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TRANSIENT ANALYSIS TECHNIQUES IN PERFORMING IMPACT AND CRASH DYNAMIC STUDIES

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Advances in technology are made typically in response to new performance requirements. The area of crash simulation is no exception. Because of the emphasis now being placed on crashworthiness as a design requirement, increasing demands are being made by various organizations to analyze a wide range of complex structures that must perform safely when subjected to severe impact loads, such as those generated in a crash event.

The ultimate goal of crashworthiness design and analysis is to produce vehicles with the ability to reduce the dynamic forces experienced by occupants to specified acceptable levels, while maintaining a survivable envelope around them during a specified crash event.

Figures 1 through 3 show examples of the type of impacts that must be simulated.

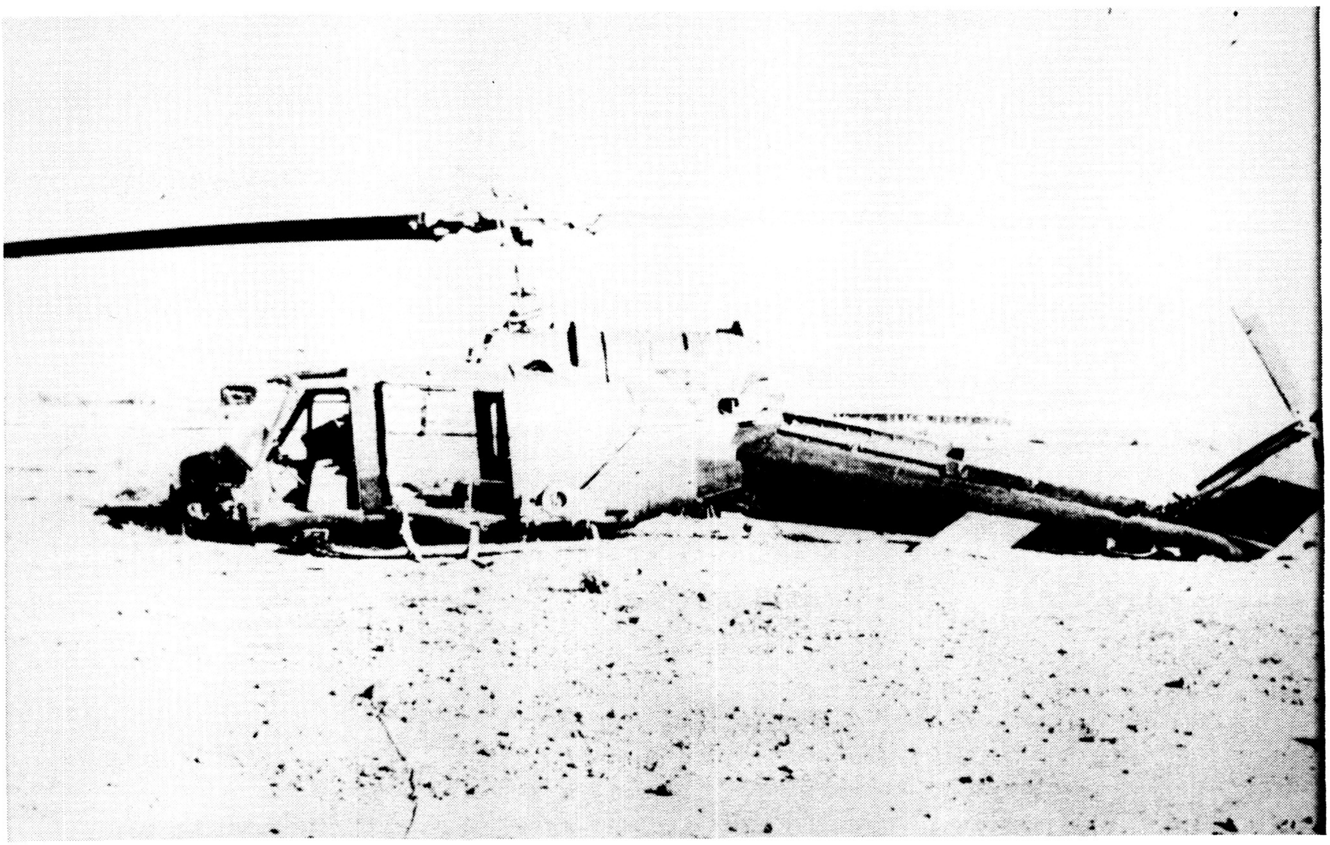

Figure 1. Vertical impact of helicopter.

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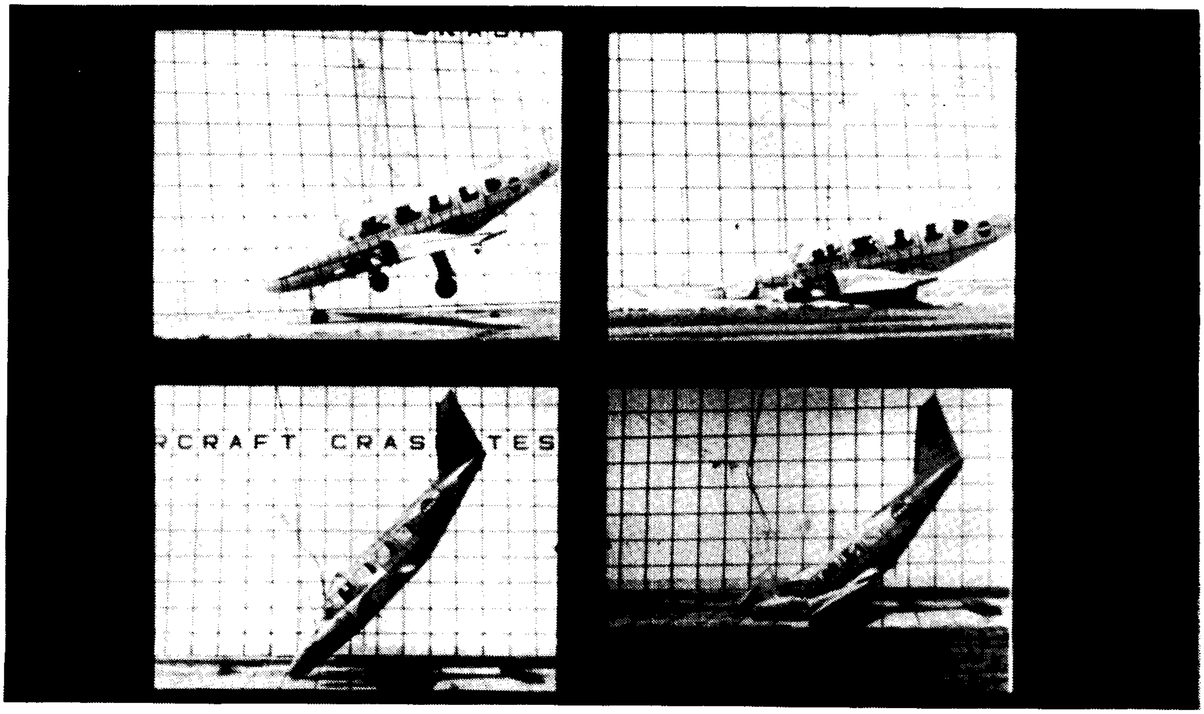

Figure 2. NASA/FAA general-aviation crash dynamics program.

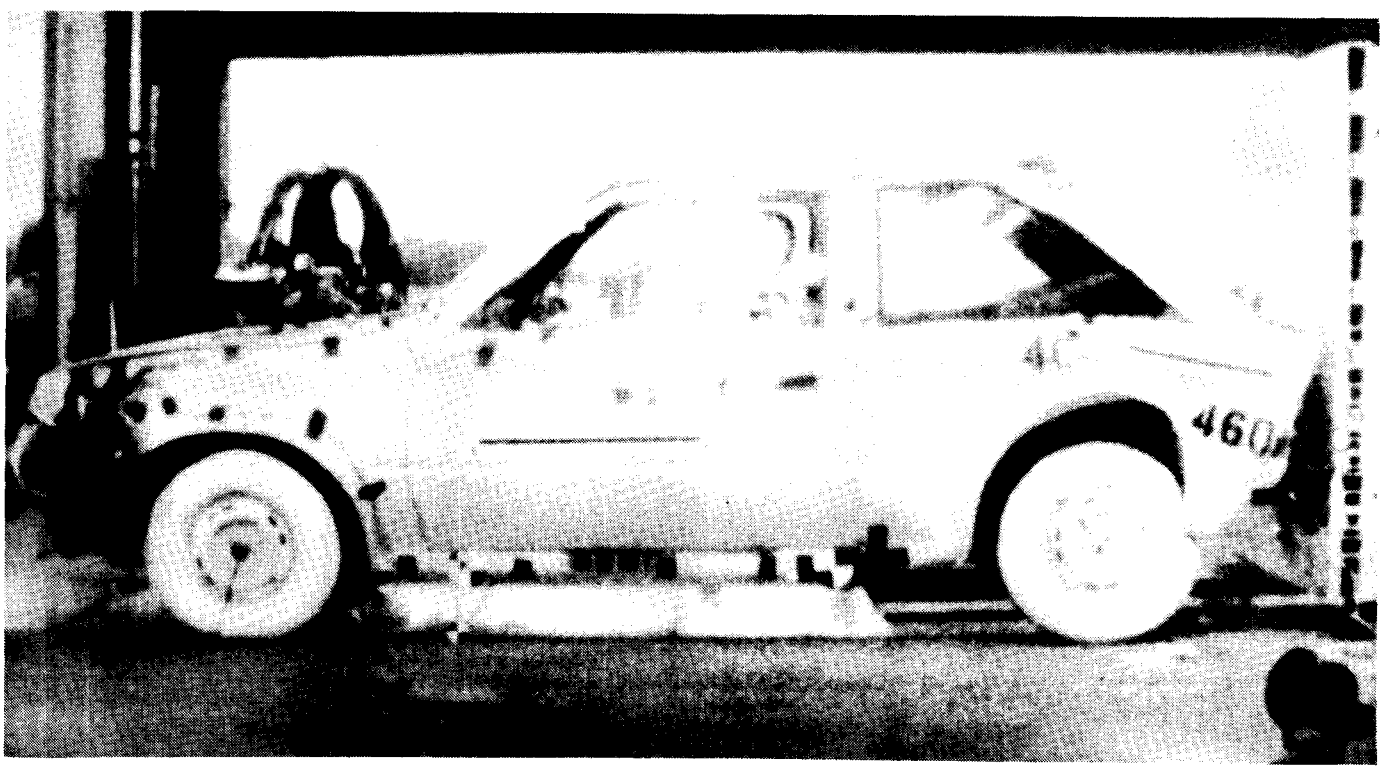

Figure 3. Rear-impact test of automobile.

The requirement for crashworthy vehicles has been a motivating force behind the development of computer programs for use in a vehicle crash simulation. Development of these programs has been the direct result of advances in both structural mechanics and computer sciences. Specifically, advances in finite-element methods, made feasible by rapid developments in computer hardware and software, form the foundation on which these programs were developed. After more than a decade of development, a number of programs are now available and are used for practical analysis and design.

The capability of one such program is reviewed and some experiences gained in the crash evaluation of automobile and aircraft structures are related.

There are a number of requirements that are essential to the simulation of a crash event (fig. 4). Although these requirements involve several areas, the most obvious are:

- A theory that treats the large elastic-plastic deformation associated with crushing of structural members including strain-rate effects where applicable

- The techniques for nonlinear boundary conditions required to simulate internal contact/rebound between structural parts or between structure and a barrier or contactor

- A capability to model a variety of structural types, typical of aircraft, and automotive structures

- Accurate and efficient numerical techniques for integrating the nonlinear equations of motion

These requirements include all of the areas that are the subject of current research in computational mechanics. However, methods to treat the essential features of all of these requirements have reached a sufficient level of maturity to be implemented into a code for crash simulation. As such, techniques that account for the essential features of each of the above stated requirements have been incorporated into our DYCAST code.

- Large elastic-plastic deformation with failure
- Variable contact/rebound
- Modelling capability for variety of structural types
- Accurate and efficient numerical techniques

Figure 4. Essential requirements of structure crash simulation.

DYCAST is a nonlinear structural dynamic finite-element computer code that started from the plans system of finite-element programs for static nonlinear structural analysis (fig. 5). It was originally developed for aircraft crash analysis with partial support by NASA Langley.

The equations of motion used in DYCAST are developed within the framework of the finite-element method and are based on the updated Lagrangian formulation for geometric nonlinearity and an incremental plasticity theory for material nonlinearity.

The updated Lagrangian approach is particularly effective for the nonlinear problem associated with crash simulation using beam, membrane, and plate elements. This is because large shape changes due to the progressive crushing and folding of the structure are accounted for by successive updating of the nodal coordinates. The nonlinearities due to the internal loads (for example, the change in stiffness due to the "beam column effect") are included so that compressive forces dominant in a crash event will act through the geometric nonlinearities to reduce the stiffness of the structure.

The following figure outlines the essential features of DYCAST. Our intent in presenting these features is to indicate our view of the necessary minimum requirements for crash analysis.

- Material nonlinearity
    - Incremental plasticity theory
    - Von Mises yield criterion
    - Kinematic hardening
    - Element maximum strain failure criterion
    - Subincremental strategy
- Geometric nonlinearity
    - Updated Lagrangian

Figure 5. DYCAST - Dynamic crash analysis of structures.

The governing matrix equation for the updated Lagrangian formulation is:

$$[M]\ \{\ddot{U}\}_{t+\Delta t} + [K_T + K_G]\ \{\Delta U\}_{t+\Delta t} + \{F\}_t = \{P\}_{t+\Delta t} \tag{1}$$

Equation 1 is the linearized equation of motion between a known equilibrium state, denoted by t, and an unknown equilibrium state, denoted by t + Δt, incrementally adjacent to it. It explicitly contains terms that reflect the current material state, and nonlinearities from the strain displacement relations.

The quantities in equation 1 are defined as follows: $\{\ddot{U}\}_{t+\Delta t}$, $\{\Delta U\}_{t+\Delta t}$ are the unknown accelerations and displacement increments, [M], $[K_T]$, $[K_G]$ are the mass, tangent stiffness, and initial stress stiffness matrices, respectively, and $\{F\}_t$, $\{P\}_{t+\Delta t}$ are the known internal and external forces at the time denoted by their subscripts.

The matrix $[K_T]$ is a function of the material behavior and therefore explicitly contains the plasticity theory implemented in the code. We have implemented the Prager-Ziegler kinematic hardening theory based on the Von Mises-Hill yield criterion for orthotropic (and isotropic) materials and used an effective stress-strain relation for multiaxial stress states. Postyield behaviors can be either: no strain hardening (perfect plasticity), linear hardening, or nonlinear hardening. Additionally, a multistep subincremental strategy has been employed to ensure that the plastic constitutive equations embodied in $[K_T]$ are never violated.

Assuming continuing and unlimited elastic or plastic deformation in a crash simulation is equivalent to assuming that a structural element will dissipate unlimited energy as it deforms along a particular load-deformation path. Obviously this can overpredict the energy that can be dissipated since actual materials will fail at some maximum deformation. To accommodate this behavior, maximum strain failure criteria have been implemented in our material model. Once these criteria are satisfied at a point, the stiffness and force contributions at that point are deleted. When a specified set of such points in an element has reached its failure strain, the element's stiffness and force contribution to equation (1) is not assembled. Provision has also been made to delete elements manually based on some other failure criterion or on engineering judgment. (Fig. 6.)

Inertia forces | Internal forces | Incremental internal forces | External forces

$$M\ddot{U}_{n+1} + F_n + \left[K_T + K_G\right]_n \Delta U_{n+1} = P_{n+1}$$

Delete failed members | Material stiffness plasticity | Geometric stiffness stress state

Update geometry
Delete failed members

Figure 6. Nonlinear dynamics - equation of motion.

At an early stage, it was clear that we should implement a variable time step integrator, i.e., one that enables the time step to be changed at different instants of the response. Such a procedure has obvious advantages over one with a constant step, particularly in complex problems arising in practical application where system nonlinearities and dynamic response are varying continuously throughout the history. Our experience has indicated that this is particularly true for crash simulation.

Variable time step integrators of both the explicit and implicit type have been implemented in DYCAST. These are the explicit Modified Adams and the implicit Newmark-Beta and Wilson-Theta methods. An explicit constant step central difference integrator is also available, as well as static, bifurcation buckling, and free vibration options.

Implementation of implicit integration in DYCAST is as follows: The technique used solves equation (1) at each step subject to the integrator recurrence relations and then performs iterations of the modified Newton type based on an imbalanced force stemming from errors in satisfying the equation of motion.

A variable time step procedure is defined by requiring that the number of iterations in each time step be less than a prescribed value. If this criterion is violated, the time step is halved. Conversely, if the solution converges in one iteration for a prescribed number of consecutive steps the time step is increased. An upper bound for the time step is also specified.

In our initial work, we used the explicit modified Adams integrator exclusively. However, we quickly found that the admissible time step for a nonuniform mesh with beam, plate membrane elements, and nonlinear springs was unreasonably small. Consequently, our current activities are associated almost exclusively with the implicit implementation. Our experience in what we describe as moderate sized problems of 1500 degrees of freedom (DOF) or less has led to a preference of the implicit method in this problem class because solution time per increment is divided almost equally between element level calculations and global solution of a matrix equation. As the number of degrees of freedom increase, the solution of the global matrix equation begins to dominate. Some later figures show examples of some typical calculations. Experience with vector processing (CRAY 1 or CDC CYBER 205) has extended this degree-of-freedom range.

Research is continuing that, hopefully, will address these issues further in such areas as mixed explicit-implicit integration, subcycling, and element-by-element solution strategies that can utilize concurrent processing. (Fig. 7.)

- Implicit
  - Variable time step
  - Modified Newton iteration
  - Newmark -β
- Explicit
  - Modified Adams — variable step
  - Central difference — constant step

Figure 7. Integration of equations of motion.

Nonlinear springs are useful to model the crush behavior of components for which data are available and whose behavior may be too complex to model otherwise (e.g., for energy-absorbing devices, for gap elements with variable contact/rebound, for nonlinear moment-rotation curves of collapsing beams, and for various other nonlinearities). (Fig. 8.)

- Nonlinear boundary conditions
  - Contact/rebound simulated with special "gap springs"
  - Contact element with simple friction

Figure 8. DYCAST - Dynamic crash analysis of structures.

The capabilities outlined represent the essential requirements for a crash simulation. There are other requirements that can be described as operational features, which nevertheless, are essential to the performance of a simulation in an efficient and timely manner. The most important of these is an efficiently designed restart procedure. In keeping with the path dependent nature of nonlinear analysis, this capability enables an analyst to perform a crash analysis in manageable time segments and to examine intermediate results to see if they appear meaningful before deciding if the analysis should be continued.

Adjunct to this is the manner used to display the results. Because the volume of data that is generated can easily overwhelm an analyst, selected summary tables of results along with graphical display are important. Postprocessing graphics include the display of the deformed model at any time and plot histories of displacements, velocity, and acceleration for any nodal degree of freedom.

Access to the restart file by a peripheral program to selectively print additional data is also desirable.

Figures 9 through 12 show necessary process control for restart and some examples of postprocessing graphics.

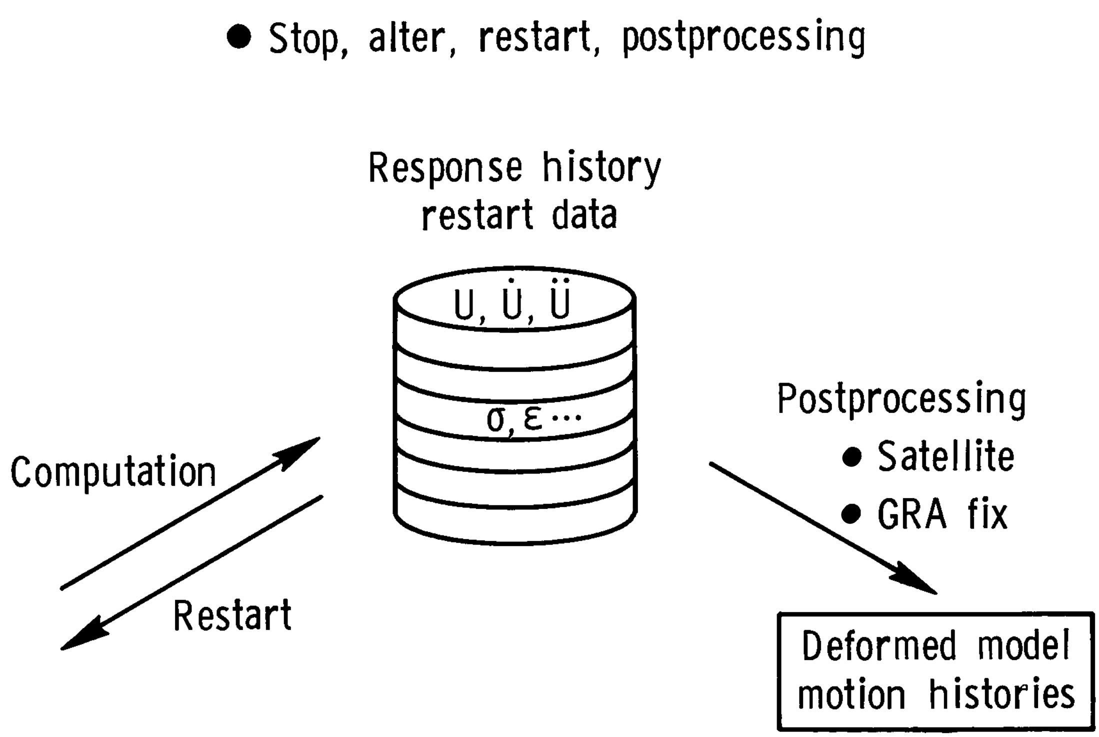

Figure 9. DYCAST - process control.

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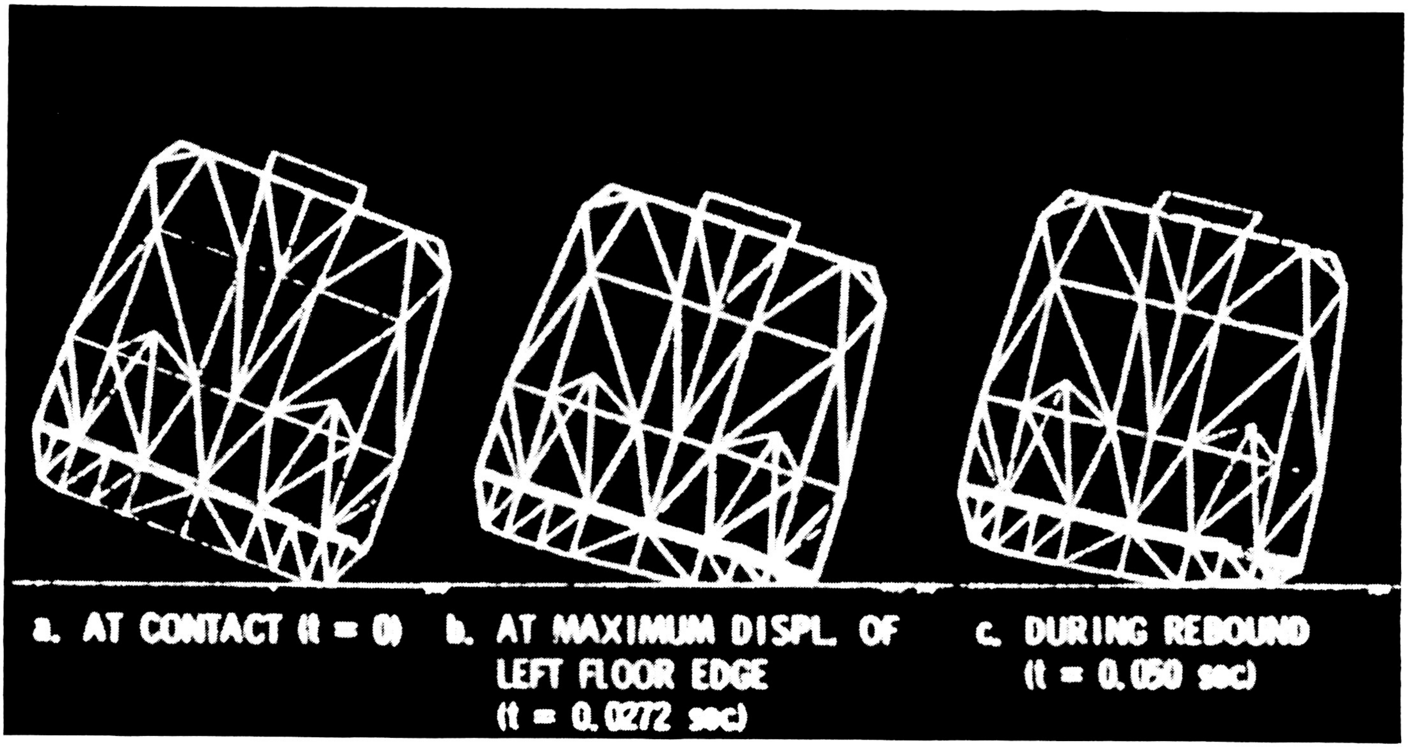

Figure 10. Deformed DYCAST model roll drop.

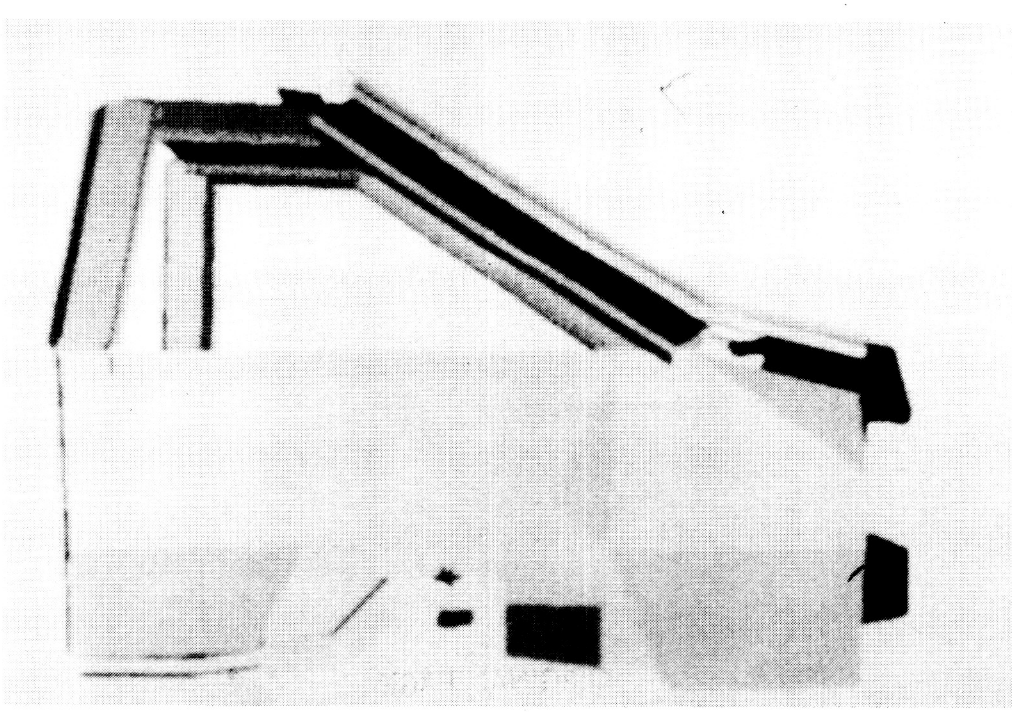

Figure 11. Undeformed model.

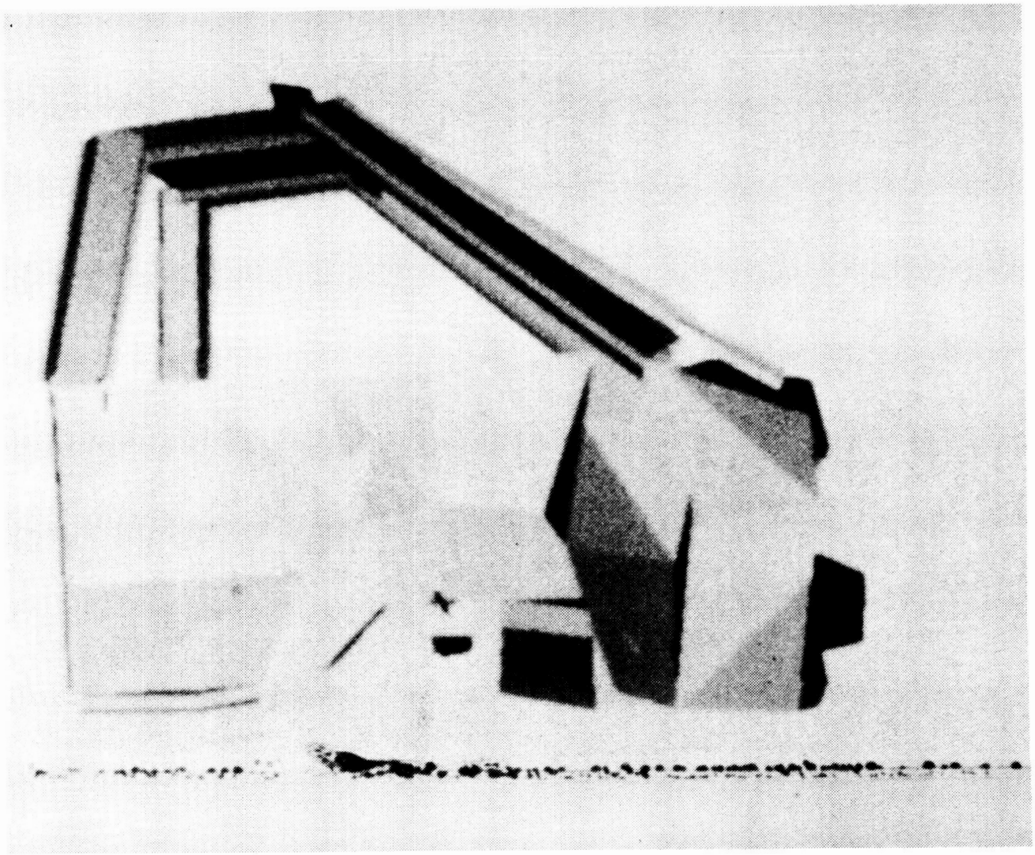

Figure 12. Undeformed model.

Experience with mathematical crash simulations has shown that, while using an adequate nonlinear dynamic computer code is essential, it is not enough. The analyst must also have some expertise in the art of modelling the structure for the nonlinear crash analysis in order to produce sufficiently accurate results within an acceptable time and cost range.

The total costs of an analysis are composed of the labor involved in creating the model and evaluating the results and the costs of using the computer. Although the modelling labor cost can be large, it is rarely discussed in the technical literature, probably because of its variability. A first-time full-vehicle finite-element model could require from one to four man-months of effort to prepare and verify, depending on factors such as the convenience of the vehicle geometry data (digital data base or drawings on paper), the use of computer graphics, and the experience of the personnel. In any case, modelling labor costs are dependent on the model size and complexity (quantity of nodes, elements, and DOF). However, after preparation and verification of the finite-element model is complete, it can be modified easily, at small cost, enabling the investigation of the effects of structural modifications.

The computational costs are dependent on model size and complexity. If it were "the best of all possible worlds" we might produce a model as shown in figure 13 for the crash analysis of an automobile. This is the type of model frequently used for linear analysis. Because of the limitation of current computers, a nonlinear dynamic analysis of this type of model is currently not feasible. However, to do so is our goal!

At the present time we consider a nonlinear vehicle crash model of 1500 DOF to be large for use on even the fastest scalar computers such as the IBM 370/3081 or CYBER 760. From two to ten restarts could be required to complete such a crash simulation. However, the new vector computers such as the CRAY-1 and the CYBER 205 allow at least a twofold to fourfold increase in overall computation speed coupled with increased memory size. In the future, improvements in both software and hardware should continue to reduce computer expense to allow more detailed models to be analyzed in smaller time periods.

Examples of computer time for two representative structures for our code are shown in figures 13 through 16.

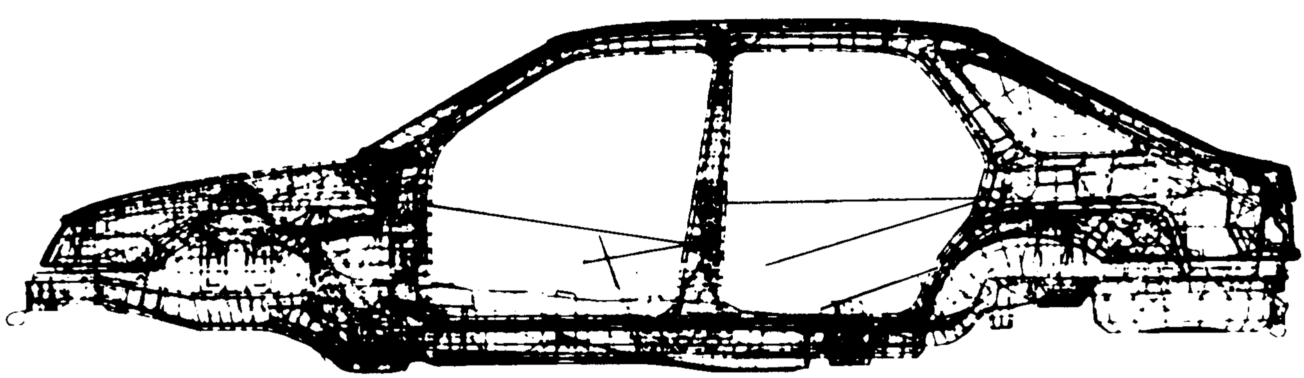

Figure 13. Detailed finite-element model of automobile.

679 DOF, 156 SBW, 263 elements

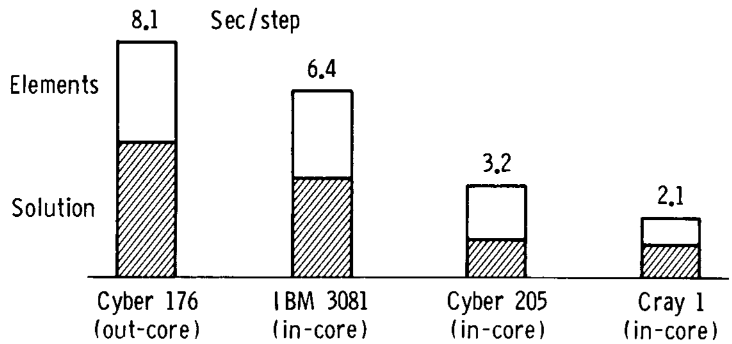

Figure 14. Scalar versus vector computers-autos rear impact.

1431 DOF, 813 SBW, 1460 elements

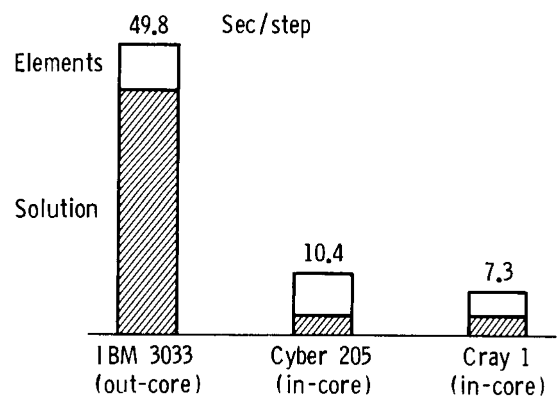

Figure 15. Scalar versus vector computer-helicopter drop.

SBW/DOF	Degrees of freedom										
	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000
0.10	25614	55564	97389	151089	216664	294114	383439	484639	597714	722664	859489
0.20	46889	103414	182439	283964	407989	554514	723539	915064	1129089	1365614	1624639
0.30	65664	145639	257489	401214	576814	784289	1023639	1294863	1597963	1932938	2299788
0.40	81939	182239	322539	502839	723139	983439	1283739	1624039	2004339	2424639	2884939
0.50	95715	213215	377590	588840	846965	1151964	1503840	1902589	2348215	2840714	3380090
0.60	106989	238564	422639	659214	948289	1289864	1683939	2130514	2629589	3181164	3785239
0.70	115764	258289	457689	713964	1027114	1397139	1824039	2307814	2848464	3445989	4100389
0.80	122039	272389	482739	753089	1083439	1473789	1924139	2434489	3004839	3635189	4325539

Figure 16. Core requirements for matrix assembly and solution.

In the early use of nonlinear finite-element models for crash analysis, a purely theoretical approach was attempted in which all the behavior was modelled using the finite elements. However, in the solution of practical problems involving actual vehicle structures, it quickly became apparent that some hybrid elements would be required in which the user specifies the nonlinear stiffness that is derived either from test data or a separate analysis. In the simplest case this would involve the modelling of a specific energy-absorbing component by a nonlinear spring with a user-specified crush curve. In the more complex cases, the collapse of a section of structure could be represented by a hybrid element, either because the crush test data were already available or because the nonlinear behavior of a component would be so complex and so localized that it would require too much computational effort in a small part of the vehicle.

This led to a modelling strategy in which we recognize three distinct behavioral zones in a vehicle structure when preparing a nonlinear finite-element model for crash analysis. These are linear behavior, moderately nonlinear behavior, and extremely nonlinear behavior zones. In the linear behavior zones, no nonlinear behavior is expected, and these zones are modelled as lumped masses or as rigid bodies with finite dimensions, or occasionally with a small number of deformable finite elements. In the moderately nonlinear zones, plasticity, material failure, and large deflections are expected, but the large deformations are not confined to highly localized regions. These zones are represented by a distribution of nonlinear finite elements in sufficient quantity and of the types required to allow for expected modes of deformation and failure. Here, the attempt is made to minimize the complexity while still approximating adequately the necessary stiffnesses. In the extremely nonlinear zones locally large deformations occur, such as: the collapse of a thin-wall hollow beam into accordion bellows-type folds, the complete local flattening of the cross section of a thin-wall hollow beam to form a weak "hinge" at a bend, and the collapse of a sheet metal panel into very short waves of accordion-type folds. The theoretically accurate modelling of such components requires a large number of plate elements involving thousands of DOF for each collapse zone. The added details of these local collapse models could increase the analysis costs by orders of magnitude. A practical approach for these components is to model them as simple nonlinear spring elements that require an input curve of force versus displacement or moment versus rotation. Thus, this local hybrid method requires the analyst to specify the expected nonlinear behavior. This method's great advantage is that only one DOF is added for each such nonlinear spring. However, if the conventional hybrid method is used, these nonlinear collapse curves are specified a priori without regard to the interactive effects of other loads acting in combination at the collapse zone. Since these combined loads can greatly reduce the collapse strength, they should somehow be taken into account.

In the case of a collapsing hinge forming in a thin-wall hollow beam, we have used a semiempirical interactive method involving the use of nonlinear rotary springs imbedded between beam elements in a full-vehicle model. The rotary springs are at first rigidized and the analysis using DYCAST is begun. The beam elements indicate the instant when lateral collapse begins as a plastic hinge forms. The analysis is then restarted at an earlier time with a revised moment versus rotation curve for the rotary spring element. This revised rotary spring curve rises to the collapse moment, then it decays rapidly with increasing rotation angle. The collapse moment, is determined interactively by the beam elements in the DYCAST analysis, and the shape of the rotary spring curve is taken from test experience. Typical results with this method in auto crashes predict collapse moments of hollow beams in the range of 10 to 50 percent of the theoretical fully plastic limit moment from bending acting alone.

This reduced peak moment is primarily caused by the presence of a large compressive force in the beam, acting together with the hinge moment, although the other moments also have an effect. (Fig. 17.)

- Linear zone
    - Elastic
    - Small deflection
- Modelled with:
    - Rigid bodies with lumped mass
    - Relatively few elastic finite elements
    - Substructure with most DOFs omitted
- Moderately nonlinear
    - Plastic
    - Large deflections on a global scale
- Modelled with:
    - Nonlinear finite elements
    - Allowance for possible collapse modes
- Extremely nonlinear
    - Large deflection on local scale
    - Requires fine model (>1000 DOF)
    - Special energy absorbing devices
    - Crushable nonstructural parts
- Modelled with:
    - Nonlinear spring elements
    - Spring properties from test or other analysis

Figure 17. Behavior zone characteristics.

A representative all-composite fuselage cabin section was designed, built, and crash-tested by Bell Helicopter and analyzed by Grumman using the DYCAST code. Two separate fuselages were built. One fuselage was tested in a flat drop at 30 ft/sec (9.1 m/sec) vertical velocity onto a flat, rigid surface, and the other in a 20-deg rolled altitude under the same conditions. Finite-element models of these two test cases were prepared for analysis by DYCAST, and the results were compared to those of the tests.

The fuselage section (fig. 18) was a structure composed of solid and sandwich panels made of epoxy resin reinforced by continuous fibers of graphite, Kevlar, and glass. The primary energy-absorbing structure was the honeycomb sandwich panels forming the vertical webs of the subfloor beams and bulkheads at the rear third of the fuselage under the fuel, passenger, and transmission masses. Additional amounts of such vertical sandwich material were placed in the forward subfloor forming the transverse bulkhead under the crew masses. The entire test article weighed 3530 lb (1600 kg), of which only 462 lb (210 kg) was for the structure and the remainder was from the added masses (transmission, fuel, crew, passengers, seats, ballast, cameras, and wiring).

The full cabin finite-element model is shown in figure 19. For the flat drop case, only the left half of the fuselage model was used in accordance with the symmetry of the structure and the impact conditions. The full structure model was used for the case of the impact of the 20-deg rolled attitude.

The structure was modelled with a combination of nonlinear springs, orthotropic membrane triangles, stringers, and beam elements. Nonlinear crush springs were vertically oriented within the structure to represent the crush behavior of the subfloor vertical panels of both the energy-absorbing sandwich and the nonabsorbing (breakable) type. Nonlinear gap springs controlled the impact and rebounded at the rigid ground surface.

The flat drop model contained 276 nodes, 716 elements, and 587 DOF and required 50 msec of event time, 241 time steps, and 43 CPU mins on an IBM 370/3033 computer.

Figure 20 shows a comparison of certain vertical accelerations for the DYCAST analysis and for the test. The acceleration predictions were generally in good agreement with the test data. The maximum predicted crush deformation of 4.4 in. (112 mm) in the subfloor structure was approximately 15 percent greater than that measured in the test. In addition, the deformation modes of the analysis agreed very well with those of the test.

The 20-deg roll model contained 504 nodes, 1470 elements, and 1431 DOF. It used 60 msec of event time, 760 time increments, and 450 CPU mins on an IBM 370/3033 computer. The increase by a factor of 10 in the CPU time for the rolled impact compared to the flat impact was caused partly by the doubling of the model and partly by the smaller time step required to follow some highly nonlinear local behavior.

A sampling of the data for the 20-deg roll case is shown in figure 21. The front view of the deforming structure (fig. 22) shows the crush of the lower left subfloor, the rotation of the fuselage about the impact point, and the lack of distortion in the upper bulkheads. This figure does show a distortion of the transmission mounting fixtures, caused by the inertial resistance of the transmission mass to the sideward acceleration of the roof when the vehicle was rotating. The predicted maximum vertical crush of 6.1 in. (155 mm) in the subfloor was approximately 10 percent less than that of the test. The predicted accelerations showed a mixed

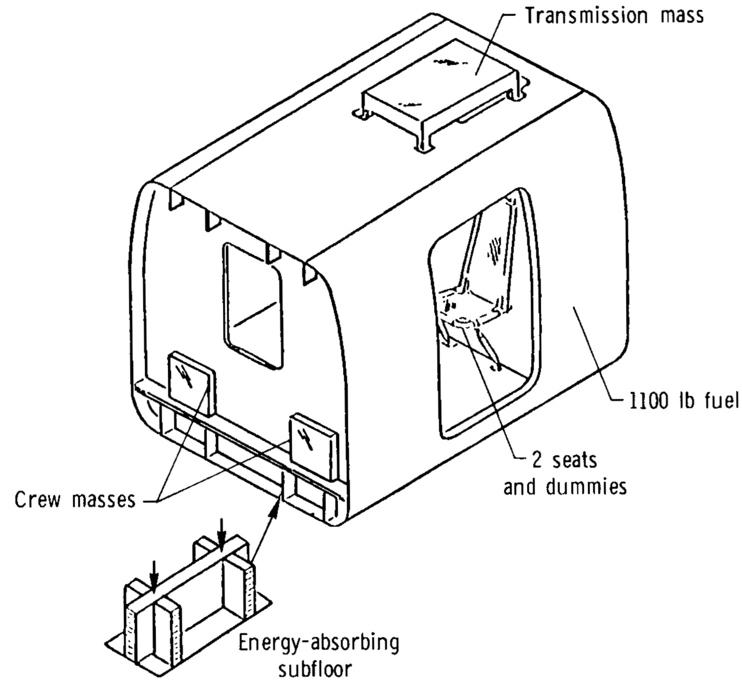

Figure 18. Composite helicopter cabin structure, external view.

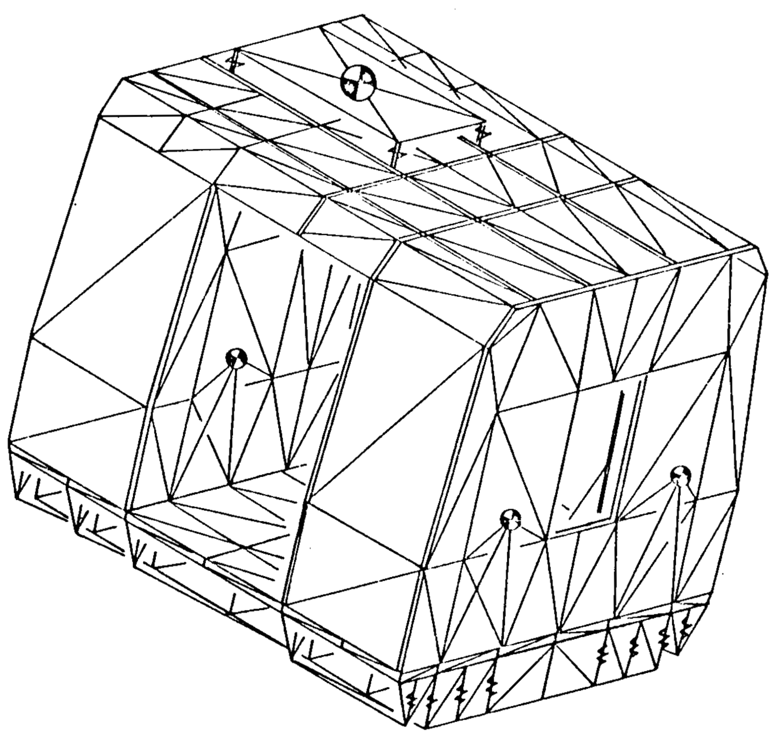

Figure 19. Finite-element model for 20° roll drop.

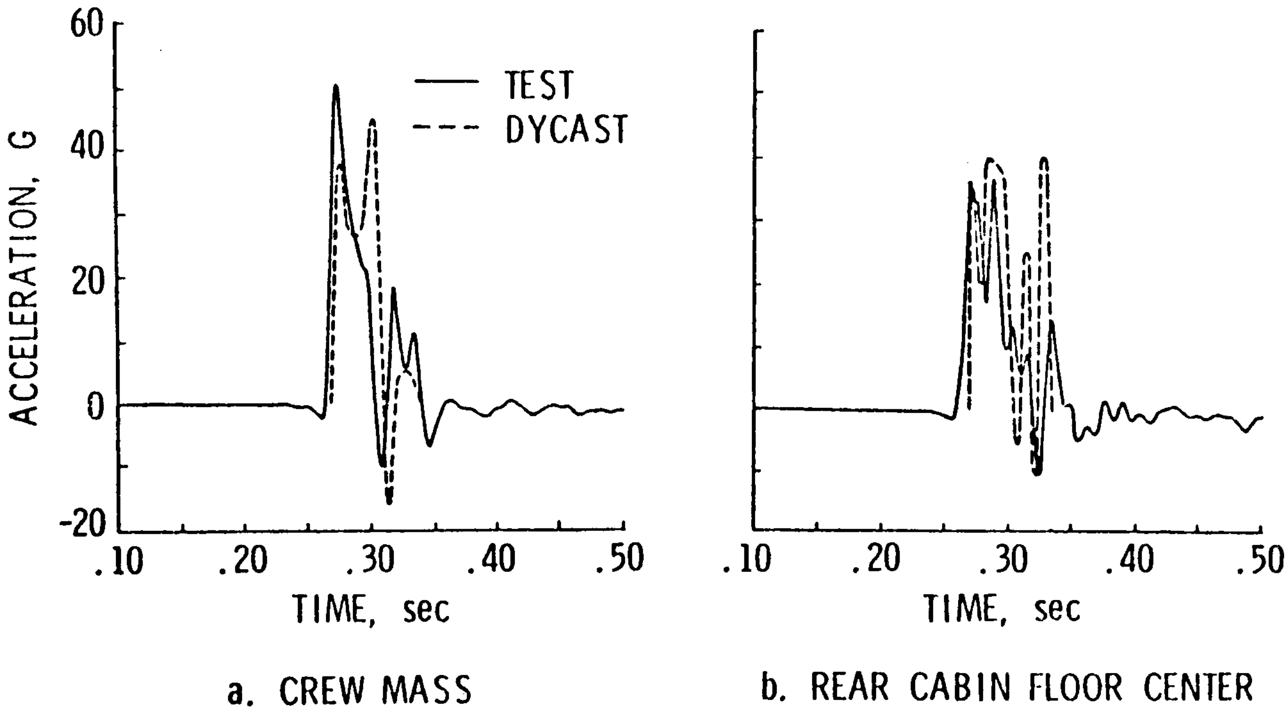

Figure 20. Vertical accelerations for flat drop.

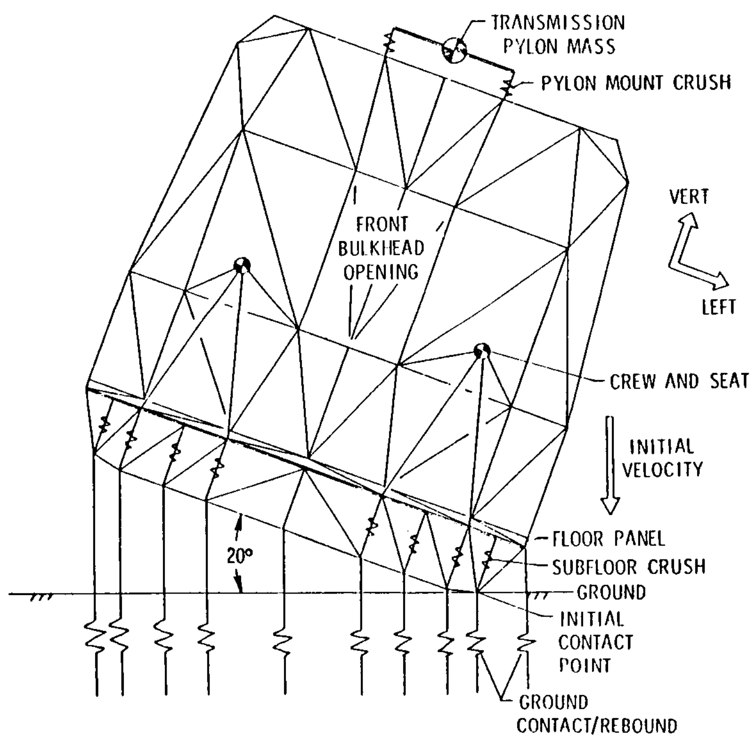

Figure 21. Front view of roll drop model.

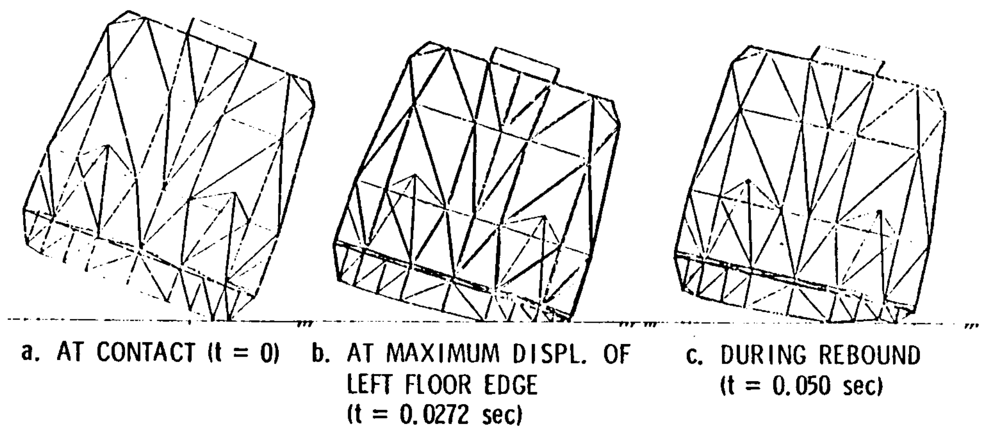

Figure 22. Deformed DYCAST model - roll drop.

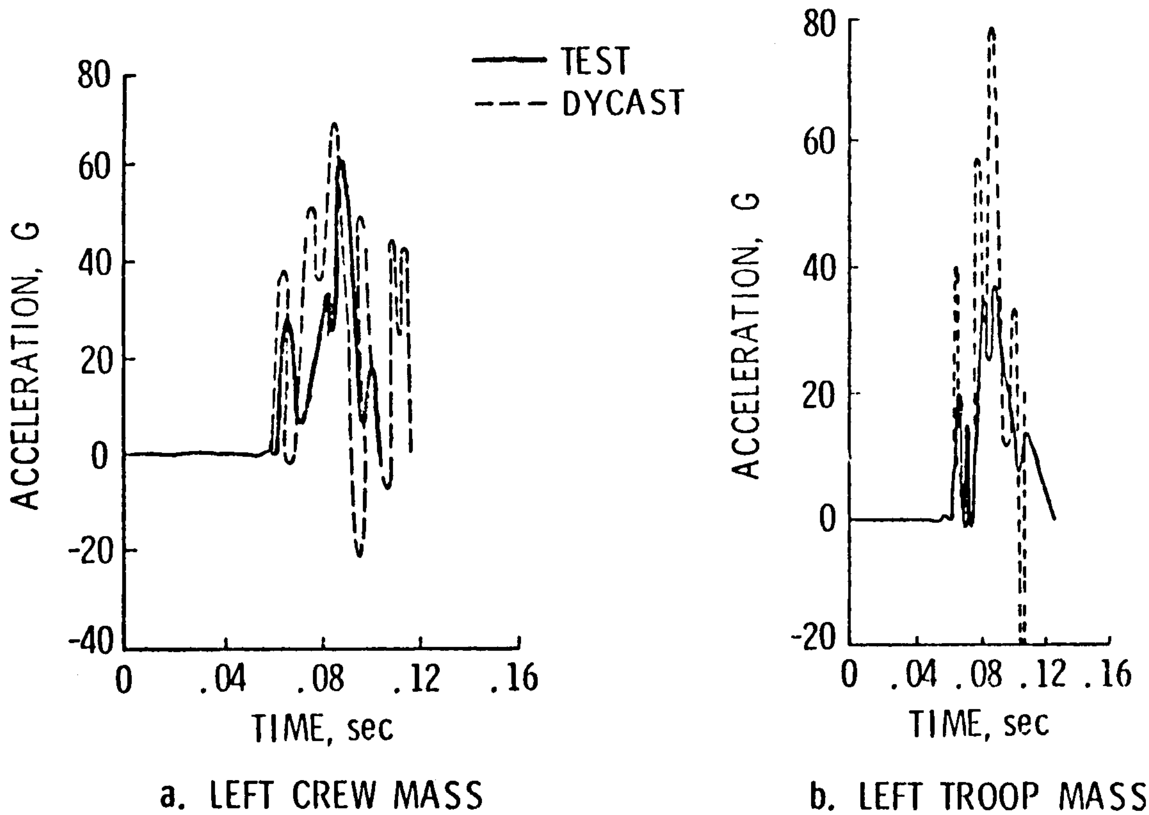

Figure 23. Vertical accelerations for roll drop.

correspondence with the test data. The left crew mass acceleration agrees well with the test data, but the left passenger mass peak acceleration is overpredicted by a factor of 2 (fig. 23).

Figures 24 through 27 outline a front barrier impact of an early prototype version of the 1984 Chevrolet Corvette, a two-seat front engine sports car with a steel frame, and a fiberglass reinforced plastic body shell. Figure 25 shows the steel frame for the analyzed vehicle, and it should be noted that the production vehicle's frame is significantly different, so that the discussion here pertains only to the early prototype and not to the final production vehicle.

The three-dimensional finite-element model involved only the left half of the car to take advantage of the symmetry. The structure was modelled all the way to the rear because it was anticipated that the engine and driveline would become a major load path to the rear of the frame (figs. 26 and 27).

The finite-element model included the frame, plus the other structure (engine bulkhead, front floor, etc.), driveline, and mechanical parts described previously. The fiberglass body was not modelled because, in the previous auto crash analysis, the fiberglass body absorbed a negligible amount (less than 5 percent) of the total kinetic energy.

The model used 157 nodes, 220 elements, and 597 DOF. The elements included 98 beams, 63 membranes, 12 stringers, and 47 nonlinear springs.

One complete simulation of 100 ms consumed 200 min of CPU time on an IBM 370/3033 computer system, required 2000 time steps using the Newmark-Beta implicit method, and was performed in four consecutive overnight segments using restart.

A complete discussion of this analysis is found in reference 1. Conclusions are found in figure 28.

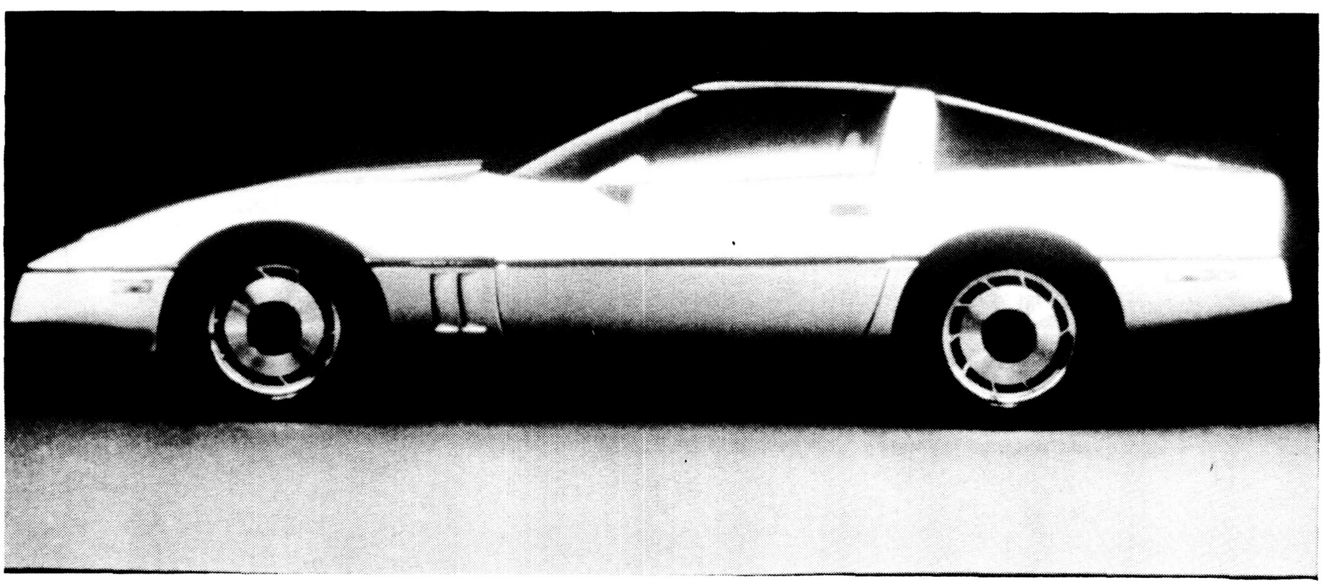

Figure 24. 1984 Chevrolet Corvette.

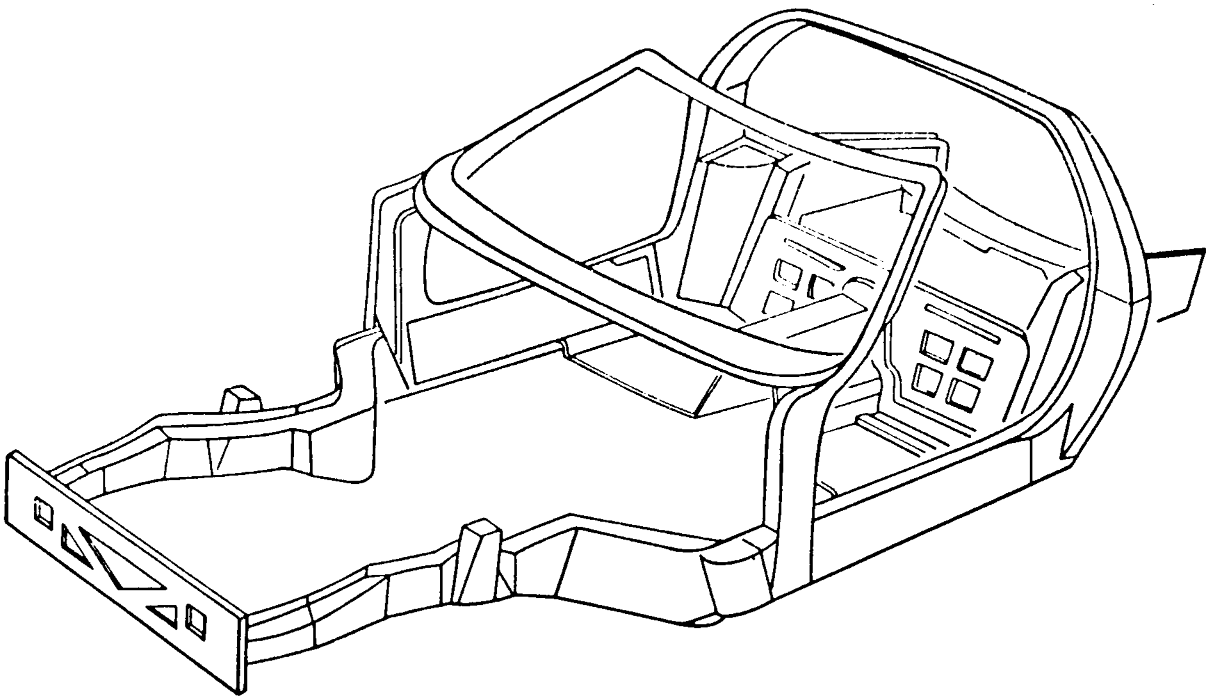

Figure 25. Welded steel frame of prototype.

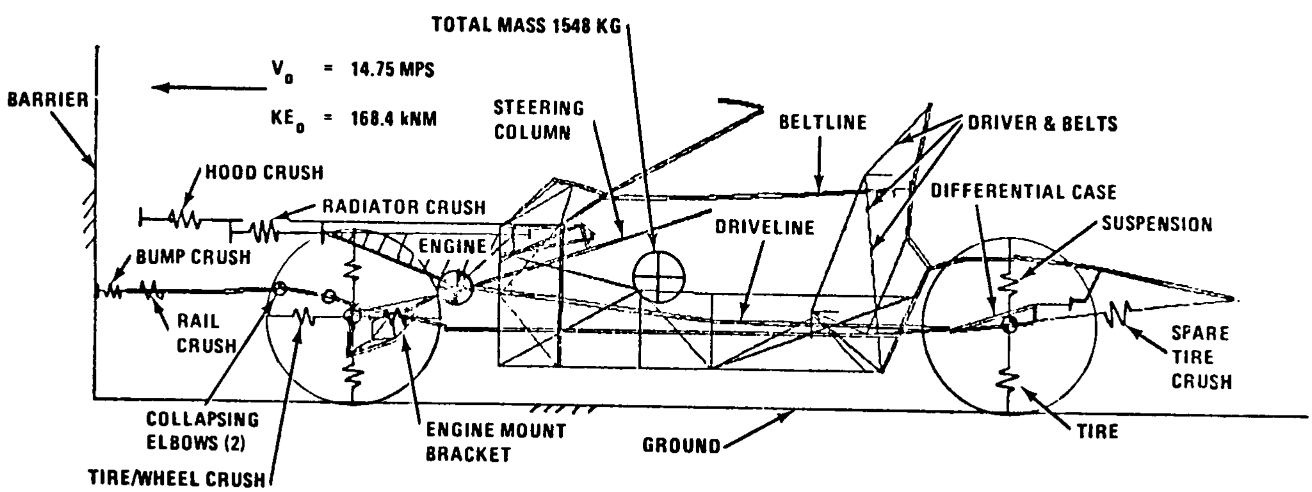

Figure 26. Side view of finite-element model.

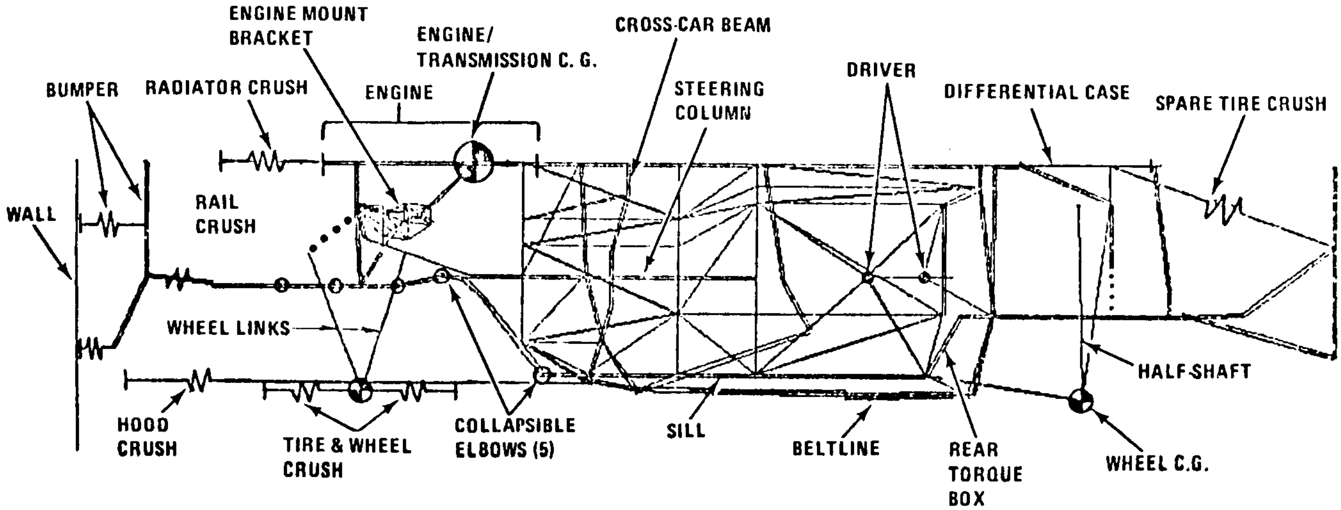

Figure 27. Top view of finite-element model.

- Full vehicle finite element analysis is currently feasible but requires expertise in modelling "art"
- Future goals (or wishful thinking)
    - Include detailed model of extremely nonlinear zones in full vehicle model
    - Same fine model for linear and nonlinear analysis

Figure 28. Conclusions.

REFERENCE

1. Winter, R., Crouzet-Pascal, J., and Pifko, A. B.: Front Crash Analysis of a Steel Frame Auto Using a Finite-Element Computer Code, SAE Fifth International Conference of a Vehicle Structural Mechanics, Detroit, April 1984.