Crashworthiness analysis in the automotive industry

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Abstract: Computer simulation of crash events has become a valuable and necessary tool in the design and improvement of automotive structures for better crashworthiness behaviour. Crashworthiness analysis has evolved through rapid improvements in computer technology, as well as through theoretical improvements in solid mechanics concerning large deformation dynamics and contact. Significant applications involve full car analysis such as frontal, rear, and side impact, as well as the analysis of components such as longitudinal beams, steering wheel systems, knee bolsters, seat belts and airbag–dummy interaction. The theoretical background needed in crashworthiness analysis is outlined. Current capabilities are shown on real applications; limitations and future technologies are discussed.

Key words: car impact, crashworthiness simulation, dynamics, finite-element method, large deformation, occupant safety systems.

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1 INTRODUCTION

For a long time crashworthiness simulation suffered from the complexity of the underlying problems involving an arbitrary form of the structure, large deformations and highly dynamic behaviour. With the rapid development of computer hardware since the mid-seventies, in particular since 1980, and simultaneous developments in software [22, 25, 49, 58, 61] more realistic and much more detailed simulation technologies became available and are now standard in most automotive companies.

The new simulation technology is based on finite-element (FE) methods and replaces the analysis of car structures and components with simple models. The simple models are still useful for preliminary studies and the analysis of car bodies, where the interaction between the parts of the body is not important and where the shape of the components is not too complicated. For an overview of the latter methods, see [39, 46, 71] and for recent applications [64, 66]. However, the restrictions and assumptions to be made are rather wide-ranging and the results are only valid for limited cases. The finite-element method, which is solely based on solid mechanics and structural mechanics, does not demand such problem-oriented assumptions and provides a qualitatively as well as quantitatively reliable answer with a high degree of accuracy. The latter is mainly dependent on the amount of computational effort spent in the analysis.

Depending on the components present in an automotive structure, the following elements are needed in the analysis: 3D solid elements for components such as the engine block and other massive parts, or for very detailed investigations; 3D shell elements for the car body, crash girders, bumpers etc.; beam elements for slender parts such as the steering wheel and steering column; springs and dashpots at various locations; rigid body mechanics components to simplify the analysis of parts with limited flexibility or small deformations. In addition, strategies for the simulation and investigation of contact behaviour are essential. Finally, the dynamics of the structure involving short duration impact have to be handled properly.
To present the background to the methodology, the necessary theoretical foundation concerning structural and temporal discretization is reviewed in Section 2. Real applications in the automotive industry are presented in the subsequent sections. Full car analysis, including frontal central impact, frontal offset impact, side and rear impact, is discussed in Section 3. Certain details of the crashworthiness analysis of selected components are given in Section 4. Section 5 is dedicated to car–occupant interaction simulations, involving steering wheel systems, restraint systems and airbag-passenger studies. Finally, a number of conclusions are drawn and future trends are discussed.

2 FE FOR CRASHWORTHINESS SIMULATIONS

Crashworthiness simulation is to a great extent dominated by the importance of efficient analysis, as the structures are very difficult to simplify and detailed models are preferred. Thus, 20,000 - 100,000 element models are becoming standard and even on supercomputers a compromise between mechanical 'accuracy' and efficiency has to be achieved. This has resulted in the current use of so-called 'explicit finite-element codes' using explicit integration for the time variables, together with spatially under-integrated structural elements requiring 'hourglass' control.

The large deformations observed during the crash event are treated with theories including geometrically and materially nonlinear effects, in particular large strains and materials such as plastifying steel and crushing composites, glass for windows, as well as anisotropic textiles for belts and airbags. It is also apparent that contact plays a dominant role and that the rather unpredictable nature of the crash process requires efficient methods to determine correct contact locations and to accurately enforce the contact conditions. Efficiency in the enforcement of contact conditions and contact searching again has some implications for the type of elements and their surfaces, which should be as simple as possible. For finite-element analysis purposes, the momentum balance for bodies in contact can be written in a weak form

\[ \int_B \dot{S} \delta \epsilon \, dv + \int_B \rho_0 \dot{\delta} \delta u \, dv - \int_B \rho_0 b \delta u \, dv \\
- \int_A F_\delta \delta u \, dA + \text{contact + friction} = 0 \]  

where \( B \) is the volume of the bodies, \( A \) is the loaded surface and \( A' \) is the contact area, \( \int \dot{S} \delta \epsilon \, dv \) represents the internal work, \( \int \rho_0 \dot{\delta} \delta u \, dv \) the apparent work of the inertia forces, \( \int \rho_0 b \delta u \, dv \) the body force term, \( \int F_\delta \delta u \, dA \) surface traction. The last terms of the energy expression refer to normal forces and friction forces arising during contact, respectively. Often, also discrete dampers are added, representing the real dashpots in the automotive structure \( \sum c_j \dot{x}_j \delta u_j \) (\( c_j \) is the viscous damping value, \( \dot{x}_j \) is the relative velocity between damper connections). The various terms of equation (1) will be discussed in detail in the following subsections.

2.1 Spatial discretization

Automotive structures contain many types of elements: solid parts, which can be treated by 3D continuum theory, thin plate and shell type parts, for which shell theory can be used, beam type elements for which beam theory is the adequate tool, and so-called discrete elements—springs, dampers and masses—at various locations. In addition, many parts are known not to deform (or undergo only little deformation) and are therefore best modelled by rigid body dynamics with specific joint definitions.

To accommodate the geometrical effects, an 'updated Lagrangian' description is used, where the reference state is updated every time step [2]. Due to the small time steps in an explicit time integration, and thus small deformation changes within a time step, linear strain displacement relations (Green–St. Venant strains) or linear rate of deformation tensors for finite deformation analysis, respectively, are sufficient.

2.1.1 Solid elements

The discretization of solid parts in the automotive structure, such as engine blocks, is preferably performed by solid elements with linear shape functions. The main reasons are efficiency and simplicity, in particular when considering contact analyses, which would be rather difficult to handle if, for example, curved surfaces based on quadratic shape functions are present. For an extensive discussion, see [19]. Only a short description of the solid elements is given here.

The current geometry of the element is described in a Lagrangian description as:

\[ x(X, t) = x(X(s, n), t) = \sum_i N_i(s) \tilde{x}_i(t) \]  

where \( X \) is the reference frame; \( s = \{ \zeta, \eta, \zeta \} \) local variables; \( n \) is the number of nodes in an element; \( N_i \) is the shape function; and \( \tilde{x}_i(t) \) is the location of node \( i \) at time \( t \).

Displacement field:

\[ u = x - X \]  

Thus, following standard procedures [12, 34] the internal work term leads to the internal forces for an element:

\[ f^e = \int_V B^e \sigma \, dV \]
with $B = V \Phi$ the strain displacement matrix in the current configuration; the symmetric stress tensor $\sigma = \{\sigma_{11}, \sigma_{22}, \sigma_{33}, \sigma_{12}, \sigma_{13}, \sigma_{23}\}^T$. And:

$$\nabla \Phi = \frac{\partial \Phi}{\partial x} = \frac{\partial \Phi}{\partial x} \frac{\partial x}{\partial s} = \frac{1}{|J|} J^{adj} \frac{\partial \Phi}{\partial s}$$

(5)

where $J = \partial x/\partial s$ and $J^{adj}$ are the Jacobian and the adjoint Jacobian matrix, respectively; $\Phi = \{N_1, N_2, \ldots\}$ is the matrix of the shape functions.

With the volume $v = \int_0^1 |J| \, ds$, the evaluation of the product:

$$\bar{B}_e = (V N_e \cdot v)_{s=0} \quad (6)$$

at the centre of the element leads to an antisymmetric property for the 8-node elements:

$$\bar{B}^1 = -\bar{B}^7; \quad \bar{B}^2 = -\bar{B}^8; \quad \bar{B}^3 = -\bar{B}^5; \quad \bar{B}^4 = -\bar{B}^6 \quad (7)$$

Thus, only half of the $B$-matrix has to be determined if one-point integration is utilized. However, this advantage is accompanied by the appearance of kinematic modes, so-called zero energy modes. An efficient remedy of these modes, named hourglass control, was developed by [17] and [19], and was further improved by [6] avoiding erroneous stiffening for distorted meshes. Hourglass control is achieved by counteracting the hourglass modes developing by so-called hourglass forces. These modes are known in advance and can be almost fully removed without interacting with the correct strains of the solid. If the element volumes become heavily distorted during the deformation process, or the element shape is far from being a rectangular brick, an exact volume integration [17] leads to more accurate results. The under-integration (being cheaper by a factor of about 16 compared to the fully integrated element) also helps to avoid rather stiff solutions, which are known from the multipoint integration ($2 \times 2 \times 2$) of the standard 8-node element. However, it must be noted that hourglass control does not fully remove the kinematic modes. In particular, well-structured, rather coarse meshes and meshes loaded with large nodal forces, resulting either from boundary conditions or from contact, are susceptible to hourglassing despite hourglass control. Fine meshes and unstructured meshes are less affected, as the higher mode excitation in these cases requires more energy. Tetrahedral solids, which are not prone to hourglassing, are less attractive as they tend to lock in many deformation modes, in particular when constant volume bending has to be represented.

Thus, the efficiency criteria dominates the selection of the element integration and the compromise ‘one-point integration with hourglass control’ is considered to be the optimal choice.

2.1.2 Shell elements

Shell elements are used to model the major parts of the car body, as these parts are made out of rather thin sheets. Modelling the thin sheets with solid elements is impossible due to the effect that several solid elements of the simple form, as shown in Section 2.1.1, would be required over the thickness, and the discrepancy between area and through thickness extension would lead to numerical ill-conditioning. In addition the time steps would become too small (see Section 2.2). Thus, the development of simple and effective shell elements was the major step towards real full-car simulations. The first breakthrough was achieved with the implementation of the Hughes-Liu element [36] into the explicit code DYNA3D [27]. This element formulation follows the so-called ‘degenerated’ concept, which is closely related to Reissner-Mindlin theory for plates and shells. The preference of Reissner-Mindlin theory to Kirchhoff theory can be attributed to the fact that the addition of the shear term, and thus the rotations as independent variables, allows the development of $C^0$-elements, for which only continuity in the displacements (translations and rotations) is required. These elements are based on the standard kinematical assumptions of shell theory:

- Displacements in the shell are described by translations and rotations of mid-surface geometries.
- Stresses perpendicular to the shell surface are neglected.

Figure 1 Solid elements, possible hourglass modes. After [17].
The internal force term in the weak form of the momentum balance equation is:

\[
\begin{align*}
\int_\Omega \sigma \varepsilon \, d\Omega + \int_\Gamma t \gamma \, d\Gamma &= 0 \\
\int_\Omega \sigma \varepsilon \, d\Omega &= 0 \\
\text{Kirschhoff additional for Reissner-Mindlin} \\
\end{align*}
\]

(10)

Within the strain displacement equation written in local form

\[
\varepsilon = \begin{bmatrix}
\varepsilon_{11}^l \\
\varepsilon_{22}^l \\
\varepsilon_{12}^l
\end{bmatrix} = \begin{bmatrix}
\frac{\partial u_1^l}{\partial x_1} + \frac{\partial u_2^l}{\partial x_2} \\
\frac{\partial u_2^l}{\partial x_1} + \frac{\partial u_3^l}{\partial x_2} \\
\frac{\partial u_3^l}{\partial x_2} + \frac{\partial u_1^l}{\partial x_2}
\end{bmatrix} \\
\gamma = \begin{bmatrix}
\gamma_{13}^l \\
\gamma_{23}^l
\end{bmatrix} = \begin{bmatrix}
\Theta_1 + \frac{\partial u_3^l}{\partial x_1} \\
\Theta_2 + \frac{\partial u_3^l}{\partial x_2}
\end{bmatrix}
\]

(11)

To satisfy the plane stress assumption, \( \sigma_{33} = 0 \) is assumed, while for simplicity thickness changes are ignored, i.e. \( \varepsilon_{33} = 0 \). However, the thinning effect should be considered e.g. in metalforming problems, c.f. [35].

Updating of the rotational quantities poses a major difficulty in standard shell and beam analysis if relatively large steps are used. In explicit integration, however, steps are very small and simple updates are sufficient.

For the shell finite elements used in explicit time integration, low order shape functions are preferred due to the time step limitations, see Section 2.2. Bilinear shape functions are chosen. For triangular elements, well-behaving elements in bending can be found [3]. However, the membrane action for the latter elements is rather too stiff, as the constant strain assumption is incorporated. The linear shape functions constitute a major difficulty for 4-node elements, which show a severely overstiff behaviour in bending, if all energy terms are fully (2*2) integrated. Several remedies of this problem, which is known as shear locking in low-order Reissner-Mindlin elements, can be found in the literature. Either uniform underintegration (1*1), selective integration (2*2 membrane and bending, 1*1 shear) or transverse shear strain interpolation (2*2 integration) is possible. The last suggestion, which is a development based on the ideas of [15] and [37], is currently the most popular element in nonlinear shell analysis and could be used for the solution of most problems. However, due to the 2*2 integration its use is very costly, which makes an efficient analysis of most time-dependent problems in industrial practice with these elements almost impossible. The same is true for the selective reduced integrated (SRI) element. The only alternative is the uniformly reduced integrated element, which is very cost-effective, but possesses kinematic modes (hourglass modes, Figure 4), which have to be removed in order to obtain reliable results. A reasonable and very effective hourglass control scheme was suggested by [4, 9]. Hourglass control introduces artificial stiffness and damping terms into the system.
depending on whether stiffness control or viscous control is chosen [25]. These artificial terms are to some extent chosen to match the physical quantities of the stiffness expected in the real structure.

The limitations of hourglass control are described in the previous section. The computational advantages of uniformly underintegrated shell elements over other bilinear elements are considerable. Based on operation counts, a factor of 9 (Hughes-Liu vs. SRI elements) and a factor of 20 + 30 (Belytschko-Tsay vs. SRI elements) in efficiency difference are found, if 5 integration points through the thickness are assumed. Thus, economic considerations dictate the compromise between reliability and efficiency. However, hourglass energies can be monitored [25] and in the event of difficulties appropriate action for prevention can be started.

**Remark:** Triangular elements are only useful as an hourglass remedy at places where membrane action is very small, as these elements are too stiff in the membrane direction. Otherwise, SRI elements or mesh refinement is suggested.

**Difference in shell formulations**

The shell formulations mentioned above differ not only in the type of integration, but also in their basic development. The Hughes-Liu shell elements are normally developed from 3D-continuum mechanics using a Lagrangian reference frame, allowing arbitrarily large deformation in the element and arbitrary initial shapes of the elements. The Belytschko-Tsay shell elements are developed for initially flat (or only slightly warped) elements following a corotational frame. The latter approach results in computationally very effective elements, which are more efficient by a factor of three than a corresponding Hughes-Liu element. However, for warped or severe out-of-plane distortion of an element, the Belytschko-Tsay element is less qualified [11]. Nevertheless, in real practical solutions the disadvantages of the Belytschko-Tsay element are surprisingly less pronounced than expected, and it has proved to be a very reliable workhorse for standard crashworthiness analysis.

**2.1.3 Beam, truss and discrete elements**

Beam, truss and discrete elements are used only rarely in crashworthiness analysis and then only when a comprehensive global response is expected. Two simple approaches for beams are usually provided; a degenerated beam following the concept of the Hughes-Liu shell with further degeneration into a second direction, and a stress-resultant form [7] based on a corotational formulation. As the first of these is developed by degeneration, the incorporation of nonlinear materials into beam analysis follows standard continuum mechanics with regard to the constraints \( \sigma_{33} = \sigma_{22} = 0 \). The integration can then be performed by choosing the appropriate number of integration points in the cross-section directions (Figure 5).

It is fairly obvious that the Hughes-Liu beam requires a substantial numerical effort in an analysis, and it is therefore only suggested for situations where accurate information during plasticization of arbitrary cross-sections is desired. Otherwise, stress-resultant formulations are preferred due to their computational advantage, as no through-the-cross-section integration is needed. However, the cross-section information has to be given in advance and plasticity has to be treated in a global fashion via the Ilyushin yield law, which is less accurate than the 3D-formulation used in the Hughes-Liu beams.

Truss elements are members carrying loading only in the axial direction and are generally treated in a corotational frame. To account for nonlinear material the use of the Hughes-Liu element with one-point integration is recommended.

Only two-node elements are provided for beams and trusses because of computational simplicity and time step-size considerations (see Section 2.2).
Springs and dampers
Springs and dampers are discrete structural members, for which the stiffness (force/displacement) or the damping constant (force/displacement rate) are given either by constant values or by curves describing the corresponding relationship (Figure 6). These relations are defined only for specified directions or rotations. In a very simple fashion, they allow arbitrary relationships involving plasticity, energy absorption or failure [25].
For arbitrarily large deformations of the structure, the orientation of the translational and rotational springs/dampers has to follow the deformation of the connection points. This is rather important, as very often springs and dampers are defined between nodes at the same geometrical location. Their direction of action then has to be explicitly defined and rotated accordingly.

2.2 Time Integration

2.2.1 Assembly of the equations
The momentum balance equations for single elements in weak form have to be combined to the equilibrium equations of the complete system considering the connectivities, constraints and contact conditions.

With \( u = \Phi v; \ \delta u = \Phi \delta v \) being the actual and virtual displacements, respectively, and \( \delta \epsilon = B \delta v \) being the strain vector, the spatially discretized system of equations is obtained as

\[
\sum_{e=1}^{ne} \int_{v} (m^e \ddot{v}^e + c^e m^e \dot{v}^e + p^e = f^e + f^e_{c}) = 0
\]

where \( \sum \) is the connectivity operator, including rigid body constraints; \( ne \) is the number of elements; \( m^e = \int_{v} \rho_0 \Phi \Phi^{T} \Phi \ dv \) is the consistent mass matrix; \( c^e m^e = \int_{v} \rho_0 \Phi \dot{\epsilon}^{T} \Phi \ dv \) is the mass proportional damping matrix; \( p^e = \int_{v} B^{T} \sigma \ dv \) is the internal force vector; \( f^e = \int_{v} \rho_0 \Phi^{T} b \ dv + \int_{A} \Phi^{T} f_0 \ dA \) is the body forces + surface loading; \( f^e_{c} \) is the contribution of contact forces (see Section 2.5).

The nodal displacements \( v \), velocities \( \dot{v} \) and accelerations \( \ddot{v} \), and thus the internal force vector and the contact forces, are generally time-dependent. This may also be the case for the surface traction and body forces usually given.

After assembly, the equations can be written at time \( t_n \)

\[
M_n \ddot{u}_n + C_n \dot{u}_n + P_n - F_n + R_n = 0.
\]

Due to the high resolution required in the whole structure, the evolving matrices may be very large. More than 200,000 unknowns are very common and a very efficient time integration scheme is necessary.

2.2.2 Central difference scheme
According to the general literature, e.g. [34], multistep methods set excessive demands on core memory. Thus, the simple one-step central difference (CD) method is chosen. The standard CD scheme is given in the following equations:

\[
\ddot{v}_{n+1/2} = \dot{v}_{n-1/2} + \Delta t_n \ddot{v}_n
\]

\[
v_{n+1} = v_n + \Delta t_n \dot{v}_{n+1/2}
\]

where \( \Delta t_n \) is the time step size at time \( t_n \).

Using the last equation, the geometry can be updated:

\[
x_{n+1} = x_n + \Delta t_n \dot{x}_n_{n+1/2}.
\]

In the case of viscous damping, the velocity term \( \dot{v}_n \) in equation (14) is replaced by the velocity \( \dot{v}_{n-1/2} \) in order to maintain the simplicity of the above cited scheme and to keep the number of vectors needed for one state (3 for CD) as low as possible. The viscous damping force is then slightly modified, which can be justified by the crude assumptions made usually for the damping matrix and by the small change in velocity due to the very small time step in explicit time integration. For problems governed by rate constitutive relations, the midpoint velocity gradient is required, thus making the midpoint formula preferred.

The algorithm is called explicit, as the equilibrium equations at time \( t_n \) are used to determine the information at time \( t_{n+1} \) and \( t_{n+1/2} \). A major attribute of explicit methods is that only the mass matrix has to be inverted for the solution. If the mass matrix assumes a diagonal form (mass lumping), the solution of equation (14) may be advanced without matrix inversion. Due to this fact and the observation that neither the stiffness matrix nor linearization of the nonlinear equation is
needed, the explicit integration is very attractive for crashworthiness simulations. These solutions are currently not possible with implicit schemes from the Newmark family [34], even on the largest supercomputers currently available.

**Stability**

Suitable time stepping schemes have to be accurate and convergent; thus they have to be stable and consistent. Though computationally very efficient, explicit time integration of the hyperbolic equations has to be executed under severe time step size restrictions in order to achieve reliable solutions. For linear problems, the following results can be stated by applying simple Fourier methods, see [34] and [5]:

- The CD algorithm is conditionally stable; thus the time step has to meet

\[ \Delta t \leq \frac{2}{\omega} \left( \frac{1}{\xi^2 + 1} \right) \]

where \( \omega \) is the highest frequency in the system; and \( \xi \) is the fraction of critical damping of the corresponding eigenmode.

- The CD algorithm is second order accurate and convergent. Thus the error decreases quadratically with \( \Delta t \to 0 \).

Although the above cited stability criterion applies strictly only to linear systems, it also provides a useful stability estimate for nonlinear problems. [5] proposes an 80% to 90% reduction of the linear time step to maintain stability in most nonlinear problems. However, it is important to check the energy balance throughout the complete solution sequence: any significant increase or loss in the total energy (5% or more) indicates an instability, unless it can be traced to other causes. In [5] it is also shown that stability cannot be maintained with a constantly increasing time step, even if the highest frequency is decreasing, as might be the case due to material nonlinearities or large strains.

### 2.2.3 Mass approximation—critical time step estimates

Since the time step size is crucial for the efficiency of the central difference scheme, a short discussion of structural elements and their mass and stiffness representation is appended. It can easily be shown that the stability criterion for a simple linear truss element with a lumped mass matrix gives a critical time step of

\[ \Delta t \leq \frac{l/c}{A} \]

This time corresponds to the traversal time for a wave across an element of length \( l \) (\( c = \) speed of sound), which is the well-known Courant-Friedrichs-Lewy (CFL) condition for stability (Figure 7). If consistent mass matrices are used, the critical time step is reduced below the CFL condition, which along with the necessity to invert then \( M \) in equation (14) constitutes a serious drawback in its use in explicit integration. In addition, numerical experience indicates that lumped matrices tend to increase the fundamental periods compared to the analytical model, whereas the central difference integrator shortens these periods. These observations suggest combining the central difference method with the lumped mass approximation (diagonal mass matrices) in order to obtain an optimally 'matched method' [34].

The examination of higher-order elements reveals that the critical time step is always below the value for elements with linear shape functions. This makes higher order elements rather unattractive for explicit integration. For a further detailed discussion on discretization effects, including the mentioned effects, see [50].

While the above considerations were concerned with membrane action only, the situation is far more complicated for structural elements, such as shell elements. In this case, the highest frequencies belong to rotational modes, resulting in time steps considerably below the Courant criterion, in particular for small elements. In order to make shell elements accessible for explicit time integration with reasonable time step size, the rotational lumped-mass coefficients are modified according to a proposal in [34] based on a suggestion in [43]. With this modification, the critical time step is determined by the transit time corresponding to the bar-wave velocity and the plane-stress-wave velocity, respectively.

The above considerations should explain why continuum elements are inefficient in thin shell analysis with explicit integration. The transit time through the shell thickness, which is very small compared to the extension in the other directions, would lead to a time step size rendering explicit methods very inefficient.

### 2.3 Nonlinear material formulation

Explicit codes are best suited for the incorporation of arbitrary material laws into an analysis, even if sharp
gradients and changes in material properties are present. This is due to the fact that the time steps, and thus the displacements and strain increments in one time step, are usually very small. The preferred approach when treating inelastic constitutive equations in the case of large deformations is based on constitutive rate equations. The large strains, e.g. those occurring in folding parts, can then be handled correctly, c.f. [26, 38, 40] for further details.

The major topics in large strain analysis are related to the integration of the constitutive equations and in rate formulations to the type of stress rate used. Usually, the constitutive equations are integrated at the midpoint geometry with centred stresses and strains, and are then further rotated to the final configuration using the appropriate transformation matrix [26, 38]. However, in explicit time integration the displacement and rotation increments are very small, typically two orders of magnitude less than in an equivalent implicit integration, and centring of the geometry is not needed. Therefore, the full rotational update of the Jaumann rate is added before evaluating the constitutive relations. Furthermore, the rotational contribution is approximated by the spin tensor at \( t_{n+1/2} \) instead of the rotation matrix. In shell analysis, the stresses and strains must be rotated into the shell local system in order to apply the stress constitutive law. For an efficient integration of the constitutive equations, implicit integration (Euler backward scheme) is used, which leads to the radial return algorithm proposed by [45].

In plane-stress situations, however, simple radial return produces stresses normal to the shell surface, which violates the shell theory assumption. To avoid major errors, the plane stress condition is enforced in the radial return procedure either iteratively [27] or directly [67]. Both approaches are equivalent, but the iterative procedure is computationally more efficient, c.f. [27].

Remark: It is well known that the Jaumann rate form of constitutive equations does not perform so well in cases of kinematic hardening and pure shear, which is mainly the case for 3D-continuum problems. In these cases, the Green-Naghdi rate, for example, should be chosen.

\[ l_{ned} = V / \text{max} A \]
\[ l_{ned} = \min a_i \]

Figure 8 Critical time step for continuum and beam/truss elements \( \Delta t = \frac{c}{l_{ned}} \); \( c \) is the sound wave speed in continua, and bar wave velocity, respectively. \( V \) is the volume of solid, \( A \) is the maximum diagonal area.

Materials

Standard materials in automobiles, such as steel and aluminium, and their nonlinear material laws, including strain rate dependency, are available in most known computer codes for crashworthiness simulations. Other materials, such as fabrics for airbag simulation, which cannot carry any load in compression, or rubber and composites, which are designed especially for energy absorption [65], laminated glass for window modelling and metallic honeycomb, are also often available. In addition, temperature effects can be considered.

The incorporation of failure and damage into materials such as composites or foam, however, poses a difficulty, which has not yet been solved in research and practice as the results are mesh-dependent to a certain extent. Honeycomb and foam materials, mainly used in bumper design, also still lack a sound foundation in material science, and should therefore be applied with great care.

2.4 Rigid bodies and rigid body dynamics

In many crashworthiness simulations, only certain parts of the automotive structures deform, whereas others remain almost unaffected. For efficiency reasons, it is desirable to model those parts as rigid; thus only the rigid body displacements, including translations and rotations, have to be calculated. It must be noted, however, that such an approach is only valid if wave propagation is negligible, as the rigid bodies reflect the waves at their connections or contact to flexible parts.

The standard approach in incorporating rigid body dynamics into an explicit finite element program follows the proposal of [12]. The rigid bodies are defined in a finite element type fashion and are connected to structural finite elements (Figure 9).

The following equations for the rigid body are then valid

\[ M_R \ddot X_{R_i} = F_i^X \quad (19) \]
\[ J_{ij} \dot \omega_j + \varepsilon_{ijk} \omega_i J_{kl} \omega_l = F_i^U \quad (20) \]

where \( M_R \) is the mass of the body, \( J \) the inertia tensor, \( X \) the location of the center of mass, \( \omega \) the angular velocity of the body, and \( F^X \) and \( F^U \) the generalized forces and torques acting on the rigid body. Three

Figure 9 Combining rigid bodies and structural finite elements.
central issues are associated with the implementation of the above equations in a structural dynamics finite element program:

- The calculation of $M_R$ and $J$ from the mass defining the body.
- The calculation of $F^s$ and $F^e$ from the forces acting on the rigid body part, from contact and constraints and from the finite elements being contiguous with the rigid body.
- Updating the displacements, velocities and the inertia tensor in a manner that does not deform the rigid body.

For details, see [12]. It must be noted that the inertia tensor is calculated from the mass matrix of the rigid body, which is assembled from single nodal masses. As a diagonal mass matrix is used, the determination is fairly straightforward. Scaling of the rotational masses, which is the usual procedure for increasing the allowable integration step size for flexible elements, should not be performed for accuracy reasons and because rigid body elements do not affect the size of the time step.

The rigid body velocities are determined by the standard difference approach, after the accelerations have been calculated. The rigid body translations are updated accordingly, where the incremental rotation matrix has to be calculated using the correct algorithm, e.g. as proposed by [26, 38]. Finally, the displacements of the nodes of the rigid bodies can be updated using the rigid body translation and rotation. The velocity of the nodes is calculated by differentiating the displacements. Both velocities and displacements are not determined directly from the rigid body accelerations, as this is less efficient and less accurate.

The generality of the procedure outlined above allows definitions of rigid bodies interfaced with flexible elements in an arbitrary fashion. However, the connections of rigid bodies with other parts have to be defined correctly. If rigid bodies are connected via joints, the appropriate definitions of the type of the joints have to be given in order to avoid contradictions. A selection of standard joints is depicted in Figure 10.

2.5 Contact

Contact simulation is a most essential part of crashworthiness analysis, where all types of contact occur, such as contact with rigid walls, contact with bodies, two-sided contact with shells and other multiple contacts, as well as folding of some shell parts of the car body on itself. All these possibilities of contact are generally defined as boundary-value problems, in which two bodies $B^1$ and $B^2$ interact according to the principles of the mechanics of continuous media. Contact problems are defined with two conditions: the first is the mechanical contact condition, where material points on the common boundaries of $B^1$ and $B^2$, the contact surfaces, may coalesce and separate during the motion of the bodies. The second is the impenetrability condition, where the two bodies $B^1$ and $B^2$ are constrained not to penetrate each other in any configuration. Both conditions and the way they are determined and enforced are decisive for an accurate and efficient analysis of contact problems. As in all contact strategies, first the contact surfaces are roughly determined (= contact search), after which the impenetrability condition is checked and the mechanical contact conditions are satisfied. The latter can be described on a surface as:

(a) The contacting points move with the same displacement and velocity in the direction normal to the contact surface during the contact.

\[ v_n = v^1_n = v^2_n \quad \text{displacement} \]

\[ \dot{v}_n = \dot{v}^1_n = \dot{v}^2_n \quad \text{velocity} \]

$n^*$ is the normal vector on contact surface of body $B^*$. (b) The momentum is balanced on the contact surfaces

\[ t^1 + t^2 = 0 \]

$t$ is the resultant stress in the contact surface. Thus, normal and frictional forces $t$ have to be equal for both bodies.

Figure 10 Joints connecting rigid bodies, from [25].

Figure 11 Bodies in contact.
No tensile traction can occur on the contact surface.

\[ r^x n^x \leq 0 \quad x = 1, 2 \]  \hspace{1cm} (24)

### 2.5.1 Contact search

In a spatially discretized problem, the bodies and their surfaces are described by elements connected at the nodes. As long as the impenetrability condition is not violated by the location of the nodes of either body, the contact conditions are satisfied. Thus, the determination of contact is restricted to checking whether nodes penetrate or contact the surface of the other bodies. The candidate contact surface, which is tested against penetration of a node, is named the 'master' surface and the penetrating node is called the 'slave' node, see [28].

The searching procedure has to be accurate enough to locate all possible contacts, and it has to be efficient by avoiding unnecessary searching. To achieve a good balance between efficiency and accuracy, a wide variation of search strategies have been developed. Their basic steps involve a crude global search, before an accurate local search is performed. The two most successful search strategies in crashworthiness analysis differ in the way searching is conducted:

Hallquist [22, 23, 24, 26] bases his global search on simply determining distances between the slave node and the closest master nodes to this slave node, see also [28]. Standard one-dimensional bucket sorts are performed, which are very efficient and which provide initial information about the contacting partners. The surfaces, which consist of 3-node triangular or 4-node quadrilateral contact segments, are then searched for the master segment to touch the slave node and a simple penetration check is performed by approximating the segment surface with a flat surface. If penetration is found, or is the distance between the node and the segment is within a certain tolerance, a local search is performed. In order to safeguard the crude algorithm, several segments around the closest master node are checked (Figure 12a).

The local search of [26] is simply a minimization of the distance \( g_s \) between the slave node and the master segment (Figure 12b).

\[ \min g_s = \| x(\xi, \eta) - x^0 \| \]  \hspace{1cm} (25)

where \( x(\xi, \eta) \) is the location vector of the point on the face; and \( x^0 \) is the location vector of the slave node.

The location \((\xi, \eta)\) of the slave node \( n_s \) on the surface and the penetration or gap \( g_s \) can then be calculated with sufficient accuracy.

Zhong [73] introduces in his HITA algorithm (Hierarchy-Territory algorithm) a system of contact hierarchies with five levels: contact bodies, surfaces, segments, edges and nodes. A territory is defined for each hierarchy to obtain an efficient indication of the position of the corresponding hierarchical object.

Two further categories are introduced according to their nature, i.e. the primitive contact hierarchy containing segments, edges and nodes, and the composite contact hierarchy containing contact bodies and surfaces. The composite hierarchy is only used for the global search, whereas the primitive hierarchy is used for the local search as well as for the final evaluation of the contact condition. In evaluating a contact-impact interface, a contact node and its corresponding target, which belongs to the primitive hierarchy objects (segment, edge or node), form a contact pair.

The efficiency of the HITA algorithm is fairly obvious, as intersections of the higher hierarchies are tested first, followed by a more detailed check on the lower level hierarchies. While connectivities are of no interest on the high hierarchy level, they need to be introduced for correct representation of the structure into the lower levels. Thus surfaces are resolved into connected segments, for which a so-called target neighbour concept is developed in order to provide a search, which facilitates finding the correct contact in all situations. Three major target neighbourhood categories are depicted in Figure 13; each contains one of the low level contact hierarchies and all its neighbouring segments.

For the treatment of more general situations involving branching parts, see [53].

Within the local search the differences between the strategies are minor. For example in the HITA algorithm, the territory of the low level hierarchies is
expanded to some extent in order to capture all possible contact and penetration situations, which is important for edge situations.

2.5.2 Contact conditions
In general, two different approaches are possible, i.e. the Lagrange multiplier method and the Penalty method. The so-called Perturbed Lagrangian method [72], is a combination of the first two methods. An algorithmically slightly modified version of the perturbed Lagrangian method is known as the Augmented Lagrangian method.

The formulation of the discrete problem is derived via the perturbed Lagrangian functional, from which the corresponding variational equations are developed.

\[
\delta \mathbf{\pi}(\mathbf{u}) + \lambda^T \delta \mathbf{g} = 0 \tag{26}
\]

\[
\delta \lambda^T \left( -\frac{1}{\varepsilon} \lambda + \mathbf{g} \right) = 0 \tag{27}
\]

\(\delta \mathbf{\pi}(\mathbf{u})\) corresponds to the variational terms of equation (1) derived through the principle of virtual work. \(\lambda^T\) is the vector of nodal contact forces and \(\mathbf{g}\) is the vector of nodal gaps (including penetration). For a simple illustration, see Figure 14. All developments are restricted to the case of bilinear elements.

The contact forces in the virtual work equation can be further identified for the slave and master nodes:

- slave node contact force \(R_n = \lambda\)
- master node contact forces \(R_{cm,i} = -h_i \lambda\)

where \(i\) is the node number; and \(h_i = h_f(\xi_c, \eta_c)\) is the corresponding shape function of the contact segment.

The classical Lagrange multiplier method is achieved from equation (27) if \(\varepsilon \rightarrow \infty\), which yields \(\mathbf{g} = 0\) is satisfied in every step. To satisfy equations (26) and (27) in the case of a perturbed Lagrangian method, a coupled set of equations involving all nodes in contact and all nodes connected to elements in contact has to be solved. This is the case when explicit methods are used. Thus, the great advantage of the explicit methods is lost in this case. In addition, the size and profile of the system of equations vary between time steps.

If equation (27) is solved for \(\lambda\) and substituted into equation (26), the so-called penalty method is obtained. The resulting set of equations can then again be solved by explicit methods directly, thus retaining the major advantage of these methods.

The open choice in the penalty formulation is the penalty parameter \(\varepsilon\), which is equivalent to a spring between the contact node and the contact surface. This factor determines the satisfaction of the contact conditions and the size of the force needed to push the nodes back to the surface. The contact force is obtained as:

\[
\lambda = \varepsilon \mathbf{g} \neq 0 \quad \text{only for} \quad \varepsilon < 0 \tag{28}
\]

No tensile forces are allowed!

Large values of \(\varepsilon\) yield to an accurate satisfaction of contact condition.

Small values of \(\varepsilon\) yield to a noticeable penetration.

Neither too small values nor too large values may be chosen for \(\varepsilon\). Too small values often violate the physics of the system, but lead to fairly smooth solutions. Large values of \(\varepsilon\), indicating a stiff element between the contacting bodies, may introduce high frequencies into the system, which in some cases could determine the critical time step size or lead to unnecessary oscillations. To avoid this, the penalty parameter has to be chosen according to the stiffness of the contacting segments. The standard factors used in explicit codes are:

- solid elements \(\varepsilon_{se} = \alpha \frac{A_s K}{h} = \alpha \frac{A_s^2 K}{V_s}\)
- shell elements \(\varepsilon_{sh} = \frac{A_s K}{l}\)

where:

- \(A_s\) is the area of segment;
- \(h\) is the thickness of brick element;
- \(V_s\) is the volume of brick element;
- \(l\) is the (thickness or the longest diagonal of shell element; whichever is longer);
- \(\alpha\) is the scaling factor (\(\approx 0.10\));
- \(K\) is the bulk modulus.

The solution of equation systems is not needed for the Lagrange multiplier technique, if a node-to-node contact is present. Zhong [73] and Oldenburg [56] exploit this fact in their DENA algorithm, which is based on an approximate Lagrange multiplier technique. In the DENA algorithm, each off-diagonal term is approximated by its value of the previous step, which is achieved by the construction of a so-called defence node on the master surface such that the above node-to-node contact results. This algorithm is implemented in LS-DYNA3D [73] and works extremely well, as numerous examples applying the HITA-search and DENA algorithms show.

2.5.3 Friction
Many friction laws have been proposed to describe the evolution of tangential forces in contact due to tangential displacements and velocities. For an extensive discussion, see [54] and [55]. As the surface conditions in real contact situations can only be roughly estimated and vary locally, the classical Coulomb friction model is sufficient for most cases.
In explicit codes, friction is provided for stick-slip situations by an algorithm tailored after the radial return algorithm in plasticity [48] (Figure 15):

Maximum frictional force

\[ F_Y = \mu |f_n| \]  

(29)

Tangential relative displacement

\[ \Delta e = x(\xi^{n+1}, \eta^{n+1}) - x(\xi^n, \eta^n) \]  

(30)

Trial forces

\[ f^* = f^n - k \Delta e \]  

(31)

Friction forces at time \( n + 1 \):

\[ f^{n+1} = f^* \quad \text{if} \quad |f^*| \leq F_y \]  

(32)

\[ f^{n+1} = F_y \frac{f^*}{|f^*|} \quad \text{if} \quad |f^*| > F_y \]  

(33)

where \( f_n \) is the vector of the normal contact force at time \( n \); \( \mu \) is the Coulomb factor; and \( k \) is the interface stiffness.

To account for the velocity dependency of the Coulomb factor [44], a smooth transition between a static coefficient \( \mu_s \) and a dynamic coefficient \( \mu_d \) is often suggested:

\[ \mu = \mu_s + (\mu_d - \mu_s)e^{-v \text{el}} \]  

(34)

where \( v = 1/\Delta t \Delta e \); and \( c \) is the exponential decay coefficient.

### 2.5.4 Contact definitions

Multiple definitions of contact surfaces are possible (Figure 16):

- Contact with rigid walls
- Tied surfaces
- Nodes tied to surfaces
- Shell edge tied to shell surfaces
- Sliding surfaces
- General contact (sliding, impact, separation)
- Folding

Contact definitions, such as tied surfaces or nodes tied to surfaces, are mainly introduced to allow mesh refinements in an 'incompatible' fashion, which offers engineers a great variation for the meshing of difficult situations without the specific introduction of constraints. In addition, the forces in the tied interface can be monitored, which allows the introduction of failure criteria, for example as necessary to model spot welds.

For structural elements with a rotational degree of freedom, a different algorithm is needed for tying in order to achieve continuity in the rotations. Thus, the 'shell edge tied to shell surface' interface is introduced.

Contact surfaces defined as sliding-only are useful for the simulation of experiments carried out on rails and are also of some use for fluid structure interaction.

Contact with rigid walls is devoted to the simulation of crash experiments such as frontal and rear impact. For side and offset impacts, rigid walls can also assume the shape of rectangular prisms, cylinders, spheres and finite flat surfaces. These walls can also be defined as moving with a prescribed motion or with an initial velocity. In the algorithmic treatment of a rigid wall contact, simple fully plastic impact is assumed. The accelerations and velocities of the nodes impacting the walls are set to zero or to the corresponding values in a moving wall, and the contacting nodes are returned to the surface of the wall. It must be noted that in a fully plastic impact the energy of the contacting nodes is lost in the analysis, as it is known from standard impact analysis.

General contact, also known under the name master and slave concept, can be defined in two ways: (a) unsymmetric contact treatment, when one side is
treated as a master and the other as a slave, or (b) symmetric contact treatment, when case (a) is performed first. This is followed by a reversal of the search (= exchange of masters and slaves) in the same cycle. The general contact type includes contact, impact and separation in a fashion as described in the previous sections. Obviously, the numerical effort needed for an unsymmetric treatment is about 50% of the effort for a symmetric treatment. If the accurate satisfaction of the contact conditions is important, symmetric treatment is absolutely necessary, in particular if both contact surfaces are not equally meshed.

Folding requires the contact surface to be checked against penetration into itself. To achieve this, a special contact search known as the 'single-surface contact' has been developed, see [13]. Checking this self-contact is very time-consuming, even on large computers. From the viewpoint of removing workload from the engineers, the single surface capability also provides the simplest way of defining contact surfaces, since all surfaces can be specified as a single contact surface at the expense of computer time.

Better ways for making automatic contact searches are provided through the HITAC algorithm and the box algorithm [25], where contact windows or boxes are specified. Within these windows/boxes, contacting bodies and surfaces are automatically defined, reducing the user effort considerably.

### 3 FULL CAR ANALYSIS

Full car analysis seems to provide the best information about the crashworthiness behaviour of the automotive structure, in particular as most parts are arbitrarily connected in the fully assembled vehicle and as the structural parts interact heavily with the surrounding parts during an impact. However, due to limitations in computer resources and the huge amount of data arising with fine meshes for full car analysis, investigations on components have to be performed in parallel to full car analysis, see Section 4.

With respect to the limitations imposed by the computer resources, different models are used for the various crash events. These can be subdivided into the following categories with their share in total events after [33]:

- frontal impact 61%
- rear impact 9%
- lateral impact 20%
- rollover 10%

Out of the 61% of events in the frontal impact 17% cannot be classified, while the other 44% can be further subdivided into: central impact 11%, right-hand offset impact 11%, and left-hand offset impact 22%.

In principle, very general crash simulations are possible, involving many rigid or flexible walls or barriers, arbitrary impact angles, head-to-head collisions, rollover impact, multiple car collisions and so forth. No limitation from the methods point of view and from the aspect of the finite element programs exists. As was previously pointed out, the limitation of computer resources might allow single studies of fairly general events. However, for design improvements and parametric studies with selected modifications, general events are less useful than clearly defined load cases [33]. For frontal impact, these loads are defined by rigid walls, against which the cars impact with velocities of max. 55 km/h, and for lateral and rear impact by moving barriers or impact cars with different shapes and soft parts at velocities also of max. 55 km/h.

**Frontal impact**

Typical models of frontal impact are depicted in Figure 18 taken from [31] and [51]. The mesh density usually decreases from the front to the rear according to the expected amount of deformation. The longitudinal girders are meshed with a rather large number of elements as they have to absorb the majority of the energy by folding and since they are heavily deformed. The engine blocks could be modelled either by solids or by shells. The latter facilitates simpler modelling and contains fewer variables, but the inertial and mechanical properties have to be adjusted. The meshes typically contain 5000 - 120000 elements. This allows a good representation of overall quantities such as impact forces or passenger accelerations and distribution of energy absorption. The good quality of the simulations is demonstrated by comparison with experimental results. While such quantities are very well suited for general construction evaluations, much finer meshes are required to capture the detailed response. To achieve results, comparable to static models with similar large deformations, meshes with 100000 - 500000 elements would be needed.

Many software and hardware improvements are visible in comparing the CPU time needed for the analysis models in 1988/1989 with the times used in 1990. For both analyses (up to 90 ms) around 22 ± 35
CPU hours were needed on the 1988 CRAY X-MPs compared to around 3 + 4 CPU hours for similar models on a CRAY Y-MP 4/216 in 1990.

**Rear impact**

The rear impact model depicted in Figure 19 shows the necessity of including details, such as the spare wheel, in the analysis as their influence on load-carrying behaviour cannot be neglected. The model contains 18050 shell and 122 beam elements and is described in detail in [60]. A moving rigid barrier simulating a so-called impact car with a mass of 1814 kg impacts against the vehicle with a velocity of 50 km/h. The deformed shape after 100 ms, i.e. the end of the impact, gives a clear picture of the extent of deformation in the structure.

![Rear impact. Courtesy of Mercedes Benz AG.](image)

**Lateral impact**

Lateral impact is of dominant importance for occupant safety. Various types of loading are possible: either rigid barriers (ISO-barrier) or barriers with deformable layers, such as foam or honeycomb materials. Also skew impact is considered (NHTSA-barrier), for which frictional effects have to be taken into account. The deformed shape of a Mercedes Benz model is given in Figure 20 left (15907 shell, 114 beam elements) for impact with the EEVC barrier ($v = 45$ km/h, 142 shell, 495 solid elements). The VW model in Figure 20 right contains 22484 shell elements (+ some beams) and is impacted by the deformable NHTSA barrier. The velocity diagram in Figure 21 for the Mercedes simulation demonstrates the good agreement with experiments. Further details are given in [60].

**Offset crash**

Offset crashes are very dangerous from the aspect of occupant safety, as only a part of the frontal assembly acts to absorb energy in a crash event. Recently this effect was discussed widely in public [57], pointing out the weakness of the vehicles against this type of loading. Nevertheless, this was already known in the automotive industry [33] and has been investigated experimentally for some time. Some of the first simulations of an offset crash are described in detail in [20]. A view of the mesh used and a deformed shape are given in Figure 22. The FE model contains 20000 shell and 140 beam elements. The mesh is refined and concentrated on the side of the impact, which shows large deformations. It is obvious that the passenger cabin is still almost unaffected, which demonstrates a fairly satisfactory energy absorption mechanism. Due to the soft part with large
deformations, the impact lasts up to 100 ms, requiring 15 hours CPU time on a CRAY Y-MP 4/216. Good agreement of the deformed shapes between analysis and experiment was found.

Other cases
As cited above, further accidents can be simulated involving rollover [59] or car-to-car impact [41]. They are good examples of the ongoing development of applications. However, the considered examples also show the limits in interpretation and parameter changes due to the huge amount of data.

Remark: With regard to the good agreement of experimental and analytical results, it has to be mentioned that from a theoretical point of view, mesh convergence studies have not been performed in a sufficient fashion to allow predictions without experiments.

4 COMPONENT ANALYSIS

Despite the strong interaction of all components involved in a crash event, single components have to be studied also separately. The first reason is to find out whether the resolution with finite elements is sufficiently fine to reflect correct crashworthiness behaviour. Other reasons are to study the influence of design modifications on the corresponding component in order to reduce the effort and to gain insight into local behaviour. Small components have to be modelled by very small elements which, when included in a full car model,
will dictate a small time step size, rendering a full car analysis impractical.

### 4.1 Longitudinal girder

The single component tested separately in most automotive companies is the longitudinal girder, which has to carry the major load and absorb most of the energy in a frontal crash. Two typical models are depicted in Figure 23 after [21] and [31]. The effect of the endplate in the right girder, where buckling and folding is initiated further away from the end of the girder, whereas the open tube (right) starts to fold up directly at the point of impact, is clearly visible. A closer look at the right girder also shows two flanges, spot-welded together, where the connection starts to open during the folding process. The small amount of penetration visible in the left girder develops since no contact zones are defined behind the folding zone in order to reduce the computing time. These minor inaccuracies have no significant effect on the energy-absorbing behaviour tested in this example.

### 4.2 Composite crash element

As the behaviour of the longitudinal steel girders in a crash tends to produce a rather unsteady force deflection curve with high peak values, the favourable property of composite elements, that of producing a relatively constant load level, was taken into consideration. The unsteady curve for the metal tubes is the result of the folding behaviour, where peaks occur (Figure 24) just before the folds start to develop and where low values are observed during folding. The crushing behaviour of composites, where the total structure of the laminates is destroyed, does not show such folding behaviour if the load is applied in the longitudinal direction of a tube imposing only membrane stresses, thus avoiding bending.

The favourable location of such crash elements is just in front of the longitudinal girders shown above. For these applications, a major research project was started involving experiments and numerical simulations [47, 65]. The simulation of the real crushing would require an extremely fine meshing with solid elements. Thus, shell elements, which can only roughly represent the complex failure behaviour, were used. The sequence given in Figure 25 shows some deformed shapes. The force deflection curve given in Figure 26 displays the very good agreement with experimental data and also exposes the expected behaviour of the composites.

### 4.3 Front subframe

Many members in the frontal structure have to be designed for crash situations, e.g. parts like the front
subframe (Figure 27) of a Mercedes passenger car. In particular, the energy absorption of this member is of interest. The subframe has a wall thickness between 2.0 mm and 4.0 mm. The FE model, taking advantage of the symmetry of the geometry of the structure, contains 2120 shell elements [42]. Elasto-plastic material behaviour with hardening is assumed. In the computer simulation the front subframe is hit by a moving rigid wall (mass 750 kg, velocity 15.5 m/s) at the wide end. The expected behaviour is observed, as a hinge starts to develop at the narrow regions and the frontal part bends below the rear part. The energy absorption observed is about 10% after 30 ms. The very complicated failure involving folding and contact between

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**Figure 25** Cylindrical composite crash element; development of deformation.

**Figure 26** Force-time, force-deflection curve for composite crash element.

**Figure 27** Front Subframe. Finite element model and deformed structure after impact. Courtesy of Mercedes Benz AG.
5 OCCUPANT SAFETY ANALYSIS

Occupant safety is the ultimate goal in crashworthiness design. This is also reflected in many studies carried out to simulate passenger-structure interaction, e.g. passenger interaction with the steering wheel/column, airbags and belt systems. Mainly simulations involving only the components in the immediate vicinity of the occupant are performed. Full car analysis including occupants’ seats and steering wheels etc. is rarely carried out.

5.1 Steering wheel system

The steering wheel itself and the steering wheel mounting system can be carefully checked and improved with computer simulation. Two different models are depicted in the following figures. The first investigation [51] (Figure 28) is concerned with the absorption of energy by the steering column and is therefore loaded by a rigid, flat moving wall hitting the wheel and simulating the upper torso of a dummy. The sliced figures (Figure 29) show the deformation pattern during loading.

Figure 28 Steering wheel-column system. Courtesy of Saab Automobile AB.

Figure 29 Steering column, exterior and sliced view of details.

Another very detailed study on a steering wheel under dummy impact is reported in [16]. Excellent results were achieved in comparison to experiments.

5.2 Energy absorbing element

To reduce head injuries energy absorbing elements are incorporated into the steering wheel construction of Mercedes Benz passenger cars. Form of the central pad and material were investigated experimentally and analytically [42]. The FE model is shown in Figure 30, where a quarter sphere with the mass equivalent to the head is impacting the central pad (wall thickness 0.8 mm) with a velocity of 24.1 km/h. The finely zoned finite element model contains 1750 shell elements, where ideal elastoplastic material was assumed for the central pad and rigid material for the head. Since energy absorbing is the main objective the central pad has to deform considerably (Figure 30 right). As a result of this design study, the maximum deceleration of the head could be reduced considerably below the allowable level.

Clearly defined studies, as discussed above, display the strength of the finite element simulation, as form and material can be easily varied in order to achieve an optimal design.

Figure 30 Energy absorbing element (central pad) before and after impact. Courtesy of Mercedes Benz AG.
5.3 Rear seat under luggage impact

Part of the danger in a crash event originates from the trunk of the car, if movable goods such as heavy luggage are being transported. These goods may impact the rear panel of the back seat with a considerable amount of kinetic energy when the car is breaking. The prismatic body in Figure 31 hits the panel in a model impact with a velocity of 10 m/s. The panel consists of a composite material which is reinforced with steel at the connection joints and holding points, see [18]. The finite-element mesh chosen contains 2131 shell elements for the panel, the impacting item of luggage is modelled with 500 solid elements, most of them as rigid material. The deformations and stresses obtained in the analysis, which was performed up to 40 ms, give a good inside view into the weak points in the structure under this load case.

5.4 Airbag and restraint system

The incorporation of airbags and restraint systems into the analysis also leads to requirements on the modelling of the dummy.

Various approaches for dummy modelling exist: (a) modelling with flexible finite elements and using rigid parts around the joints; (b) modelling with rigid bodies and the joints in a FE-program; (c) modelling with rigid bodies and analysing the response using the MADYMO3D code [69] for rigid body calculations and the finite element code for contact and airbag/belting system analysis in a coupled version of the codes [59, 68]. All three possibilities have their advantages; the flexible finite element solution for the dummy is closer to reality but very time consuming, whereas the second solution is very cost-effective. However, in both choices (a) and (b), the analysis is limited to the specific joints available. If the given joints are not sufficient and if validated dummy models are preferred, the coupled approach using the sophisticated joint capabilities offered by MADYMO3D seems to be appropriate. The following examples mainly use approach (a) or (b).

The first investigation [1] is on a belted Hybrid III—dummy, which is modelled with rigid body parts connected by joints (Figure 32). The seat belt is simulated using quadrilateral flexible shell elements. The whole model including the interior is decelerated from an initial velocity of 14.8 m/s, according to a given curve simulating an impact. The analysis and the very complex model are described in detail in [1] and show good agreement with other known analyses. 1.5 hours CPU time were needed for an analysis up to 150 ms on an APOLLO DN 10000.

The same dummy is used in a simulation involving an airbag. The surrounding interior and the dummy are modelled as depicted in Figure 33. The airbag is in its inflated position at the start of the analysis, when the
car is decelerated from an initial velocity of 16.6 m/s according to a given diagram. The analysis is performed until rebound after 150 ms. The calculations required around 3.4 hours CPU time on an APOLLO DN 10000. The comparison to other analyses shows fairly good agreement, see [1].

A further informative study, in particular concerning coupled simulations, is described in [63].

The airbag model mentioned above is based on a simple inflation model suggested by [70], which allows simulation of the general behaviour of an airbag, i.e. folding and unfolding, as well as interaction simulations as shown above [30, 59]. The skin of the airbag is modelled either with membrane elements or with shell elements using one-point integration through the thickness. Standard airbags are made out of fabric materials with strong anisotropies. These materials wrinkle if compressive stresses develop. This poses some difficulties on the simulation, if non-wrinkling membranes have to be modelled without neglecting standard continuum mechanics and at the same time the energy of the system has to be preserved. The non-wrinkling or so-called smeared wrinkling models, though somewhat inconsistent with theory, are preferred in many situations, in order to use only a rather coarse mesh for the airbag skin, which also reduces the computer time needed in contact calculations. Otherwise, sufficient elements have to be chosen to represent the wrinkles correctly. However, this non-smooth surface then leads to a rather rough contact surface.

As a final example [62] describe a 50% Hybrid III dummy, which is very finely meshed with flexible solid elements (Figure 34) and is sitting in a complete passenger cabin. At the time of writing this article, it is the only FE dummy known by the authors to be meshed to such perfection. At the start, the cabin is at rest and the dummy is given an initial velocity. A total of 8000 elements are used. The airbag is located in the wheel in an initially flat position. The inflation phase is started at the beginning of the analysis so that the dummy finally contacts a fully inflated airbag (Figure 35). Due to the
wrinkling in the airbag structure, many difficulties in the contact analysis were observed, as the wrinkles formed a very rough surface for the airbag. In order to reduce the computational effort, the arms of the dummy were not considered for contact with the airbag, because already 11 CPU hours on a CRAY 2 were needed for the original model.

Airbag inflation and unfolding from real fully folded positions constitute an extremely difficult task for any contact algorithm, as multiple surfaces with very little stiffness are closely packed and any contact force results in considerable movements. Some research studies, e.g. [29], show that folding and unfolding can be modelled using a refined version of the contact algorithms described in Section 2.5.

Simulations of the inflation process with real flow and correct pressure distributions interacting with the unfolding structure in three dimensions are still in the development stage. An initial step with interfaced programs is presented in [14].

6 CONCLUSIONS AND FURTHER DEVELOPMENTS

The current state of the art in crashworthiness analysis has been presented. An overview on the theoretical background has been given, together with selected examples from applications in the automotive industry. Many details important for modelling in practical applications have not been discussed, but are available as analysis options in most standard codes, e.g. constraints, arbitrary rigid body definitions, static preloading in order to accommodate gravity forces, restart capabilities with multiple possibilities for modifications, submodelling capabilities and so forth.

However, still many fields in the theoretical basis for the analysis programs need to be improved. More efficient shell elements need to be developed, e.g., low order triangular elements with improved membrane behaviour are needed in order to simplify meshing (a general task also for standard FE programs). New and more efficient material models need to be developed, e.g. for honeycomb, foam and composite materials. Materials failure and fracture need to be incorporated into the analysis more accurately, and currently many research projects are active in meeting these demands. There is an urgent need for more efficient contact algorithms since they are mainly responsible for the long computer times needed today.

Nevertheless, the situation of software and hardware is such that the majority of applications in the automotive industry can be analysed by efficient analytical instruments. New developments in hardware, such as vector and parallel computers, are extremely well suited for the explicit algorithms used in the solutions, and considerable speed-up in turnaround times for crashworthiness analyses can be expected in the near future.

Finally, the major obstacle for more intensive application of crashworthiness analysis has to be mentioned. Today, the creation of the geometry and the meshing of the models occupy most of the engineer's time. Even much faster analysis will have only a small impact on the solution of the complete workload problem. Unless the capabilities in the generation of the models can be improved by a factor of at least 5 ÷ 10, many of the opportunities and advantages offered by the currently available analysis tools cannot be exploited.

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