requirement that the headway be greater than this emergency stopping distance (i.e., 352 ft, 107.3 m) would result in a requirement of 18 automobile lengths between vehicles compared to the 6 automobile lengths proposed by the California highway rule. Human drivers can operate at closer headways than the emergency stopping distance because the automobiles in front of them require a finite period of time to effect a velocity change. If the braking characteristics of the vehicles are identical, the minimum headway required to produce a zero-impact velocity is the distance required for a driver to perceive that the automobile in front is decelerating and to brake his or her vehicle. If this reaction time is 0.75 sec, the minimum headway would be 66 ft (20.1 m) to avoid collision, which would be about 4⅓ automobile lengths.

Equations for determining the impact velocity, V_I, as a function of headway, D_0, involved in the collision of the rear of one vehicle and the front of another vehicle with initial velocities of V_{01} and V_{02} (at t_0, time of brake initiation of the leading vehicle) respectively, constant deceleration rates of a_1 and a_2 respectively, and a driver reaction time, t_R, in the following vehicle (as shown in Figure 3) are as follows:

$$V_I = V_{I2} - V_{I1} \tag{3}$$

$$V_{I1} = V_{01} - a_1 t_I \qquad V_{I1} = 0 \text{ for } t_I \geq (V_{01}/a_1) \text{ and } a_1 = 0$$

$$V_{I2} = V_{02} - a_2(t_I - t_R) \qquad t_R < t_I$$

$$V_{I2} = V_{02} \qquad t_I < t_R$$

$$D_0 + D = V_{02} t_I - \tfrac{1}{2} a_2 (t_I - t_R)^2 \qquad D = V_{01} t_I - \tfrac{1}{2} a_1 (t_I)^2$$

$$D_0 = (V_{02} - V_{01}) t_I + \tfrac{1}{2} [a_1 t_I^2 - a_2 (t_I - t_R)^2] \tag{4}$$

Variations of impact velocity with headway distance are shown in Figure 4 for three situations. In each situation both vehicles are assumed to have the same initial velocity. Initial velocities are 60 mph (88 ft/sec), 30 mph (44 ft/sec), and 10 mph (14.7 ft/sec) [96.6 km/h (26.8 m/s), 48.3 km/h (13.4 m/s), and 16.1 km/h (4.5 m/s)]. Both vehicles decelerate at 11 ft/sec (3.7 m/s), and driver reaction time is 0.75 sec. Figure 5 shows a comparison of impact velocity with headway distance when there is no deceleration (braking failure) by the following vehicle. As might be expected from Figure 3, maximum impact velocity is the same for the three situations shown in Figure 4; with that length of headway distance, maximum impact velocity decreases as initial velocity decreases. As also might be expected from Figure 3, the initial and final portions of the plot in Figure 4 are symmetrical. Tailgating reduces impact velocity rapidly with decreasing headway below 3 ft (0.9 m). The distance involved, however, would seem impractical for the human vehicle operator; but, if dual-mode instrumentation and computer automation could be designed to detect particular closing speeds and distances and give relevant decisions, close headways are feasible. However, in the event of a major system malfunction, such as braking failure, relatively large impact velocities could occur, as indicated in Figure 5.

CRASHWORTHINESS AND CATEGORIES OF VEHICLE SAFETY

Since dual-mode system (DMS) development, even with shorter headways, must reproduce or improve on the degree of safety that is expected for the automobile, the definitions of safety in terms of past research for automotive vehicles must, at the least, be adopted in DMS crashworthiness studies. The vast amount of work that has already been carried out for automobile and rail vehicles (1, 2, 3, 4) has given rise to analytical simulation models that have proved to be important aids in the prediction of vehicle crashworthiness. The availability of such tools can also be useful for DMS studies. The more recent possibility of using scale-model crash testing should also form an important adjunct to such studies (5).

The crashworthiness of dual-mode vehicles must be designed so that it may accept controlled collision possibilities and be useful in delineating possible categories of vehicle safety that can be used in conjunction with various design and automatic control strategies. For convenience, the following qualitative categories can be defined: operational (no injury or damage), operational (no injury, minor damage), and safe (no fatalities, minor injury, nonoperational damage). Trade-off studies, using computer simulation models, can allow a more quantitative separation of these levels. Collisions at different speed ranges will be associated with the above

Figure 1. Speed of single lane of traffic versus vehicle density.

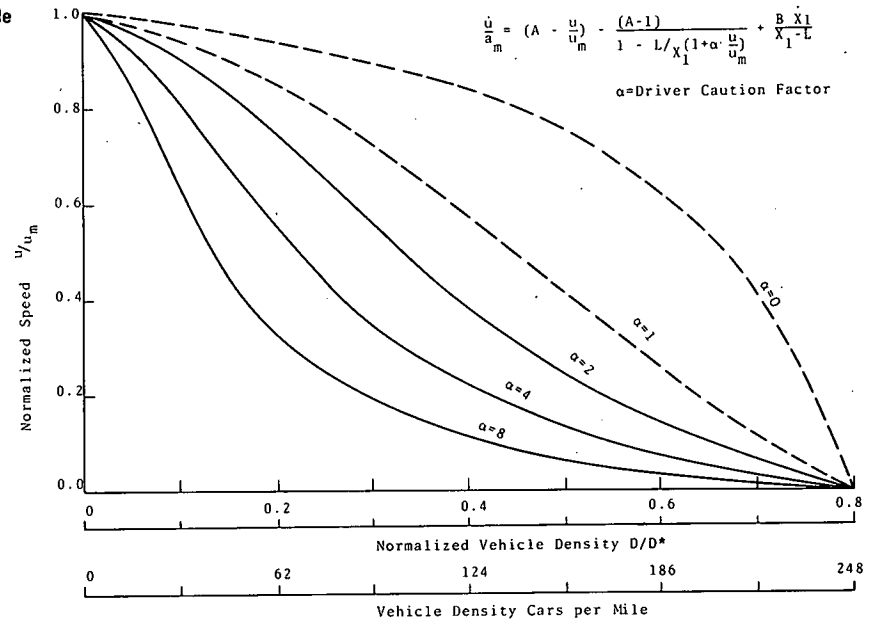


Figure 2. Traffic speed as a function of traffic flow rate.

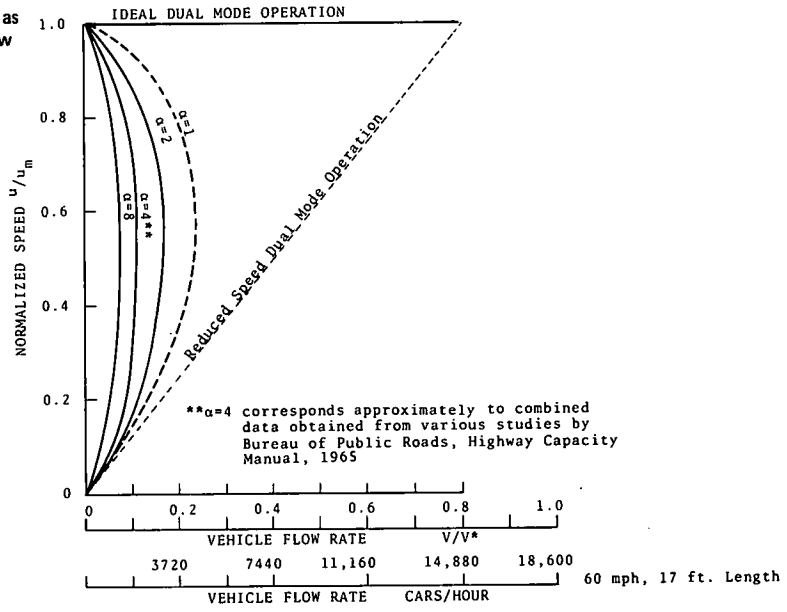
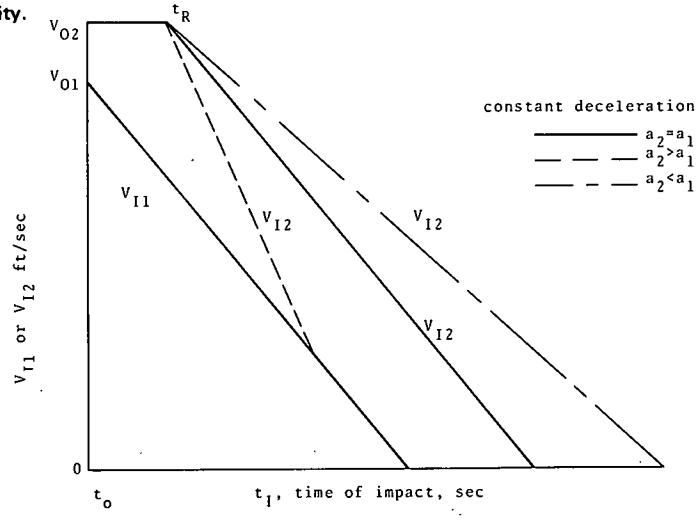


Figure 3. Impact velocity.



categories, which may or may not include passenger-restraint devices. Now, research and tests by human volunteers (6) indicate that deceleration levels as high as 35 *g* can be tolerated by humans under proper restraints and no impacts from sharp edges or packages. Automobile tests of adequately restrained bodies indicate tolerable acceleration levels as high as 35 mph (56.3 km/h) for frontal collisions with fixed objects; 10 mph (16.1 km/h) for side collisions with a tree or utility pole; and 25 mph (40.2 km/h) for side collisions with similar size vehicles.

This discussion indicates the particular importance of energy-absorption devices for optimum energy management to allow reduction of peak *g* levels. For instance, by increasing the impact deflection via appropriate crush devices, peak *g* levels can be decreased, and, together with selective use of several levels of energy absorption in the vehicle design, a nearly constant acceleration could well be attained. The various safety levels alluded to must necessarily be derived by properly correlating injury criteria with the vehicle structural response, which can be calculated by using appropriate analytic tools to perform parameter studies.

In design trade-off studies by the Transportation Sys-

Figure 4. Variations of impact velocity with headway distance for two vehicles that have similar initial velocities, deceleration speeds, and driver reaction times.

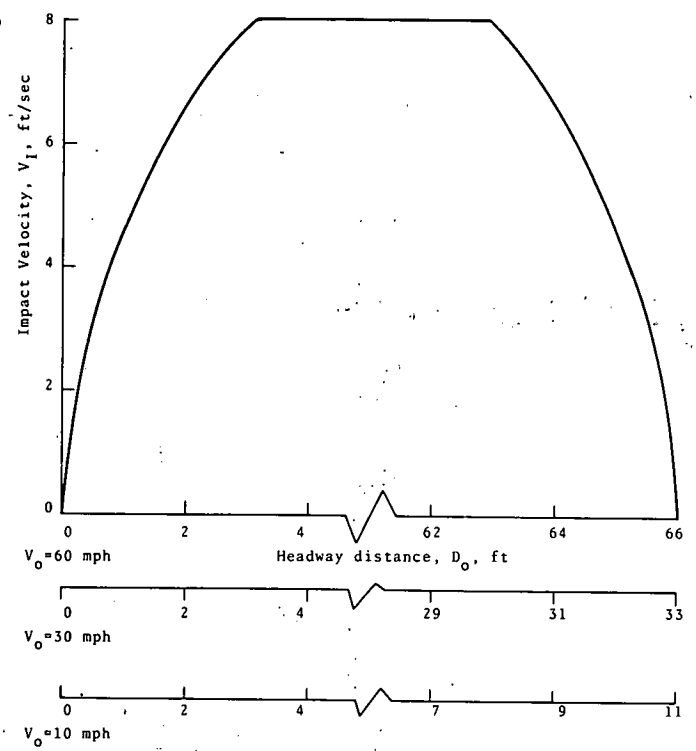
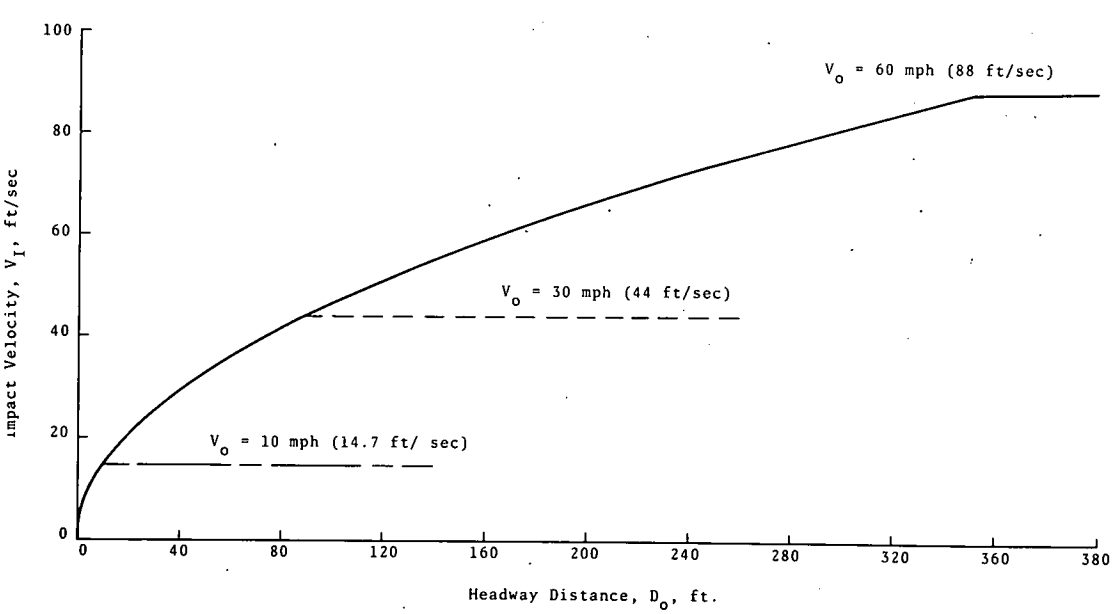


Figure 5. Variations of impact velocity with headway distance for two vehicles that have initial velocities but only one decelerates.



tems Center, some of the parameters being considered are

1. Bumper and energy absorber deformation characteristics (i.e., in terms of force levels and stroke lengths),
2. Relative locations of absorbers and large masses,
3. Restraint systems and biomechanics parameters (i.e., peak *g* levels tolerable), and
4. Overall allowable vehicle crush distances.

The available analytical tools used to conduct these studies are described in the next section.

ANALYTICAL TOOLS FOR CRASHWORTHINESS PREDICTION

In recent years various simulation programs have been developed to model the dynamic structural response under vehicle impact conditions. The models vary from the simple ones that give only average features of the overall response to the complex ones that provide greater detail in the response. Limited success has been achieved by using simplified spring-mass configurations with nonlinear resistances modeled by means of individual or group-component testing or other available information (7, 8, 9, 10, 11). Good overall agreement with vehicle-impact tests has shown that such simple models can be used successfully for specific configurations in conjunction with engineering judgment. In many cases they will be sufficient where "ball-park" predictions will serve to clarify the alternatives in any decision involving designs and standards. As will be discussed later, any dynamic response or crashworthiness analysis or both will serve to yield information helpful toward the identification of significant parameters that would allow extrapolation of available crash test data and the judicious planning of future tests; correlation with injury criteria in the development of safety limits and design standards; and proper design for energy absorption to optimize energy management during impact.

Of the simplified spring-mass models, representatives are the Tani-Emori (7) and the Kamal (8) models. In the Tani-Emori barrier impact model, there are two masses and four nonlinear resistive elements that represent gross structural properties. The model is capable of establishing general trends, and good correlation with test results of peak-body deceleration has been shown to be within 10 percent. In the Kamal model, there are three masses and eight nonlinear resistive elements. Masses represent the passenger compartment, engine transmission unit, and engine cross member. The resistances are determined experimentally by crushing gross vehicle components quasi-statically. In addition, an empirical strain rate correction factor is used in the program. The model can be used to perform parameter studies for existing designs and to predict general vehicle behavior (average values of acceleration are predicted). Good correlation with test results has been shown for body displacement and velocity.

A larger spring-mass model that can handle a greater number of parameters is embodied in the Battelle computer simulation program (9, 10). The Battelle model, shown in Figure 6 (10), handles collinear vehicle-to-vehicle as well as vehicle-to-barrier impact conditions. It can include 4 masses and as many as 35 nonlinear resistances in the form of elastoplastic springs, hydraulic energy absorber elements, and viscous dampers. The load deformation characteristics are obtained from theoretical or experimental data or both. An empirical strain rate correction factor is also included in the program. Rebound characteristics are programmed into the model for the purpose of handling impact between vehicles of different sizes in aggressiveness studies. This aspect of the program has yet to be checked out. The model has shown good simulation ability and in its present form is a useful tool for predicting general and specific behavior; for making preliminary evaluations and comparisons of vehicle energy-absorption devices; and for aiding in planning and evaluating crash tests.

With regard to the Battelle FMCCM model simulation ability, predicted values of peak vehicle crush using FMCCM runs fall close to a median line (Road Research Laboratory data curve) for a large variety of current production automobiles (Figure 7). Also, good correlation is indicated for predicted engine deceleration/time response for an FMCCM run and an experimental result obtained by Emori and Tani (7).

The next step in complexity for analytical simulation models involves the frame models (12, 13, 14, 15, 16). The Calspan model (12), which represents the earliest development in this regard, is applicable to two-dimensional frame structures. It is essentially a finite-element model with straight-beam elements, lumped masses at nodes, and localized plastic hinges at preselected nodes. The program is operational and at present can be useful for simple front frame and bumper configurations (Figure 8).

The 3-D TSC model (13), being developed at the Transportation Systems Center, can be regarded as a three-dimensional extension of the Calspan model in the sense that the assumptions on ideal plastic hinges and lumped masses at nodes are the same as those given by Shieh (12). The analysis and computer implementation of the TSC model are more heavily based on finite-element techniques (17, 18, 19, 20, 21, 22), which are being used to also include lumped parameter elements in the program by means of substructuring concepts (23, 24). Another three-dimensional frame model, developed by Lockheed and referred to as KRASH (14), has been installed and is running on TSC equipment. It is, however, operational only for specific aircraft applications at the present time. Much more work is needed for these and other more complex models (15, 16) to make them operational. By and large they are still in the prototype stage and have a limited number of check cases to their credit. It appears then that for the present the simplified mass-spring models, tailored specifically for dual-mode configurations, will form the initial analytical tools.

In summary, the simplified lumped parameter models are to be regarded as especially useful in making design trade-off studies in which many parameters are involved and larger models would be expensive. On the other hand, the large finite-element models are more suitable when a proposed design is studied for which parameter values are relatively fixed.

PARAMETER STUDIES AND ENERGY MANAGEMENT

By using analytical tools oriented specifically toward DMS configurations, parameter studies can be performed to yield useful data, especially during the feasibility stage of new ideas. Such studies will aid in identifying the important parameters characteristic of dual-mode vehicles and be useful in future crash testing, standard formulation, and structural design and optimum energy management concept development. The several design parameters will be defined, among other things, with respect to (a) bumper configurations, (b) energy absorbers and their relative locations, (c) shape and orientation of structural members (i.e., frames, compartment, engine or motor mounts), (d) overall and individual mass c.g. locations (i.e., engine or motor, pas-

Figure 6. Battelle collision models.

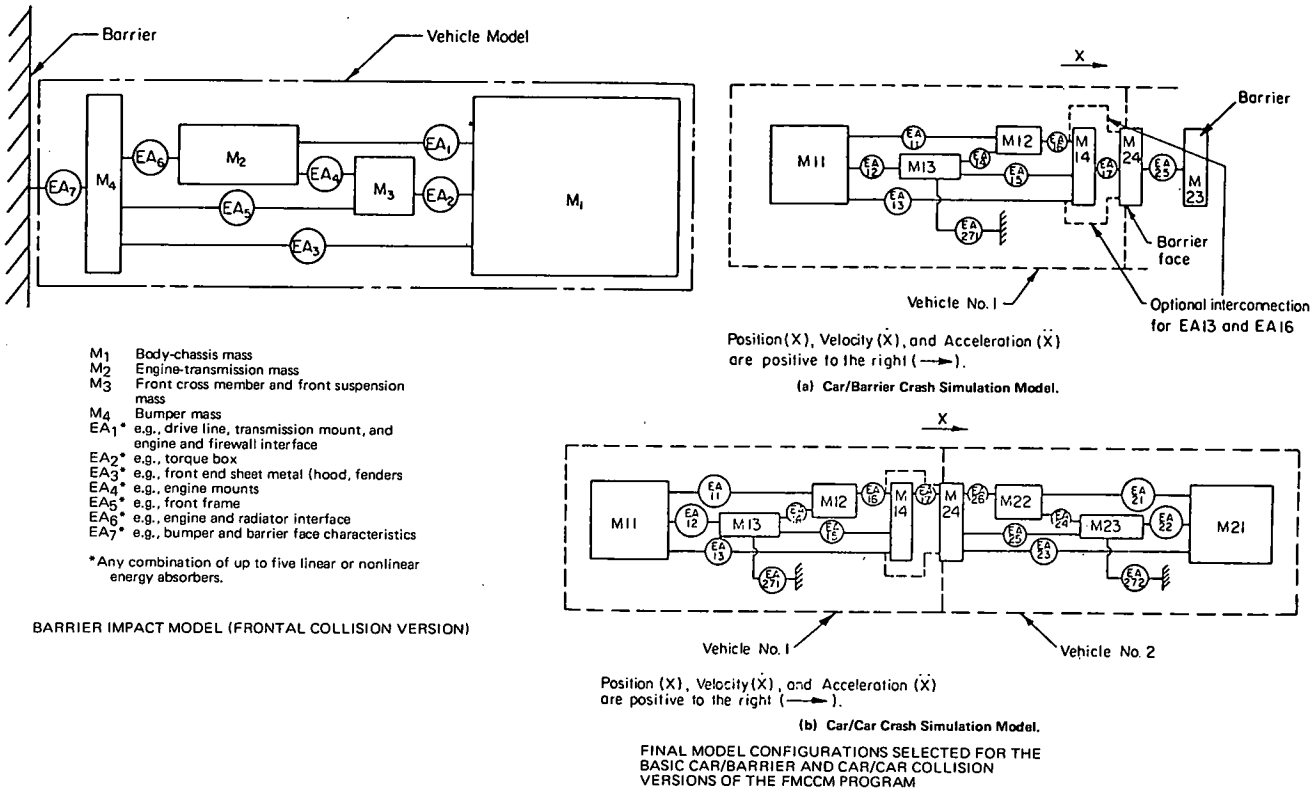
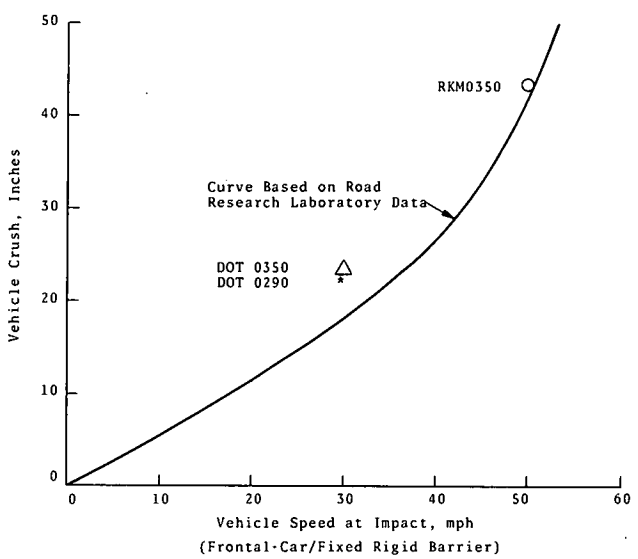


Figure 7. Vehicle crush as a function of impact speed.



senger compartment, (e) weight of payload relative to structural weight, and (f) guideway elasticity.

To contribute data for purposes of defining useful measures of safety and collision survivability, other impact parameters will consider (a) permissible overall crush distances and stroke lengths of absorbers, (b) maximum acceleration levels at selected points, (c) parameters that can be used in conjunction with injury criteria related to critical body regions (i.e., head, chest, pelvis, and femurs), and (d) restraint-system characteristics.

The model studies can be made to simulate various types of collisions, which can be classified according to the following impact arrangements: (a) impact into a rigid pole or line barrier, (b) impact into a flexible pole or line barrier, and (c) impact of two simulated vehicles that are of the same size or of different sizes and are involved in bumper-to-bumper and head-on collisions. The spring-mass models can be used in parameter studies in which stroke lengths and absorber activation forces will be design variables and the effects of different absorber characteristics and their location on peak acceleration can be evaluated.

The results of the various studies of simplified impacts between vehicles of different sizes can help in designing for least damaging aggressivity. For instance, a particular alternative might be to favor small vehicles and use different absorber force levels in each vehicle weight class. Studies can also be made to differentiate between high and low-speed impacts. For instance, in a defined "low" speed collision, the absorber will not be activated and the bumper with an auxiliary "oleo strut" device might be sufficient. It will also be possible to study the concept in which activation "multistage" absorbers can be defined to occur at certain established impact speed levels.

Proper energy management (i.e., optimum location of absorbers in the vehicle, force-deformation characteristics, proper use of restraint systems) can be instrumental in achieving safety at high-speed levels. In regard to energy-absorption devices, a general classification includes (a) innovative bumpers, (b) collapsible frames and structures, and (c) passenger-restraint systems. For high-speed collisions, devices in the second category play an important role and are discussed at length in other reports (4,25,26,27,28,29). The parameters that define different absorbers are (a) maximum energy absorbed in a given stroke (determines the speed of impact that can be decelerated), (b) peak force in a

Figure 8. Modified automotive front frame and bumper.

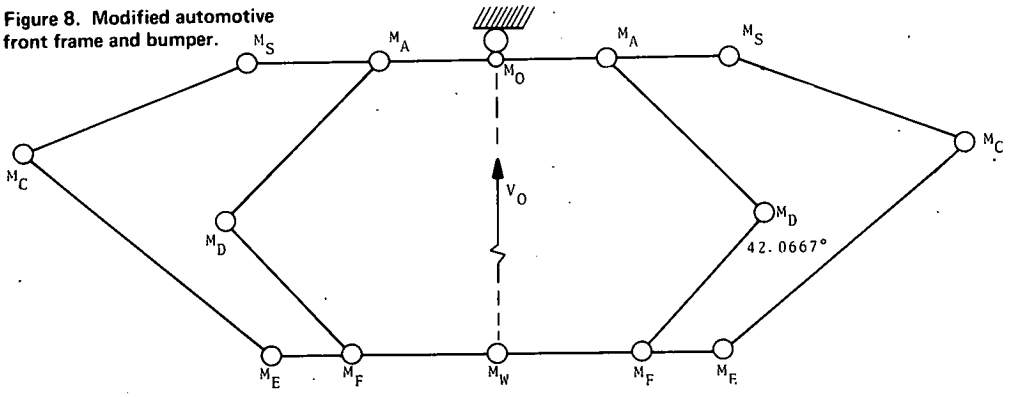


Figure 9. Energy-absorber characteristics.

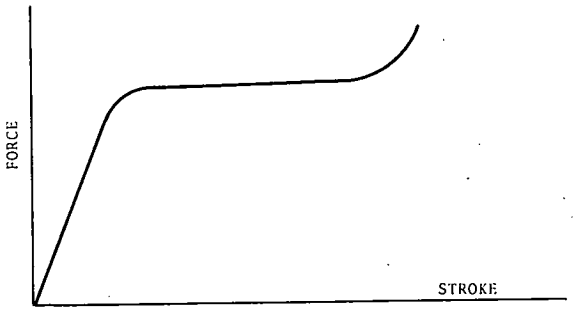


Figure 10. Passenger compartment waveform most beneficial to occupant.

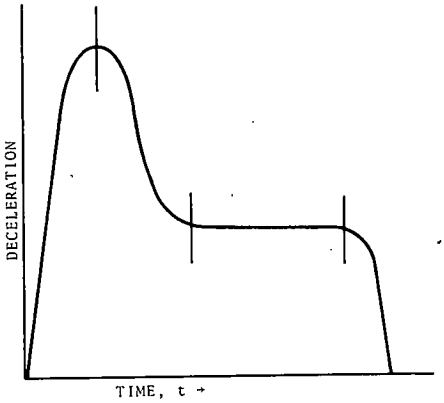


Figure 11. Head-neck model.

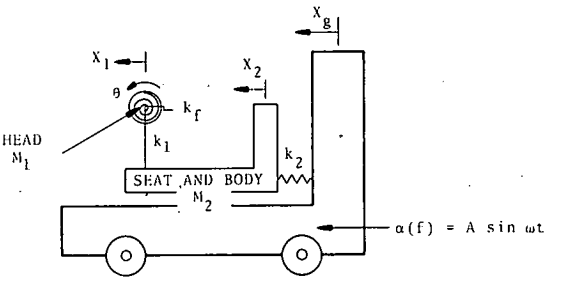


Figure 12. Crash system.

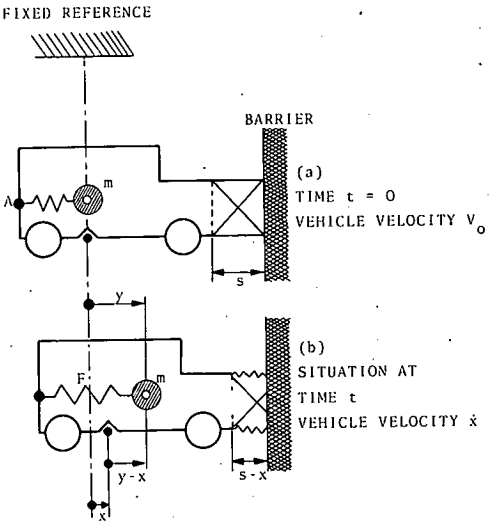
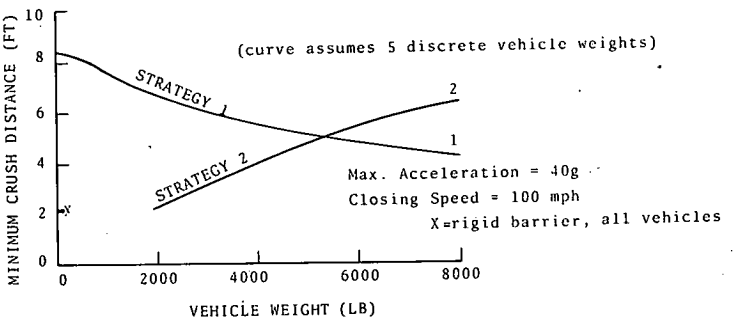


Figure 13. Minimum crush distance versus vehicle weight.



given stroke (leads to peak deceleration to be experienced by occupants), and (c) stroke length. The favored device attains least peak force for a given absorption energy and stroke.

A typical force-stroke curve is shown in Figure 9. The linear portion of the curve is desirable in that the jerk rate is limited. Another desirable feature is to have a substantial flat portion. Since the area under the curve equals the kinetic energy that can be absorbed, the flat part of the curve implies uniform force levels during absorption. Selection of an appropriate device depends on the application. For instance, a variable force device could provide lower stroking forces for low-speed impacts while avoiding the extremely long strokes necessary for high-speed impacts. A low initial force followed by a higher force (at longer strokes) could be achieved with a velocity-sensitive device, but it could also be achieved with multiple-stage, constant-force devices (4). As detailed in other reports (4, 25), selection can be made from extrusion devices, material deformation devices, and friction devices. In any case, some devices can yield shorter stroke lengths for a given amount of energy, while others are more easily adapted for reusability. Both will influence DMS design. On the basis of weight and cost, crushable honeycomb appears to be the simplest, lightest, and least expensive system that can provide a constant deceleration (4).

BIOMECHANICS DYNAMIC MODELS AND ASSESSMENT OF INJURY POTENTIAL

Well-defined ranges for certain parameters that will allow quantitative evaluation of passenger safety need to be developed. In this regard biomechanics dynamic models are essential so that the structural dynamic response can be related to injury criteria. As such, mathematical models of different aspects of the acceleration response of the human body have been developed. Some results to date have yielded the acceleration waveform for the passenger compartment most beneficial to the occupant, as shown in Figure 10. Simple studies of an articulated model of the occupant bear this out (30, 31). In a frontal impact, for a fully lap- and shoulder-belted occupant, these studies show that an early high deceleration pulse followed by a lower sustained deceleration level (Figure 10) is least damaging to the passenger. Experimental investigations have substantiated these studies (33). Excellent reviews on various models are given elsewhere (34, 35). For instance, the mathematical model of the head and neck that is described by Martinez and others (36) and shown in Figure 11 can be viewed as a development aimed ultimately at direct injury prediction in the sense that it simulates the body mechanism (i.e., the head-neck action) associated with the specific injury of interest, whiplash. As more definitive injury criteria are developed for this type of injury, the analytically predicted actions can be compared with injury thresholds. The head-neck model (36) is limited to planar motions in rear collisions and to linear system characteristics.

Another highly simplified situation considers point-mass models that have been used in fundamental studies (37, 38) of the behavior of a restrained occupant in a frontal automobile collision (Figure 12). The evaluations of injury potential (38) were based entirely on consideration of the occupant (i.e., point mass) deceleration. The evaluations of relative injury hazards (37) were based on predictions of the velocity with which the simulated occupant would strike the vehicle interior, with consideration also given to the level of occupant deceleration produced by the restraints. These models, although simplified, can be used with analytical computer simulation of vehicle structural dynamic response to obtain useful indexes of injury.

RECENT STUDIES IN PREDICTING OPTIMUM CRUSH CHARACTERISTICS OF VEHICLES

An important area, being studied at the Transportation Systems Center, involves the ability to design for optimum crush characteristics of vehicles of various sizes. It is related to energy management concepts and the proper use of energy-absorption devices. For instance, depending on the strategy, lighter vehicles in a collision may require absorbers that must deform with larger crush distances than may be required by heavier vehicles. Two possible strategies are indicated in Figure 13. In strategy 1, all new vehicles should be capable of collision with a rigid 8000-lb (3629-kg) vehicle closing at 100 mph (160.9 km/h) and colliding at 50 mph (80.5 km/h) with a rigid barrier. In strategy 2, all vehicles should be capable of collision with a rigid barrier at 50 mph (80.5 km/h), and all smaller automobiles that are closing at 100 mph (160.9 km/h) and are designed to this criterion must be protected. Other trade-offs are currently under study at the Transportation Systems Center.

REFERENCES

1. C. F. Vail. Dynamic Modelling of Automobile Structures From Test Data. Shock and Vibration Symposium on System Identification of Vibrating Structures, ASME, New York, 1972, pp. 149-177.
2. I. K. McIvor. Modelling and Simulation as Applied to Vehicle Structures and Exteriors. Proc., Conference on Crashworthiness, March 30, 1973, National Highway Traffic Safety Administration.
3. K. J. Saczalski. Structural Problems Associated With the Prediction of Vehicle Crashworthiness. Surveys of Research in Transportation Technology, ASME, AMD Vol. 5, 1973, pp. 87-108.
4. J. W. Larson and others. Engineering Program for Assessment of the Crashworthiness of Existing Urban Rail Vehicles. Calspan Corp., Rept. ZL-5369-K-1, Oct. 1973.
5. B. S. Holmes and J. D. Colton. Scale Model Experiments for Safety Car Development. International Automotive Engineering Congress, Detroit, Jan. 1973, SAE, Paper 730073.
6. T. J. McGean and N. W. Lutkefedder. Application of Automotive Crash Survivability Research to Close Headway PRT Systems. Conference on Transportation, Sept. 1973, ASME, Paper 73-ICT-52.
7. M. Tani and R. I. Emori. A Study in Automobile Crashworthiness. Automotive Engineering Congress, Detroit, Jan. 1970, SAE, Paper 700175.
8. M. M. Kamal. Analysis and Simulation of Vehicle to Barrier Impact. International Automobile Safety Conference, Detroit, May 1970, SAE, Paper 700414.
9. J. T. Herridge and R. K. Mitchell. Development of a Computer Simulation Program for Collinear Car/Car and Car/Barrier Collisions. Battelle Columbus Laboratories, Rept. DOT-HS-800-645, 1972.
10. J. T. Herridge and R. K. Mitchell. Program for Collinear Car/Car and Car/Barrier Collisions. Intersociety Conference on Transportation, Denver, May 1973, ASME, Paper 73-ICT-34.
11. P. R. Spencer. Parametric Study of Side Impact Vehicle Aggressiveness. 18th Annual Meeting of Institute of Environmental Sciences, March 1972.
12. R. C. Shieh. Basic Research in Crashworthiness

II: Large Deflection Dynamic Analysis of Plane Elasto-Plastic Frame Structures. Calspan Corp., Rept. YB-2987-V-7, Dec. 1972.
13. J. N. Rossettos, H. Weinstock, and S. Pasternack. Analytical Finite Element Simulation Model for Structural Crashworthiness Prediction. Transportation Systems Center, Cambridge, Mass., Interim Rept. DOT-TSC-NHTSA-73-12, 1974.
14. G. Wittlin and M. Gamon. Experimental Program for Developing Rotary Wing Aircraft Structural Crashworthiness Criteria and Concepts. Lockheed California Co., Final rept.
15. J. W. Young. CRASH: A Computer Simulation of Nonlinear Transient Response of Structures. Philco-Ford Corp., Rept. DOT-HS-09-1-125B, March 1972.
16. J. E. Thomson. Vehicle Crush Predictions Using Finite Element Techniques. SAE, Paper 730157, Jan. 1973.
17. O. C. Zienkiewicz. The Finite Element Method in Engineering Science. McGraw-Hill, New York, 1971.
18. I. Holand and K. Bell, eds. Finite Element Methods in Stress Analysis. TAPIR, Technical Univ. of Norway, Trodheim, 1969.
19. C. S. Desai and J. F. Abel. Introduction to the Finite Element Method. Van Nostrand, New York, 1972.
20. J. T. Oden. Finite Elements of Nonlinear Continua. McGraw-Hill, New York, 1972.
21. J. F. McNamara and P. V. Marcal. Incremental Stiffness Method for Finite Element Analysis of Nonlinear Dynamic Problems. International Symposium on Computer Methods in Structural Mechanics, Urbana, Ill., Sept. 1971.
22. J. A. Stricklin, W. E. Haisler, and W. A. von Riesemann. Formulation, Computation, and Solution Procedures for Material and/or Geometric Nonlinear Structural Analysis by the Finite Element Method. Sandia Laboratories, Albuquerque, Rept. SC-CR-72-3102, Jan. 1972.
23. J. S. Przemienieki. Matrix Structural Analysis of Substructures. AIAA Journal, Vol. 1, No. 1, Jan. 1963.
24. H. C. Martin and G. F. Carey. Introduction to Finite Element Analysis. McGraw-Hill, New York, 1973.
25. M. C. Shaw. Designs for Safety: The Mechanical Fuse. Mechanical Engineering, Vol. 94, No. 4, April 1972.
26. E. J. Brabin. Devices Used in Crash Testing. Motor Industry Research Association Bulletin, Great Britain, 1968, p. 11.
27. J. R. McGehee, M. E. Hathaway, and E. J. Zavads. Frangible Tube Energy Dissipation. U.S. Patent 3 143 321, 1964.
28. S. W. Blank. Deformed Shock Absorbing Devices. U.S. Patent 3 437 367, 1969.
29. A. Blake. Soaking Up Shock With Rolling Diaphragms. Machine Design, Vol. 46, No. 2, Jan. 1974.
30. D. E. Martin and C. K. Kroell. Vehicle Crush and Occupant Behavior. Automotive Engineering Congress, Detroit, Jan. 9-13, 1967, SAE, Paper 670034.
31. D. F. Moore. Minimization of Occupant Injury by Optimum Front End Design. International Automobile Safety Conference, Detroit, May 13-15, 1970, SAE, Paper 700416.
32. D. N. Renneder. A Basic Study of "Energy-Absorbing" Vehicle Structure and Occupant Restraints by Mathematical Model. SAE Trans., Vol. 76, Paper 670897, 1967.
33. J. J. Ryan. Reduction in Crash Forces. Proc., Fifth Stapp Conference, Univ. of Minnesota, Sept. 1961.
34. V. O. Roberts, E. L. Stech, and C. T. Terry. Review of Mathematical Models Which Describe Human Response to Acceleration. ASME, Paper 66-WA/BHF-13, 1966.
35. H. E. von Gierke. Biodynamic Response of the Human Body. Applied Mechanics Reviews, Dec. 1964.
36. J. L. Martinez and others. The Whiplash Injury: A Study of Head-Neck Action and Injuries in Animals. ASME, Paper 65-WA/HUF-6, Nov. 1965.
37. G. Grime. Safety Cars. U.K. Road Research Laboratory, Rept. 8, Harmondsworth, 1966.
38. A. Egli. Stopping the Occupant of a Crashing Vehicle. Automotive Engineering Congress, Detroit, Jan. 9-13, 1967, SAE, Paper 670038.